MECHANISM OF COMBUSTION AND PISTON SLAP INDUCED NOISE -- ANALYSIS, CONTROL AND DIAGNOSIS

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I. INTRODUCTION

The high frequency vibrations of IC engines are caused by a number of sources -- combustion, piston slap, and valve clatter among others. This vibration is a major source of engine noise, particularly for diesels. It is also a signal that has potential for revealing internal operating characteristics of the engine for purposes of engine control and fault diagnosis.

We have been studying those features of engine vibration that are related to structural excitation by combustion pressures [1,2] and piston/cylinder wall impact [3,4]. Our studies have emphasized these phenomena in small diesel engines. The applications have been to engine noise reduction, and reconstruction of combustion pressure pulses from vibration signals. This reconstruction would be used either for fault diagnosis or engine combustion control.

We have made extensive use of vibration transfer functions -- the ratio of vibratory response at one location to force excitation at another. We have been particularly concerned with evaluating the limitations of this technique -- to what extent one may limit the number of variables, and to what extent one can treat the engine as a linear time invariant structure.

Vibrations due to piston slap (and other mechanical impacts) are determined by the dynamics of the impact, and the vibration transmission path. In our studies of piston slap, and the alleviation of this noise component, we have made a detailed study of the collision dynamics, and their effect on the noise spectrum.

II. CYLINDER PRESSURE AS A VIBRATION SOURCE

A major source of vibration in diesel engines is the rapid pressure rise during the early stages of combustion in the cylinder. Examples of such pressure traces, measured in a John Deere 4219 diesel are shown in Fig. 1 [1].

There is a compression pressure of several atmospheres amplitude which is relatively unaffected by the amount of combustion, on which a smaller amplitude combustion pulse "rides".

The spectral implications of this behavior are shown in Fig. 2, which compares the no-load spectrum (compression-expansion cycle only) with the spectrum for 60% and 100% load (curves B and C in Fig. 1). The combustion pulse appears to have most of its energy in the frequency range above 500 Hz, dropping off at about 12 dB/octave. Obviously, the combustion pulse has the potential for greatly increasing engine vibration levels in the most audible bands. Alternatively, if we wish to use vibration data to infer combustion performance, our processing should concentrate on the frequency range from 500 Hz to a few kilohertz [1].

Cylinder pressure also affects piston slap excitation in the engine. A simplified diagram of the forces on a piston during the compression-firing-power stroke part of the cycle is shown in Fig. 3. In this case, the cyclic pressure plays an important role in driving the piston into the cylinder wall, with the impact coming earlier in the cycle and with more momentum as the combustion component of the pressure is increased [3,4].

The cylinder pressure governs primarily the magnitude of the force pulse produced by piston slap, but the shape of the spectrum is primarily due to the dynamics of the piston-cylinder wall collision. We have been studying this problem, and our conclusions are as follows [4]:

(a) The piston motion is a combination of sliding and rocking

(b) The bottom skirt of the piston makes first contact with the wall, followed by rotation of the piston about the gudgeon pin and crown impact.

(c) The principal high frequency energy produced by the impact occurs when the crown impacts the cylinder wall.

III. VIBRATION TRANSFER FUNCTIONS

The excitation of the engine casing by the combustion pulse can occur through two "paths" -- upward through the head, and downward through the piston connecting-rod crankshaft assembly [2]. The transfer function relating casing vibration to force -- either on the top of the piston or the combustion chamber surface in the head has been measured for a non-running engine and are shown in Figs. 4 and 5. Comparing Figs. 4 and 5, we note that this P/CR/CK path dominates below 2.5 kHz. If we add the magnitudes of Figs. 4 and 5 (assume the signals combine by energy addition), we get the total transfer function for a stationary engine (SVTF) shown in Fig. 6.

Since we are really interested in the vibration produced during engine operation, we have also measured a dynamic transfer function (DVTF) between casing vibration and cylinder pressure shown in Fig. 7 [1]. There is not much difference in the magnitude of these two transfer functions, so that we can use the SVTF for problems related to noise control. The phase functions shown in Fig. 8 are quite different, however, and that affects the utility of the SVTF in reconstruction of the cylinder pressure waveform, as we shall see.

It is difficult to directly measure the force between the piston and cylinder wall that results from piston slap since that force is distributed over an area and is fairly nonuniform. We have measured the spectra of vibration velocity due to piston slap at various points on the block surface and the transfer mobilities from the upper cylinder wall to the same points. Typical data are shown in Figs. 9 and 10 for a four-cylinder Isuzu engine. A comparison of these figures suggest that either the force spectrum has a strong peak at about 1.5 kHz or there is another transfer path. Measurements on the piston of a motored engine show a rocking motion at about 1.5 kHz in response to the impact. Experiments are currently being performed which will indicate whether or not the connecting rod and the crankshaft comprise an important transfer path for piston slap noise. A comparison of expected vibration spectra for piston slap and combustion pulse for this engine is shown in Figs. 11.

IV. APPLICATIONS TO NOISE REDUCTION

Since the vibration due to cylinder pressure in the audible frequency range results from the combustion pulse, reduction of this noise at the source requires modifying the combustion. We have avoided this by concentrating on the vibration transmission. We have constructed a transfer function by combining component transfer functions. For example, the P/CR/CK/BLK chain can be divided into a piston/connecting rod component, and crank and block (or casing) components.

A comparison between the measured SVTF and one constructed from component VTF's is shown in Fig. 12. The agreement is good, up to about 3 kHz. We believe that this is due to the neglect of moments and angular velocities at component junctions -- moments should be more important in transmitting vibration at higher frequencies. An analysis of the admittance characteristics of the various junctions indicated that the best place to put some resilience in the system to reduce the transfer function was between the crank bearing and the bearing support structure. This was done for the non-running engine, and a comparison of predicted and measured reduction in the SVTF is shown in Fig. 13. The 10-15 dB reduction in transmitted vibration should result in a similar reduction in radiated noise *due to this path,* but, of course, would not do so with other sources present, the total noise reduction may be less.

The reduction of vibration through this latter path should be achievable by the same resilient bearing support as described for the combustion induced vibration. Reduction of the high frequency components may be most effectively achieved by modification of the cylinder wall. We are presently studying this possibility.

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V. APPLICATIONS TO CONTROL AND DIAGNOSTICS

We have been studying the reconstruction of the cylinder pressure trace, shown in Fig. 1, by using the inverse of the vibration transfer function. We saw from Figs. 6, 7, 8 that the magnitudes of the VTF's for stationary and running engines were rather similar, but there were significant differences in their phases. Our study has shown that reconstruction of the combustion pulse is very sensitive to use of the correct phase, but rather insensitive to getting the amplitude spectrum just right.

A typical vibration pulse on the engine casing arising from a combustion pulse is shown in Fig. 14. When this pulse waveform is used to reconstruct the combustion pulse using both correct and incorrect (wrong cylinder) DVTF's, we get the results shown in Fig. 15. Clearly, this procedure is able to perform fairly well in reconstructing the pressure pulse, *and* in descriminating against the wrong signal for reconstruction.

We are exploring the use of this procedure for diagnostics (fault determination) and control (parameter estimation). We believe that modern data processing can make it possible to use highly modified data for sensing of engine behavior, procedures that have heretofore been conceivable, but impractical.

REFERENCES

1. A. Ordubadi, "Component and Fault Identification in a Machine Structure Using an Acoustical Signal", PhD Thesis, M.I.T., Dept. of Mech. Eng., May, 1980.

2. R. DeJong, "Vibrational Energy Transfer in a Diesel Engine", ScD Thesis, M.I.T., Dept. of Mech. Eng., August, 1976.

3. T. Oguchi, "Piston Slap Noise -- Its Trasnfer Through an Internal Combustion Engine, MSc Thesis, M.I.T., Dept. of Mech., Eng., Jan., 1979.

4. J. Slack, Thesis in Preparation.

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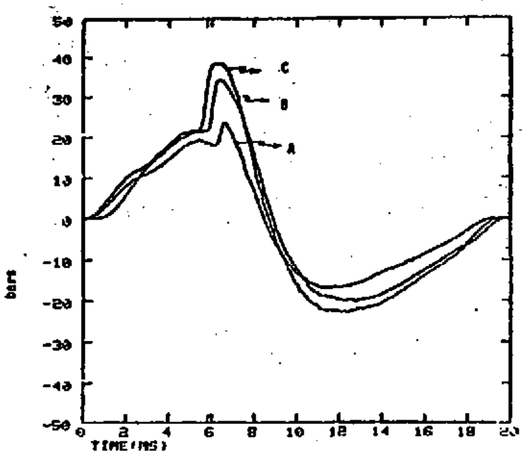

Fig. 1: Cylinder Pressure Measured at 1500 RPM (a) No Load, (b) 60% Full Load (c) Full Load

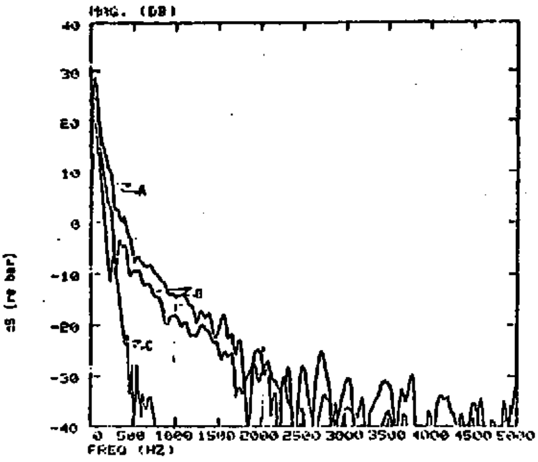

Fig. 2: Spectrum of Cylinder Pressure at 1500 RPM and (a) Full Load, (b) No Load, (c) injector shut off

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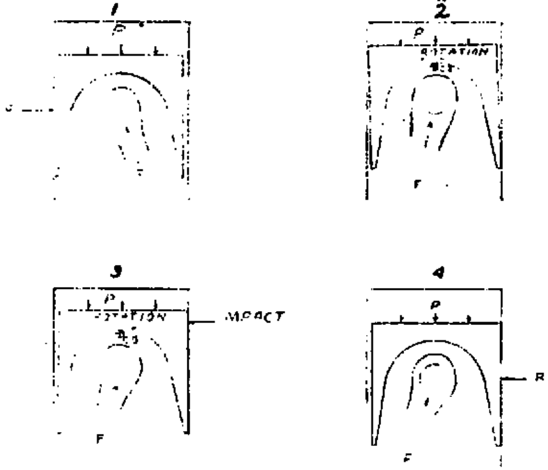

Fig. 3: Motion of Piston During Piston Slap

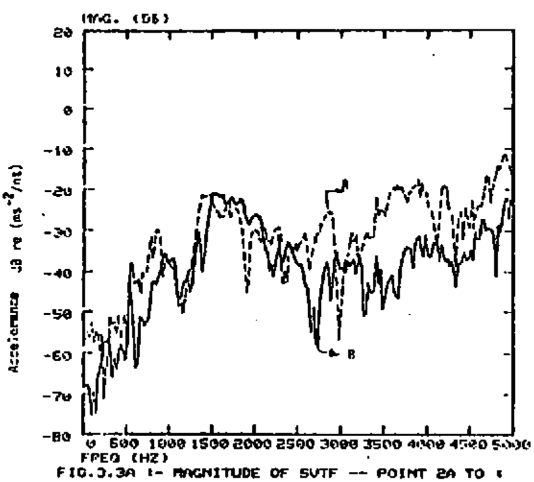

Fig. 4: Magnitude of SVTF -- Point 2A to: (a) Piston #1, (b) Piston #4

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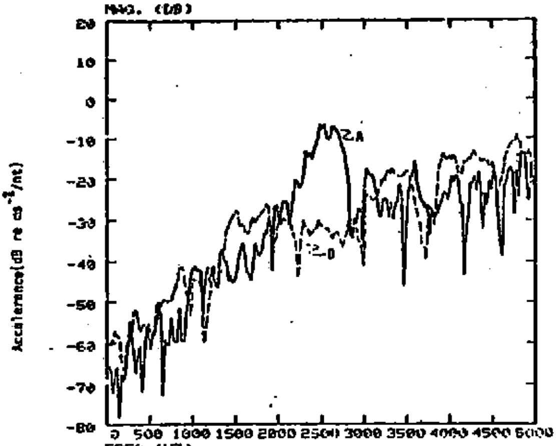

Fig. 5: Magnitude of SVTF through Head -- Point 2A to (a) Cylinder #1 (b) Cylinder #4

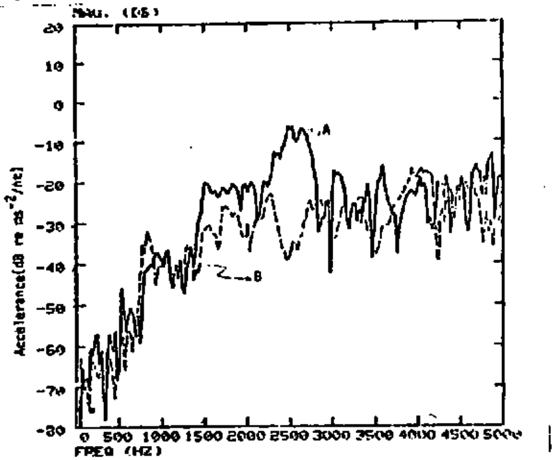

Fig. 6: Magnitude of Total SVTF -- Point 2A to (a) Cylinder #1 (b) Cylinder #4

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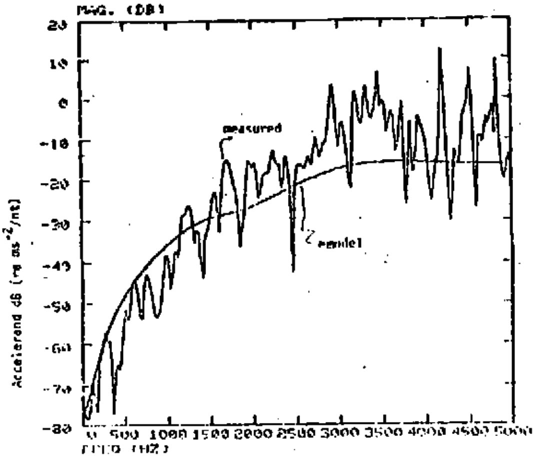

Fig. 7: Magnitude of Measured and Model Vibration Transfer Function of the Operating Engine -- 1500 RPM, 60% Load

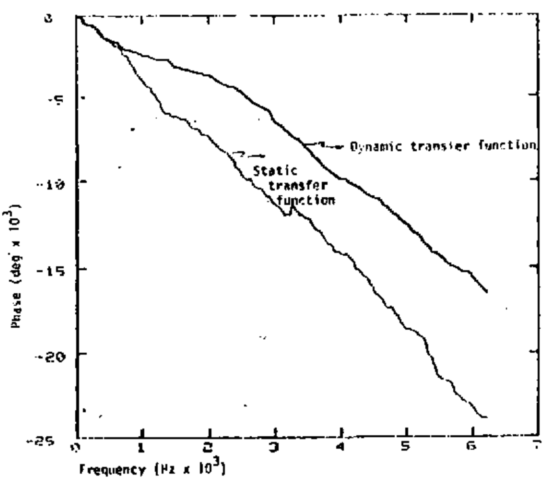

Fig. 8: Phase of Static and Dynamic Transfer Functions Measured Between Cylinder #1 and a Point on the Engine Block. Point is Located on Bearing Cap of Cylinder #4

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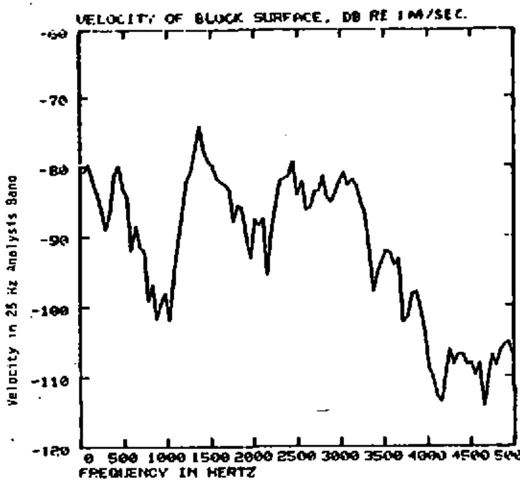

Fig. 9: Vibration Spectrum of Block Surface

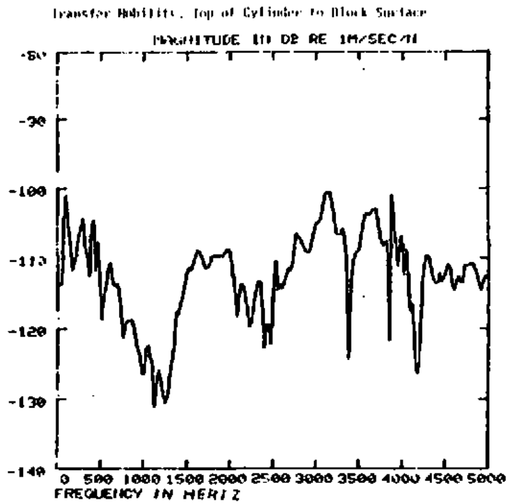

Fig. 10: Transfer Mobility: Cylinder Wall to Casing

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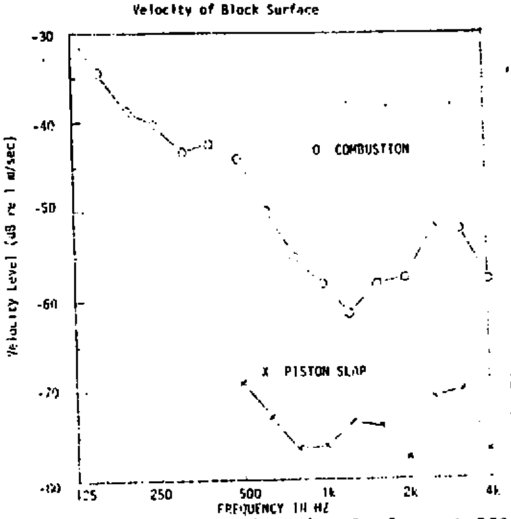

Fig. 11: Predicted Velocity Spectrum of Engine Surface at 3600 RPM Full Load

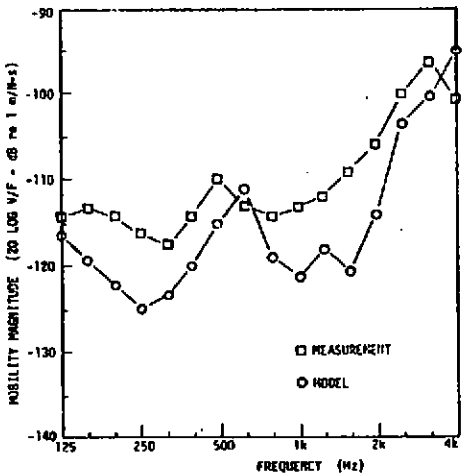

Fig. 12: Total Average Transfer Mobility from the Combustion Forces to the Engine Block Surface

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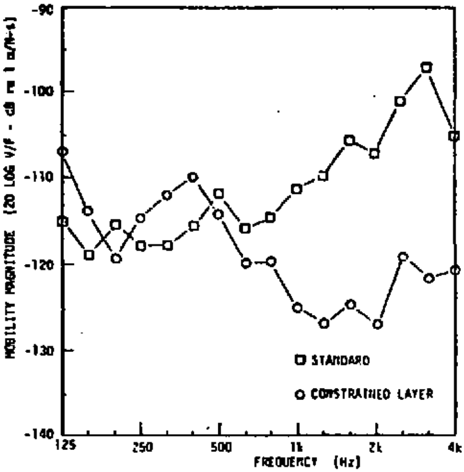

Fig. 13: Transfer Mobility Magnitude T2-S1 with Constrained Layer Bearing Rings

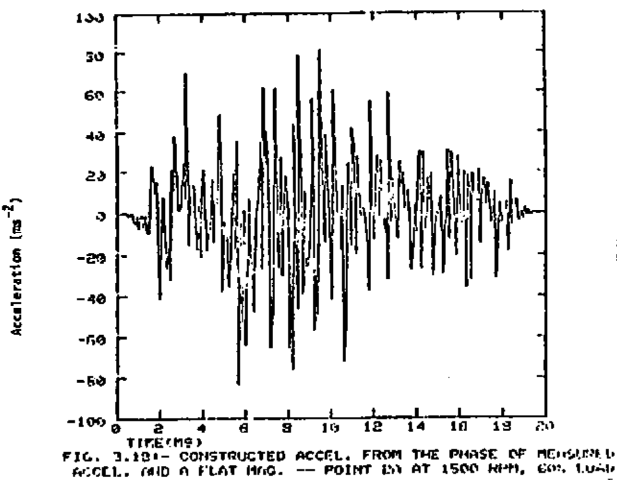

Fig. 14: Constructed Acceleration from the Phase of Measured Accel. and a Flat Magnitude -- Point 2A at 1500 RPM, 60% Load

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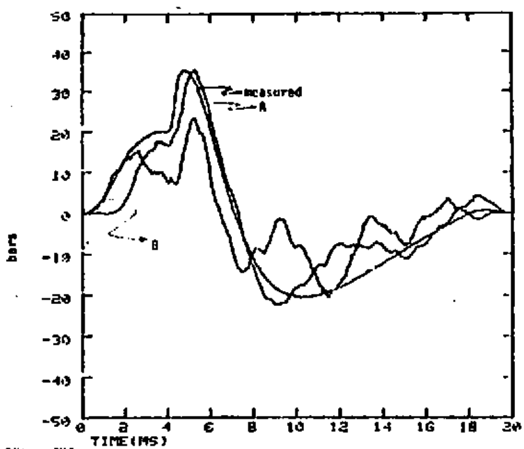

Fig. 15: Estimated Pressure Using DVTF1 at 1500 RPM and 60% Load in --
(a) Cylinder #1 and (b) Cylinder #4