N-96-01
II-A-206

EPA 550/9-82-334

ANALYSIS AND CONTROL OF
MECHANICAL NOISE IN
INTERNAL COMBUSTION ENGINES

UNITED STATES · ENVIRONMENTAL PROTECTION AGENCY

JULY 1982

FINAL REPORT

CONTRACT NO. 68-01-4737

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Natan E. Parsons

Richard G. DeJong

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

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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. Detailed reports covering specific phases of this work have been previously published (see EPA 550/9-82-335 and EPA-550/9-82-336). The authors wish to acknowledge the Massachusetts Institute of Technology which served as a subcontractor on the project, the Cummins Engine Company, the Detroit Diesel Allison Division of General Motors Corporation, and the John Deere Company. The cooperation and assistance of these organizations has been an essential ingredient to this project.

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I. INTRODUCTION

During the period from December, 1977 to June, 1982 Cambridge Collaborative, Inc. performed an engineering study for the Environmental Protection Agency (Contract No. 68-01-4737) to demonstrate methods of noise control for internal combustion engines. The program was carried out in two major phases. The goal of the first phase was to understand in depth the process of mechanical noise generation within an internal combustion engine through the development, usage, and verification of a modeling technique called transfer path analysis. This technique made it possible to evaluate the transmission of vibrational energy from the various mechanical excitation sources, such as combustion and piston slap, to the external radiating surfaces of the engine.

The goal of the second phase of the program was to utilize transfer path analysis to determine what changes could be made within an engine structure to yield a significant overall noise reduction without incurring penalties in other engine parameters such as fuel economy, emission, weight, and durability. Two demonstration engines were constructed to demonstrate the reduction of in-truck noise.

In the performance of the program Cambridge Collaborative studied and tested seven different engines. Under the first phase, five engines were tested in seven configurations. In the second phase two engines were redesigned and tested. The test engines were the following:

<u>Phase I</u>

a. 4-cylinder gasoline engine for a passenger car

b. 6-cylinder gasoline engine for a passenger car

c. 4-cylinder diesel engine for construction and farming equipment

d. 6-cylinder, in-line configuration diesel engine for construction equipment and trucks

e. 6-cylinder, vee configuration diesel engine for construction equipment and trucks

Phase II

a. 6-cylinder vee configuration diesel engine for truck classes 6-7

b. 6-cylinder in-line configuration diesel engine for truck classes 6-7

This final report summarizes the work that was performed on all of the engines and the various results that were derived. The report is organized according to the outline below:

A. Discussion of theory of vibration transmission and control in diesel engines

B. Description of transfer path analysis

C. Summary of work on two gasoline engines

D. Summary of work on the John Deere 4219D engine

E. Summary of work on DDA 6-71 and 6V-71 engines

F. Summary of verification work on the Camaro engine with transmission

G. Summary of verification work on the John Deere 4219D engine

H. Summary of work on the DDA 6V-92TTA engine

I. Summary of work on the Cummins NTC-350 engine

J. Conclusions.

II. BASIC THEORY OF VIBRATION TRANSMISSION AND CONTROL IN DIESEL ENGINES

2.1 Basic Theory of Vibration Transmission

Within a diesel engine there are three major noise sources: exhaust, intake, and mechanical noise. The first two noise sources can be controlled through the use of various mufflers and chamber resonators. The noise reductions achievable through the use of such devices appear satisfactory at the present time. This is not the case for mechanical noise, however, since methods to control mechanical noise are complex and highly dependent on other engine parameters. Mechanical noise, which is radiated by the external surfaces of the engine, is excited by a number of different mechanisms such as combustion, piston slap, gear mesh, injection apparatus, and other mechanical impacts within the engine. The relative order of importance of these mechanisms and their contribution to overall mechanical noise is dependent upon the parameters listed below:

1. the configuration of the diesel engine (in-line, vee, opposed piston, etc.);

2. the type of combustion process (2-stroke, 4-stroke);

3. the type of combustion chamber (pre-chamber, single chamber, fast swirl, low swirl);

4. the type of injection apparatus utilized (unit injector, pump and nozzle type injection, pressure time system);

5. the injection pressures;

6. the ratio of bore to stroke of the piston;

7. the piston to liner clearance and piston to liner expansion rates;

8. the type of gears used (straight out, helical, tapered helical, and double helical);

9. the type of material used for the engine construction;

10. whether the engine is turbocharged or naturally aspirated.

It is generally recognized that in direct injection, 4-stroke, naturally aspirated, conventional diesel engines, combustion and piston slap are considered to be the major noise sources [1]. These mechanisms generate forces containing substantial amounts of energy in the frequency range from 500 to 3000 Hz. This frequency range is one to which the human ear is highly sensitive, and it is also a frequency range in which the external radiating surfaces of the engine block generally radiate efficiently.

The reason why both the combustion forces and piston slap generate forces at this particular frequency range can be attributed to separate mechanisms. In the case of the combustion system, when there is injection directly into the combustion chamber with relatively high injection pressures (10 to 17 kpsi), the process of combustion occurs at a relatively rapid rate in time. This in turn generates forces with high frequency content. The need for the rapid combustion process stems from the relationship that exists between the duration of combustion and the fuel consumption of the engine. In general, the faster the combustion process, the more efficiently the fuel is burned, which in turn leads to better fuel consumption. This is in contrast to a slow rate of pressure rise in the combustion chamber which would be preferable from a noise point of view.

The mechanism that generates the high frequency components in the spectra of piston slap is the impact between two solid bodies of the piston and the liner. The shape and level of the frequency spectrum generated during the impact is controlled by three factors: (a) the engine speed, (b) the mass of the piston, and (c) the local stiffness of both the piston and liner at the point of impact. The magnitude of the impact is also controlled by several factors: (a) the piston mass and location of the center of gravity, (b) the local stiffness of the piston and liner which is associated with the geometry and material of these elements, (c) the clearance between the piston and liner, and (d) the local damping of the piston-liner interaction.

It should be emphasized that the clearance between the piston and liner has a dominating effect. For example, for two identical systems of piston and liner having different gaps, the magnitude of the impact will be higher for the larger piston to liner clearance. This can be expressed in the following mathematical expression for the momentum of the impact:

$$\text{momentum} = M_{piston} \times V_{impact}$$

$$V_{impact} = \frac{c}{2} \left(\frac{6d}{c}\right)^{2/3}$$

c = a constant
d = piston to liner clearance
M_{piston} = mass of the piston
V_{impact} = velocity of piston at impact

Based on this mechanical expression it is evident that from a noise point of view it would be preferable to have as small a liner to piston clearance as possible. From the point of view of engine endurance requirements and wear rates of piston rings and liners, however, it is preferable to have larger piston to liner gaps.

As a result of the requirement for larger piston to liner clearances and the shortest duration of combustion from an engine performance standpoint, it is evident that the control of these noise sources is very complex and highly dependent on other engine criteria. In certain cases such as those described above, these other engine criteria can oppose noise requirements.

The forces that are generated internally within the engine structure are transmitted through various engine components to the external surfaces which vibrate and radiate noise. The transmission of vibrational energy is a relatively complex process where for one given excitation source there will generally be a number of transmission paths to the external radiating surfaces. Fortunately, in most cases each transmission path has its own characteristics. That is, when two transmission paths are compared with the same force input, there will be a difference in their transfer characteristics. One path will transmit energy in one frequency range while the other path will be a major contributor in another frequency range. The controlling factors that determine the response of a particular transmission path to a force within a given frequency range are: (a) the vibration characteristics of each element in that particular transmission path, (b) the method by which each element in the path is coupled to the next element, and (c) the damping values of each element in the transmission path.

In conventional direct injection, 4-stroke, naturally aspirated engines the trancmission of combustion forces is primarily through main paths. The first path is through the head to the block, and the second is through the piston-connecting rod-crankshaft to the lower block. The two main transmission paths of piston slap excitation are: liner-inner block-block radiating surfaces, and in certain cases, piston-connecting rod-crankshaft-lower block.

In conclusion, the three determining parameters that control the magnitude and the frequency content of radiated engine noise due to combustion forces are: (a) the shape of the pressure signal within the combustion chamber and its frequency content, (b) the characteristics of the various transfer paths from the combustion sources to the external block surfaces, and (c) the radiation efficiencies of the external surfaces in the particular frequency range in which they are excited.

2.2 Principles in Noise Control of Diesel Engines

Generally, three basic approaches are taken toward the control of overall engine mechanical noise. They are:

1. treatment of radiating surfaces of the engine

2. alterations of the excitation mechanisms

3. alteration of the transmission paths.

The following paragraphs discuss each of these approaches in more detail.

2.2.1 Treatment of radiating surfaces of the engine

In principle, this approach consists of two techniques to reduce emitted noise from the external radiating surfaces of the engine. The first method uses covers and shields that are fitted over the engine's external surfaces. These covers are usually constructed of a mass layer with relatively high damping values backed by some type of absorption material such as foam or fiberglass. The covers are mounted on the engine using various types of resilient mounts in order to prevent short circuiting of the treatment. This method has several limitations such as cost, relatively short life cycle, and the fact that it often interferes with ease of accessibility to the engine for servicing purposes.

The second method used to control noise emission from the radiating surfaces has two elements. The first is an attempt to change the radiation efficiency characteristics of these surfaces by reducing the thickness of the outer walls, thus raising the coincidence frequency (critical frequency) beyond the frequency range of interest. This step creates substantial problems, however, relating to the block structure integrity. The second element involves shifting resonances of the radiating surfaces to higher frequencies by stiffening the surfaces. As a result the modal response of the surfaces in the frequency range of interest will be reduced. This approach can result in substantial increases in engine weight which goes against present engine design philosophy. The most common technique used to evaluate these changes is dynamic finite element modeling.

2.2.2 Alterations to the excitation mechanisms

This approach presents substantial difficulties in achieving overall noise reduction without having to make sacrifices to other engine parameters such as fuel economy, durability, emission, and desired wear behavior.

In the case of the combustion process, it is evident from the previous discussion that in regard to noise, a long pressure pulse with a slow rise rate is preferable in order to reduce the high frequency content. This, however, is contrary to the requirements for fuel economy. It is also contrary to CO emission requirements, but not to the emission of NOx pollutants which requires longer combustion duration resulting in lower peak combustion temperature and lower levels of NOx formation. It has been shown that the use of an indirect injection system, i.e., pre-chamber, reduces the rate of pressure rise in the combustion process which has positive effects in terms of noise but hampers the specific fuel consumption. The only method currently used that contributes significantly to the reduction of combustion excitation without penalties to fuel economy is the method of turbocharging.

Several attempts have been made to reduce piston slap excitation as a noise source but all of the methods used appear to have adverse effects on other engine performance parameters. The following list describes these methods and their inherent problems:

1. Reduction of piston to liner clearance results in substantial noise reductions but presents problems in the areas of piston ring and liner wear, and difficulties in the mass production of systems with relatively small tolerances. It also presents problems in meeting an SAE engine endurance requirement for heavy duty diesel engines that calls for the piston motion to be maintained for a period of ten minutes with no lubrication supply.

2. Coating pistons with various substances such as teflon or tin in order to reduce clearances presents problems associated with rapid wear rates for these materials [2].

3. Off-setting the piston pin towards the major thrust side (the side the piston impacts) appears to be a beneficial method for noise reduction but has adverse effects on fuel economy.

4. Reduction of piston mass also shows a noise reduction potential but contradicts the need for minimal material quantity needed to control the thermal expansion of the piston.

As can be seen from this list, there are substantial restrictions imposed on the changes that can be made in order to reduce the contributions from these excitation mechanisms. The same is basically true for other excitation mechanisms such as gear mesh and injection mechanisms. It is believed that some progress will be made in the future regarding excitation control but not immediately, primarily because of the important dependence on the development of new materials.

2.2.3 Alteration to transmission path

The third approach taken toward noise reduction of internal combustion engines is alteration of the transfer path of vibrational energy from the excitation sources to the radiating surfaces. This approach utilizes the analytical technique of transfer path analysis. By use of this technique it is possible to identify the main transmission paths for a given frequency range. By using the baseline data acquired with this technique for a given engine, it is possible to determine what changes in the transfer path are needed in order to achieve a desired noise reduction. There have been several papers published that demonstrate the feasibility of this approach without substantial penalties to the other engine parameters [3]. The methods and techniques of transfer path analysis will be discussed later.

When an attempt is made to achieve a reduction of vibration transmission in a given transfer path such as piston to crankshaft, it is necessary to know the dynamic characteristics of each element of the transfer path in the frequency range of interest. By creating mismatches in the vibration responses of these elements, there is an effective isolation in the transfer of vibrational energy through this path. Instead of the vibrational energy being transmitted through the system, it is reflected back toward the excitation source which in turn results in higher vibrational energy levels near the source. Since there are significant damping values associated with the piston to liner inteaction, a larger portion of the vibrational energy generated at the source will be dissipated as heat.

III. TRANSFER PATH ANALYSIS

3.1 Measurement of Vibration Transmission Paths

The measurement of the transmission of vibration in any mechanical structure involves the measurement of the response (velocity) at one point for a given excitation (force) at another point. The ratio of velocity to force is called the mobility:

$$Y = V/F. \tag{1}$$

The mobility (inverse of impedance) is convenient to use as the transfer fraction of a mechanical system because the resonances of the system appear as peaks in the mobility function. The ratio of velocity to force is a function of frequency and a complete representation involves both magnitude $|Y|$ and phase ϕ.

The development of digital signal analysis equipment has greatly facilitated the measurement of vibration transmission. The instrumentation used for the measurements in this study is shown in Figure 1. A force impulse is used to excite the structure and an accelerometer is used to record the response. The impulse hammer is equipped with a force gauge to measure the input force signal. Figure 2 shows a typical time history and frequency spectrum of the force impulse. The force spectrum indicates that the impulse has enough energy to excite the structure at all frequencies from 0-5000 Hz.

The transient force and response signals are recorded and transferred to a mini-computer where the frequency spectra of each signal and the mobility ratio are calculated. The results are plotted and stored on disk for future usage. For the measurements reported here the frequency resolution of the spectral data is approximately 25 Hz constant bandwidth.

Figures 3-6 show examples of the measured mobility functions for several components in a 4-cylinder diesel engine. Each component was measured while removed from the engine and freely suspended by flexible cords. The piston-connecting rod assembly

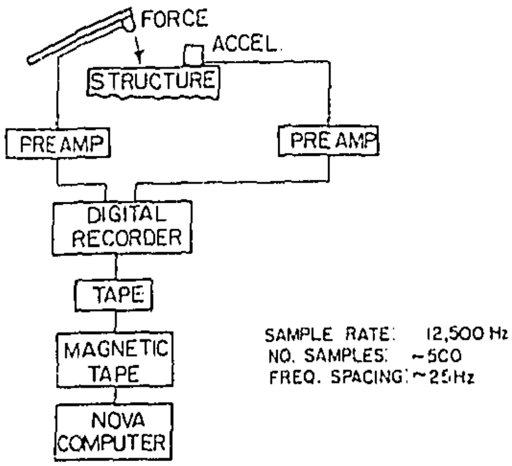


Figure 1. Instrumentation for Mobility Measurements

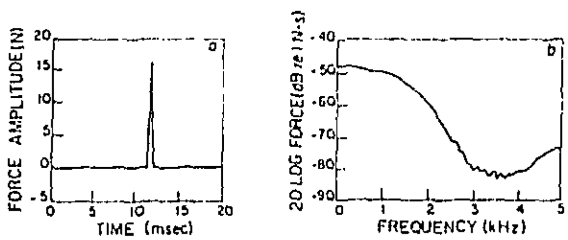


Figure 2. Typical Force Impulse Used to Excite Vibration in Engine Structures: a) Time History, b) Frequency Spectrum

(Figure 3) shows a resonance at about 4000 Hz. Below this resonance the vibration transmission characteristics of the piston-connecting rod assembly is controlled by the stiffness of the connecting rod. Figure 4 shows the drive point mobility (force and velocity measured at the same point) of the piston top. Comparing this with Figure 3 indicates the difference in the velocity level at either end of the piston-connecting rod assembly.

The crankshaft (Figure 5) has several resonances in the frequency range of interest. To aid in understanding the vibration of the crankshaft a shaker was used to excite it at the resonances and the mode shapes of the vibration were measured. Below 3000 Hz these resonances were found to be bending modes with the crankshaft acting basically like a beam. Above 3000 Hz the mode shapes were difficult to measure because of the complicated motion coupling torsional, bending, and axial modes. This complexity would undoubtedly increase for crankshafts in engines with more than 4 cylinders.

The transfer mobility of the oil pan (Figure 6) shows a large number of resonances which are too close together to count from a measurement of this resolution. The average spacing between the modes can be calculated for a flat plate with the same surface area S and thickness h as the oil pan using the formula [4]

$$\Delta f = \frac{0.57\ hc}{S} \qquad (2)$$

where c is the longitudinal wavespeed in the material. Although the exact frequency value of a given mode in the oil pan will be different from that of the corresponding mode in the flat plate, the average spacing between modes remains the same. For the oil pan measured in Figure 6 the calculated average frequency spacing is 20 Hz.

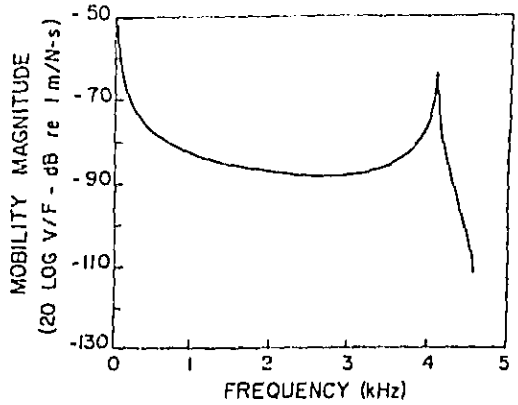


Figure 3. Measured Transfer Mobility of Piston from Top to Big End

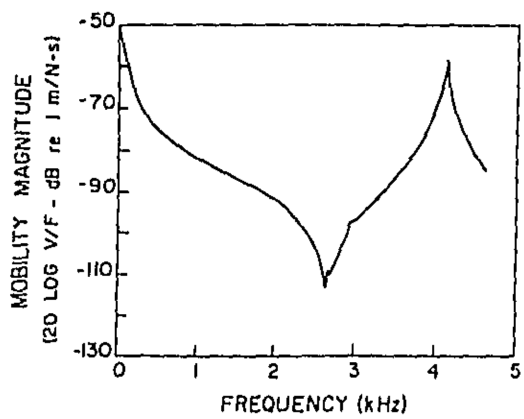


Figure 4. Measured Drive Point Mobility of Piston Top

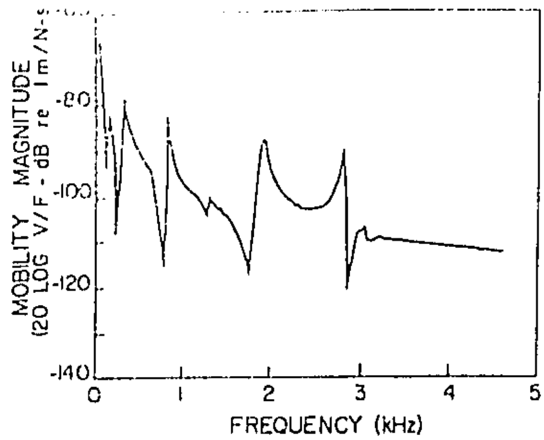


Figure 5. Measured Transfer Mobility of Crankshaft from Pin #2 to Journal #2

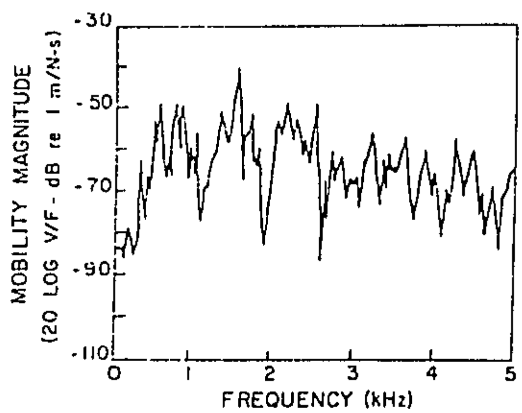


Figure 6. Measured Transfer Mobility of Oil Pan from Flange to Surface Point

3.2 Modeling the Vibration Transmission Paths

Having measured the vibration transmission characteristics of the individual components of the engine, it is possible to combine the measurements numerically to predict the overall vibration transmision of the assembled components. However, this does not give any information about how the physical designs of these components affect the vibration transmission. Therefore it is more useful first to construct analytical models of the individual components and then to use the representation of the model to predict the overall vibration transmission. In this way changes in the design parameters of the engine components can be directly related to changes in the vibration transmission within the engine.

Two types of models can be used to represent the vibration transmission characteristics of the individual components. First, when the component has relatively few resonances below the maximum frequency of interest, a lumped parameter or finite element model (depending on the complexity of the component) can be used to match the transfer mobility function over the frequency range desired. As the number of resonances increases it becomes more difficult to match the mobility function with this type of model. However, it may not be necessary to match exactly every resonant frequency if all one is interested in is the average transmission in a frequency band (such as a one-third octave band). In this case, if there are more than about three resonant frequencies in the frequency band of interest, it is possible to use a statistical model of the component using Statistical Energy Analysis (SEA) [5]. With this type of model the mobility function is matched only on the average over each frequency band. The following gives some examples of both types of modeling.

PISTON MODEL - A lumped parameter of the piston-connecting rod can be constructed using a method similar to that for synthesizing electric networks [6]. The vibrational motion is assumed to be purely longitudinal and a series of mass-spring elements are combined to match the resonance and anti-resonance frequencies in the drive point mobility of the piston top (Figure 4). Dashpots are then put in parallel to the springs to match the peak amplitudes of the resonances. The resulting model of the piston and the comparison of the measured and predicted transfer mobility are shown in Figure 7.

CRANKSHAFT MODEL - A similar model of the crankshaft can be made assuming that the motion is transverse to the axis and that the crank model is then construced to match the drive point mobility at the center journal. The resulting model of the crankshaft from the 4-cylinder engine and a comparison of the measured and predicted transfer mobility between two points are shown in Figure 8. The comparison becomes poor at higher frequencies because of the breakdown in the assumption of purely transverse motion.

A comparison of the measured and predicted mode shapes of the crankshaft is shown in Figure 9 for the first four resonant frequencies. Whereas the model is symmetric, the crankshaft does not have exactly symmetric mode shapes. However, the match in the first four mode shapes is fairly good.

OIL PAN MODEL - A statistical model is used for the oil pan because of the large number of resonant frequencies below 5000 Hz (Figure 6). Since the average resonant frequency spacing of the oil pan was calculated to be about 20 Hz, any frequency band greater than 60 Hz in width will have a sufficient number of modes for the SEA model. In using one-third octave bands this corresponds to a lower frequency limit of 250 Hz.

The average drive point mobility of a plate-like structure can be given by

$$\langle Y \rangle = \frac{1}{4\ \overline{\Delta f}\ M} \qquad (3)$$

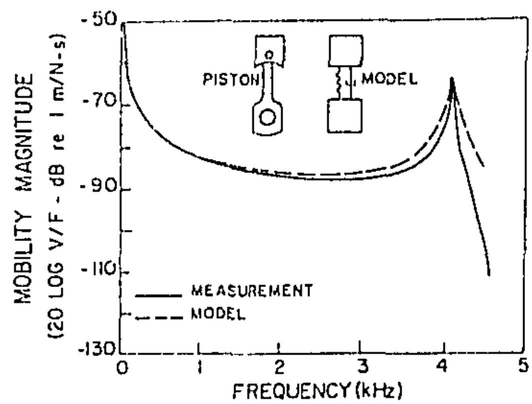


Figure 7. Comparison of Measured and Predicted Transfer Mobility of Piston

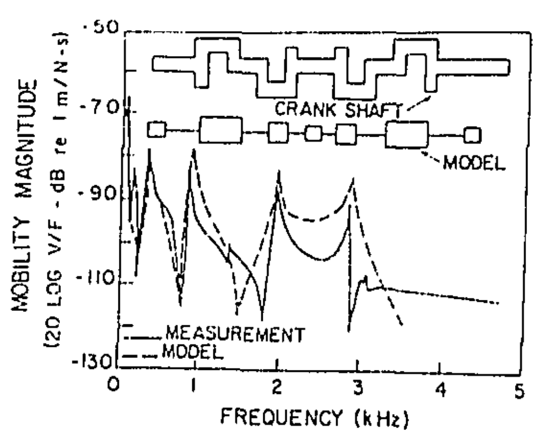


Figure 8. Comparison of Measured and Predicted Transfer Mobility of Crankshaft

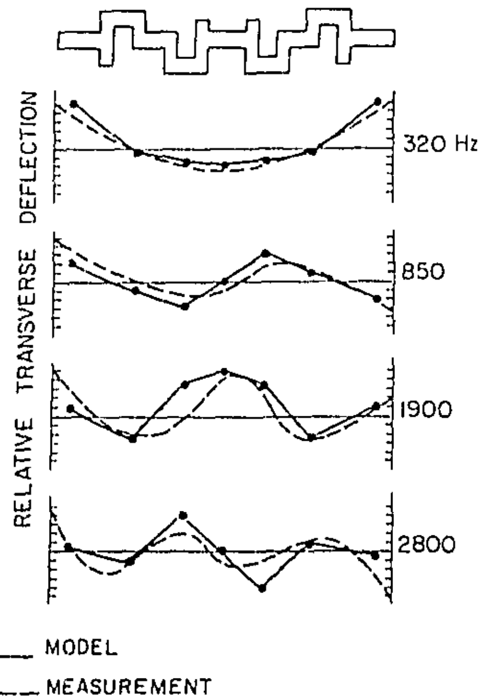


Figure 9. Comparison of Measured and Predicted Mode Shapes of Crankshaft

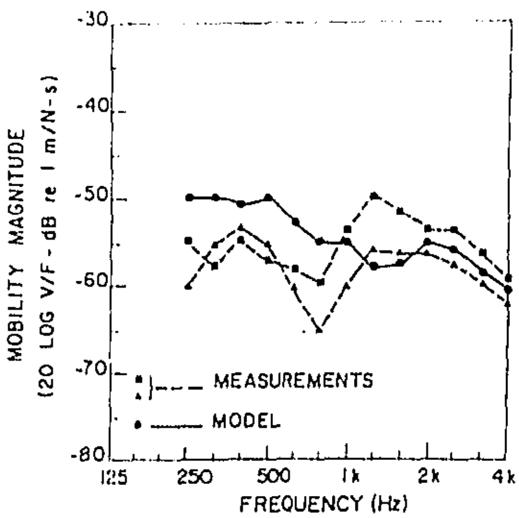


Figure 10. Comparison of Measured and Predicted Transfer Mobility of Oil Pan

where Δf is the average frequency spacing of the resonances and M is the total mass. Since the drive point mobility at the edge of a plate-like structure is approximately 4 times the average mobility [4], Equation 3 can be modified appropriately to model the average response of a structure by matching the drive point mobility at the edge. This is convenient for a component such as an oil pan which is normally connected to the engine along its edges.

In order to construct an SEA model for the oil pan, the drive point mobility was measured at seven points along the connection flange and the results were averaged. The model can then be used to predict the average transfer mobility from the edge of the oil pan to other points on the surface. Figure 10 shows a comparison between the model predictions and two measured transfer mobilities on the oil pan. The comparison becomes increasingly better at higher frequencies as the number of resonances in each band increases.

3.3 Constructing the Complete Transmission Path

The individual component models can be used to analytically connect the elements together to predict the overall vibration transmission of the engine by again using the methods of network synthesis [7]. For each component with an input and output point defined as in Figure 11, the velocity V and force F at the two points can be related by the matrix equation:

$$\begin{bmatrix} V_1 \\ V_2 \end{bmatrix} = \begin{bmatrix} Y_{11} & Y_{12} \\ Y_{21} & Y_{22} \end{bmatrix} \cdot \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} \qquad (4)$$

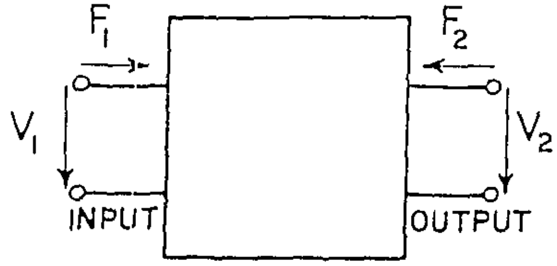


Figure 11. Model Definition of Forces and Responses

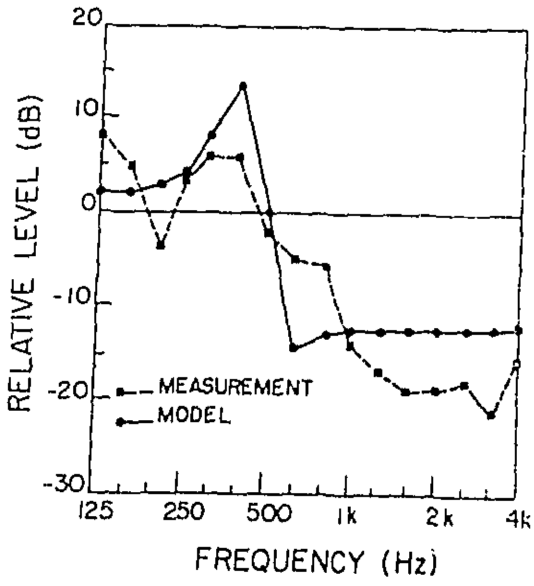


Figure 12. Comparison of Measured and Predicted Change in Vibration Transfer from Piston Top to Block Side Walls

where Y_{11} and Y_{22} are the drive point mobilities at either end and Y_{12} and Y_{21} are transfer mobilities. For linear, passive, bilateral systems $Y_{12} = Y_{21}$ [5]. Components with more than one input or output can be handled by letting the mobility functions themselves be matrices.

When two components, "a" and "b", are connected together at a point with the output of "a" attached to the input of "b", the conditions at the interface are considered to be such that $V_2^a = V_1^b$ and $F_2^a = -F_1^b$ (using superscripts to denote the component). In this case the overall transfer mobility of the assembly is given by [7]

$$Y_{12}^{(a+b)} = \frac{Y_{12}^b \, Y_{12}^a}{Y_{11}^b + Y_{22}^a} \tag{5}$$

The usefulness of this result comes from the fact that the mobility functions used in Equation 5 are obtained from analytical models in which the physical parameters have been derived from actual measurements of the components being modeled. This enables the engine designer to use the models to determine how to change the physical design parameters of the various engine components to achieve a reduction in the vibration transmission within the engine and thereby reduce the noise radiated by the engine. The following are some examples of the use of such models to identify quantitatively the amount of vibration transmission reduction which can be achieved through redesign of various engine components.

3.4 Combustion Pressure Transmission

In many direct injection diesel engines the major source of noise is the high frequency component of the cylinder pressure which excites the engine structure to vibrate. Two paths can be identified through which this vibration is transmitted to the external surfaces of the engine block. (1) The cylinder pressure exerts a force directly on the cylinder head which transmits vibration to the engine block through the head gasket and bolts. (2) The cylinder pressure exerts a force on the piston top which transmits vibration through the connecting rod, crankshaft, and bearings into the block. In several cases the path through this piston has been found to be the dominant one. This has been reported in other work [8].

The measurement of the transfer mobility through the piston-crank-bearing path was made with the piston at top dead center and an artificial static load applied to the piston with flexible couplers through holes in the head where the valves were removed. Heavy grade oil was used in the bearings during the measurements. This simulated as best as possible the conditions under which the high frequency component of the cylinder pressure, which occurs at the onset of combustion, is transmitted through the piston path. The vibration transmission was also measured with the crankshaft at various angles between ± 30 of TDC and very little change in the result was observed. Therefore, it is assumed that the vibration characteristics of the crankshaft at TDC represents the conditions during the entire combustion process. This may be true only in the case of 4-cylinder engines and other crankshafts with more complicated geometries need to be studied as well before a more general conclusion can be made. In all measurements the piston was isolated from the cylinder wall with a plastic sleeve to prevent impacts between the piston and the wall from contaminating the results.

A complete model of the vibration transmission path through the piston-crank-bearings was then constructed in order to identify a means of reducing the vibration transmission to the engine block. For simplicity only forces normal to the connection interfaces between the components were considered which is consistent with what would be expected at journal bearings. Also for simplicity, only two connections were considered between the crankshaft and the block, those being at the bearing on either side of the piston being modeled. Since a statistical model was used for the block, the output of the complete model is a prediction of the average transfer mobility from the piston top to the engine block surface averaged over one-third octave bands.

In order to gain insight into what types of design changes would reduce the vibration transmission through the piston-crank-bearing path, it is necessary to look more carefully at the interaction which occurs at the connection points between the various components as indicated by Equation 5. This equation states that the overall transfer mobility of two connected components is equal to a ratio with the product of the component transfer mobilities in the numerator and the sum of the drive point mobilities at the connection point in the denominator. In order to reduce the vibration transmission either the transfer mobilities need to be reduced or the drive point mobilities need to be increased, or both.

In the case of the connection between the piston-connecting rod assembly and the crankshaft, the drive point mobility of the connecting rod big end is generally much greater than that of the crankshaft. Therefore the latter can be dropped from the equation as an approximation. Also, over most of the frequency range the transfer and drive point mobilities of the piston-connecting rod are nearly equal in magnitude so that they tend to cancel each other out. The result is that the overall transfer mobility of the piston-connecting rod-crankshaft assembly is approximately equal to that of the crankshaft alone. This is a result of the fact stated earlier that the piston-connecting rod assembly acts like a spring below its first resonance at 4000 Hz and any force which is applied to the piston top is directly transmitted to the crank pin. Therefore, modifications to the piston-connecting rod will have little effect on the overall vibration transmission of that path unless the first resonant frequency is lowered by at least a factor of two or three. This is unlikely because of the constraints on the piston mass and connecting rod stiffness.

As a result the overall transfer mobility from the piston to the block is dominated by the connections at the crankshaft journals and the main bearings. The bearings are considered as inputs to the engine block model. Again referring to Equation 5, the interaction of these connections can be studied to determine if design changes can be made to reduce the vibration transmission. Because of the complexity of the crankshaft design it is not clear at this time if design modifications could be made which would significantly change the mobility functions. Therefore, only changes in the block design are considered here.

One means of reducing the vibration transmission is to decrease the transfer mobility of the block. This can be done by increasing the stiffness, mass, or damping of the block. Increasing the stiffness of the block at the main bearing supports has been tried by others with some success [9,10] and appears to be worth further investigation. Increasing the mass or damping of the block appreciably is an unlikely solution.

A second means of reducing the vibration transmission into the block is to increase the drive point mobility of the main bearings by increasing their flexibility. Although the bearings are by nature required to have high stiffness for load bearing capabilities, it may be possible to achieve a sufficient amount of resilience at higher frequencies. An attempt at such a design was implemented in the 4-cylinder engine by using a constrained layer of silicone rubber between the bearing rings and the bearing caps [11]. Although this design could not be directly implemented in a running engine, it was used to demonstrate the prediction capabilities of the model.

Using the resilient bearing design a change in the overall vibration transmission from the piston to the block was predicted and measured on the non-running engine shown in Figure 12. Above 1000 Hz the vibration reduction is inversely proportional to the stiffness of the bearings.

3.5 Vibration Transmission to Covers and Shields

The method of statistical modeling was applied to the noise radiated by the engine covers in a study conducted at Calspan Corp. under contract to the Department of Transportation [12].

One of the engines studied was a turbocharged, 6-cylinder, in-line diesel engine and the noise levels radiated from the individual surfaces and external components at 50% rated load were measured by the method of lead wrapping in an anechoic test cell. The results of the source identification are shown in Table 1.

The noise radiated by the block sides was treated by the addition of shields of various constructions. A variety of combinations of steel plates with and without damping treatment, with and without foam absorption on the block side, with and without isolation mountings, were tried. The measured mobility of a shield constructed with 16 gauge steel with 1/2" foam backing is shown in Figure 13 as compared with the drive point mobility of the engine block at a typical connection point.

Rather than use Equation 5 to evaluate an overall vibration transmission, in the case of the analysis of covers it is more convenient to modify the equation slightly to evaluate the ratio of the average cover velocity to the engine block velocity. This is done by multiplying Equation 5 by Y^a_{12} resulting in

$$\frac{V^b_{rms}}{V^a_{rms}} = \frac{Y^b_{12}}{Y^b_{11} + Y^a_{22}} \qquad (6)$$

where component "a" is the engine block and component "b" is the cover. In the case of lightweight covers the magnitude of the drivepoint mobility of the cover will be much larger than that of the engine block so that

$$\frac{V^b_{rms}}{V^a_{rms}} \approx \frac{Y^b_{12}}{Y^b_{11}} \qquad (7)$$

This result leads to some well known and frequently tried solutions to cover noise reduction which are the addition of damping to the cover to reduce the transfer mobility Y^b_{12} and the

RADIATING SURFACE	A-WEIGHTED SOUND POWER LEVEL (dBA RE. 10^{-12} WATTS)
BLOCK SIDES	101
VALVE COVERS	98.5
ENGINE FRONT	98
OIL PAN	97.5
ALL OTHERS	99
TOTAL	106 dBA

Table 1. Source Identification of a 6-Cylinder Diesel Engine

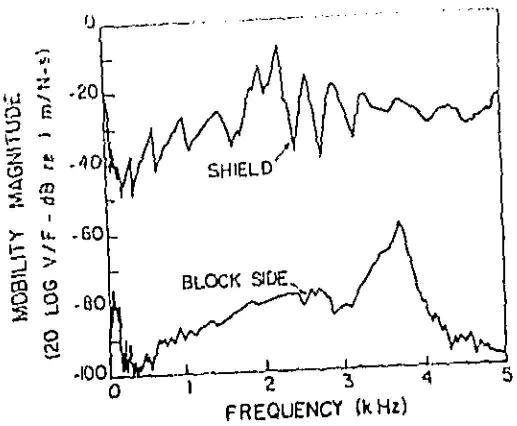


Figure 13. Comparison of Measured Drive Point Mobilities of Engine Block Side and Block Side Shielded with Foam Backing

use of isolated mountings to increase the drive point mobility Y^b_{11}. However, Equation 7 is useful in that it gives quantitative estimates of the noise reduction achievable.

In the case of the block side shields the foam gives enough damping to bring down the velocity of the shields to about 10 dB below that of the block. This will reduce the block side radiated noise proportionately provided the radiation efficiency is not significantly changed. A 10 dBA reduction in the block side source level would bring the overall engine noise level down by 2 dBA which is essentially the maximum achievable reduction possible by complete elimination of that source until other sources are also reduced. This was demonstrated in the measurements taken with the variety of block side shields mentioned previously. For all measurements taken with foam between the block and the shields an overall noise reduction of about 2 dBA was achieved. Without the foam between the block and shield the noise reduction was significantly reduced, apparently because of the buildup of noise behind the shield and leakage around the edges of the shield.

Similar measurements and analysis were done for the valve covers. The statistical model of the valve cover loses accuracy in the lower frequency range due to the lack of resonant modes. Specifications can be obtained from Equation 7 to determine the necessary treatment to reduce the valve cover source level by 10 dB. This can be achieved by using isolation mountings with a point stiffness of 5×10^5 N/m (3000 lbs/in) and the addition of damping to obtain a critical damping ratio of 0.05 in the frequency range of interest.

The oil pan was also studied with the mobility functions. This particular oil pan was made of aluminum, and it was found that by going to a steel pan with the same bending stiffness, the drive point mobility could be increased by a factor of 2. Combined with an increase in the oil pan damping factor to 0.03 critical, a noise reduction of 10 dBA was predicted for the oil pan source level. Such an oil pan was constructed and installed on the engine with a reduction in total engine noise of 1 dBA.

Unfortunately the new oil pan source level was not measured individually. However, a total reduction of 1 dBA would require at least a 10 dBA reduction in the oil pan radiation, although the actual oil pan reduction cannot be determined accurately by this method.

3.6 Conclusions

The following conclusions can be drawn from this study:

1. The vibration transmission paths in engine structures can be modeled with sufficient accuracy to predict changes in vibration transmission with changes in design parameters of the individual engine components. The component models can be constructed from measurements of the mobility on the individual components while freely suspended by flexible mounts. For components with only a few resonant modes below the maximum frequency of interest a model is used which matches the mobility functions over this frequency range. For components with many resonant modes a statistical model is used which matches the average of the mobility function in a given number of frequency bands.

2. Although the models used in this study were limited to point force transmission at the connection points, conceptually the models can handle both moment bearing and line connections. However, at this time there is no readily available method for measuring moment and line mobilities which would be necessary to construct such models.

3. The modeling of the vibration transmission has the potential to be extended to predict the total noise produced by the engine. In order for this to be

accomplished two additional things are needed. First, the force strength of the internal sources must be known along with an identification of the appropriate transmission paths to the external surfaces. Second, the nature of the noise radiation from the vibration of the external surfaces must be better understood in terms of its relationship to the physical parameters of the engine structure.

IV. SUMMARY OF WORK ON TWO GASOLINE ENGINES

This summary presents the results and analyses of tests performed on two gasoline engines during the initial portion of the program. This phase focused on the noise radiated by the surfaces of the engine and did not include exhaust, intake, or cooling.

The noise generation process of engines can be divided into three stages. The first stage involves the generation of internal vibration through pressures and forces within the engine structure. These internal sources of vibration include the combustion pressure, piston impacts, bearings, gears, pumps, and air flow through valves. The second stage involves the transmission of the vibration from within the engine to the external surfaces of the engine. The third stage involves the radiation of noise by the external surfaces.

This phase focused mainly on the second stage of the noise generation process, that of the transmission of vibration through the engine structure to the external surfaces. The goals of the investigation were to identify the important paths of vibration transmission from the internal sources to the external surfaces and to specify means of reducing the vibration of the external surfaces through changes in the design parameters of the transmission path elements. The specifications of noise reduction methods included quantitative values for the mass, stiffness, and damping of the engine components and their connection points to achieve various degrees of vibration transmission reduction.

The process of evaluating the vibration transmission was carried out in three steps. First, baseline noise emission measurements were performed to determine the amount of noise radiated by the various engine surfaces and the dependence of the overall noise on the speed and load of the engine. Second, vibration transmission measurements were performed on the non-running engine and analytical models of the transmission paths were constructed. Finally, the vibration transmission models were used to identify changes in the design parameters of the engine components which would reduce the vibration transmission.

Within this third step, two types of vibration transmission were studied. The first was the transmission of vibration from the engine block to the engine covers and the second was the transmission of vibration through the piston-crank-bearing path into the engine block. These two types of vibration transmission are important contributors in the noise generation process of the engine, but are not necessarily the only important ones.

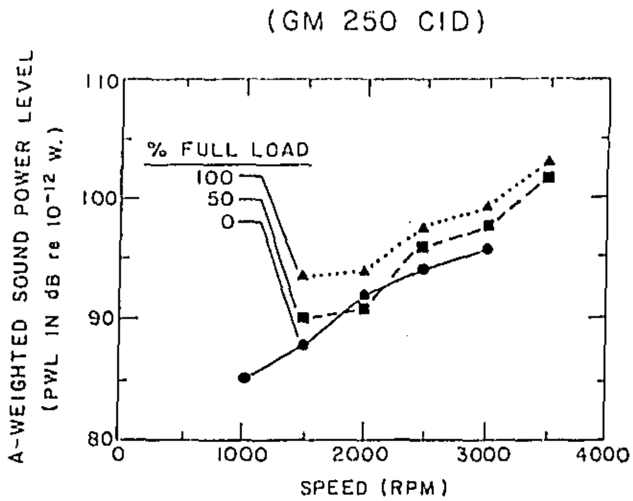
Two gasoline engines were chosen for study to allow a comparison to be made of the vibration transmission characteristics for two different engine block constructions. The first engine studied was a 6-cylinder, in-line with a cast iron block (1976 General Motors 250 CID Camaro engine). The second engine studied was a 4-cylinder, in-line engine with an aluminum block (1977 General Motors 140 CID Vega engine). Both engines were obtained from used automobiles and had less than 10,000 miles on them.

The results of the study yielded insight into the noise generation process within these engines and methods by which the noise might be controlled. The following observations were made:

1. In the case of both engines the magnitude of the noise emission is more dependent on engine speed than on engine load (see Figure 14).

2. In the case of the 6-cylinder gasoline engine the major radiating surfaces can be ranked in the following order: left block side, right block side, oil pan (see Figure 15).

3. In the case of the 4-cylinder gasoline engine the major radiating surfaces can be ranked in the following order: oil pan, left block, right block (see Figure 16).



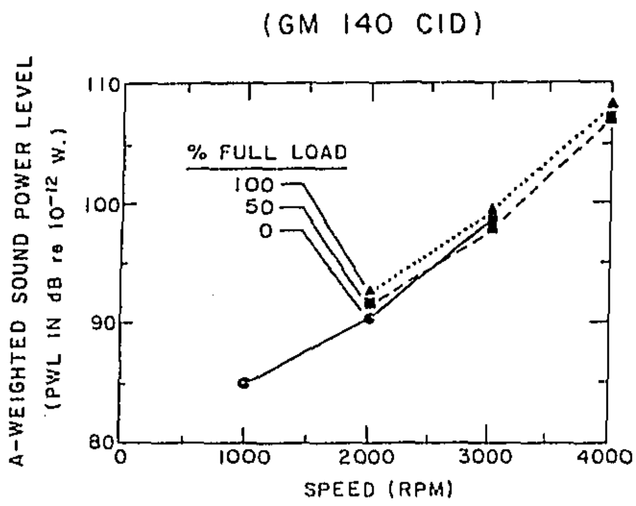


Figure 14. Noise Emission vs. Speed at 3 Load Conditions: a) Camaro Engine, b) Vega Engine

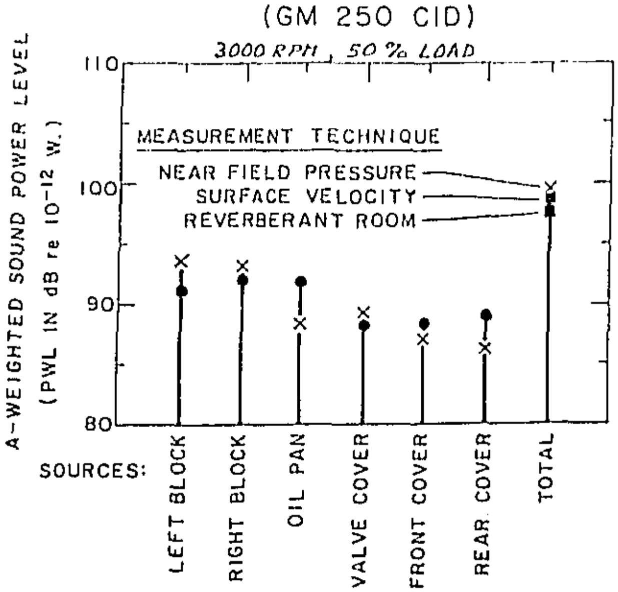


Figure 15. Noise Source Ranking of Camaro Engine

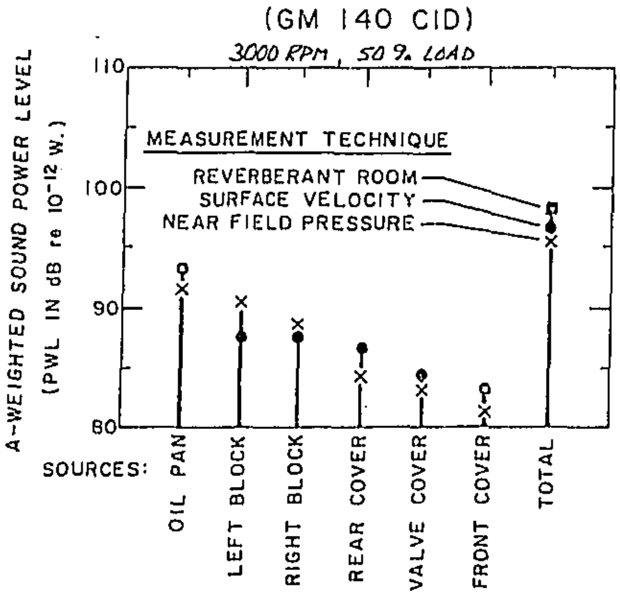


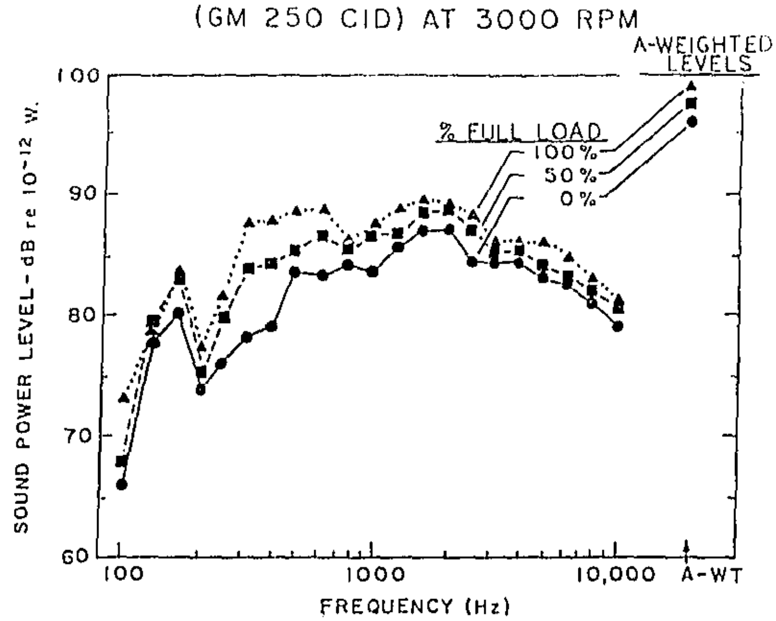
Figure 16. Noise Source Ranking of Vega Engine

4. At 3000 rpm, 50% load, the 6-cylinder gasoline engine appeared to be slightly quieter than the 4-cylinder engine (this might be attributed to the inherent balancing problems of 4-cylinder engines; see Figure 17).

5. For both engines the contribution of combustion excitation and its associated transfer paths was compared to overall noise emission. It was found that a substantial reduction in the combustion contribution would result in a significant overall noise reduction since the importance of the other excitation mechanisms is significant.

6. For both engines a reduction in the engine cover vibration levels is of limited usefulness in reducing overall noise emissions because of the significant contribution from the block side walls themselves.

7. The most promising area of noise reduction is in the reduction of block wall vibration, part of which is transmitted through the main bearings as considered here.



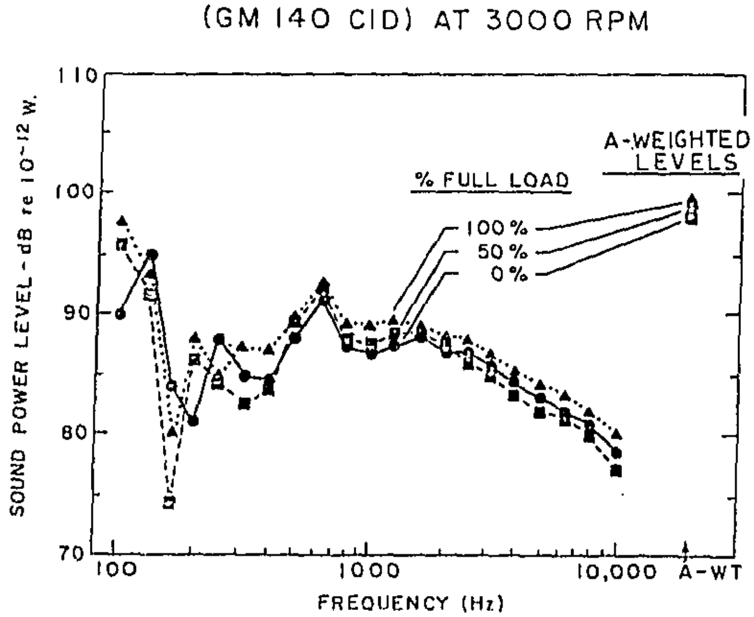


Figure 17. A Comparison of Noise Emission at 3000 RPM, 3 Load Conditions, of Camaro and Vega Engines

V. SUMMARY OF WORK ON THE JOHN DEERE 4219D ENGINE

This phase of the program focused mainly on the second stage of the noise generation process, the transmission of vibration through the engine structure to the external surfaces. The goals of the study were to identify the important paths of vibration transmission from the internal sources to the external surfaces and to specify means of reducing the vibration of the external surfaces through changes in the design parameters of the transmission path elements. The specifications of noise reduction methods included quantitative values for the mass, stiffness, and damping of the engine components and their connection points to achieve various degrees of vibration transmission reduction.

The data presented in this section are for a 4-cylinder, 4-stroke, direct injection, naturally aspirated diesel engine (John Deere 4219D) which was previously unused. This engine was chosen because it represented a typical unturbocharged engine in which the dominant internal noise source would most likely be the combustion pressure. The vibration transmission paths for this noise source were better understood than others and could be analyzed more easily.

Three steps were taken to accomplish the goals of the study. First, baseline noise emission measurements were performed to determine the amount of noise radiated by the various engine surfaces and the dependence of the overall noise on the speed and load of the engine. To carry out these running measurement tasks the engine was mounted in a reverberant test facility and a water brake dynamometer was used to simulate various load conditions. Three types of measurements were performed:

1. overall sound pressure measurements using a boom microphone

2. near field measurements of the various engine surfaces using microphones mounted approximately 1/2" from the engine surfaces

3. surface velocity measurements of the various engine surfaces using of accelerometers mounted on various external surfaces of the engine.

The engine was run at different speed and load conditions and a lead wrapping technique was employed to achieve more definite results regarding the contribution of various engine surfaces to overall noise.

The second step in the process consisted of vibration transmission measurements performed on the non-running engine for the purpose of constructing analytical models of the transmission paths. Of greatest interest were the transmission paths from the combustion chamber to the external engine block walls and from the engine block to the various covers.

The third step involved using the vibration transmission analytical models to identify changes in the design parameters of the engine components which would reduce the vibration transmission.

The results of the research yielded the following conclusions with respect to noise emission and control in the engine:

1. The major radiating surfaces are the various engine covers (see Figure 18).

2. By increasing the damping values of the oil pan from a present value of $\eta = 0.01$ to a value of $\eta = 0.1$, a 10 dB reduction in oil pan velocity will be achieved, in turn resulting in a 1 dBA overall noise reduction.

3. If proper isolation were designed for the other covers and proper damping applied to the oil pan, an overall 4 dBA noise reduction will be achieved.

4. If a larger reduction is needed, the engine block walls will have to be treated.

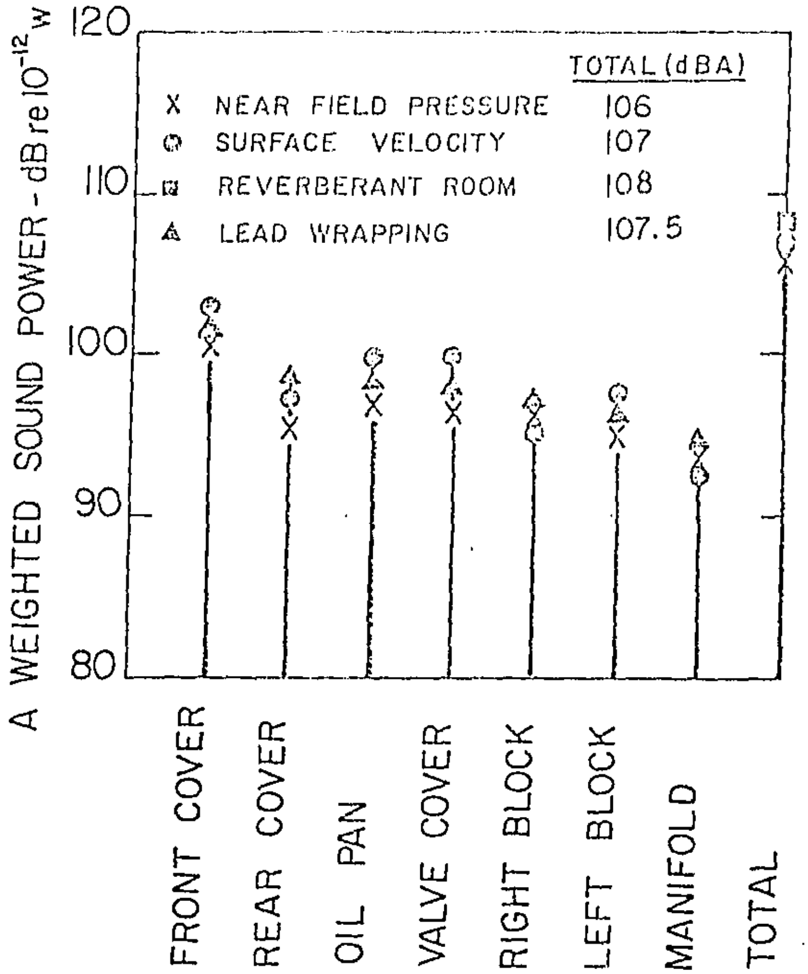


Figure 18. Noise Source Ranking of the 4219D John Deere Engine

VI. SUMMARY OF WORK ON DDA 6-71 AND 6V-71 ENGINES

The work carried out under this phase of the research program was basically similar to that performed earlier, studying the noise generation in engines with concentration on the noise radiated by the engine surfaces. Instead of gasoline engines, however, two diesel engines were examined.

Two engines were chosen which would present a comparison of the vibration transmission characteristics for two different engine block constructions. The first was a 6-cylinder, in-line diesel engine with a cast iron block (Detroit Diesel 6-71, naturally aspirated). The second was a vee-six diesel engine with a cast iron block (Detroit Diesel 6V-71, naturally aspirated). Both engines were received on consignment from Detroit Diesel Allison and had been used previously as test engines. The fact that the two engines had nearly identical cylinder geometry and combustion characteristics made them ideally suited for a comparison of the vee and in-line construction. The naturally aspirated versions were chosen in order to maximize the combustion noise and thereby highlight this noise source in the comparison of the two engines.

As was done for the gasoline engines, this research focused mainly on the second stage of the noise generation process, that of the transmission of vibration through the engine structure to the external surface. The goals of the study were to identify the important paths of vibration transmission from the internal sources to the external surfaces and to specify means of reducing the vibration of the external surfaces through changes in the design parameters of the transmission path elements. The specifications of noise reduction methods included quantitative values for the mass, stiffness, and damping of the engine components and their connection points which would achieve various degrees of vibration transmission reduction.

Three steps were taken in accomplishing these goals. First, baseline noise measurements were performed to determine the amount of noise radiated by the various engine surfaces and the dependence of the overall noise on the speed and load of the engine. This was done by mounting the engines in a reverberant test facility and performing three types of measurements:

1. measurements of overall sound pressure levels using a boom microphone

2. measurements of surface velocity by mounting accelerometers on the external surfaces of the engines

3. measurements of near field sound pressure by locating microphones very close to the external radiating surfaces of the engines.

Throughout the whole measurement process the technique of lead wrapping the engine was used to obtain the definite radiation characteristics of each engine source.

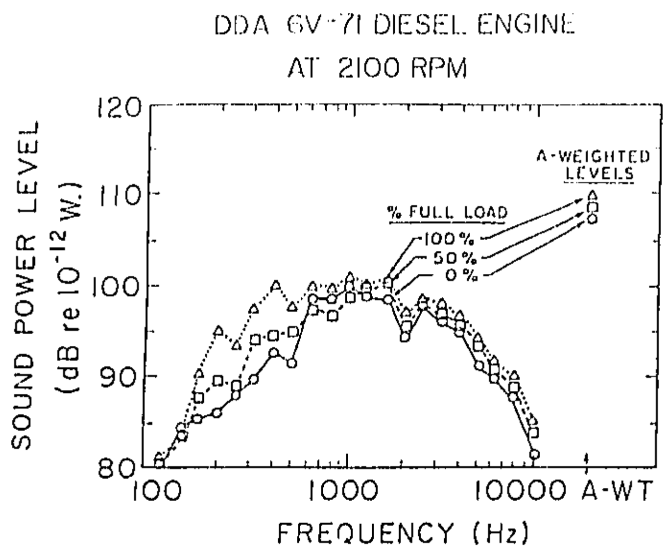
The second step in this phase was measuring vibration transmission on the non-running engines and constructing analytical models of the transmission paths. The measurements were done on both assembled and partially assembled engines (see Section III, Transfer Path Analysis, for a more detailed discussion of this measurement methodology).

Finally, the vibration transmission models were used to identify changes in the design parameters of the engine components which would reduce the vibration transmission.

Upon completion of this phase several conclusions were made:

1. The 6-71 engine appeared to be noisier than the 6V-71 (Figure 19).

2. The 6-71 engine had a more pronounced peak at 1000 Hz but a lower peak at 1600 Hz compared to the 6V-71 engine measured at 2100 rpm, 100% load.



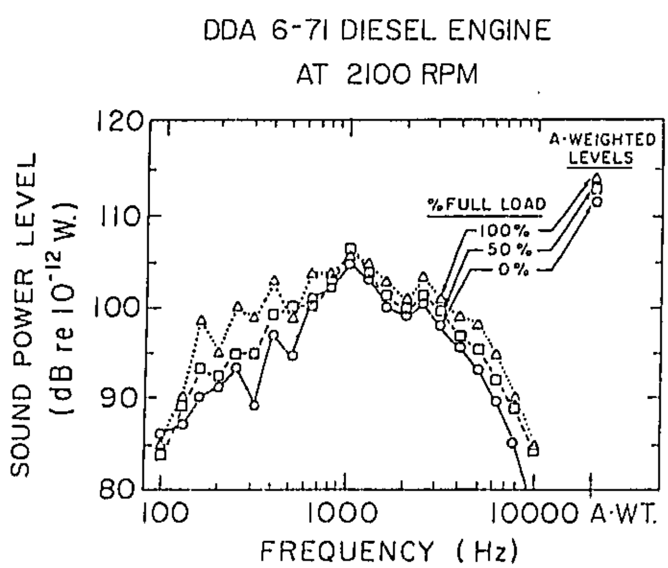


Figure 19. Overall Noise Spectra at 2100 RPM, 3 Load Conditions: a) 6V-71 Engine, b) 6-71 Engine

3. Noise level emissions for both engines appeared to be more dependent on engine speed than on engine load.

4. The major radiating surfaces of the 6V-71 are oil pan and right block (Figure 20).

5. The major radiating surfaces of the 6-71 are blower, left block, and oil pan (Figure 21).

6. It appears that the transfer path from piston top to block side is substantially stiffer in the 6V-71 than in the 6-71 engine. This is understandable since the 6V-71 has a shorter, and therefore stiffer, crankshaft.

7. For both engines, combustion is not a major noise source.

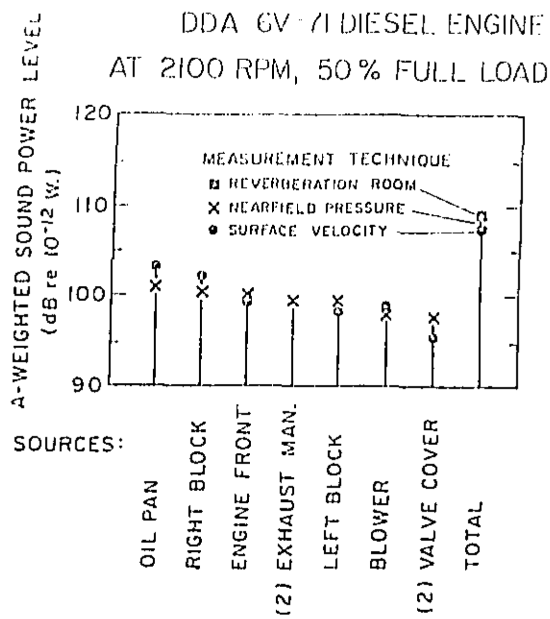


Figure 20. Noise Source Ranking of 6V-71 Engine

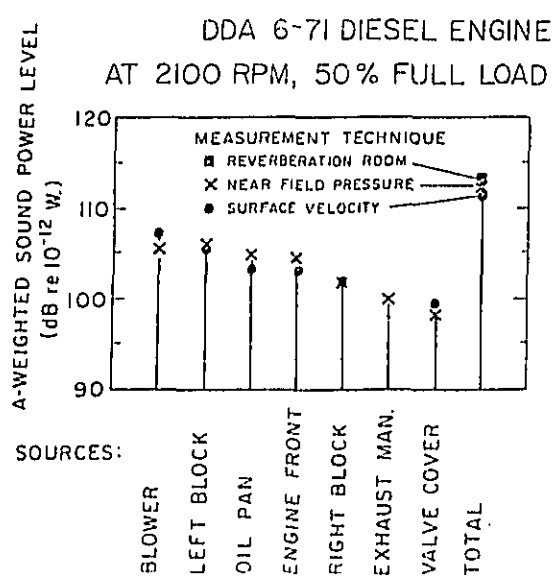


Figure 21. Noise Source Ranking of 6-71 Engine

VII. SUMMARY OF VERIFICATION WORK ON THE CAMARO ENGINE WITH TRANSMISSION

The goal of this phase in the noise identification process was to evaluate the effects of attaching a transmission to an engine. Specifically, we sought to learn whether the transmission would affect the vibrational pattern of the engine block and whether it would significantly affect the noise radiation of the engine.

To answer these questions a Camaro 250 CID, 6-cylinder engine was selected and mounted in a reverberant room with a transmission attached. The engine was run at various speed and load conditions during the measurements. For surface velocity measurements it was instrumented with accelerometers. Near field microphones were used for measuring near field pressures, and a boom microphone was used to measure overall sound pressure levels.

During the first series of measurements it appeared that the gear mesh occurring in the transmission between the gears of the main and auxiliary shafts was generating extensive noise levels which prevented evaluating the effect of the transmission structure upon engine generated noise. To alleviate this problem the auxiliary shaft was removed so that no gear mesh could occur and a second set of measurements was carried out. When the results of these tests were compared with the results of running the Camaro engine without a transmission, several conclusions were made:

1. The vibration pattern of the engine changed but not in a significant fashion. This change could be considered within the limits of experimental error.

2. The overall noise radiation did not appear to have changed in respect to first order evaluation.

It should be noted that during the period of running the Camaro engine with a transmission several attempts were made to measure cylinder pressure. These attempts were not successful, however, due to problems associated with a cavity in the mated adapter of the pressure transducer.

VIII. SUMMARY OF VERIFICATION WORK ON THE JOHN DEERE 4219D ENGINE

The purpose of the verification work performed on the John Deere 42190D engine was to validate the transmission models derived from non-running engine measurements.

The validation was carried out by actual measurements of pressure and velocity during engine operation. In particular, the measurements were made in order to determine if the vibration transmission through the moving parts of the engine could be characterized by the vibration response of these parts disassembled which are used in the combustion noise model.

Two locations were identified for vibration measurements: one on the big end of a connecting rod and the other on a main bearing cap of the block. The objective was to measure simultaneously the cylinder pressure and the component vibration level in order to measure the actual vibration transmission characteristics of the piston-crank path during the operation of the engine.

The instrumentation used to make these measurements is shown schematically in Figure 22. The vibration signal from the accelerometer mounted on the rotating big end of the connecting rod was sent out from the engine by the use of a battery-powered FM transmitter. An antenna was mounted on the inside of the block in the shape of the loop so as to maintain a constant distance between the transmitter and the antenna throughout the rotation of the crankshaft. An electronic, passive, high pass filter was fabricated and mounted on the connecting rod in order to reduce the amplitude of the low frequency acceleration signal at the engine rotation rate.

In order to mount the FM transmitter on the connecting rod and maintain sufficient clearances it was necessary to machine a recess in the side of the connecting rod thereby weakening it. An analysis of the strength of the connecting rod indicated that it would safely sustain about 25% of the rated load. Therefore all tests were limited to this condition. No detectable change in the vibration transmission characteristics of the connecting rod resulted from this modification.

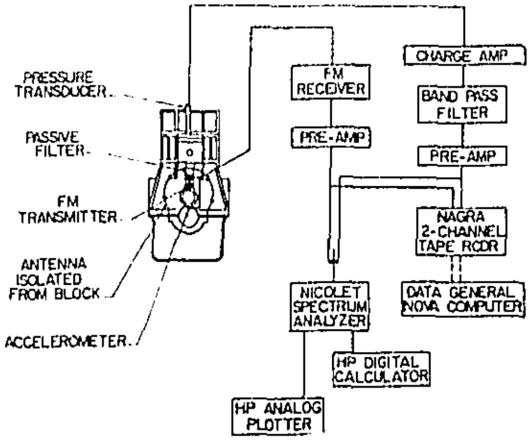


Figure 22. Schematic of Running Engine Data Acquisition System

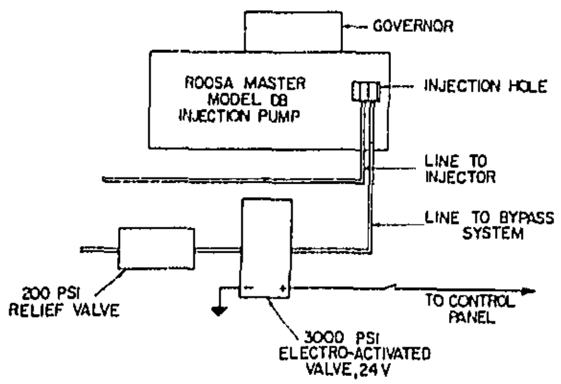


Figure 23. Schematic of Injection Control Mechanism

Initial measurements on the bearing caps indicated that the vibration resulting from successive firings of adjacent cylinders overlapped in time. As a result it was difficult to distinguish the entire vibration signal resulting from one cylinder firing in order to measure the individual vibration transfer functions. To overcome this problem a modification to the engine's injection system was designed which allowed for the selective operation of any one or combination of cylinders (see Figure 23). The principle of the modification involves the use of electrically activated valves in each line from the injection pump to the injectors which provide a bypass for the fuel. When a valve is activated the injection line is opened to a 200 psi relief valve which prevents the spring loaded injector from opening while maintaining sufficient pressure in the line for the proper operation of the pump.

In order not to affect the fuel injection adversely when the valve was not activated, the line to the valve was inserted in parallel with the regular injection line with as little additional fluid volume as possible. As a result less than 10% additional volume was added to the injection lines and less than 1% additional length was added to the lines from the pump to the injectors.

Cylinder #4 was selected for the measurements as well as bearing cap #4 (between cylinders #3 and #4). Figure 24 shows the results for the measurement of the cylinder pressure, big end vibration, and bearing cap vibration with the engine operating at 1500 rpm, 25% load. The 4-stroke engine fires once per cylinder every two revolutions (720°) with a firing order of 1-3-4-2. The vibration trace of the bearing cap shows an impulse corresponding to the firings of both cylinders #4 and #3 (540°). However, the big end vibration shows a large impulse only at the time of the firing of the cylinder. The other smaller impulses in the vibration signals are presumably due to impacts in the bearings during the rotation of the crankshaft.

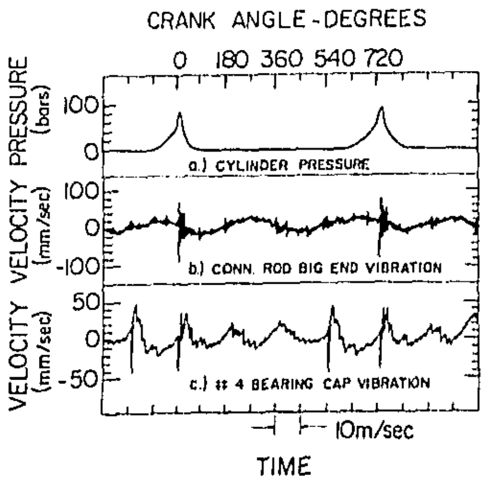


Figure 24. Measurement Signals with All 4 Cylinders Firin
(1500 RPM, 25% Load)

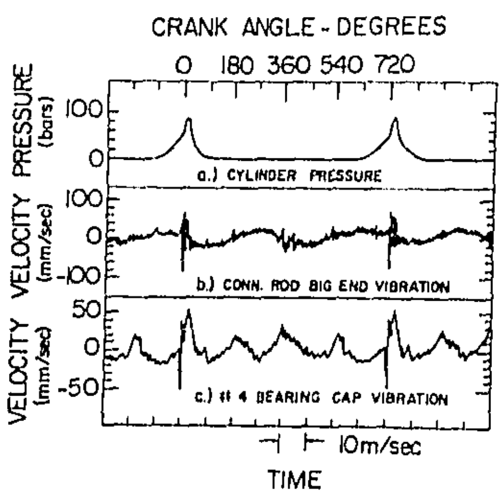


Figure 25. Measurement Signals with Only Cylinders #1 and
Firing (1500 RPM, 25% Load)

In order to isolate more fully the vibration responses due to the firing of cylinder #4, the engine was operated with only cylinders #1 and #4 firing. It was found that the engine operated very smoothly under these conditions with half the load applied compared to the operation with all cylinders (maintaining a constant load per cylinder). Figure 25 shows the measured pressure and vibration traces with only two cylinders firing with the same conditions per cylinder as shown previously. The only major difference in this case is the absence of the impulse in the bearing cap vibration at 540° where cylinder #3 is at TDC but is not firing.

The transfer functions from the cylinder pressure to the vibration levels at the two measurement locations were evaluated by selecting simultaneous 40 msec time histories for each signal centered around the firing of cylinder #4 using a cosine tapered window. Transferred mobilities were computed from the Fourier Transforms of the the signals where the cylinder pressure was multiplied by the piston surface area to obtain the force exerted on the piston.

The same transfer mobilities were constructed from the vibration transmission model using measurements of the mobilities of the individual components in the piston-crank-block path. These mobilities were cascaded together analytically assuming the components were attached with rigid pin connections [7]. Comparisons with the measured transfer mobilities are shown in Figure 26.

Reasonable agreement is found between the measured and predicted results over most of the frequency range although significant discrepancies occur at some frequencies. Further investigation is required in order to determine the cause of these discrepancies. However, the assumption that the combustion related vibration transmission through the piston-crank path occurs when the piston is near TDC appears to be valid because of the short duration of the vibration impulses.

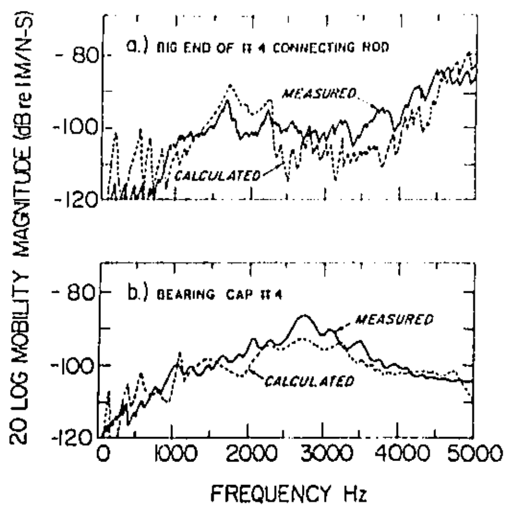


Figure 26. Comparison of Measured and Predicted Transfer Mobilities:
a) from Cylinder #4 Combustion Force to the #4 Big End Vibration Velocity
b) from Cylinder #4 Combustion Force to the #4 Main Bearing Cap Vibration Velocity

The following conclusions can be drawn from the work performed in this phase:

1. The combustion noise model has been used successfully to predict the noise radiated from a 4-cylinder, direct injection, naturally aspirated diesel engine. The combustion noise model can be used to relate the engine noise characteristics to the physical design parameters of the engine structure.

2. The modeling of the piston-crank vibration transmission at TDC appears to be valid because of the short duration of their vibration response to the rapid change in cylinder pressure at the time of combustion as compared to the period of rotation of the crankshaft. However, some discrepancies between the measured and predicted vibration levels of the components of this path warrant further investigation. Some reasons for these discrepancies include the unknown effects of the oil film in the bearings, the omission of the effects of moments in the bearings, the contamination of the measured vibration levels due to other sources than combustion being present, and possible errors in the measurements of the vibration characteristics of the individual compoents for use in the model.

3. The use of internal vibration measurements and the selective operation of various cylinders is a useful diagnostic tool in studying the importance of various paths of vibration transmission from the internal sources of the engine to the external surfaces.

IX. SUMMARY OF WORK ON THE DDA 6V-92TTA ENGINE

Piston slap is an important source of noise and vibration in many diesel engines [13] especially in turbocharged engines where the combustion noise is suppressed [14]. One method of controlling piston slap noise which has been extensively investigated is the use of tight-fit pistons [15,16]. In this design the piston to bore clearance is reduced below that which is typically used in production engines in order to reduce the magnitude of the impacts of the piston on the liner. However, there is a necessary tradeoff in this design with a decrease in mechanical efficiency due to increased frictional energy losses at the piston-liner interface [16]. There are also potential problems with increased wear and seizures of tight-fit pistons due to limitations in lubrication and production tolerance control.

This section presents the results of work to develop an alternative method of reducing piston slap noise in a particular diesel engine with a Vee-block configuration (the Detroit Diesel Allison 6V-92TTA engine). This engine was chosen for an initial diagnostic study of the noise generation mechanisms because of its compact Vee-block construction and turbocharged air intake, both of which contribute to the reduction of combustion noise [13,14].

A detailed investigation of the internal sources of noise and vibration indicated that piston slap and injector forces are the major sources in this engine. A first prototype engine was constructed to investigate methods of reducing the noise due to injectors [17]. Tight-fit pistons were also incorporated and evaluated in this engine. A summary of the results of that work is presented in the first part of this section.

Further work was then undertaken to develop a liner-block design which would reduce the noise generated by piston slap without requiring the reduction of piston-liner clearances beyond presently acceptable production values. If future developments would allow the reduction of piston-liner clearances, then these two designs could be combined to achieve additional reductions in piston slap noise.

A second prototype engine was constructed to test a particular liner-block design and to evaluate its effectiveness in reducing piston slap noise. A detailed description of this work is given in the second part of this section.

9.1 Analysis of Engine Noise Sources

The internal sources of noise and vibration in the DDA 6V-92TTA have been investigated using vibration transmission measurements on both a running and non-running engine. It has been shown that this method can be used to evaluate quantitatively the importance of an internal source of noise in terms of the overall noise and vibration of the engine [18,1]. A detailed description of the application of this method to this engine has been previously reported [17], and only a summary will be given here.

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, and injection;

2. The pressures and forces act on the 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: combustion, piston slap, and injection forces. 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.

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 cylinder pressure exerts a force both on the piston crown and on the cylinder head surface. The vibration response of the engine structure due to this pressure was 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. 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. Figure 27 presents a summary of the results of these measurements and a comparison of the predicted block vibration due to combustion with the measured vibration during the operation of the engine. This verifies that combustion is not a major source of vibration and radiated noise in this engine.

PISTON SLAP - The source level of piston slap cannot easily be measured directly. 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 outer surface of the liner at the assumed height of major piston slap near TDC. A second experiment was conducted on a non-running engine 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

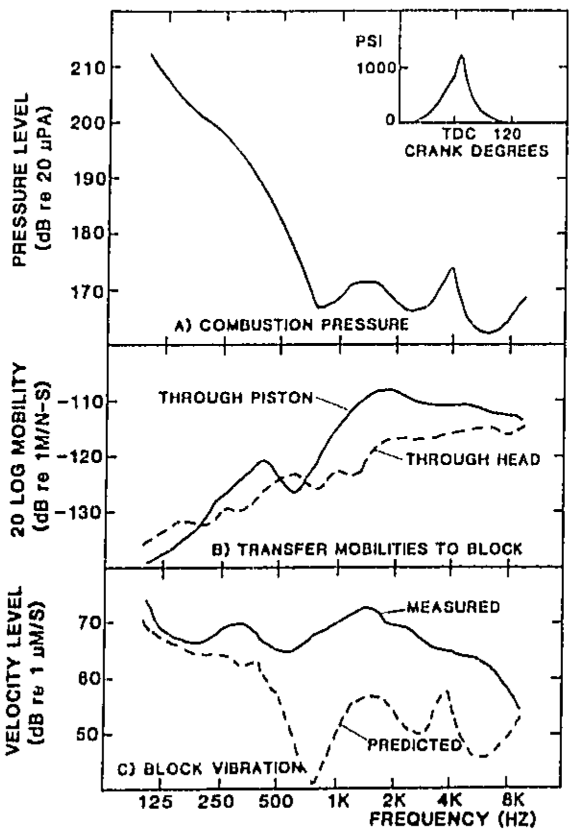


Figure 27. A Summary of the Combustion Source Model and Block Vibration for the 6V-92TTA Engine at 1950 RPM

drive-point mobility. The transfer mobility from piston slap to the engine surfaces was also measured.

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 as shown in Figure 28. This shows that piston slap is a major source of block vibration, and therefore radiated noise, in the frequency range of 500-2000 Hz.

INJECTION FORCES - The DDA 6V-92TTA engine is equipped with unit injectors, each of which injects a specific amount of fuel into a cylinder by means of a cam-driven rocker arm assembly. 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 rocker arm was measured during the operation of the engine. The force exerted on the rocker arm was obtained by dividing the measured velocity spectrum by the measured mobility of the rocker arm in its fully assembled condition.

Forces generated by the injectors are transmitted to the engine block by two paths: the first is through the injector casing to the head and the engine block; the second 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 rocker arm from the cam push rods and exciting the structure with a shaker attached to one or the other of the disconnected members.

Combining the force excitation level of the injectors and the measured transfer functions to the block gives a prediction of the engine block vibration due to the injectors, as shown in Figure 29. This shows that the injectors are a major source of vibration at and above 1000 Hz.

RADIATED NOISE - By summing the contributions to block vibration from these three sources, a nearly complete model of the engine vibration can be obtained for this engine. To

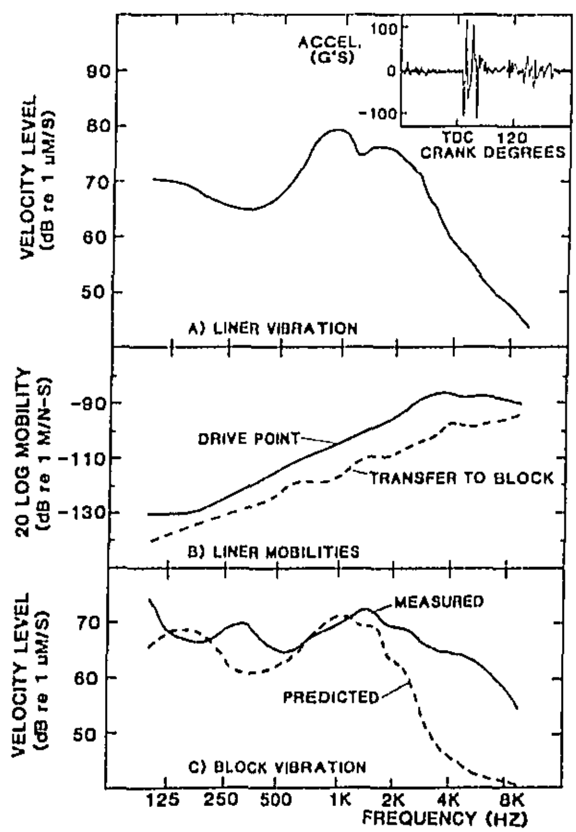


Figure 28. A Summary of Piston Slap Source Model and Block Vibration for the 6V-92TTA Engine at 1950 RPM

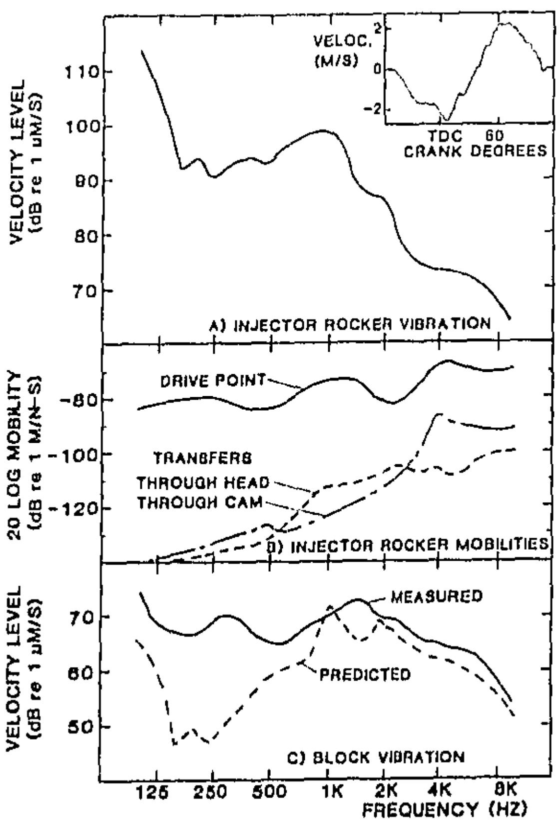


Figure 29. A Summary of the Injector Source Model and Block Vibration for the 6V-92TTA Engine at 1950 RPM

complete the noise generation model the noise radiated by each engine surface must be related to the block vibration. This was done by measuring first the vibration transmission from the block to the attached covers, and then the radiation efficiency of each surface vibration using the acoustic intensity technique.

These measured transfer functions were then used to predict the noise radiated by the engine. Results are shown in Figure 30 where they are compared to the total measured sound power levels of the engine measured in a reverberation room. There is good agreement between the predictions and measurements except in the 300-500 Hz range. In this range the engine noise is thought to be dominated by the Roots intake blower, which is required by the 6V-92 since it is a two-stroke engine. The blower has not been included in our modeling.

A summary of the contributions of each internal noise source to the overall sound power level of the engine is given in the first column of Table 2.

9.2 Evaluation of Tight-fit Pistons

Having identified injector forces and piston slap as the two major noise sources of the engine, work was done to develop design modifications to the engine structure to reduce the noise and vibration due to these sources without compromising the operating performance of the engine. A first prototype engine was constructed which incorporated design modifications to the injector train. These included the use of resilient bearings on the cam shafts and drive gears to isolate the block from the injector train vibrations. The design of these bearings is shown in Figure 31 and is described in more detail in Reference [17].

Of more interest here is the fact that tight-fit pistons were also used in this prototype engine as a means of reducing piston slap noise. An experimental set of tight-fit pistons was provided by Detroit Diesel Allison, which used a 0.002" coating of tin on the piston surface. The pistons were matched with specific standard liners to achieve a tape fit clearance of

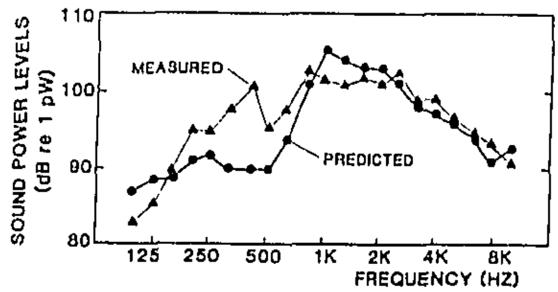


Figure 30. A Comparison of the Measured Sound Power Level with Those Predicted by the Noise Generation Model of the 6V-92TTA Engine at 1950 RPM, Full Load

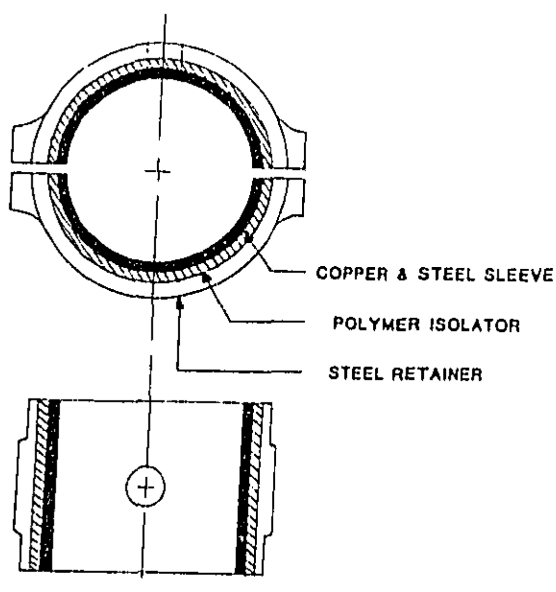


Figure 31. A Typical Design of a Resilient Cam Bearing

SOURCE		Prototype (Tight-fit Pistons)	Prototype (Modified Cylinder Liner)
Piston Slap	107 dB(A)	103 dB(A)	103 dB(A)
Injection Force	109	105	[104]
Combustion	100	100	100
Other	[99]*	[99]	[99]
TOTAL	111.5 dB(A)	108.5 dB(A)	108 dB(A)

* The numbers in brackets are inferred levels.

Table 2. Summary of the Contributions of the Internal Noise Sources to the Overall Sound Power Level of the DDA 6V-92TTA Engine at 1950 RPM, Full Load

0.0030"-0.0035". This is a 40-65% reduction from the production tolerances of 0.0051"-0.0097".

Measurements were made on this prototype engine to evaluate in detail the effectiveness of the tight-fit pistons. The liner vibration was measured as before and an estimation was made of the magnitude of the piston slap force using the measured liner mobility. The frequency spectrum of the estimated impact force of the piston is shown in Figure 32 for both the tight-fit pistons and the standard design. It can be seen that the reduced clearance lowers the piston impact force by an average of about 5 dB over the entire frequency range except around 500 Hz where the force actually increases. The low level of piston slap force around 500 Hz in the standard engine is due to a resonance of the piston mass on the contact stiffness between the piston and liner which limits the force which the piston can exert on the liner in this frequency range. The tin coating appears to reduce this resonance to around 300 Hz.

The noise radiated by the prototype engine was measured and analyzed as before. Figure 33 shows the contributions of injector forces and piston slap to the total noise of the prototype engine. These results are summarized in the second column of Table 2. Both sources have been reduced by 4 dB(A) for an overall engine noise reduction of 3 dB(A).

9.3 Development of a Modified Cylinder Liner Design

As an alternative to tight-fit pistons, a cylinder liner design was developed which would reduce the amount of vibration transmission from the liner to the block, thereby reducing the radiated noise. The standard liner is mounted in the engine bore by a lip on the upper rim of the liner which is clamped between the head and the block (see Figure 34a). A high compressive load is applied to this lip in order to maintain the proper seal for combustion.

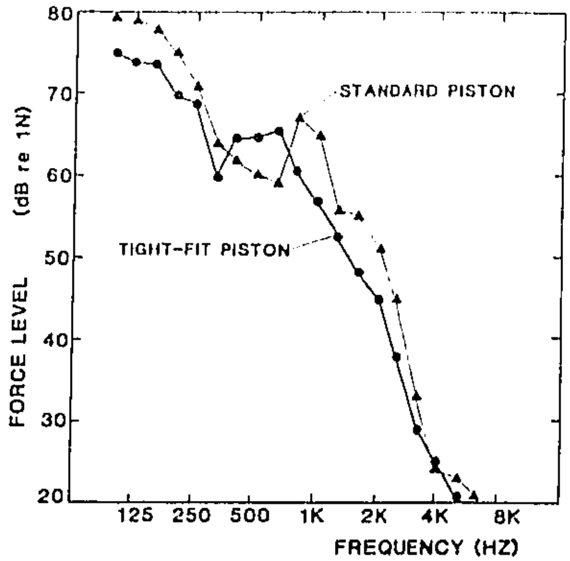


Figure 32. A Comparison of the Estimated Piston Slap Force Spectra for Standard and Tight-Fit Pistons in the 6V-92TTA Engine at 1950 RPM, Full Load

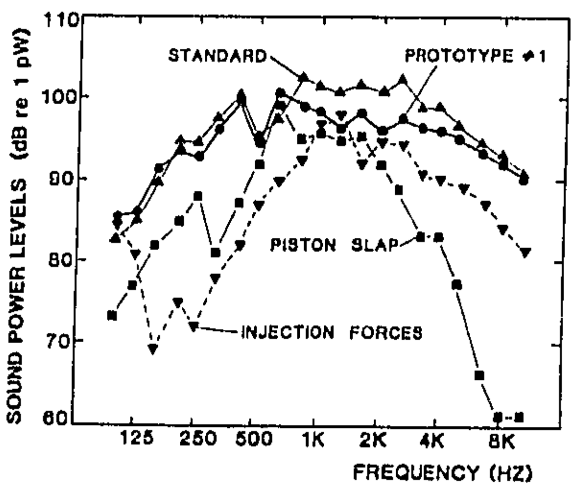


Figure 33. A Comparison of the Measured Sound Power Level of the First Prototype Engine with Those Predicted by the Noise Generation Models and Measured with the Standard 6V-92TTA at 1950 RPM, Full Load

The vibration transmission through the clamped lip is very efficient in the 800-2000 Hz range because of several vibration modes of the engine block which are strongly excited at this location (see Figure 34c). This was overcome by a liner design which lowers the mounting in the block (Figure 34b) to a point where the block modes are not as strongly excited (see Figure 34c). This design has the added benefit of increasing the liner wall thickness in the region of the piston slap which reduces the level of the liner vibration due to the piston impact.

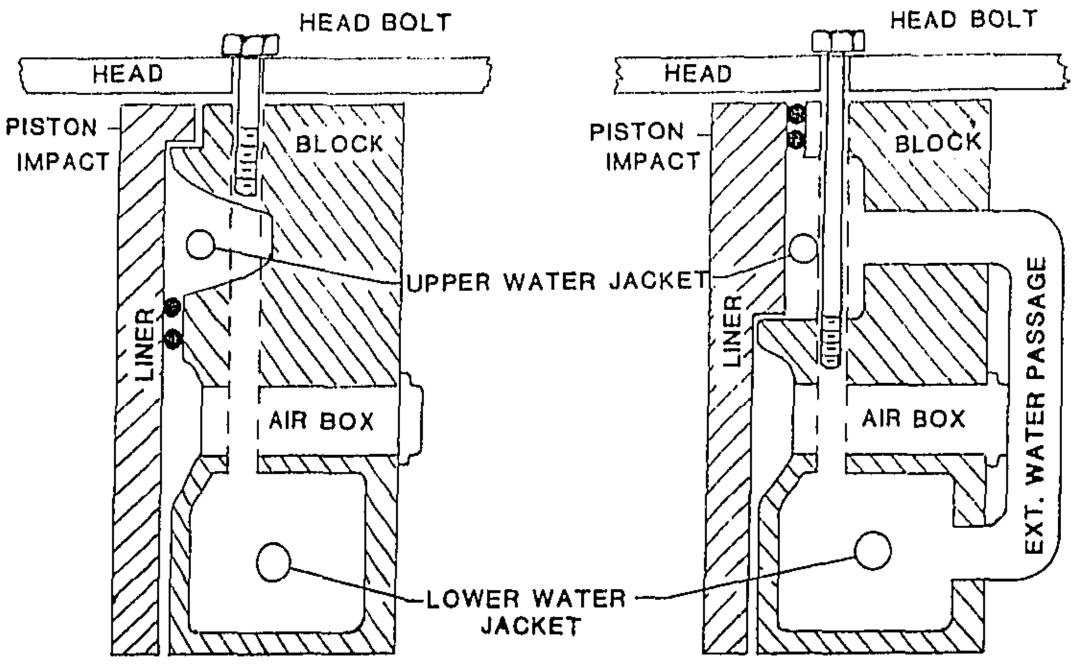
In order to carry the compressive load on the liner at the lower mounting point in the block, the head bolts were extended down below this point. This necessitated the re-routing of the water passage from the lower to the higher sections of the block. This was accomplished by using external water passages which were designed to be an integral part of the air box covers currently part of the standard design (see Figure 34b).

The effectiveness of this liner design in reducing piston slap noise was verified by performing vibration transmission measurements on a non-running engine with the modified liners installed. The measured mobilities of both the standard and modified liners are shown in Figure 35. A 5-8 dB reduction in the vibration responses of the engine with modified liners is achieved in the 800-2000 Hz range.

9.4 Evaluation of the Modified Cylinder Liner Design

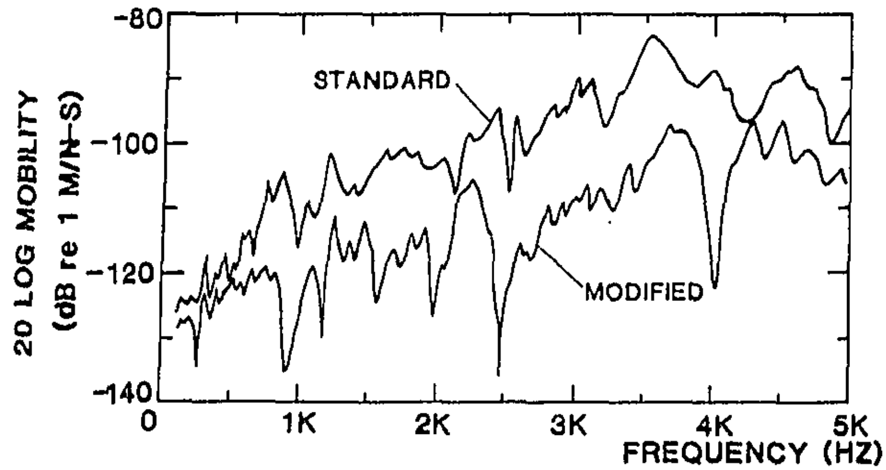
A second prototype engine was constructed which incorporated the modified cylinder liner design as well as the injector train modifications used in the first prototype. The piston-liner clearances were carefully controlled to be between 0.006" and 0.007" which is in the middle of the range of production tolerances.

The analysis of piston slap noise was repeated for this engine so as to compare the effectiveness of the modified liner design with the standard and tight-fit piston designs. The measured liner vibration levels are shown in Figure 36 as compared



a) Standard Liner Design

b) Modified Liner Design



c) Vibration Response of Engine Block at Point of Liner Attachment

Figure 34.

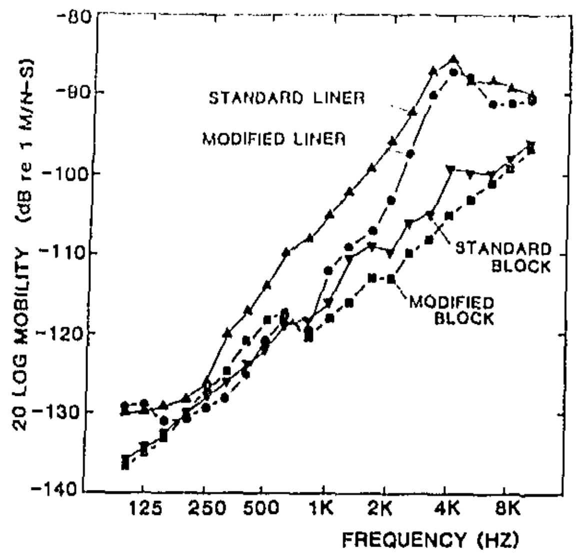


Figure 35. A Comparison of the Measured Vibration Response of the Cylinder Liner and Block for the Standard and Modified Liner Designs in the 6V-92TTA Engine

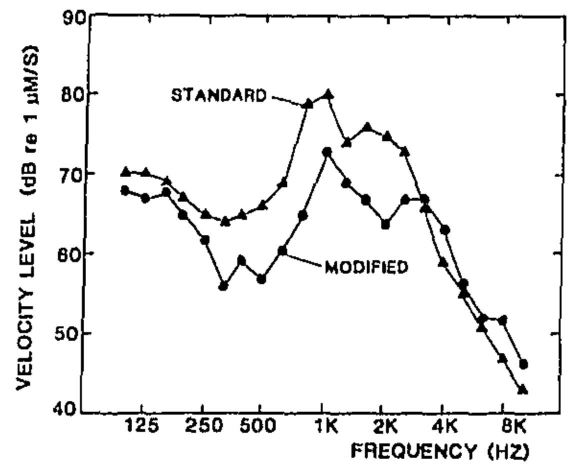


Figure 36. A Comparison of the Measured Cylinder Liner Vibration Levels for the Standard and Modified Liner Designs in the 6V-92TTA Engine at 1950 RPM, Full Load

with those of the standard design. The vibration levels of the modified liner are reduced by 5-10 dB from 800-2000 Hz primarily because of its increased wall thickness and therefore reduced mobility to piston slap (see Figure 35). There is no significant change in the force levels of the piston impact computed from these results when compared with those in the standard engine. The piston slap noise reduction achieved by the modified liner is then primarily due to the change in the vibration transmission from the liner to the block.

The measured noise of this second prototype engine is shown in Figure 37 as compared to the original engine. A noise reduction of 3.5 dB(A) is achieved as shown in the third column of Table 2. The predicted levels for piston slap and injection forces are also included in Figure 37 for comparison. The modified liner design results in a 4 dB(A) reduction in piston slap noise which is the same as that obtained for tight-fit pistons.

It should also be noted that the additional noise reduction achieved in this engine above 2500 Hz is believed to be the result of a small change in the design of the resilient thrust bearings on the cam shaft after the tests on the first prototype engine. An inspection of the original resilient bearings indicated that there was excessive wear on the thrust surfaces of end cam bearings which was probably due to a breakdown in lubrication. The dimensions of these bearings were modified to enlarge the thrust surfaces so as to increase their load bearing capabilities. If the measured noise reduction above 2500 Hz in the second prototype engine can be attributed to a reduction in the injector source level (which has not been verified yet), then the injector noise level has been reduced by 5 dB(A).

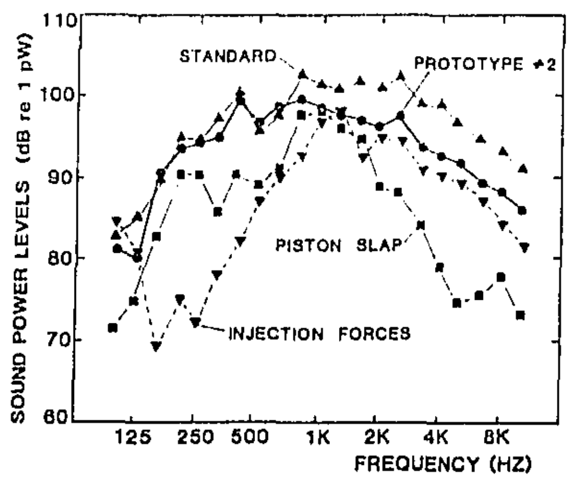


Figure 37. A Comparison of the Measured Sound Power Levels of the Second Prototype Engine with Those Predicted by the Noise Generation Models and Those Measured with the Standard 6V-92TTA Engine at 1950 RPM, Full Load

9.5 Conclusions

The following conclusions can be drawn from this work:

1. A design modification for the cylinder liner and block bore in the DDA 6V-92TTA engine has been successfully developed which reduces piston slap noise by 4 dB(A) without the adverse effects associated with tight-fit pistons. This reduction in piston slap noise is the same as that achieved by the use of tight-fit pistons in this engine.

2. Since the noise reductions obtained from tight-fit pistons and the modified cylinder liner are separate and additive, a design incorporating both of these treatments in the DDA 6V-92TTA engine could potentially achieve an 8 dB(A) reduction in piston slap noise. This would correspond to a total noise reduction of approximately 4.5 dB(A) in the second prototype engine design.

3. The design modifications used for the reduction of both the piston slap and injection noise levels were made under the requirements that they not adversely affect the performance of the engine. The modifications were designed into an existing block structure with only small modifications required in the block casting and no significant added weight.

4. The development of vibration transmission models, using vibration measurements on a running and non-running engine, is a useful analysis tool for identifying the types of design modifications which will achieve significant reductions in the radiated noise of the engine.

X. SUMMARY OF WORK ON THE CUMMINS NTC-350 ENGINE

The goal of the research in this phase was to demonstrate a 5 dB reduction in radiated noise by retrofitting an existing, popular heavy duty truck diesel engine, the Cummins NTC-350 Big Cam One engine. 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 already been demonstrated [19].

The work was carried out in five major stages:

1. Determining the engine noise characteristics.

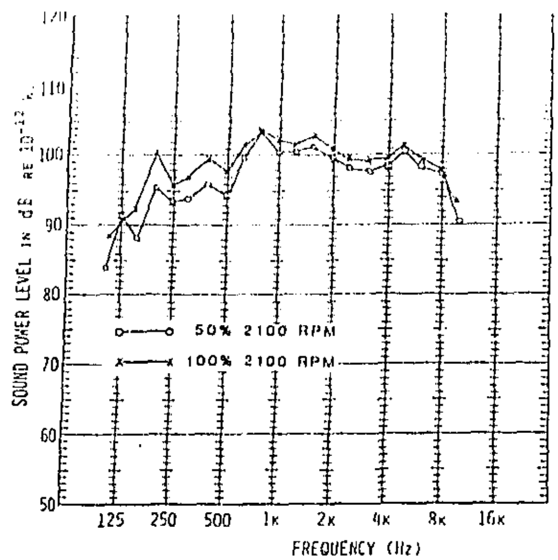
2. Developing a model which relates internal forces to external vibration or noise levels.

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

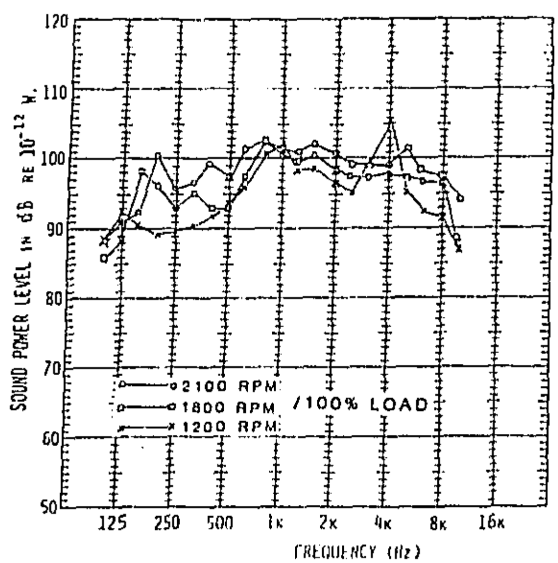
4. Developing a demonstration engine.

5. Testing and evaluating the demonstration engine.

The baseline noise characteristics were determined which relate the total sound radiation spectra of the engine to the speed and load of operation. It became apparent that the noise emission is controlled equally by speed and load (Figure 38a,b). Next, the magnitude of the sound intensity radiation and the vibration characteristics from each of the various engine surfaces were measured in order to rank them according to their importance in the noise generation process. These measurements determined the basic noise emission characteristics of the engine in its standard configuration.



a) NTC-350 Noise Spectra at 2100 RPM, 2 Load Conditions



b) NTC-350 Noise Spectra at 100% Load, 3 Speed Conditions

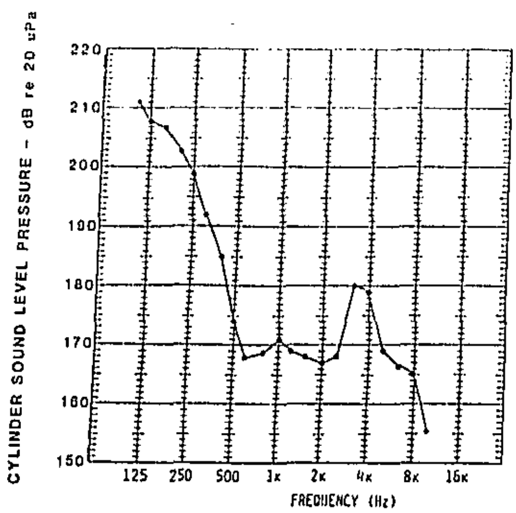
Figure 38.

Upon completing the overall noise emission survey, a noise generation model for the engine was developed. First, the levels of various internal sources (such as combustion pressure, piston impacts, injectors, bearings, gears, pumps, and air flow through valves) were determined through direct and indirect measurements (Figure 39a,b,c). Second, the characteristics of the vibration transmission through the engine structure, from each source location to the external surfaces, were determined from vibration response measurements on the disassembled engine (Figure 40a,b,c). Third, the sound radiation characteristics of each vibrating surface were determined by relating the sound intensity measurements to the vibration levels of the engine measured during operation of the engine in its standard configuration. Combining the results of the three procedures gave us a means of estimating quantitatively the contribution of each internal source to the total radiated noise.

From the outcome of the research, it appeared that piston slap is the major noise source with contribution from injectors at 2 kHz and combustion at 3.8 kHz which is associated with combustion chamber resonance (Figure 41). When all the sources were combined and compared to the measured data, generally good agreement was obtained (Figure 42).

Based on the guidelines derived from the modeling effort, it became clear that to achieve a 5 dBA overall reduction in truck noise it would be necessary to treat the excitation of piston slap primarily, with some treatment to the other sources. The most appropriate way of approaching piston slap excitation is to reduce the transmission of vibrational energy from the liner to the block through an impedance mismatch.

Several methods were investigated for treating piston slap. The most promising method was to increase the liner thickness for a higher local impedance, resulting in a reduction of power flow from the piston to the liner in conjunction with attaching the liner to the head and isolating the head from the block (see Figure 43 for schematic). This technique appears to have great noise reduction potential (see Figure 44) without incurring



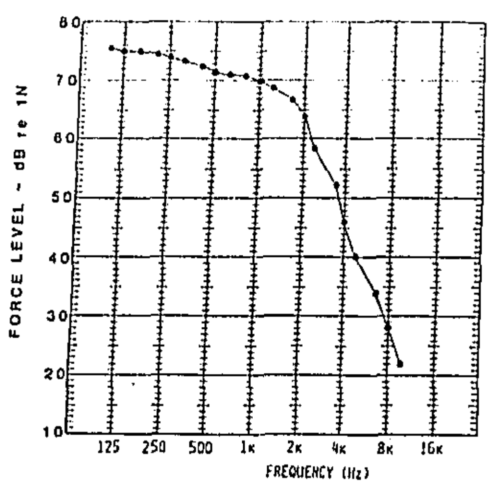
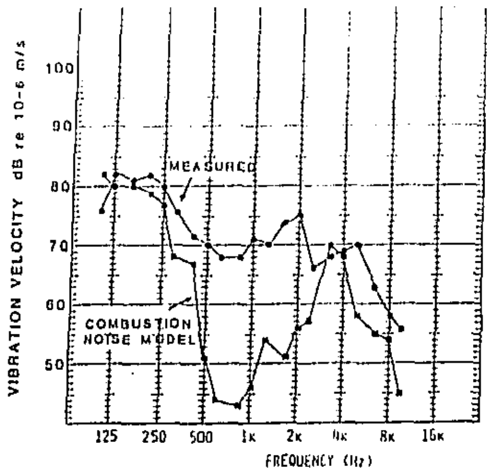
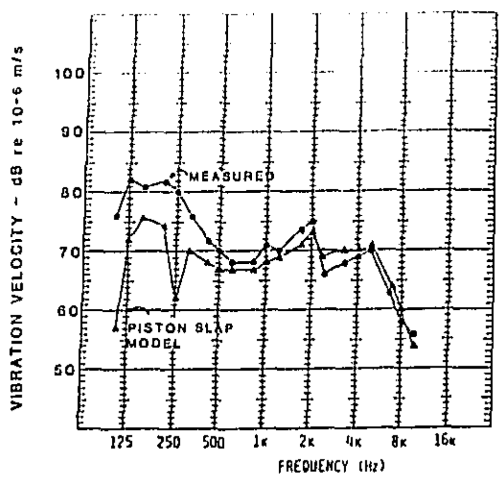


Figure 39. a) NTC-350 Measured Cylinder Pressure

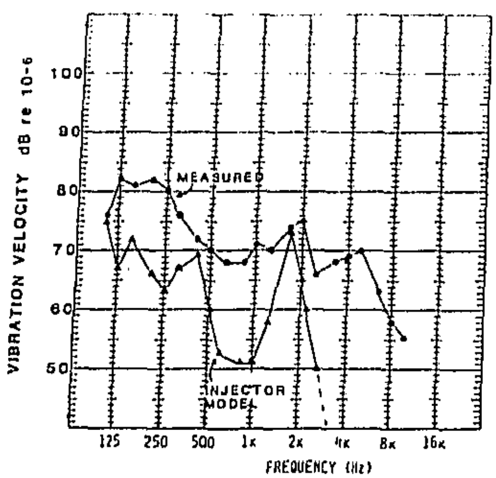
b) NTC-350 Piston Slap Inferred Force



a) Comparison of Vibration Transmission from Combustion with Overall Vibration on NTC-350 Engine at 2100 RPM, Full Load



b) Piston Slap Vibration Levels vs. Measured Block Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load



c) Injector Vibration Levels to Lower Left Block vs. Measured Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

Figure 40.

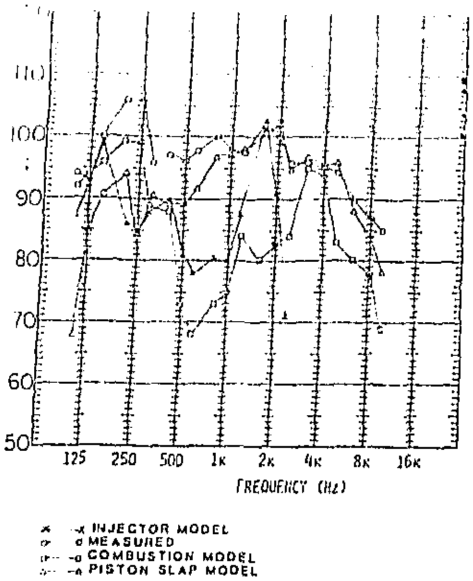


Figure 41. Contribution of Sources to Overall A-Weighted Truck Sound Power Levels on NTC-350 Engine at 2100 RPM, Full Load

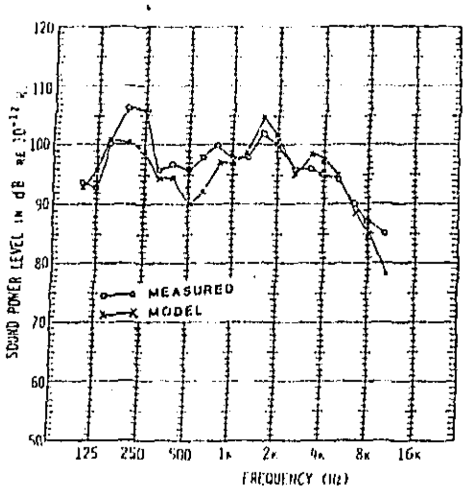


Figure 42. Lower "In-Truck" Sound Power Levels vs. Total Measured Sound Power Levels on NTC-350 Engine at 2100 RPM, Full Load

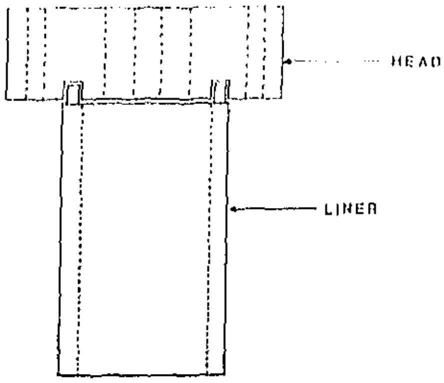


Figure 43. Modified Liner Schematic

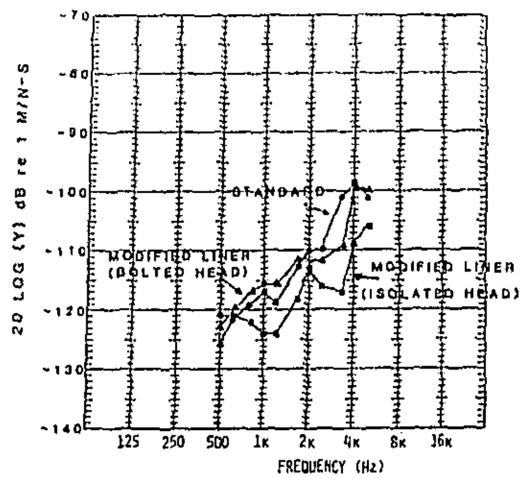


Figure 44. Transfer Mobilities of Piston Slap Force to Block Vibration with New Liner Attachment on NTC-350 Engine

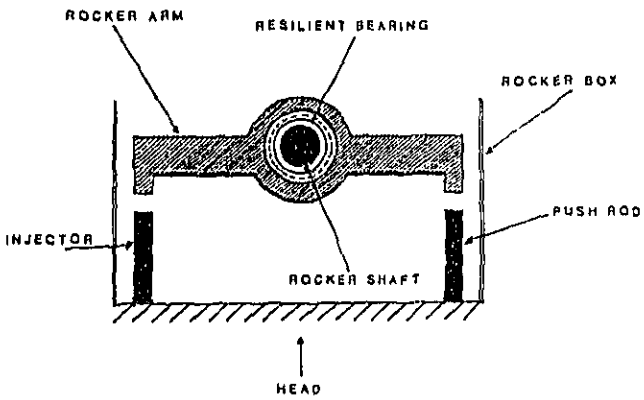


Figure 45. Injector Rocker Isolation Schematic

significant penalties to other engine performance parameters, although the approach is slightly unorthodox. The technique will also benefit the transmission of vibrational energy from combustion excitation since one of the major paths for this source is through the head to the block.

To complete the noise treatment for combustion, a design for isolation of the crankshaft from the block was created using a resilient bearing concept. The same resilient bearing principle was applied to the injector rocker shaft in order to deal with the contribution of the injection process to overall noise at the 2 kHz band (see Figure 45 for design schematic).

In conclusion, the new design concepts mentioned here should result in an overall noise reduction greater than 5 dBA, but due to major delays in fabrication, the modified engine was not tested in a running configuration.

XI. CONCLUSIONS

This report has described and demonstrated various methods to model and reduce internal combustion engine noise which were developed and tested under a five-year research program at Cambridge Collaborative. The primary objective of the program was to evaluate techniques of reducing overall engine noise that would not result in penalties to other engine parameters such as fuel economy, emissions, weight, maintainability, etc. This objective was demonstrated in the work on the DDA 6V-92TTA engine in which an existing block was modified and an overall noise reduction of 4.5 dBA was obtained. It is our assessment that if these design approaches were to be implemented in a development program for new engines and new blocks could be casted, an even greater noise reduction could be achieved, on the order of 10 dBA.

One of the most important techniques developed during the research program was the method of transfer path analysis of vibration transmission within an engine structure. Transfer path analysis was found to be a viable and dependable method for understanding the transfer of vibrational energy from the excitation mechanisms within an engine to the radiating surfaces. Although transfer path analysis cannot be used to predict overall noise emission of an engine based on drawings and designs alone, it can predict noise emissions from an engine and their dominant sources based on measurements of a non-running engine quite accurately and rapidly. If transfer path analysis is utilized during the developmental stage of engine design, it could enable the designers to create a substantially quieter engine as well as prove to be a very cost-effective approach to noise control.

The conclusions of our extensive research have indicated that lower overall engine noise can be achieved through internal structural changes to an engine without sacrificing performance in other areas by using transfer path analysis as an evaluation tool for design changes.

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

1. REPORT NO. EPA 550/9-82-334	2.	3. RECIPIENT'S ACCESSION NO.
4. TITLE AND SUBTITLE Analysis and Control of Mechanical Noise in Internal Combustion Engines		5. REPORT DATE July 1982
		6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S) Natan E. Parsons, Richard G. DeJong, 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, Massachusetts 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 12/77 - 6/82
		14. SPONSORING AGENCY CODE EPA-ONA
15. SUPPLEMENTARY NOTES		

16. ABSTRACT

This report reviews the state-of-the-art for internal combustion engine noise reduction and presents new techniques for reducing engine block vibration and radiated noise. A vibration analysis technique based on measured mobilities was developed as a diagnostic tool for identifying noise sources and vibration transmission paths. This technique makes it possible to identify and rank order the sources of noise within the engine. New design techniques using resilient bearings and modified cylinder liners are also described.

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 93
	20. SECURITY CLASS *(This page)* Unclassified	22. PRICE

EPA Form 2220-1 (9-73)