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DEMONSTRATION OF NOISE CONTROL FOR THE CUMMINS NTC-350 HEAVY DUTY TRUCK DIESEL ENGINE

UNITED STATES ENVIRONMENTAL PROTECTION AGENCY

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Prepared for

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

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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. The authors wish to acknowledge the Massachusetts Institute of Technology which served as a subcontractor on the project, and the Cummins Engine Company. The cooperation and assistance of these organizations have been an essential ingredient to this project.

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I. INTRODUCTION

This report presents the results of an engineering study to identify and design methods for reducing diesel engine block vibration and radiated noise. The methods identified are such that noise reductions could be achieved without degrading the engine's performance, fuel economy, or emissions. The goal of the study 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 [1].

The work was carried out in five major phases:

1. Determining the engine noise characteristics.
2. Developing a model which relates internal forces to external vibration or noise levels.
3. Designing noise controls with constraints provided by the basic design and by the manufacturer of the engine.
4. Developing a demonstration engine.
5. Testing and evaluating the demonstration engine.

The report is organized into sections discussing each of these phases separately. Section II presents a discussion of the engine's design and performance features as well as a detailed description of the noise characteristics. First, the baseline noise characteristics are determined which relate the total sound radiation spectra of the engine to the speed and load of operation. Second, the magnitude of the sound intensity radiating from each of the various engine surfaces is measured in order to rank them according to their importance in generating noise. These measurements determine the basic noise emission characteristics of the engine in its standard configuration.

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

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

II. ENGINE DESIGN AND PERFORMANCE

The Cummins NTC-350 Big Cam One is a heavy duty truck engine designed to be used primarily in truck classes 6 and 7 for long range hauling. The designation "NTC-350 Big Cam One" refers to the following characteristics of the engine:

N - New

T - Turbocharged

C - Custom-rated

350 - 350 horsepower

Big Cam One - The engine has a larger diameter cam than the standard NTC-350 engine. In this version the cam is increased from 2" to 2.5" in diameter which results in shorter injection duration and lower combustion temperature and yields lower emissions and better fuel economy.

The NTC-350 Big Cam One's rated horsepower is 350 at 2100 rpm. It is a six-cylinder, in-line, four-stroke engine which is turbocharged and intercooled. The NTC-350 Big Cam One utilizes a PT fuel system (P - pressure and T - time). This fuel system incorporates a low pressure metering fuel pump that controls the fuel delivery rate to the injectors located in the head at the center of the combustion chamber. The injectors are activated through a rocker arm push rod assembly by the camshaft. These injectors perform the same task as unit injectors in that they receive fuel at a low pressure and raise the pressure significantly through the use of a plunger activated by the camshaft to achieve proper fuel spraying into the combustion chamber. The injectors are different, however, from the so-called unit injectors because the metering of fuel is done by the PT pump rather than by the injector (see Figure 1 for a schematic of the fuel system).

The NTC-350 Big Cam One engine comprises a single block with three separate heads, each head serving two cylinders. It utilizes a cast aluminum oil pan and rocker boxes and stamped

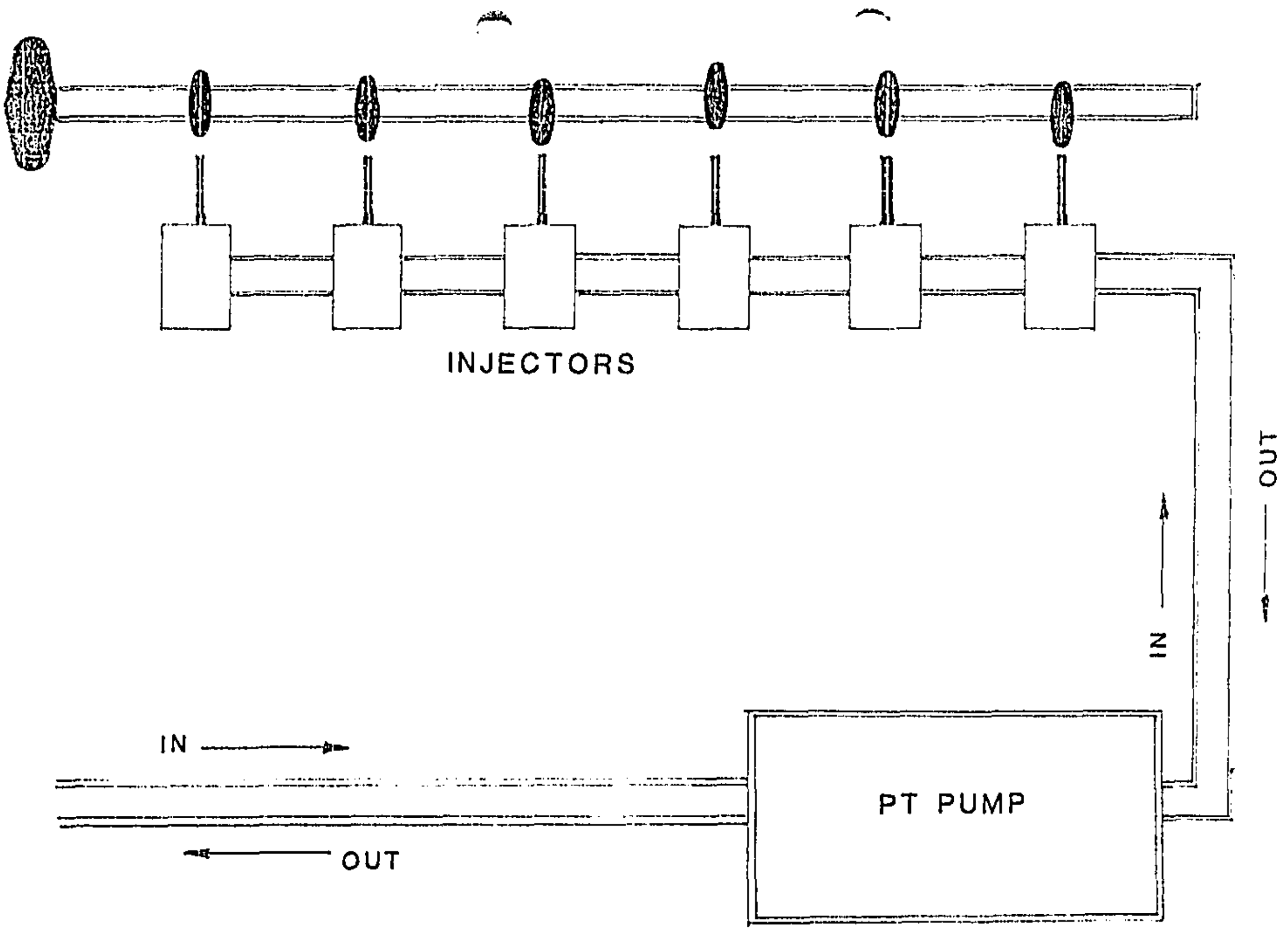

Figure 1. Fuel System of NTC-350 Engine

steel valve covers and is equipped with an air compressor driven by the same shaft as the PT fuel pump. The engine uses a 24V electric starter attached to the SAE #2 flywheel housing.

The specifications of the NTC-350 Big Cam One are as follows (see Figure 2 for engine performance):

power rating	350 bhp
governed rpm	2100
peak torque	1065 lb-ft at 1400 rpm
nominal torque rise	22%
no. of cylinders	6
bore and stroke	5.5 x 6 in.
piston displacement	855 in^3
lube system oil cap	11.5 gallons (U.S.)
net weight with standard accessories, dry	2580 lbs.

The NTC-350 Big Cam One incorporates the following elements: single piece aluminum pistons, cast iron liners attached to the block at the top flange, a single piece head gasket per head, cast iron block, cast iron heads, and a positive displacement oil sump pump.

A. Baseline Noise and Vibration Characteristics

The noise radiated from the NTC-350 Big Cam One engine was measured in one-third octave bands in the frequency range from 100 to 10,000 Hz. A 200 m^2 reverberant test cell which was calibrated for sound power level measurements of noise sources with puretones according to American National Standard ANSI S1.21 and International Standard ISO 3742 was used for the measurements. The instrumentation required for the measurements is shown schematically in Figure 3. Using the rotating microphone boom, the sound pressure in the room was measured and averaged over time and space. The sound power level is computed from the average sound pressure level by comparing it to the measured sound

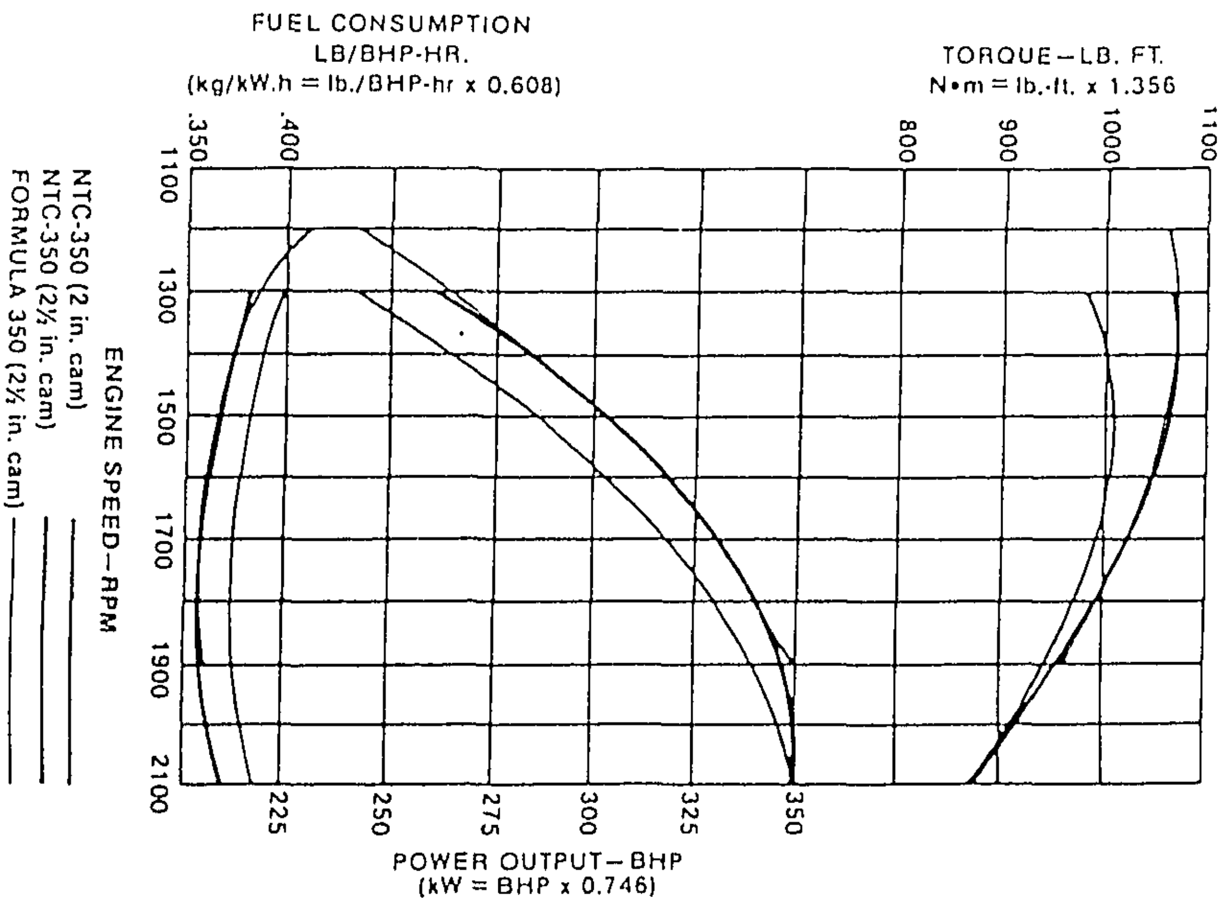

Figure 2. Performance Curve for NTC-350 Engine

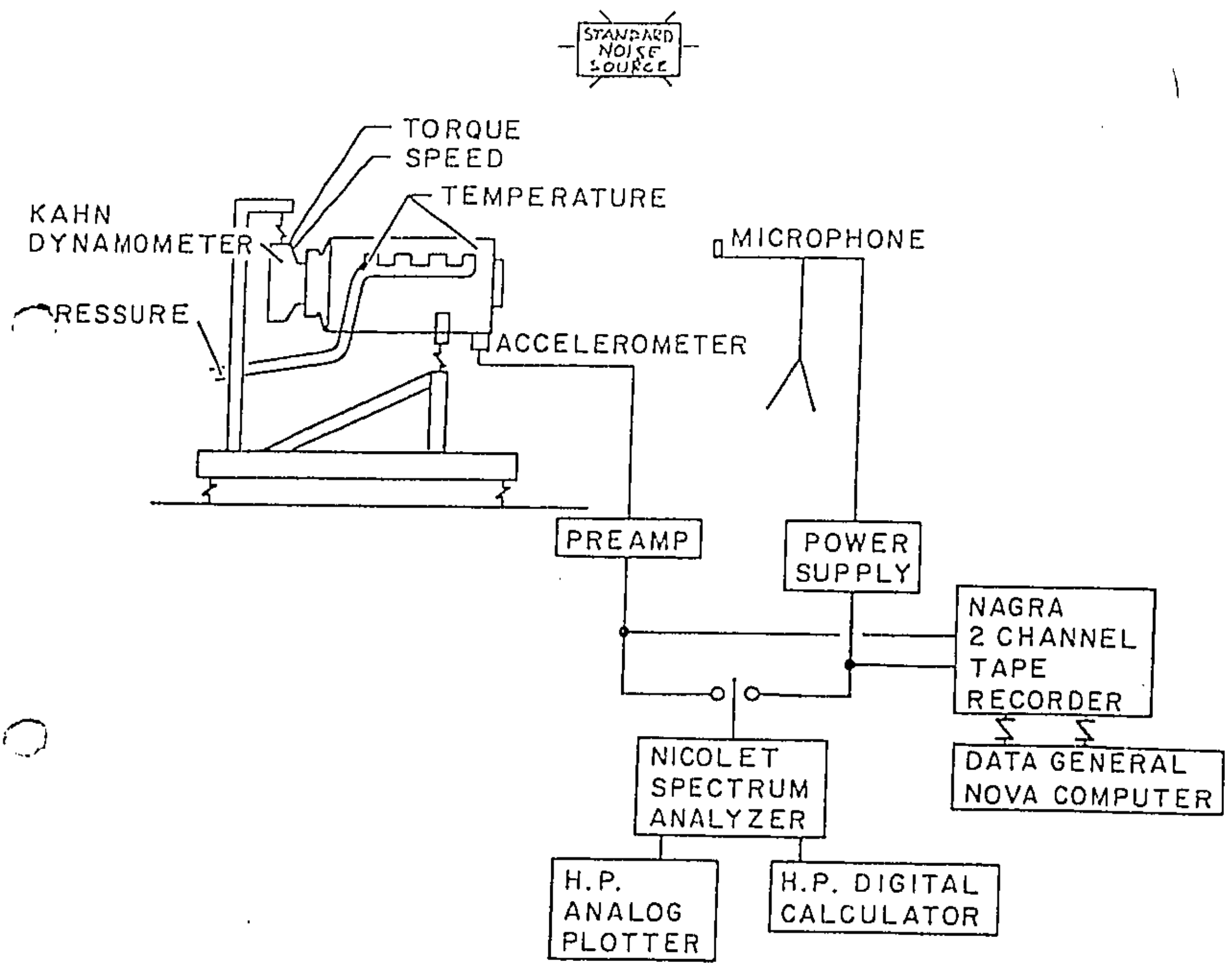

Figure 3. Noise Measurement Instrumentation

pressure level of a standard noise source with a constant sound power output (a centrifugal fan made by ILG Manufacturers, Chicago was used for this project). The standard noise source was measured after each engine measurement sequence to account for changes in the temperature and humidity of the air.

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

A load was applied to the engine with a Go-Power DT-2000 Waterbrake Dynamometer attached to the clutch housing. The dynamometer indicates both speed and torque during the operation of the engine. Other engine parameters such as temperatures and pressures were also monitored during the tests to assure that the engine was operating properly.

A photograph of the reverberant room facility is shown in Figure 4. A photograph of the engine mounted in the facility is shown in Figure 5. In order to reduce the intake and exhaust noise, the intake and exhaust pipes were wrapped with fiberglass and lead during the noise test. The exhaust was cooled with a water spray. The dynamometer and stand were also wrapped to reduce their radiated noise. The engine was run without a cooling fan to eliminate that noise source. The total sound power radiated by the engine was measured as a function of speed and load.

Figure 6 shows a summary of the noise emissions of the diesel engine. The A-weighted sound power level is plotted versus speed for 50% and 100% of the full load at each speed. The noise emissions show a dependence on both speed and load. The sound power level is approximately proportional to 25 log N where N is the engine speed in RPM. A more detailed representation of the noise emissions is given in the next two figures. Figure 7 shows the frequency spectra of the sound power levels at 2100 RPM for

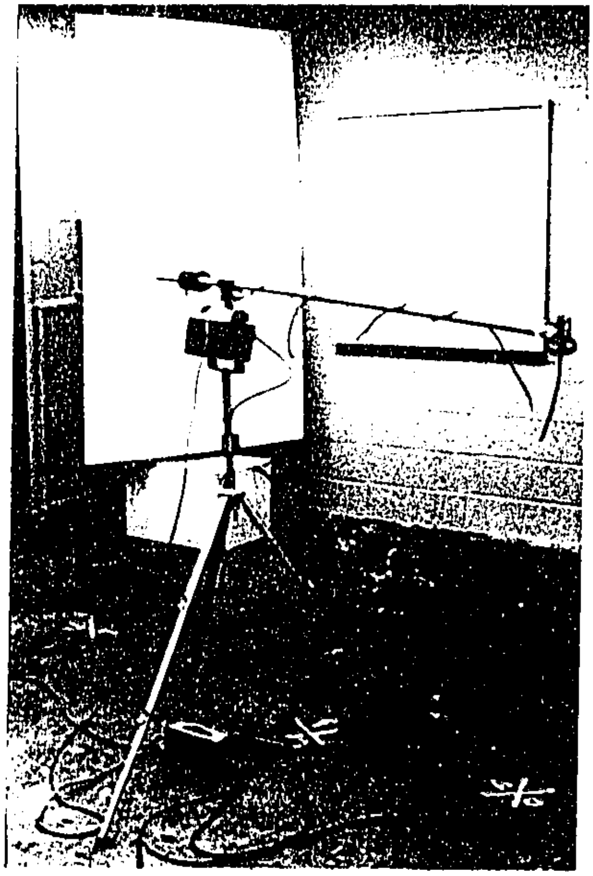

Figure 4. Reverberant Room Facility

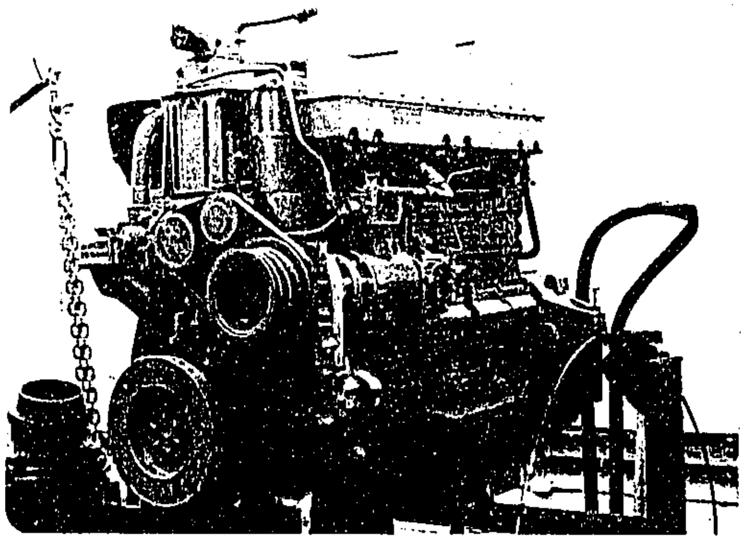

Figure 5. NTC-350 Engine Mounted in Test Cell

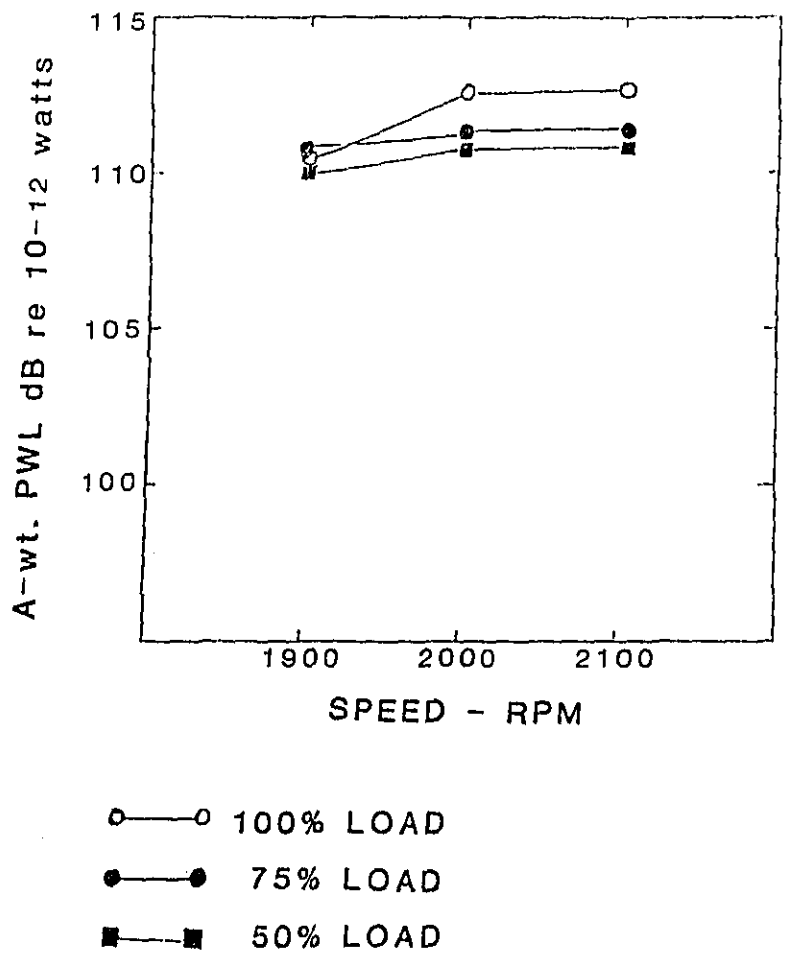

Figure 6. A-Weighted Sound Power Levels of NTC-350 vs. Engine Speed at Three Load Conditions

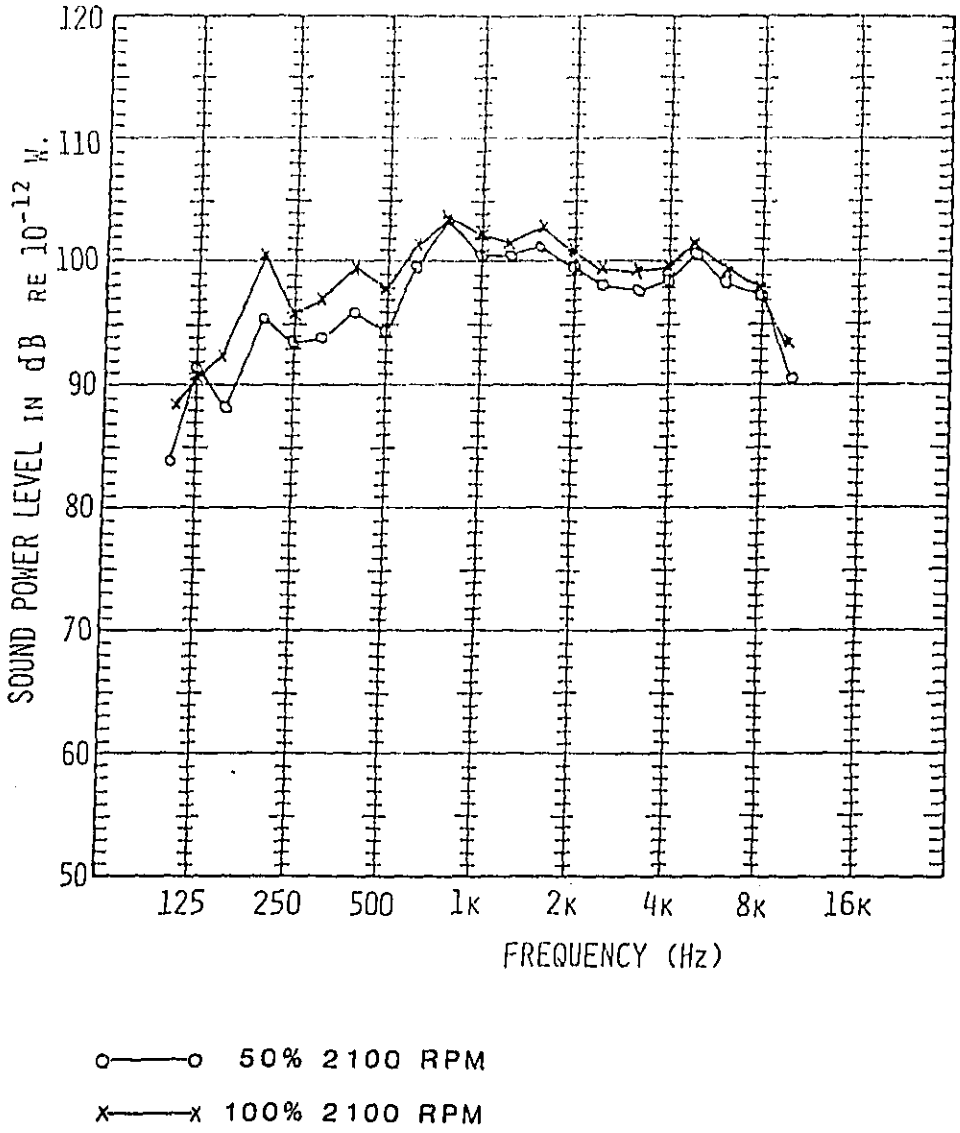

Figure 7. Frequency Spectra of Sound Power Levels of NTC-350 Engine at 2100 RPM, Two Load Conditions

the two load conditions. At higher frequencies the increase in sound level with load is small but fairly uniform with frequency. Figure 8 shows the frequency spectra of the sound power levels at full load for three speeds. Except at low frequency the increase in sound level with speed is fairly smooth with frequency.

B. Engine Vibration Characteristics

After the conclusion of overall engine noise measurements, the NTC-350 Big Cam One was instrumented with accelerometer mounting pads at several points on the external engine surfaces. The locations of the measurement points can be seen in Figure 9. The engine was run at 2100 RPM, 100% load and an accelerometer was moved from location to location. Accelerometer output for each point was analyzed with an FFT analyzer, averaged over time, and then plotted using an X-Y plotter.

After all the data were collected they were divided into the following six areas: front, right side, left side, oil pan, valve covers, and the after cooler. The data were then used to obtain the average surface velocity for each surface. Greater emphasis was placed on the surface velocity calculations for the lower right block, lower left block and oil pan surfaces since the major objective of the program was to evaluate the noise and measured vibration patterns of these surfaces within a truck engine compartment. Figure 10 represents the lower right and left block surface velocities and Figure 11 depicts the oil pan surface velocity.

It can be seen in Figure 10 that the lower left block has greater vibrational energy levels than the lower right block in the frequency range from 100 Hz to 2 kHz with a pronounced peak around 2 kHz. Since the left side of the engine block is the major thrust side with respect to piston motion, piston slap is expected to produce greater vibration on the left block side. In Figure 11 it can be seen that the cast aluminum pan exhibits fairly large vibration levels in the frequency range from 800 Hz to 2.5 kHz.

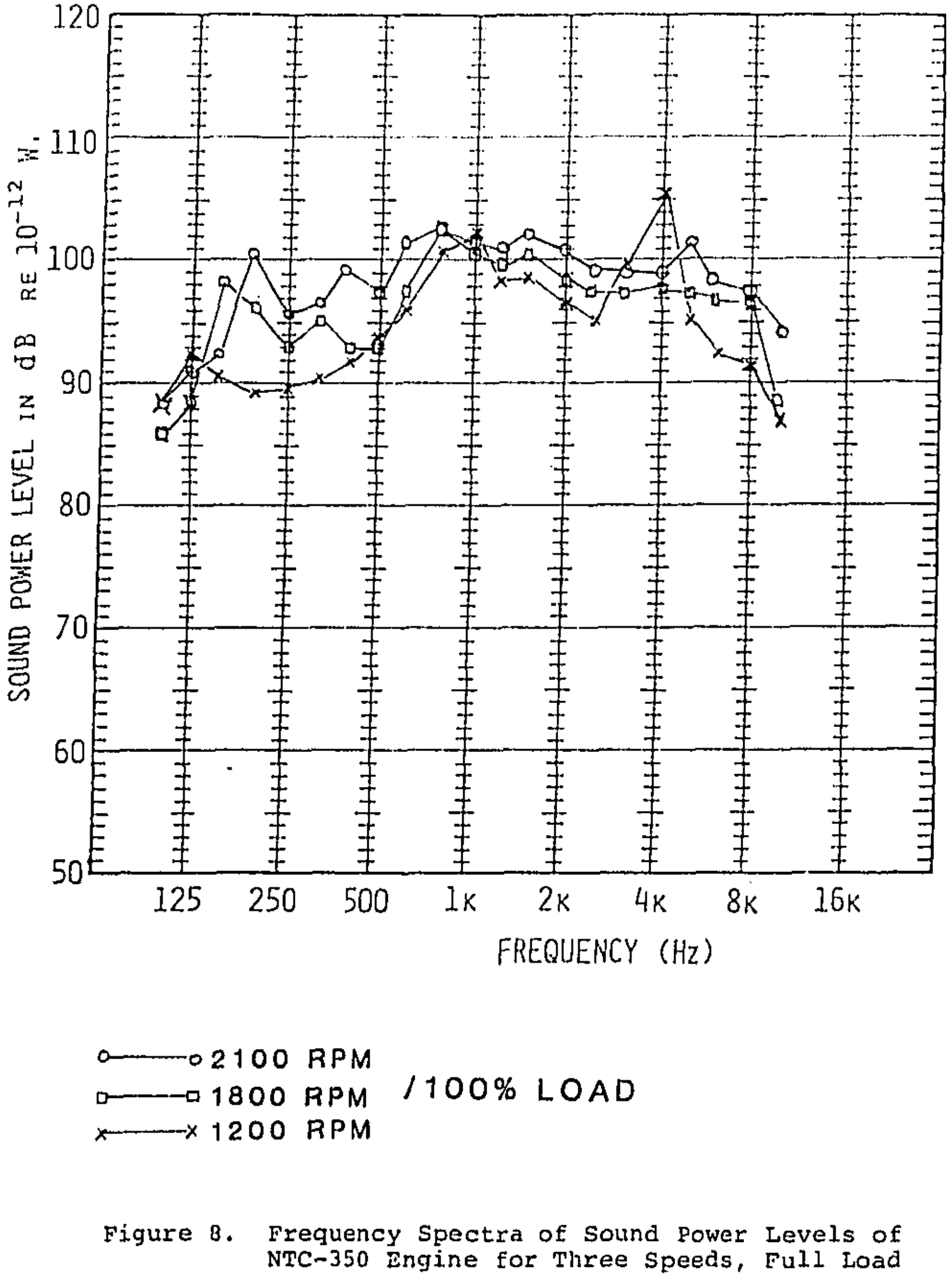

Figure 8. Frequency Spectra of Sound Power Levels of NTC-350 Engine for Three Speeds, Full Load

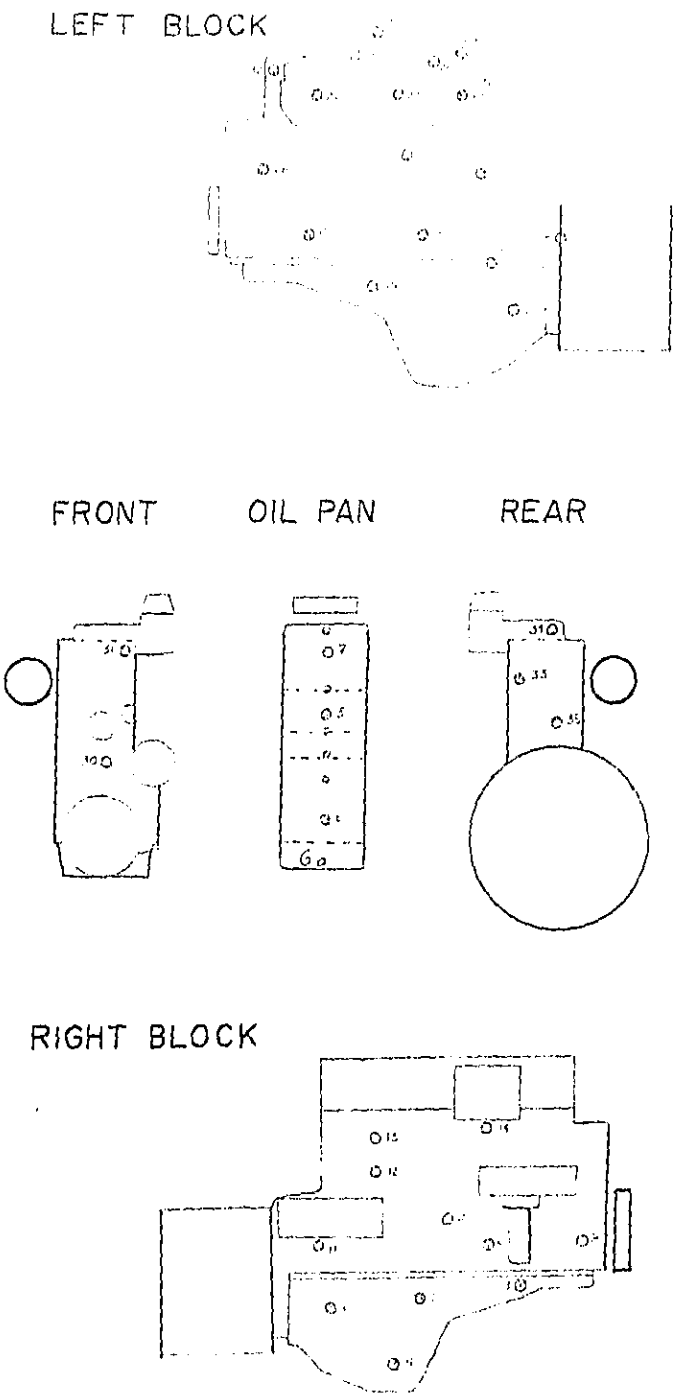

Figure 9. Vibration Measurement Locations on External Surfaces of NTC-350 Engine

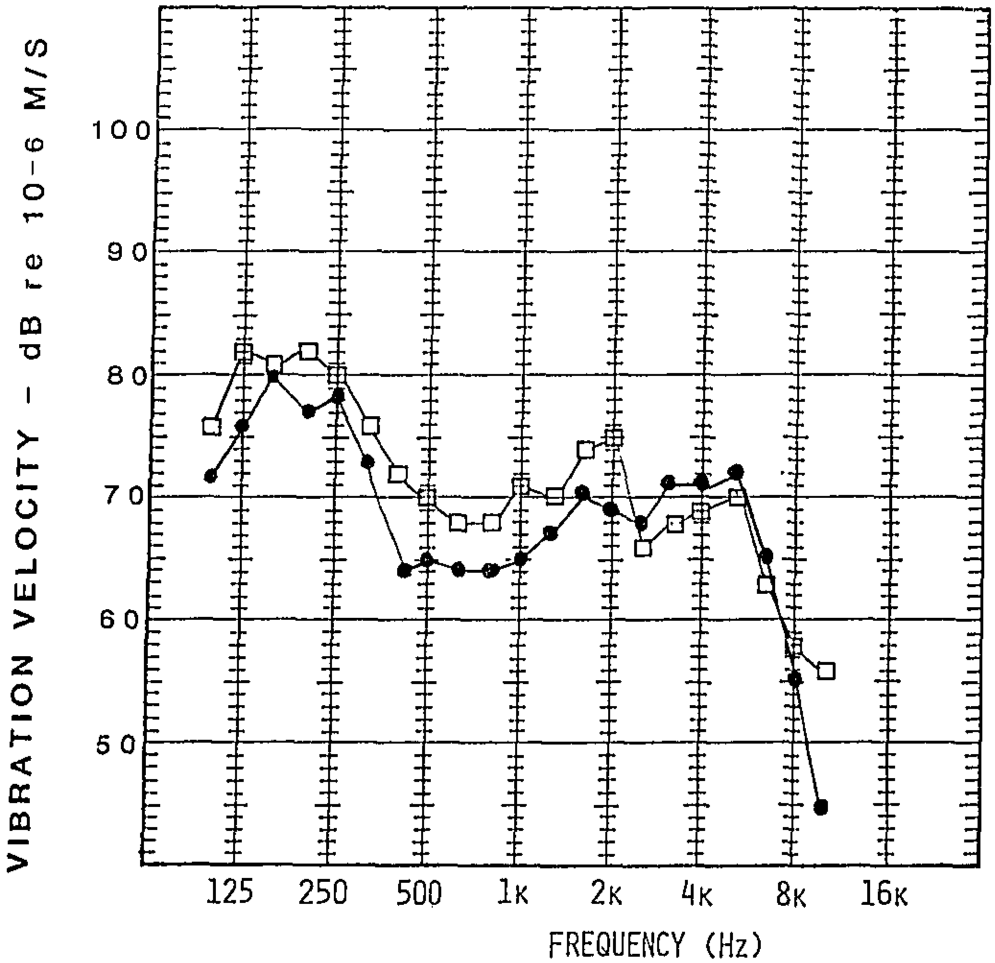

Figure 10. Lower Right and Left Block Surface Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

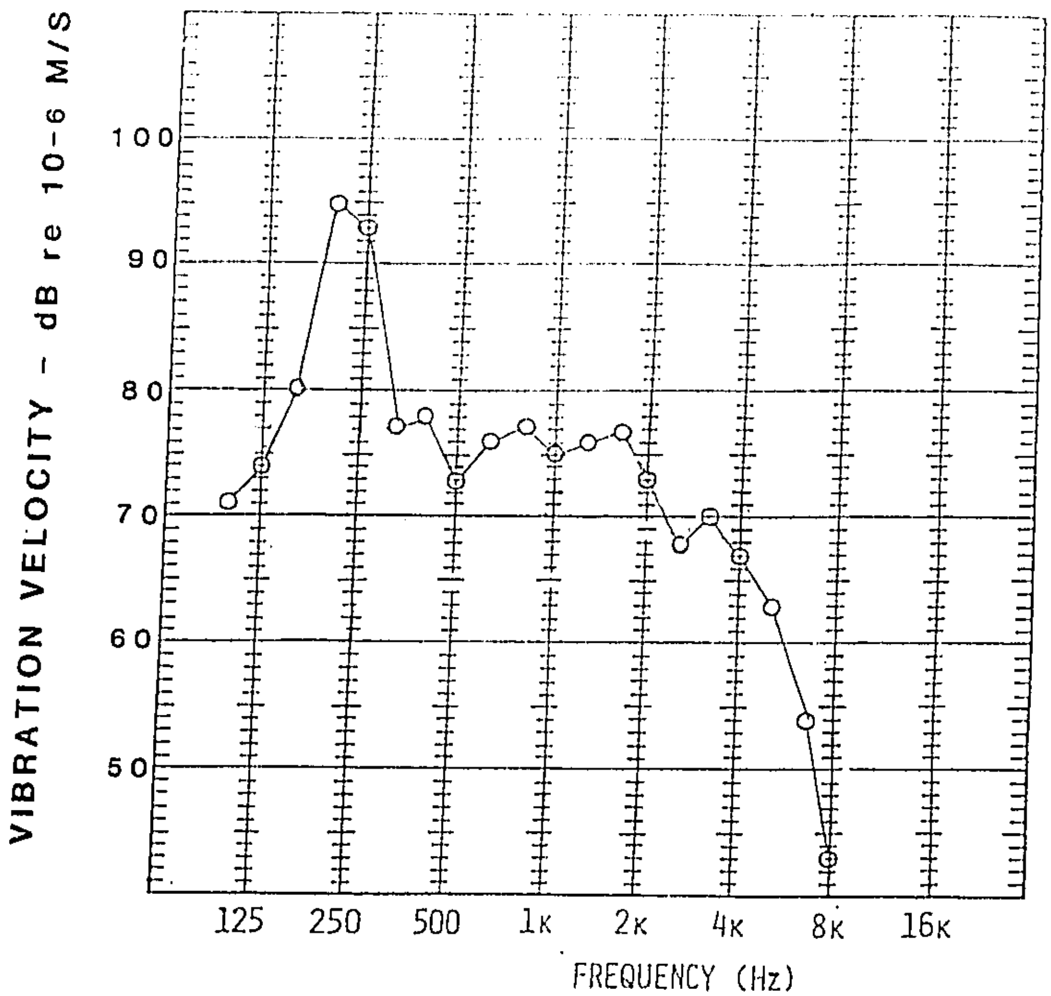

Figure 11. Oil Pan Surface Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

III. DEVELOPMENT OF A NOISE GENERATION MODEL

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

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

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

1. dynamic pressures and forces are produced within the engine by internal sources such as combustion, piston slap, fuel injection, gear mesh, etc.;

2. the pressures and forces act on the internal engine structure and cause a local vibration;

3. the engine structure transmits the vibration to external surfaces of the engine;

4. the vibrating external surfaces of the engine produce radiated noise.

Based on previous test results from Vee and in-line 6-71 engines as well as the DDA 6V-92TTA engine [2], three internal sources were chosen for study: combustion, piston slap, and injectors. For each source the level of the excitation and the vibration transmission to the engine surfaces have been measured. The results have been combined into a noise generation model which predicts the noise radiated by these engine sources and can be compared to the measured noise levels of the engine. The following paragraphs outline the procedures for studying each of these three sources and their associated vibration transmission paths. A more detailed discussion of the procedure is presented elsewhere [3].

A. Combustion Pressure

The source level for combustion noise was measured directly during operation of the engine with a pressure transducer in the combustion chamber of one of the cylinders. The transducer was water-cooled and flush-mounted in the cylinder head. Figure 12 shows the measured frequency spectra of the pressure signal. This cylinder pressure exerts a force both on the piston crown and on the cylinder head surface, indicating that there are two paths of vibration transmission. The first path is through the piston, connecting rod, crankshaft, main bearings, and into the block. The second path is through the head and into the block.

The vibration transmission characteristics of the engine structure are determined from measurements taken in a non-running engine of the transfer mobilities of the various components along each vibration transmission path. The transfer mobility is a transfer function which measures the ratio of the vibration velocity at one point on the structure to the force exerted at another point. The mobility is measured with the instrumentation shown in Figure 13.

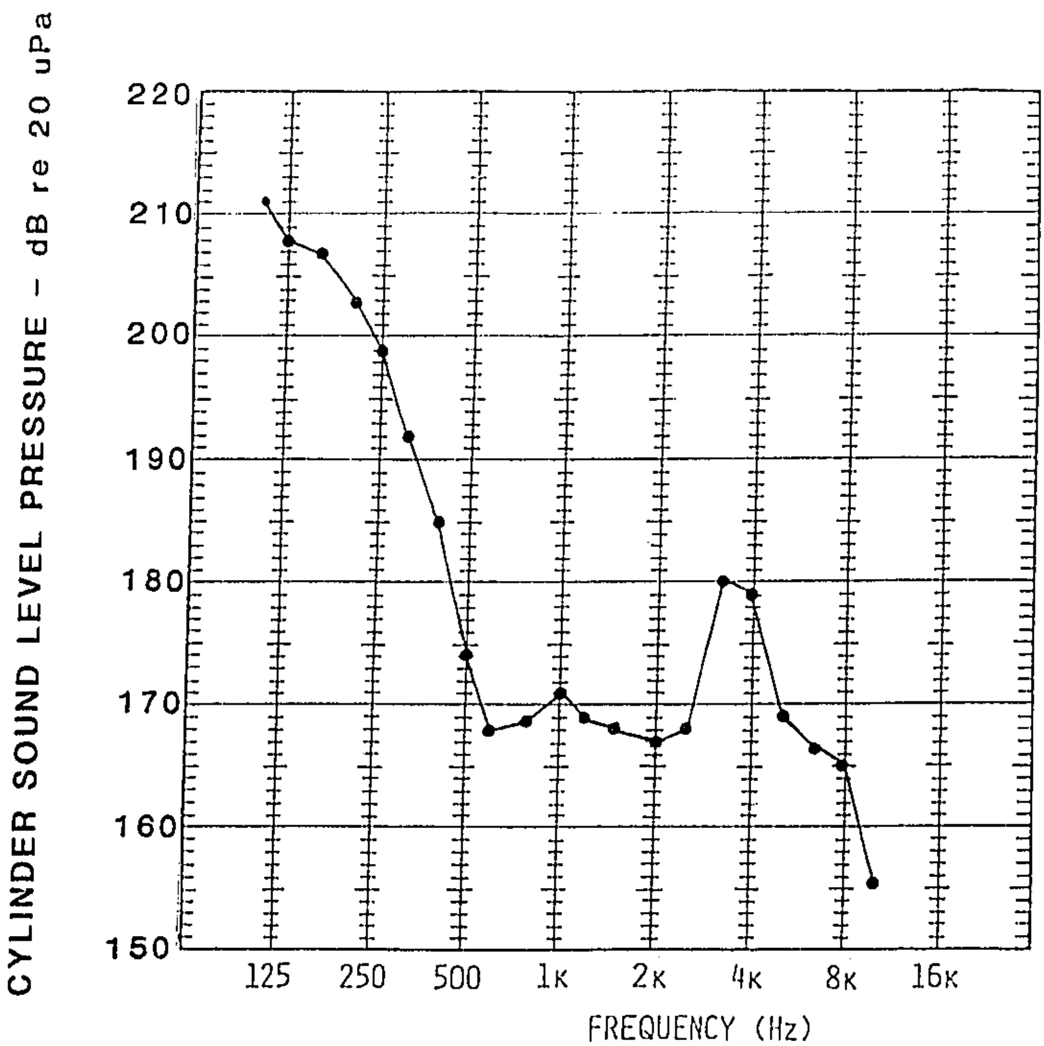

Figure 12. Frequency Spectra of Measured Cylinder Pressure on NTC-350 Engine at 2100 RPM, Full Load

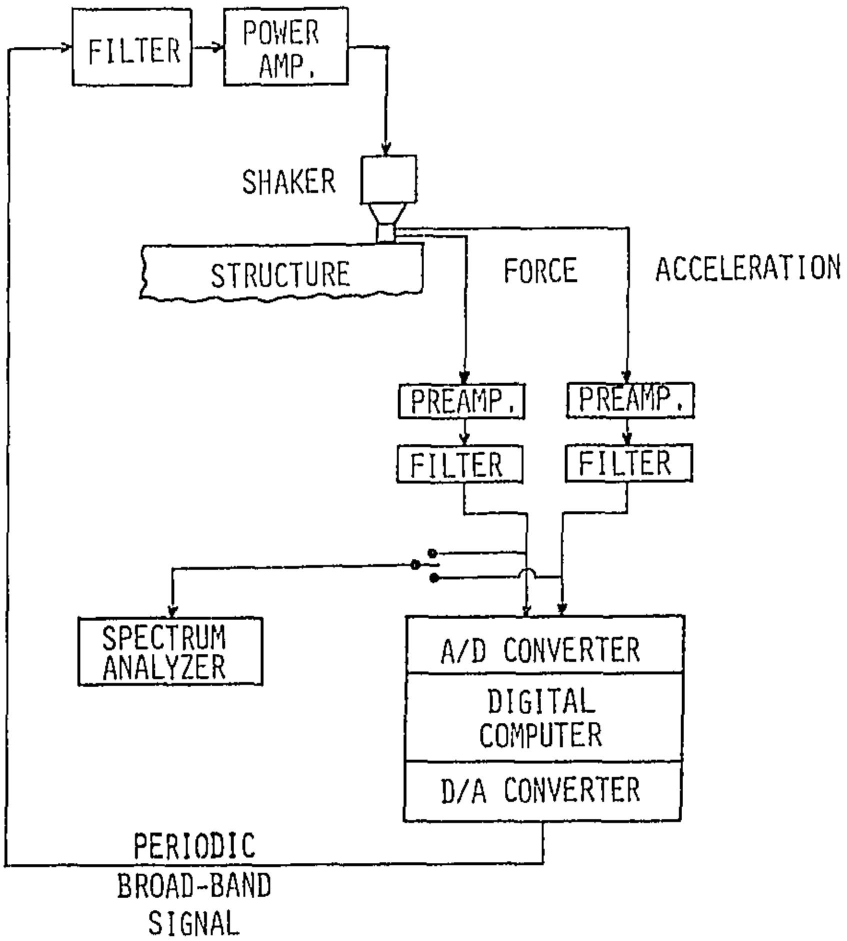

Figure 13. Instrumentation for Mobility Measurements

The two transfer mobilities for the paths by which combustion noise is transmitted, i.e. the head-block path and the piston-connecting rod path, are measured mobilities of one-third octave band averages of mobilities measured between eight points on the block and three cylinders. A prediction of the block vibration due to combustion when the engine is running is obtained by multiplying the total mobilities for each path by the combustion force (pressure times piston area) and adding the contributions from each path. A comparison of this prediction of block vibration due to combustion with the measured vibration during operation of the engine is shown in Figure 14. The comparison shows that except for the frequency band at 3200 Hz, combustion is not a major source of block vibration and therefore is not a major source of noise for this engine.

B. Piston Slap

The source level of piston slap cannot be measured directly by any known methods. Therefore, an indirect measurement procedure has been used to determine the magnitude of the force generated. This was done by measuring the local vibration response of the cylinder liner at a point on the liner where piston slap is believed to occur. Figure 15 shows a typical time response of the liner vibration during operation of the engine and the corresponding velocity spectrum. A second experiment was conducted to measure the drive-point mobility at the same point on the liner where the vibration response was measured. Since the drive-point mobility is the ratio of the velocity to the applied force at the point where the force is applied, the force generated by piston slap can be determined by dividing the measured velocity level by the drive-point mobility. Results are shown in Figures 16 and 17.

The vibration transmission path from piston slap to the engine surfaces is primarily through the liner and block side walls for the NTC-350. The transfer mobility between the cylinder liner and the block side walls is shown in Figure 18.

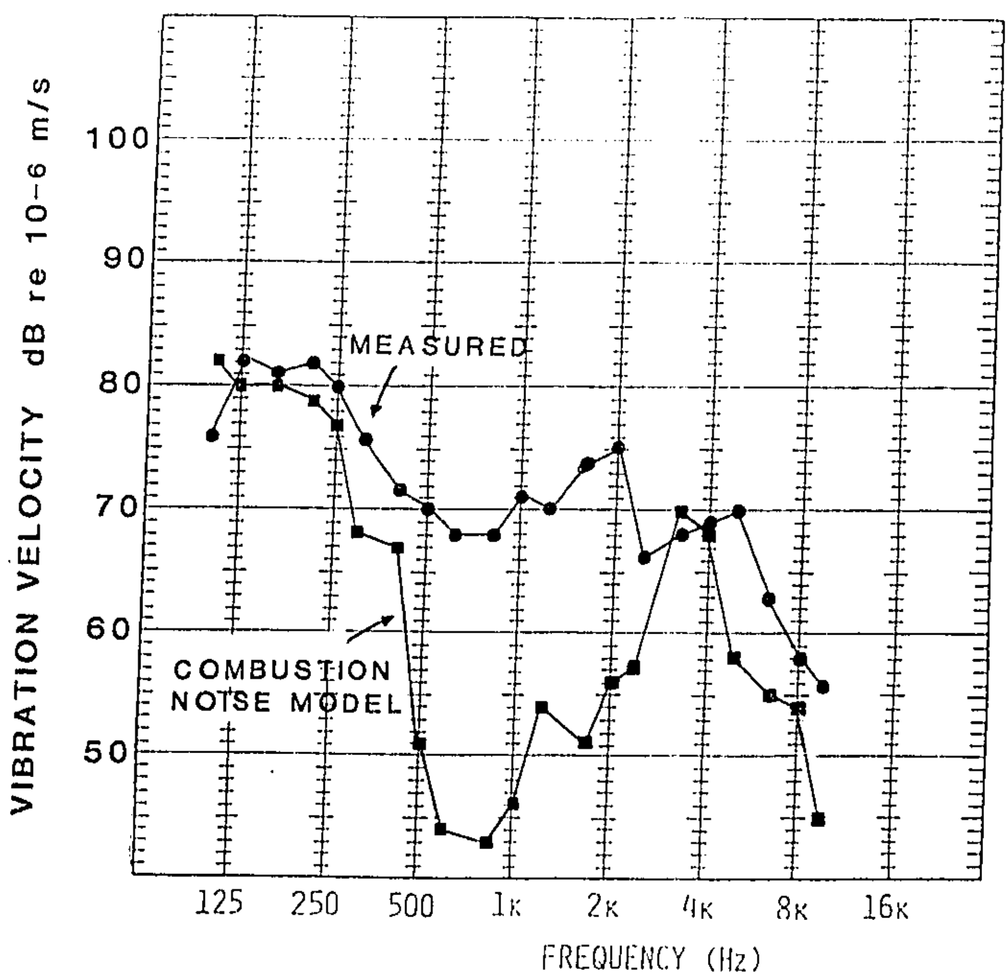

Figure 14. Comparison of Vibration Transmission from Combustion with Overall Vibration on NTC-350 Engine at 2100 RPM, Full Load

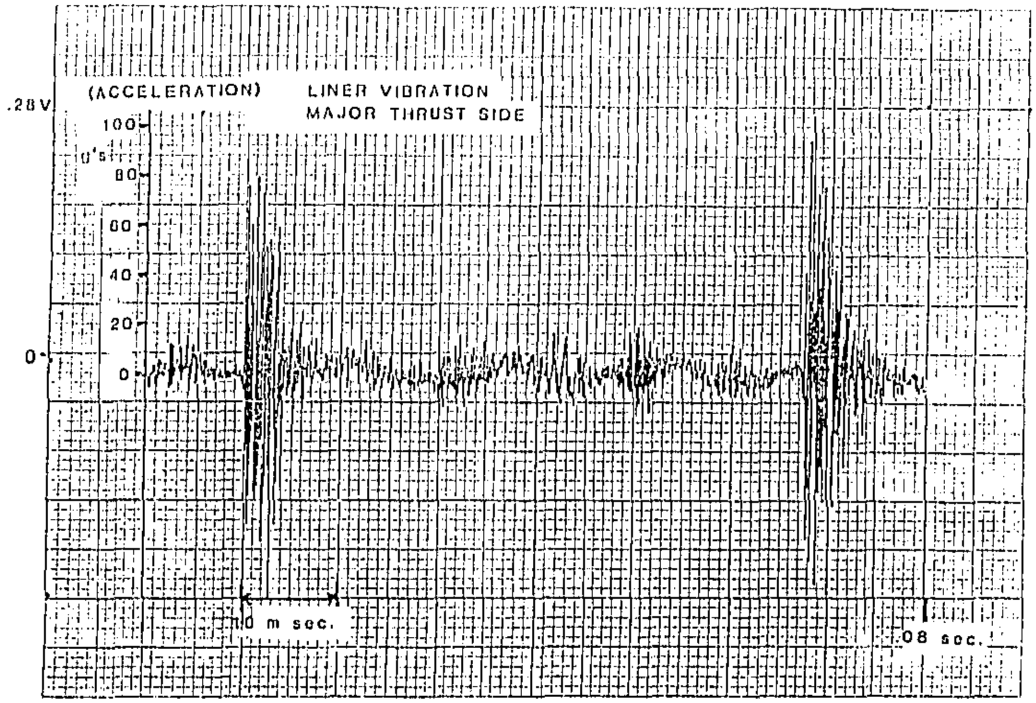

Figure 15a. Time Response of Liner Vibration, Major Thrust Side, on NTC-350 Engine at 2100 RPM, Full Load

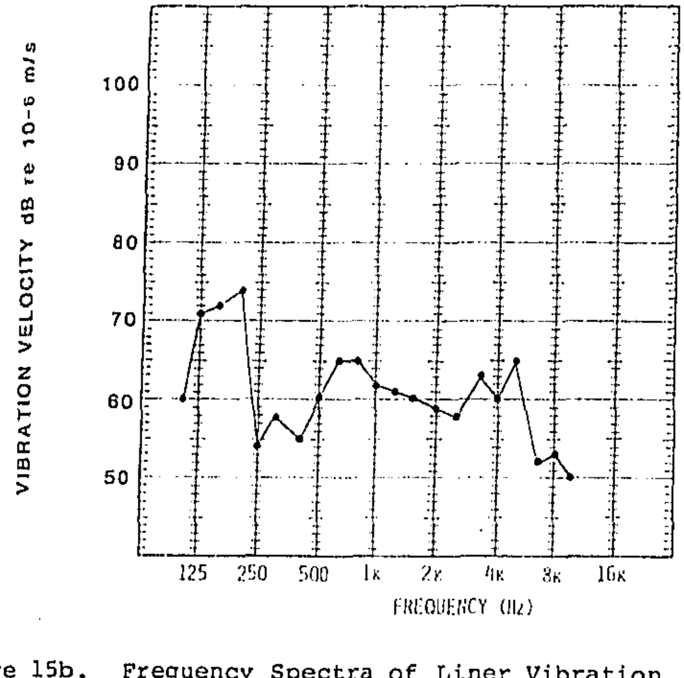

Figure 15b. Frequency Spectra of Liner Vibration, Major Thrust Side, on NTC-350 Engine at 2100 RPM, Full Load

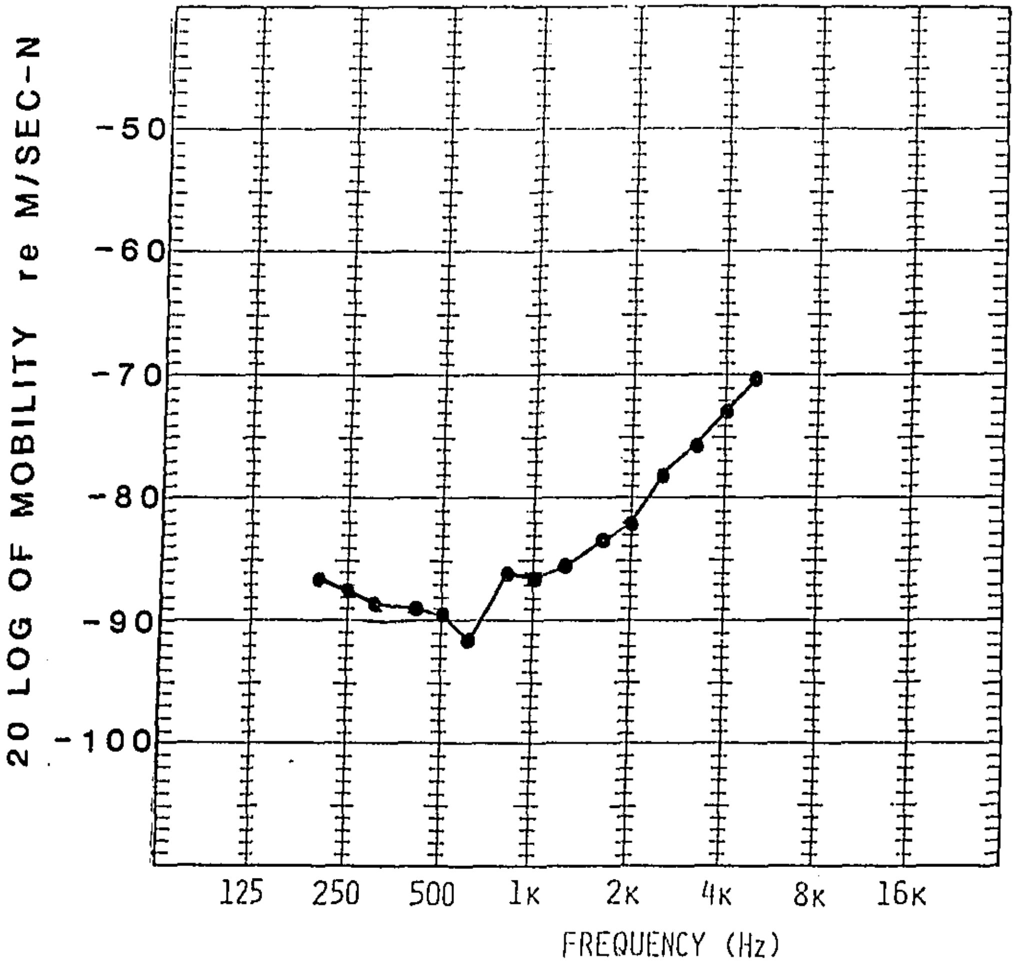

Figure 16. Drivepoint Mobility of Cylinder Liner on NTC-350 Engine

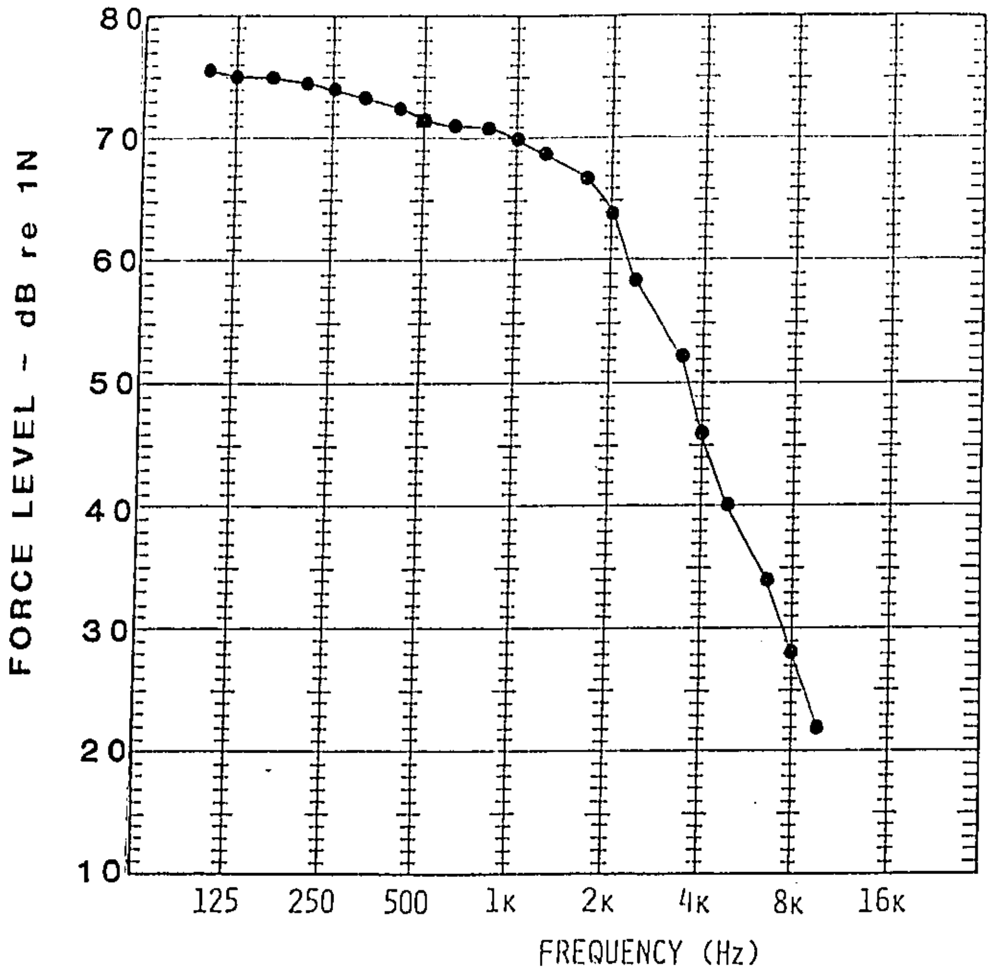

Figure 17. Estimated Force of Piston Slap on NTC-350 Engine at 2100 RPM, Full Load

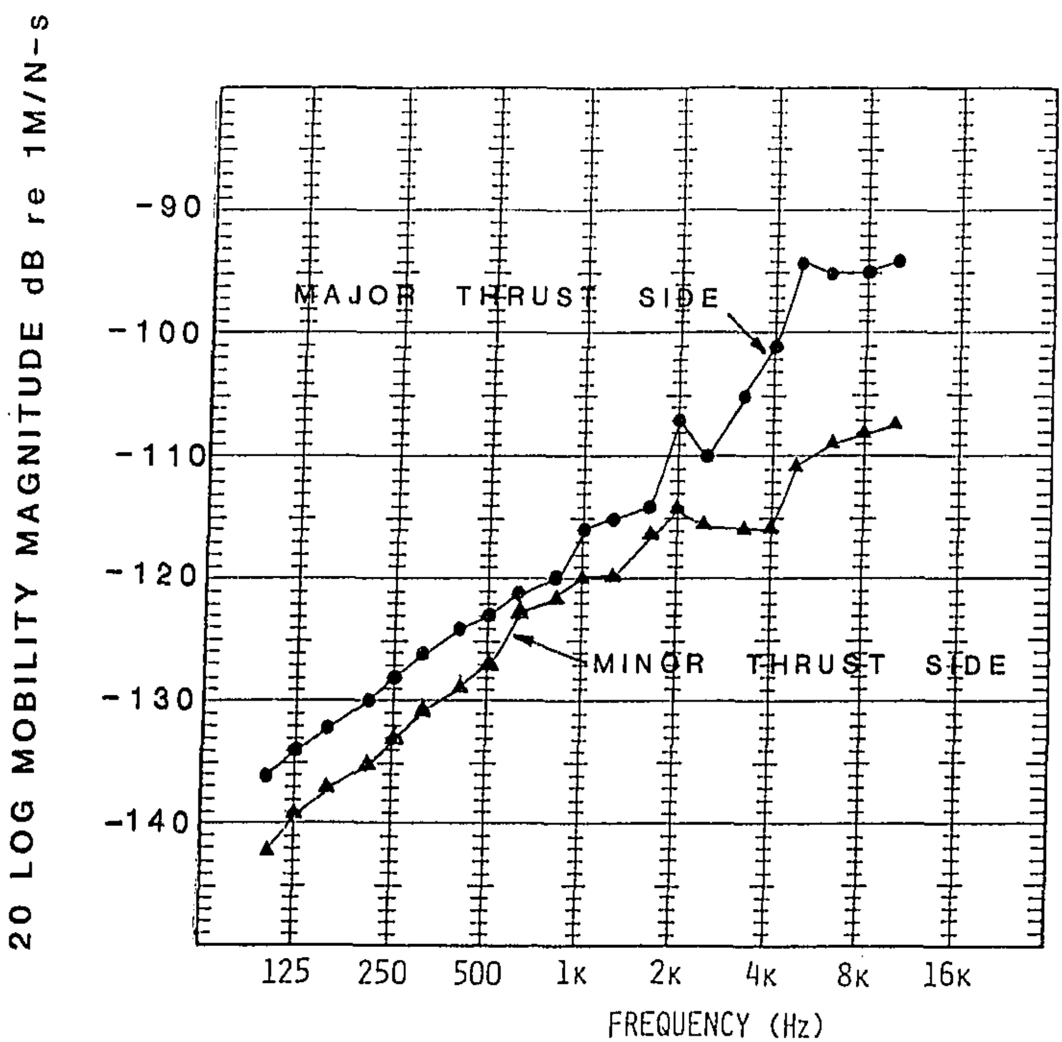

Figure 18. Piston Slap Vibration Transmission from Cylinder Liner to Block on NTC-350 Engine

Combining the force excitation level of the piston slap with the measured transfer mobility gives a prediction of the engine block vibration due to piston slap. Figure 19 compares this prediction with the measured vibration levels for the left block side, which is the major thrust side and the major radiating surface. This comparison shows that piston slap is a major source of block vibration, and therefore radiated noise, in the frequency range from 500 to 5000 Hz with a peak at 2000 Hz.

C. Injector Forces

The NTC-350 Big Cam One is equipped with PT injectors, each of which injects fuel into a cylinder by means of a cam-driven plunger. A direct measurement of the dynamic force generated during injection is difficult. Therefore, an indirect measurement of the injector source level was obtained in a procedure similar to that done for piston slap. The velocity of the injector rocker was measured during the operation of the engine and is shown in Figure 20. The designed velocity level of the injector cam follower is also shown by the dashed line. It can be seen that the plunger exhibits a large vibration near the end of injection with a large 1600 Hz component in the spectrum and a sharp drop beyond that point in the frequency spectra.

The force exerted on the injector plunger was obtained by dividing the velocity spectrum of the plunger by the measured mobility of the plunger in its fully assembled condition (see Figure 21).

Forces generated by the injectors are transmitted to the engine block by two paths: the first through the injector rocker box to the head and the engine block, and the second through the cam push-rods, camshaft, bearings, cam gears to the block. The vibration transmission through each path was determined by physically disconnecting the injector plunger from the cam drive train and exciting the structure with a shaker attached to one or the other of the disconnected members. The resulting measured

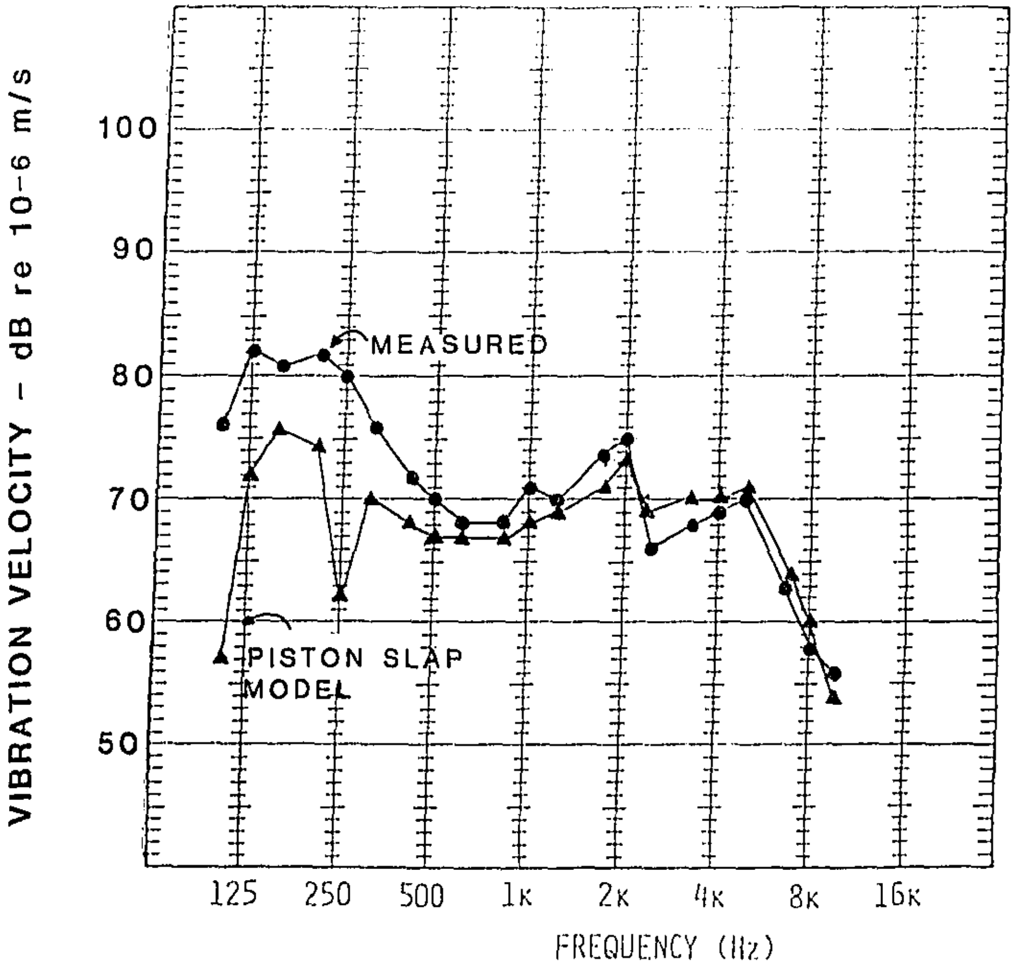

Figure 19. Piston Slap Vibration Levels vs. Measured Block Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

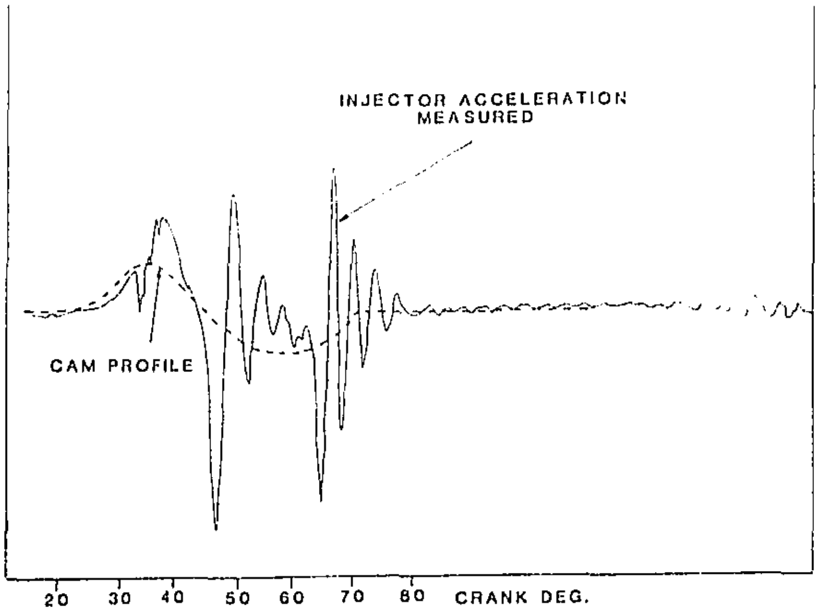

Figure 20a. Injector Rocker Velocity, Cylinder #3, on NTC-350 Engine at 2100 RPM, Full Load

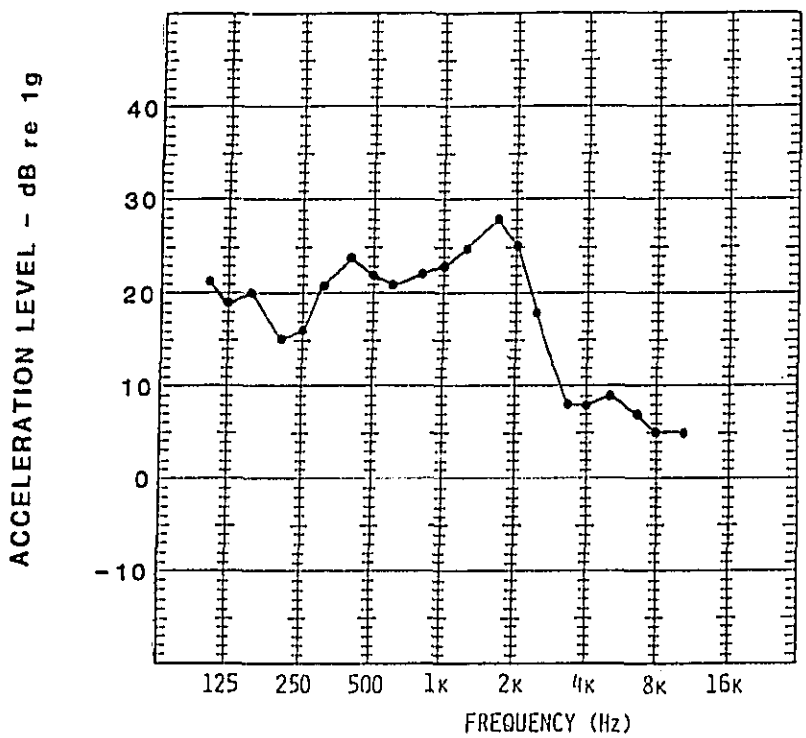

Figure 20b. Frequency Spectra of Injector Vibration Level on NTC-350 Engine at 2100 RPM, Full Load

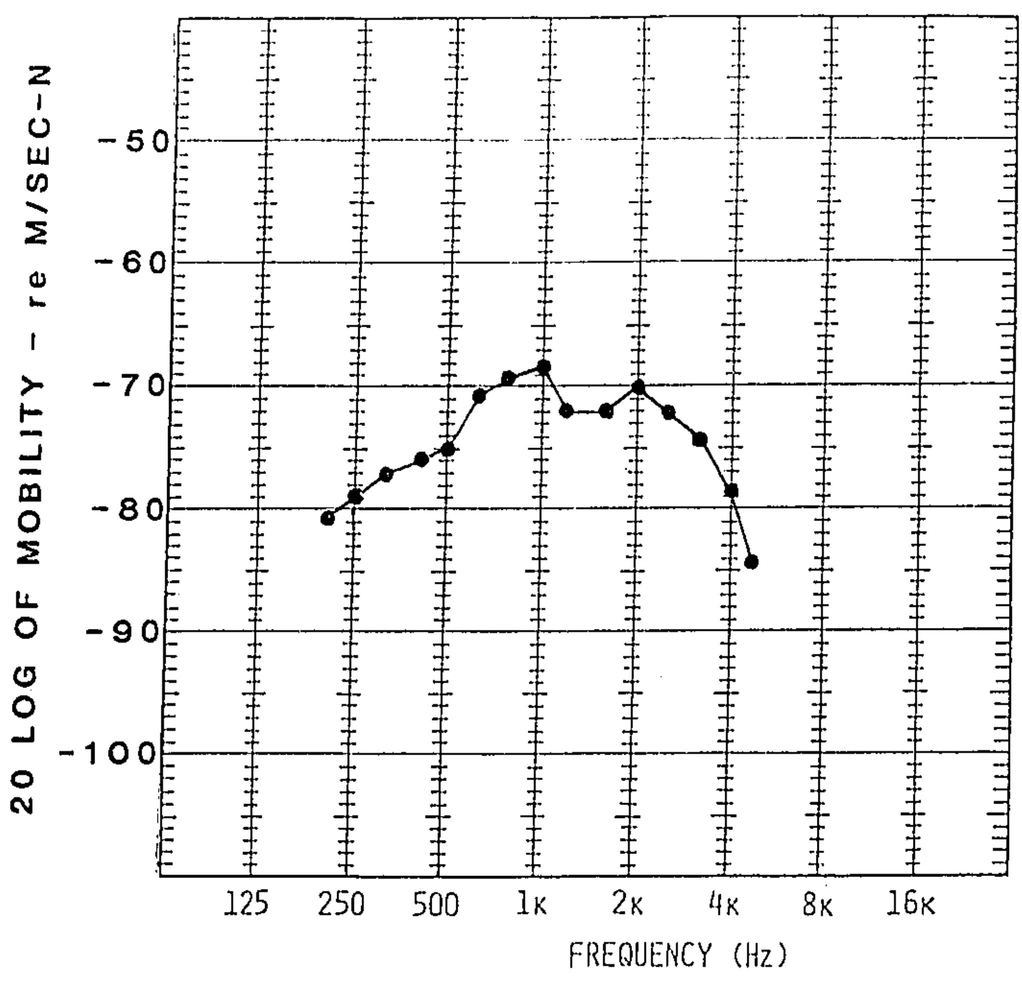

Figure 21. Drivepoint Mobility of Injector on NTC-350 Engine

transfer mobilities to the block are shown in Figure 22. The path through the camshaft dominates the vibration transmission above 2000 Hz.

Combining the force excitation level of the injector and the measured transfer functions to the block gives a prediction of the engine block vibration due to the injectors. Figure 23 compares this prediction with the measured block vibration levels during operation of the engine. The comparison shows that the injectors are a major source of vibration at 1600 Hz.

D. Vibration Transmission of Non-Load Bearing Covers

By summing the contributions to block vibration from the major sources, piston slap and injectors (see Figure 24), a nearly complete model of the engine vibration can be obtained for this specific engine. In order to complete the model it is necessary to relate the vibration of non-load bearing covers, such as the oil pan, valve covers and front cover to the block vibration since these covers can be major radiating surfaces. Following the mobility approach the vibration transmission from the block to covers has been determined by combining measured mobilities on a non-running engine with measured block vibration levels. Examples of the mobilities for the oil pan are shown in Figure 25. A prediction of the ratio of velocities of the oil pan and the block is shown in Figure 26 along with a ratio determined from measured data. The comparison shows that the oil pan has higher vibration levels over most of the frequency range of interest.

E. Noise Radiation

To complete the modeling, the noise radiated by each engine surface must be related to the predicted velocity. The approach chosen was determining a radiation efficiency by simultaneously measuring the surface vibration and the sound power radiated using the acoustic intensity technique. The measured values of

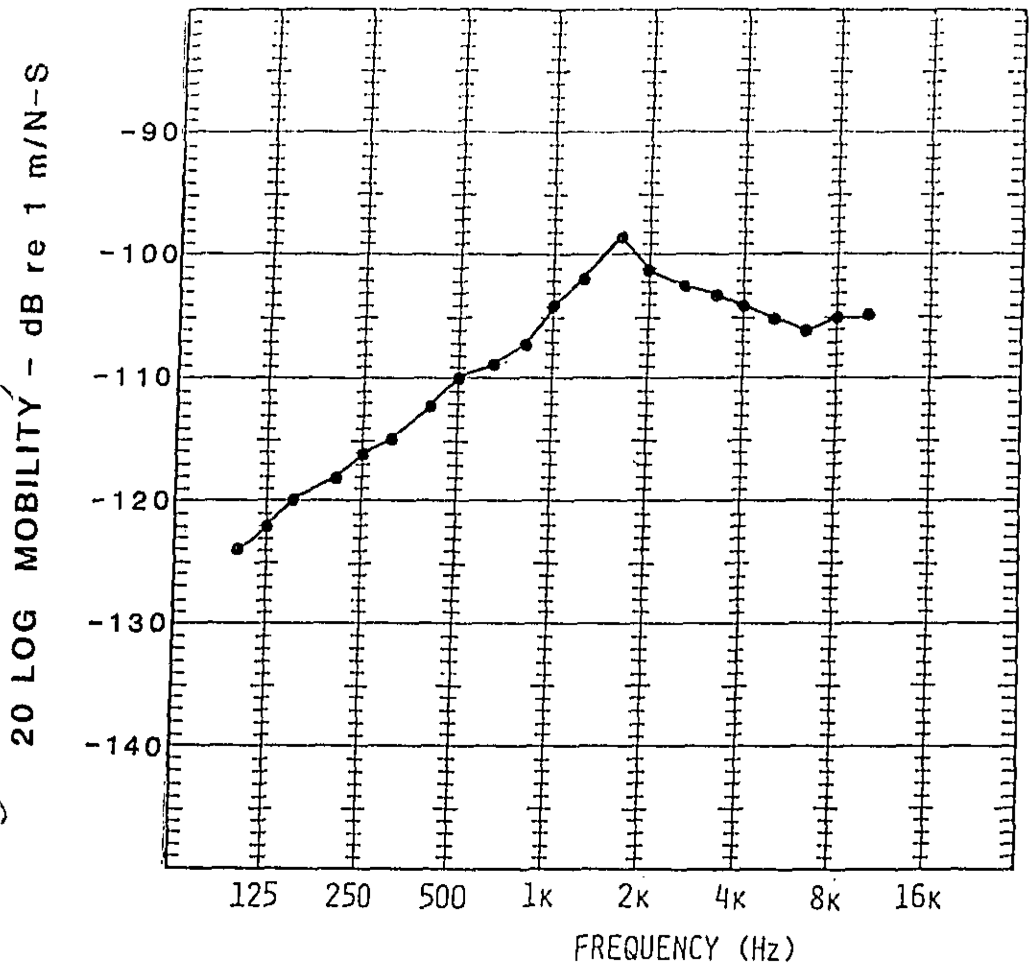

Figure 22. Vibration Transmission from Injector to Block on NTC-350 Engine

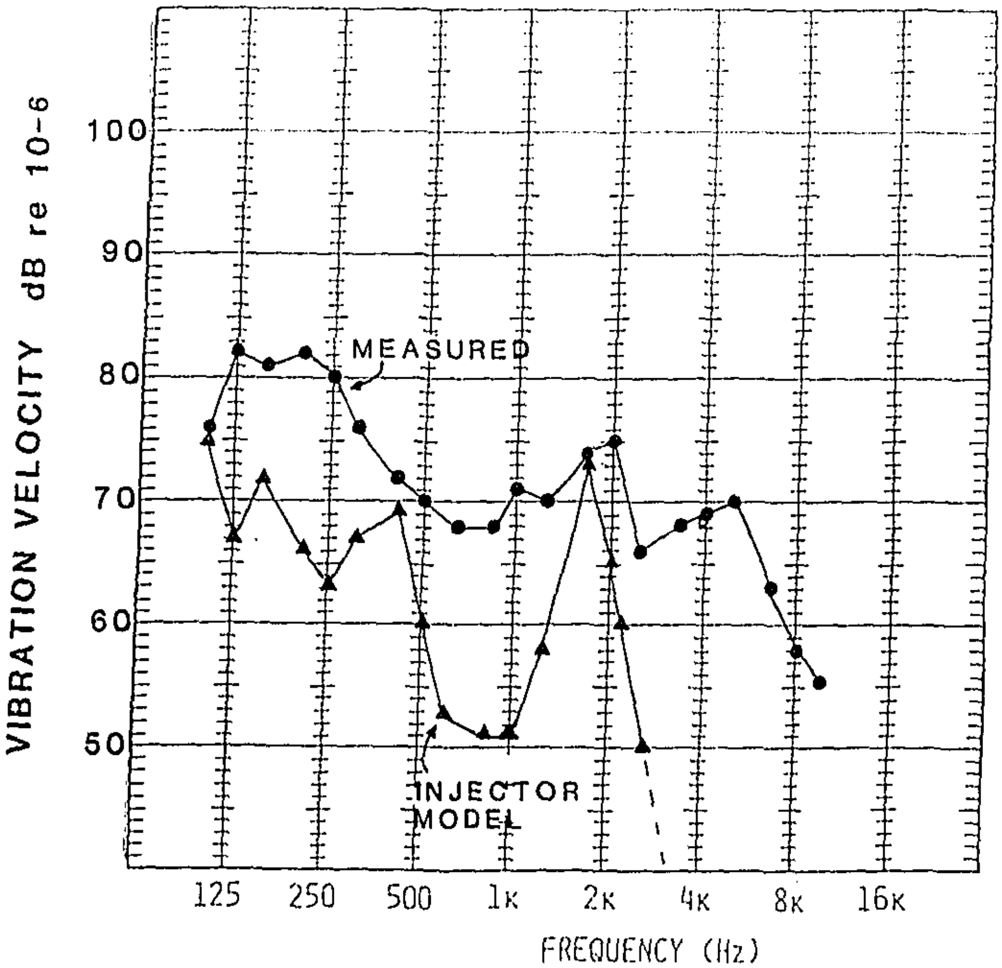

Figure 23. Injector Vibration Levels to Lower Left Block vs. Measured Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

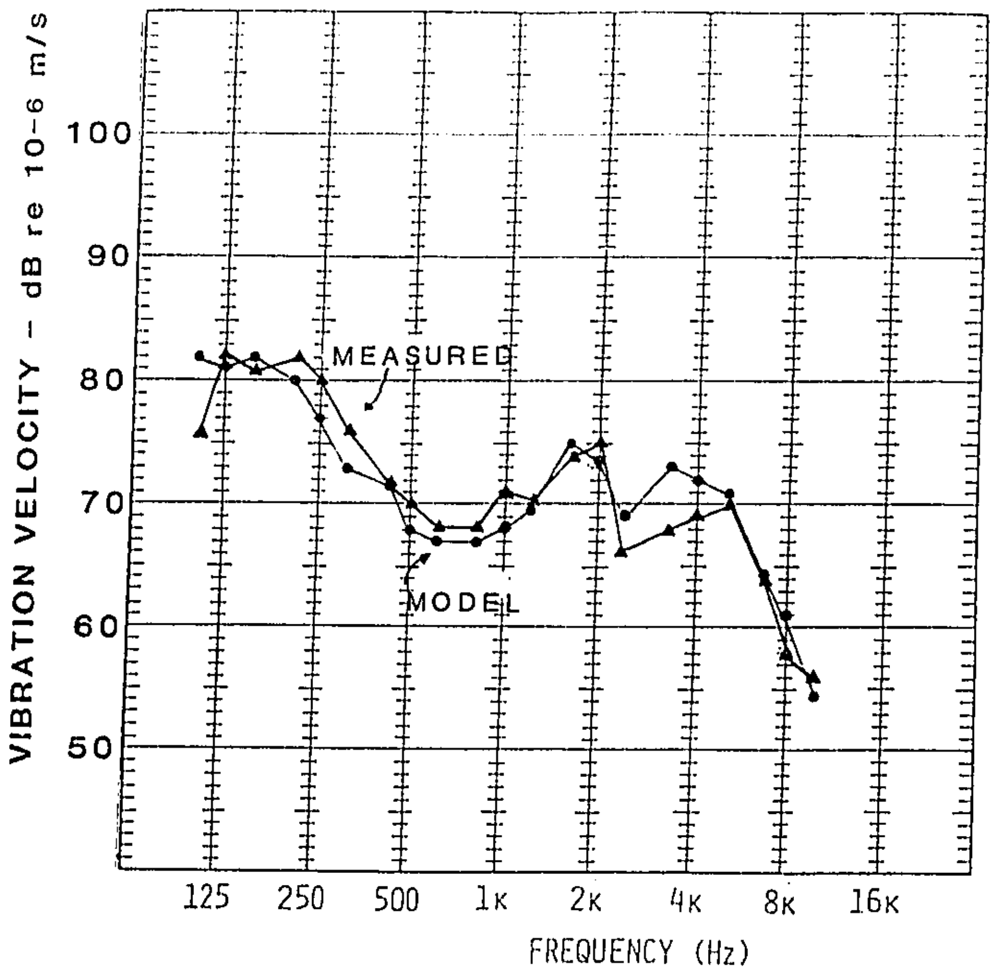

Figure 24. Sum of Piston Slap and Injector Vibration Levels to Lower Left Block vs. Measured Vibration Levels on NTC-350 Engine at 2100 RPM, Full Load

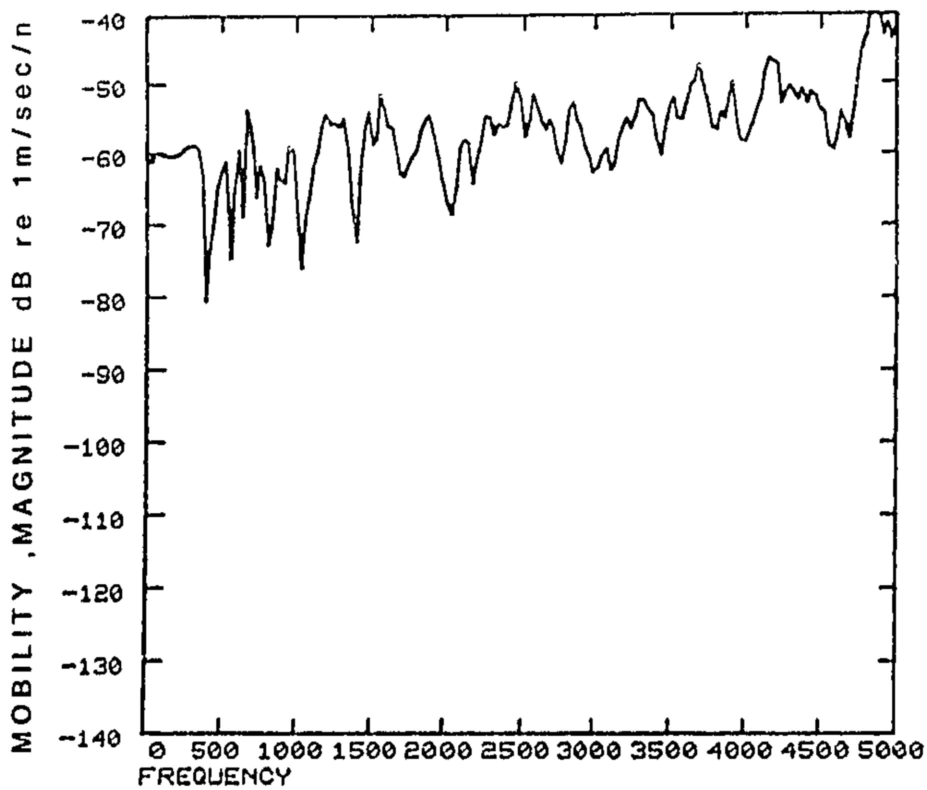

Figure 25. Oil Pan Mobilities, Free, on NTC-350 Engine

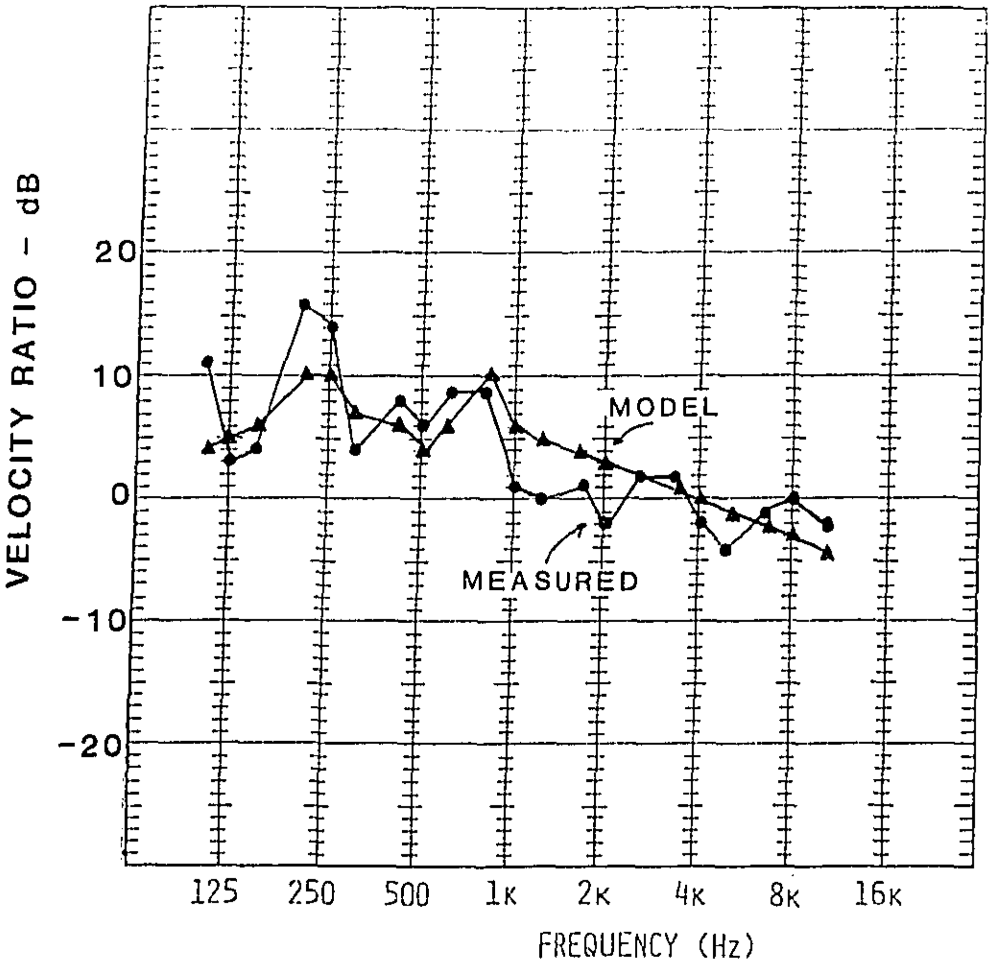

Figure 26. Oil Pan to Lower Block Vibration Ratio on NTC-350 Engine at 2100 RPM, Full Load

radiation efficiency were combined with the prediction of block and cover vibration to generate a prediction of the noise radiated by the engine. Results are shown in Figure 27 and are compared to the total measured sound power levels. There is good agreement between prediction and measurement except in the 300 to 500 Hz range.

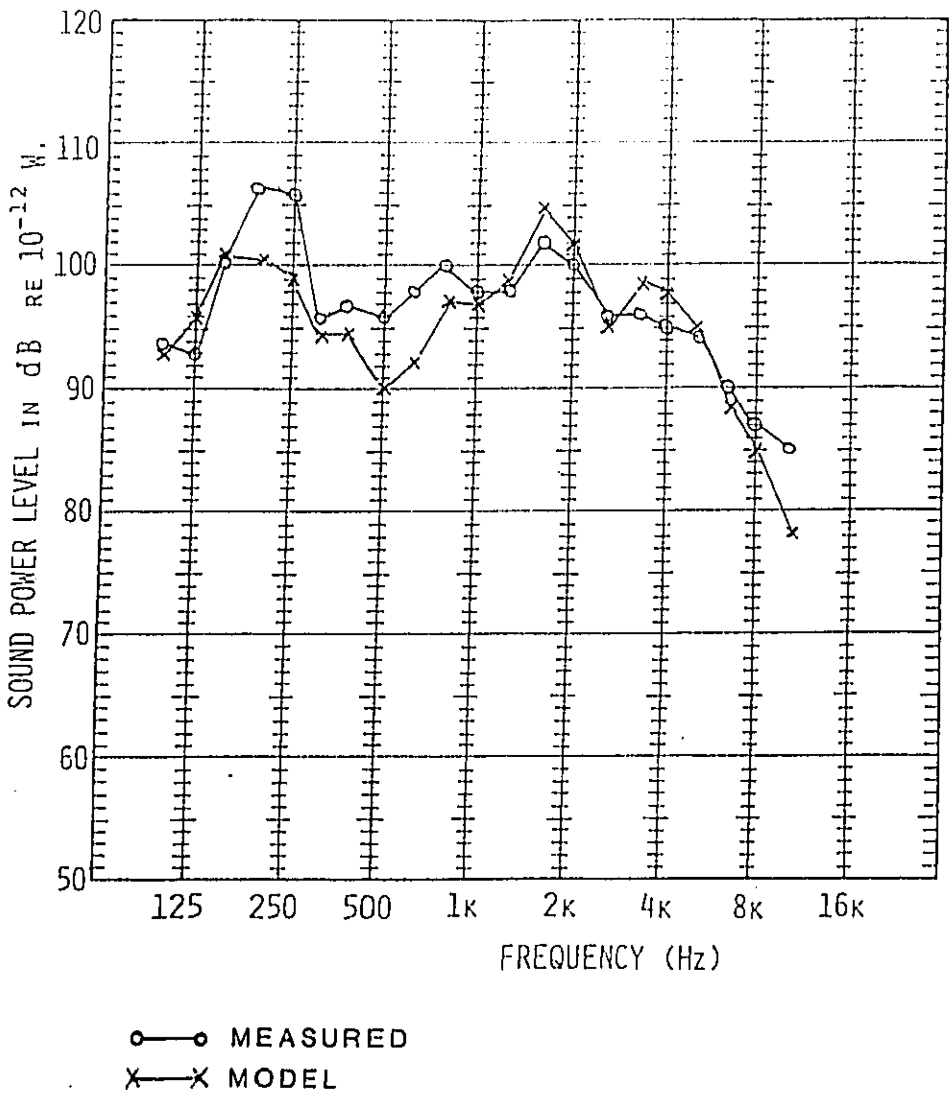

Figure 27. Lower "In-Truck" Sound Power Levels vs. Total Measured Sound Power Levels on NTC-350 Engine

IV. NOISE REDUCTION DESIGNS

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

Based on the results obtained in the modeling phase, it is apparent that the major source of high frequency vibration for the NTC-350 Big Cam One is piston slap, with lesser contributions to overall A-weighted truck sound power levels from injectors and combustion. As can be seen in Figure 28, piston slap dominates the spectrum from 500 Hz to 8 kHz with equal contributions by injectors at the 1600 Hz bandwidth.

The combustion source is important below 500 Hz and at frequency bands of 3.15 to 4 kHz. The limited contribution of the combustion process to overall A-weighted sound power levels is common to turbo-charged engines and the peak in the frequency spectrum of the combustion model can be correlated to the cavity resonances of the combustion chamber.

Despite the overall domination of piston slap as the major noise contributor, it was felt that to achieve an overall 5 dB reduction in truck noise it would be necessary to treat all three sources, piston slap, injection, and combustion, since the expected reduction in transmission of piston slap excitation would result in a 4 dBA overall noise reduction. Our approach to reducing the noise from these sources was to modify the engine structure to reduce the vibration transmission from each source to the radiating surfaces. Other methods for achieving the desired noise reduction included: (a) modifying the sources to reduce the impulsive forces generated during injection and piston slap, (b) using engine covers and enclosures to reduce the noise radiation from the vibrating engine surfaces, and/or (c) using damping and vibration isolation treatments to reduce the

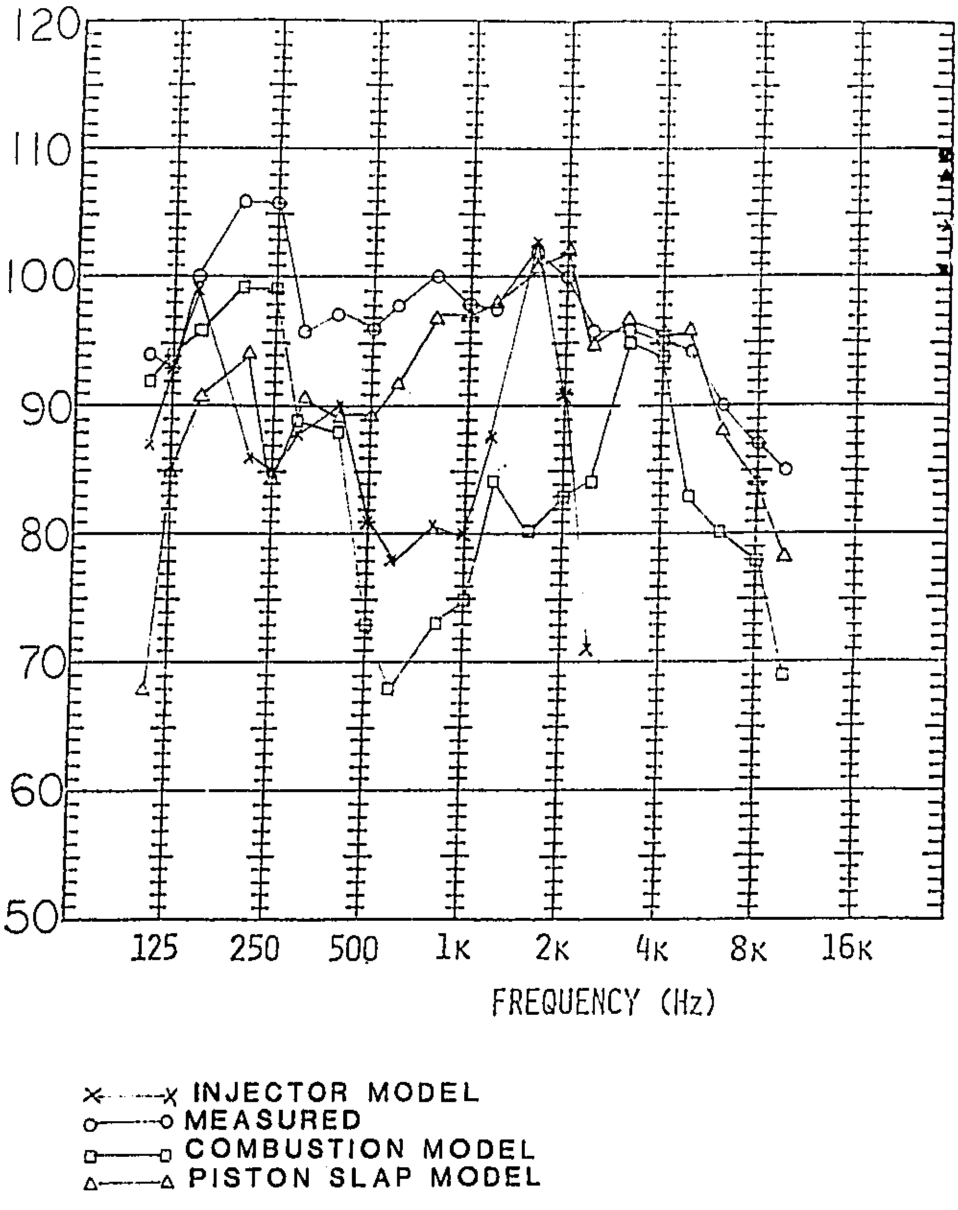

Figure 28. Contribution of Sources to Overall A-Weighted Truck Sound Power Levels on NTC-350 Engine at 2100 RPM, Full Load

vibration and noise radiation from non-load bearing covers and shields. These other techniques are valid noise control procedures and have received a great deal of attention by others working in the area of engine noise reduction. In our work we have not eliminated these techniques from consideration, but have not spent any time developing them because of the work already done by others and because of our belief that modification of the engine structure is the most cost-effective means to achieve our 5 dB noise reduction goal.

A. Piston Slap Noise Reduction Designs

In order to reduce the contribution of piston slap excitation to overall noise generation, the three primary paths of vibrational energy transmission from the excitation point to the radiating surface were examined. It became apparent that the major transmission path is from the top liner attachment point to the upper block and from there to the lower radiating surfaces. In seeking to alter this path two other possible arrangements for attaching the liner to the engine structure were considered.

The first alternate approach involved suspending the liner midway in the block right under the top water jacket. The second approach called for attaching the liners to the head through some fastening arrangement. To use either approach we found that it would be necessary to increase the wall thickness of the liner to achieve a higher local impedance resulting in a decrease of power flow from the piston to the liner during piston impact. A more detailed discussion of both approaches is given in the following paragraphs.

1. Midway Suspended Liner

The primary reason for suspending the liner midway on the block is the relative ease of implementing this design change. Since the engine block is an in-line configuration, however, and the lower attachment point is not necessarily stiffer than the original attachment location (as is the case for an engine block of vee construction [1]) certain doubts were raised based on our fundamental design principles. We believe that to achieve a major reduction in the contribution of an excitation mechanism to overall noise by making a structural change in the transmission path, it is necessary to create a substantial impedance mismatch along a particular path. The mismatch is needed to achieve the reflection of vibrational energy back toward the excitation mechanism and the high damping values possibly associated with this occurrence.

In the case of the transmission path for the midway suspended liner, the increase of the liner stiffness resulted in the need for a stiffer attachment point to the block so that an impedance mismatch could be achieved. Achieving a softer attachment point for the purpose of an impedance mismatch presented the problem of selecting an isolation material that would have the proper stiffness and the proper load capabilities to sustain the relatively high clamping loads applied to the liner top in order to achieve combustion seal. These clamping loads are in the magnitude of 40,000 to 60,000 PSI.

In determining the appropriateness of the midway attachment of the liner to the block, several measurements of the drivepoint impedance and the transfer impedance were conducted at the proposed liner attachment point and from that location to six lower block locations. The results shown in Figure 29 support our argument that the transfer of vibrational energy through this attachment point is not reduced and in certain cases actually increases. This design approach was rejected, therefore, in favor of the attaching the liner to the head.

2. Liner Attachment to the Head

When the design having the liner attached to the head was originally conceived, it appeared to have great potential for reducing the contributions of piston slap and combustion to overall noise reduction. However, it required major changes in

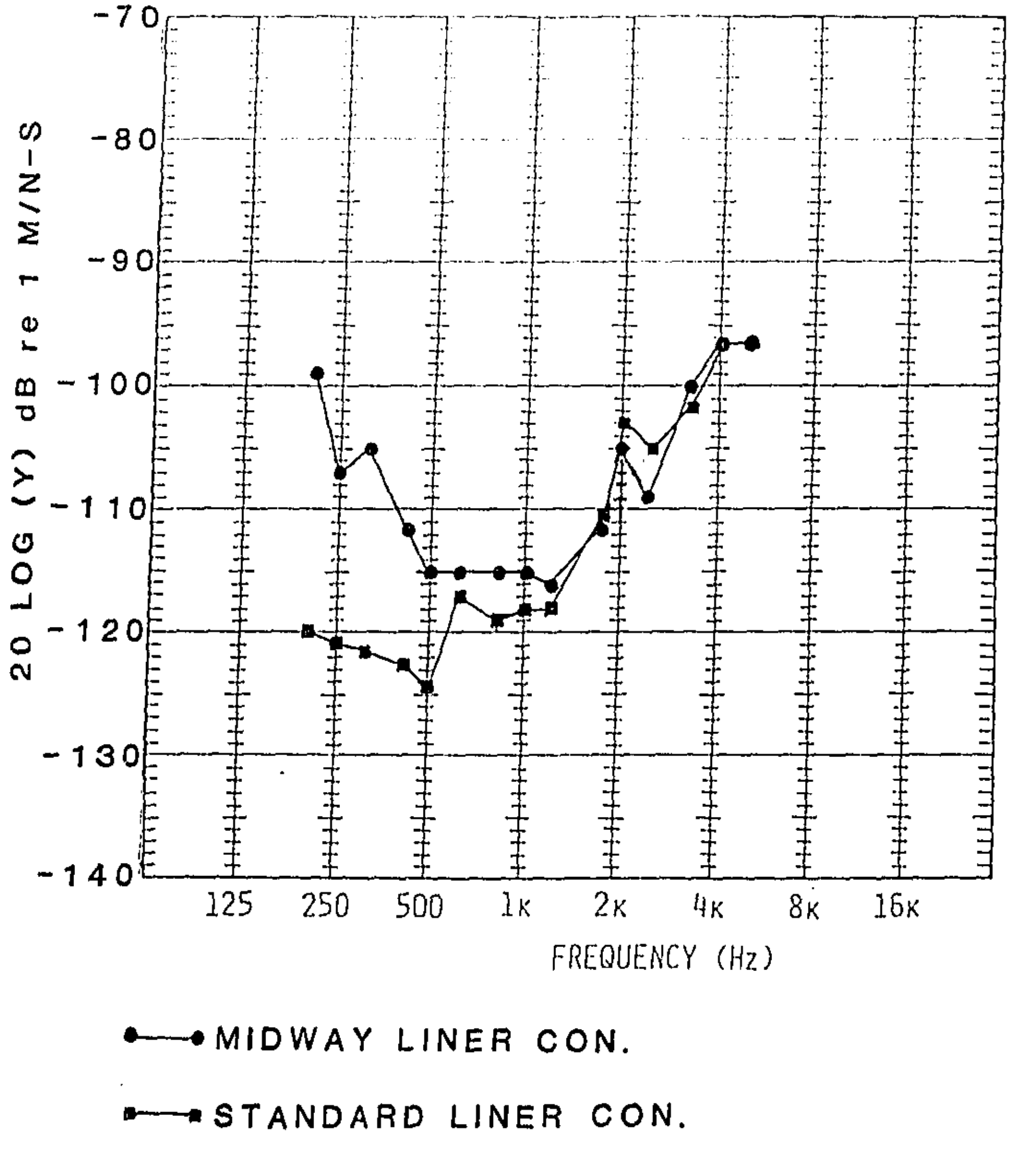

Figure 29. Comparison of Midway Suspended Liner with Standard Liner Attachment on NTC-350 Engine

regard to engine maintenance and assembly procedures. The basic design involves the attachment of the top portion of the liner to a machined groove in the fire deck of the head (see Figure 30). The fastener of the liner to the head for combustion sealing and alignment was designed to be a threaded connection using a low taper 10° thread pattern which enables a better load distribution characteristic than standard 45° threads. The primary reason for having liners attached to the head which can be separated for maintenance as opposed to casting the liners and head as an integral part, is that most heavy duty engines are designed with the intention of replacing liners when they are worn down rather than over-sizing the bores.

Another feature of this design approach is isolating the head from the block by using a relatively compliant material between the fire deck of the engine head and the top deck of the block as well as isolating the head bolts from the head (see Figure 31 for bolt isolation design).

The attachment of the liner to the head eliminates the need for applying a high preload on the head gasket by the head bolts in order to achieve a compression seal. This allows installation of a softer gasket material in between the head and the block so that an effective isolation can be achieved. The isolation, coupled with the lower power flow into the liner resulting from the higher local impedance, should yield a substantial reduction of piston slap contribution to overall noise.

The primary change concerning engine assembly caused by this design approach is the necessity for installing the pistons in the liner prior to installing the liner head assembly to the engine block. In order to facilitate production of a demonstration engine within cost and time constraints, we found that the following compromises had to be made:

a. the production of the prototype liners from retrofitted standard liners rather than from new liner castings, preventing us from having substantially thicker liners as originally requested.

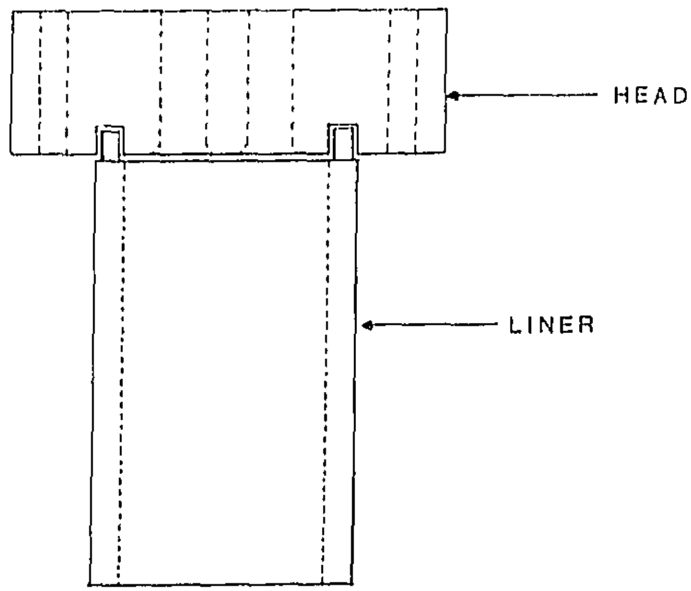

Figure 30. Proposed Liner Attachment to Head on NTC-350 Engine

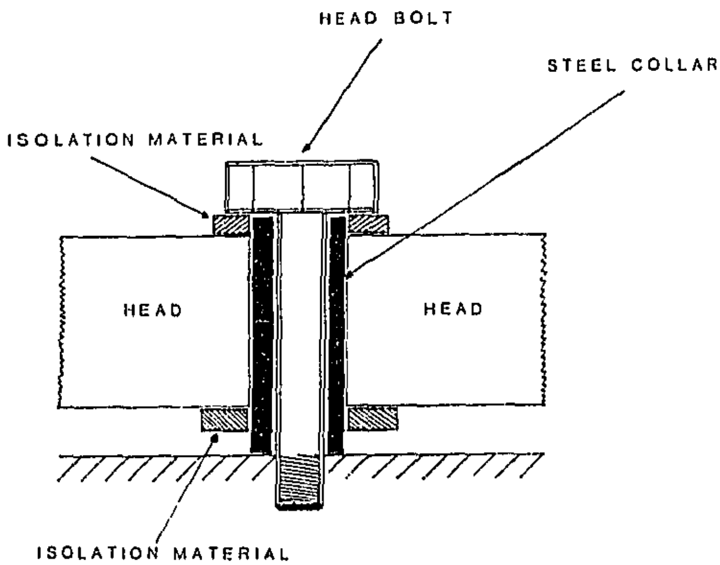

Figure 31. Bolt Isolation Design for NTC-350 Engine

b. the attachment of the liners to the head by soldering rather than by threading, which created substantial machining problems.

c. a small shift in the standard compression ratio by about 2% as a result of using standard retrofitted liners.

A mock-up assembly of the liner-head attachment was constructed to evaluate its dynamic effectiveness in a non-running engine. Transfer path analysis was carried out to four lower block points from piston slap excitation location and the results are shown in Figure 32. It can be seen that in the frequency range from 800 Hz to 5 kHz there is an average of 8.5 dB reduction in the vibrational energy transfer compared to a standard liner configuration. An additional 3 dBA reduction could be achieved by increasing the liner thickness as specified by our original design.

B. Combustion Excitation Source Reduction

Based on the conclusions derived from the modeling effort, it is apparent that treatment of piston slap excitation alone yields on the order of a 4 dBA noise reduction. If this reduction were not sufficient, it would be necessary to treat the contribution of combustion excitation to overall noise. The primary contribution of combustion excitation and its associated transfer paths to the noise spectra lies in the frequency range from 3.15 to 4 kHz. The two major transmission paths are the piston-connecting rod-crankshaft-lower block and the head-upper block-lower block. Since the path through the head can be treated by noise reduction designs for piston slap, the focus is shifted on treating the path through the piston.

To treat the transmission path through the piston, a design was developed involving the application of a set of resilient bearings. The resilient bearings are installed at the main

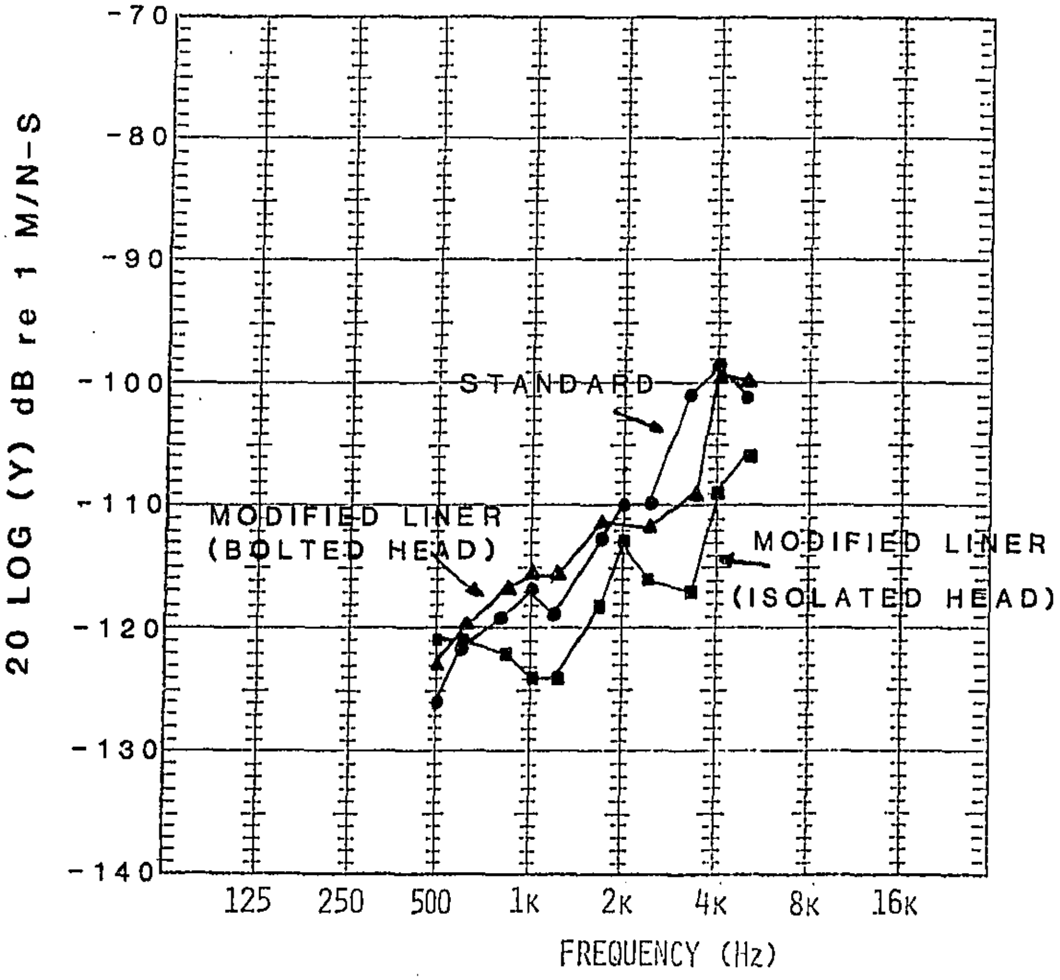

Figure 32. Transfer Mobility of Piston Slap Force to Block Vibration with New Liner Attachments on NTC-350 Engine

bearings of the crankshaft and result in the reflection of vibrational energy back toward the source. The high damping values attributed to the piston motion also create a substantial benefit. The treatment using resilient bearings could potentially yield a reduction of vibrational energy transfer excited by piston slap. This has already been demonstrated on other engines where in particular frequency regions the transmission of piston slap excitation is dominated by the path through the piston rather than by through the liner.

C. Injection Excitation Source Reduction

A final step in reaching a 5 dB or more truck noise reduction involves reducing the excitation generated by the injection process at the 2 kHz bandwidth. Since the main transmission path of this excitation is through the rocker box, a design was created to isolate the rocker arm shaft from the rocker box using a frequency tuned resilient mounting bearing. This alleviates the problem without affecting the injection performance requirements (see Figure 33 for the modification schematic).

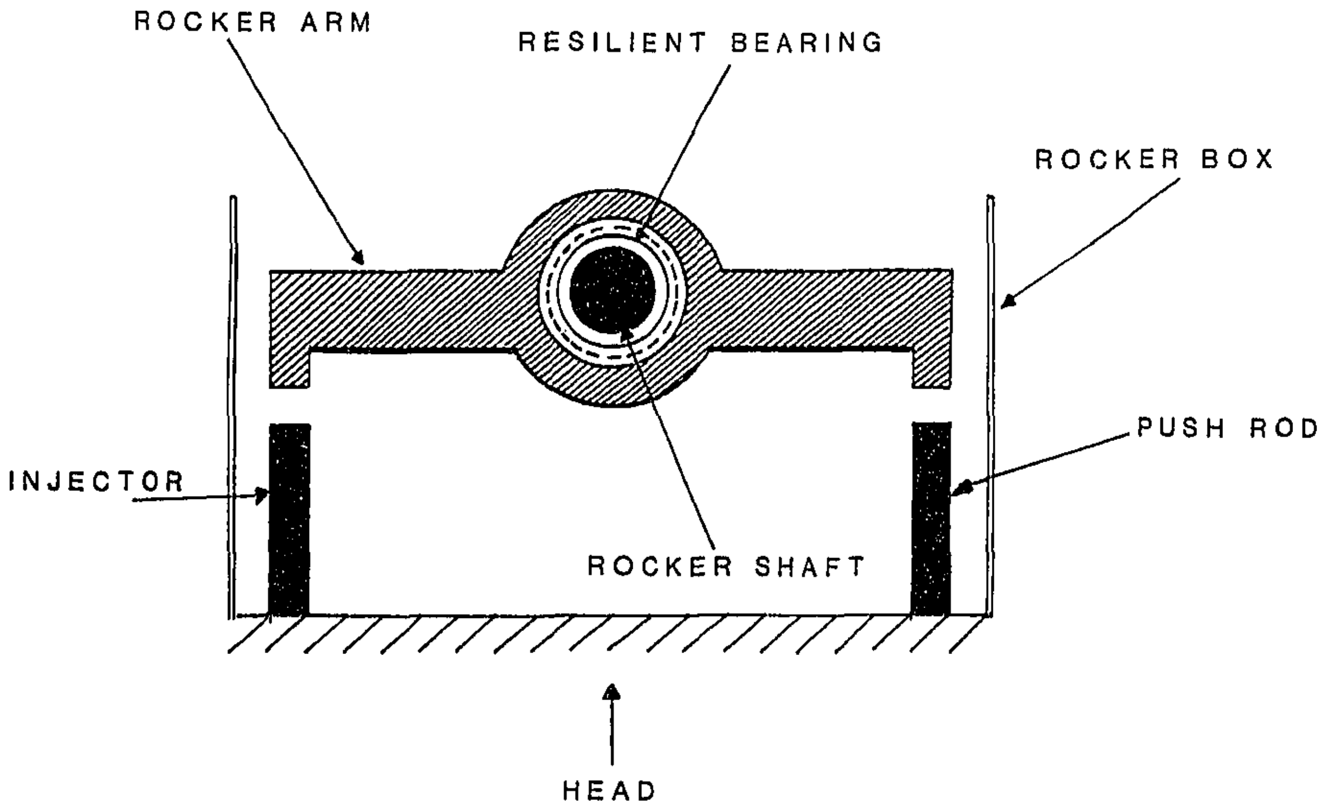

Figure 33. Injector Isolation Design on NTC-350 Engine

V. CONCLUSIONS

This report has described a demonstration of the use of transfer path analysis in the investigation of major noise contributors and their transmission paths within a heavy duty diesel engine, the NTC-350 Big Cam One. The report also describes suggested methods for reducing the contribution from the various sources to overall noise emission which involve alterations to the transmission paths of the sources.

The following briefly summarizes our conclusions from this study:

1. Transfer path analysis is a valuable tool for gaining an understanding of the noise emission mechanisms in diesel engines.

2. In the case of the NTC-350 Big Cam One diesel engine piston slap is the major noise contributor and by adequate treatment applied to this source a 4 dBA overall noise reduction can be achieved.

3. The proposed method for reducing the piston slap noise contribution by attaching the liners to the head and isolating the head from the block appears to be a viable technique having significant noise reduction potential even though it has not yet been tested in a running engine and is considered to be a somewhat unorthodox practice in common engine design.

VII. REFERENCES

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16. ABSTRACT

This report presents the results of an engineering study to design and demonstrate a method of reducing piston-slap noise for an in-line, heavy duty truck diesel engine. The Cummins NTC-350 engine was selected for the demonstration. The engine heads and cylinder liners were structurally modified to reduce the vibration transmission to the engine block and the radiated noise. The modifications were designed to achieve a 5 dBA noise reduction without degrading engine performance, fuel economy, or emissions.

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