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Optimum knock sensor location through experimental modal analysis of engine cylinder block

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ABSTRACT

The knock sensor is provided on an engine cylinder block to detect abnormal engine combustion (knocking) and to provide feedback to engine control unit (ECU). The ECU then modifies the engine input and avoids knocking. A commonly used knock sensor is an accelerometer that detects cylinder wall vibration and estimates knocking of the engine. Selecting the location of a knock sensor in many cases involves a challenging trial and error approach that depends upon the measurement of the knock signal at many locations on engine structure. However, a cylinder block exhibits many structural resonances. Thus, a large vibration signal at the surface of cylinder block can be either due to knocking of the engine or due to the resonances of the cylinder block structure because of normal excitation forces. Hence, this conventional method does not always yield reliable results. The aim of the work reported in this paper is to experimentally determine the inherent dynamic characteristics of a cylinder block and to combine this with a calculation of the fundamental knock frequency and, thus, to identify the optimum location for the knock sensor.

INTRODUCTION

Engine Control Units (ECUs) have been incorporated in almost all modern vehicles. Knock sensing is regarded as one of the major inputs to the ECU for combustion control. Knock control allows the engine to be calibrated close to its optimum operating point under all usage conditions. Hence, the knock sensing device within the engine system becomes critical for successful combustion control. A commonly used Knock sensing device is an accelerometer that detects cylinder wall vibration and estimates knocking of the engine. When a knock event takes place, huge pressure fluctuations within the combustion chamber cause the cylinder block to vibrate. A knock sensor detects the vibration and accordingly provides feedback to the ECU. The ECU then modifies the engine input for example by retarding the spark advance in the case of a gasoline engine or controlling the pilot injection in the case of a diesel engine and, thus, avoids combustion knock. The location of a knock sensor on the engine, hence, becomes important to correctly assess the engine knocking condition.

Selecting the location of a knock sensor in many cases involves a challenging trial and error approach that depends on the measurement of the knock signal under engine running condition at many locations on the engine structures. The location of a strong signal is generally taken as the location for the knock sensor placement. However, under normal operating conditions a cylinder block is subjected to large excitation forces over a wide frequency range and, hence, exhibits many structural resonances. Thus, a large vibration signal at the surface of the cylinder block may be due to either knocking of the engine or due to resonances of the cylinder block structure because of normal excitation forces i. e. normal combustion. Hence, this conventional method does not always yield reliable results.

The specific objectives of the work reported in this paper are: (i) to experimentally understand the inherent dynamic characteristics of a cylinder block; (ii) calculate the resonant frequency of the acoustic cavity; and (iii) then determine the optimum location for the knock sensor. A brief review of current understanding of the structural vibrations of an engine block and the knock frequency characteristics of an engine is given. An experimental modal analysis of a cylinder block of the test engine is described. Hence, a reference surface on the cylinder block is identified for a possible sensor location. Frequency Response Function (FRF, accelerance) measurement is carried out initially using electromagnetic exciter and later by impact hammer at 96 different locations on the reference surface. Modal frequencies and mode shapes are determined by curve fitting on a set of measured FRFs using proprietary software. Resonant peaks are found to occur across all of the frequency range, which is due to the very complex and heavily damped structure of cylinder block. The 1st and 2nd modes of vibrations are found to be torsional and bending modes. The mode shapes at higher frequencies exhibit complex deflection patterns.

The frequency of engine knocking is known to be related to the acoustic cavity resonance of the combustion chamber. Hence, it is a function of the bore diameter and the local speed of sound. From empirical relations, the fundamental knock frequency for the test engine is determined. Mode shapes at resonant frequencies close to the fundamental knock frequency are studied. Then an optimum location for knock sensor on the cylinder block reference surface is determined.

CYLINDER BLOCK STRUCTURAL DYNAMICS

A comprehensive study on engine structural vibration was presented by N Lalor in reference [1]. The way in which the engine structure vibrates under running conditions was examined and differences were related to the overall design. It was shown that the first mode of vibration occurs at a few hundred hertz and consists of a torsional motion about an axis parallel to the crankshaft. At higher frequencies, almost up to 1000 Hz, the engine starts to bend along its length like a homogeneous beam. Above this frequency the structure ceases to behave as a solid body and the panels forming the sides of the bays begin to vibrate independently.

This phenomenon was illustrated with the results of a finite element analysis of a 6-cylinder in-line engine crankcase. Although the structural response is very complex it was found that it falls into two distinct groups. Firstly, whole body modes which typically occur between 0 Hz to around 1000 Hz to 1500 Hz. In this frequency range the engine behaves as if it were an elastic plate. Thus, each point on one side of the block will be in phase with its corresponding point on other side. Secondly, panel modes which occur at higher frequencies. In this frequency range each panel of the block structure vibrates independently. The specific natural frequencies of these panels will depend on their size and shape and, therefore, each engine will have a range of frequencies where its different panels vibrate in their fundamental mode.

The influence of engine block design features on resulting noise and vibration was investigated by, amongst others, Busch, Maurell, Meyer and Vorwerk [2]. Experimental modal analysis was undertaken to identify the mode shapes of the engine block. Experimental modal analysis was carried out on 4-cylinder in-line diesel engine by Honda et al. [3], who found 8 modes of vibration in the frequency range from 0 to 2000 Hz.

PREVIOUS WORK ON KNOCK SENSOR LOCATION

A review of knock sensor selection criteria is given by Dues et al. in reference [4]. Knock phenomena, types of sensor and application issues were discussed. An experimental study on knock sensing for a spark ignition engine was conducted by Jang et al. [5]. The study was carried out in order to understand the knocking of a spark ignition engine and to determine the design parameters for a knock sensor. An analysis of cylinder pressure fluctuations and engine block vibrations was undertaken during the knocking condition. As the knock intensity increased, the more the knock frequency was observed in the lower frequency range of 6 to 8 KHz. Therefore, it was suggested that the knock sensor, which measures engine block vibration, should be designed for detecting knock frequency in this fundamental knock frequency range.

Identification of the optimum knock sensor location was attempted by Forbes et al. [6] using laser holography to identify compliant regions of the engine block and then combining this with empirical engine data. The determination of the knock sensor location on a heavy-duty natural gas engine was reported by Soylu and Van Gerpen in reference [7]. Six piezo-electric knock sensors were located on suitable regions of the block and a comparison of the response signals under knocking and non-knocking conditions used to identify the optimum sensor location. Knock sensor location for a single-cylinder test engine was undertaken by Topfer et al. [8] using eight optical sensors located around the spark plug. Bengisu [9] developed a finite element model of the engine block to predict normal surface vibration levels and, thus, the optimum knock sensor location. Shi [10] applied the finite element method to determine knock sensor location and considered specifically the uncertainty of accurate prediction due to the presence of background noise.

EXPERIMENTAL MODAL ANALYSIS OF THE CYLINDER BLOCK

In this paper, experimental modal analysis [11,12] of cylinder block is carried out to determine its vibration characteristics. A 4-cylinder in-line diesel engine block (Bore/Stroke: 110/132 mm) is used for the analysis.

Test set-up

The cylinder block is suspended from a rigid frame to simulate freely mounted conditions so that the modal information of test object alone could be obtained. The cylinder block is excited initially using electromagnetic-exciter from underneath cylinder block and later by impact hammer.

FRF measurements using an electromagnetic exciter

The front surface of cylinder block (intake side) is considered as reference surface for measurement locations since this surface alone had scope for the knock sensor mounting position. The number of measurement points is restricted due to the fact that the response accelerometer needs a flat and smooth mounting surface. A total of 27 points are chosen for measurement, nine points in each of three rows at different heights on the cylinder block.

An electromagnetic-exciter is used to apply random waveform excitation to the cylinder block over a wide range of frequencies. Measurements are carried out for a frequency range from 0 to 10,000 Hz. At each response point, measurements are taken with 100 averages. The FRF data is then analysed by a commercial software package. The modal parameters of the structure are obtained using the well-known curve fitting method.

A total of 8 modes are identified in the selected frequency range of 0 to 2000 Hz. Analysis of 1st & 2nd mode shapes at 595 & 1420 Hz showed combination of both bending and twisting motions and further higher frequency modes showed highly complex deflection pattern. Hence it is decided that measurements along only 3 different heights are not sufficient and measurement locations must be increased.

FRF measurements using an impact hammer

Impact hammer testing allows many locations on the irregular surface of cylinder block to be excited easily. Hence, it allows an increased number of FRF measurements to be made. The reference surface of the cylinder block for impact hammer testing is shown in Figure 1. The surface is divided into a grid of twelve vertical columns and eight horizontal rows, giving a total of 96 FRF measurement points. This number of points being expected to provide a sufficiently good spatial resolution for modes up to 5500 Hz.

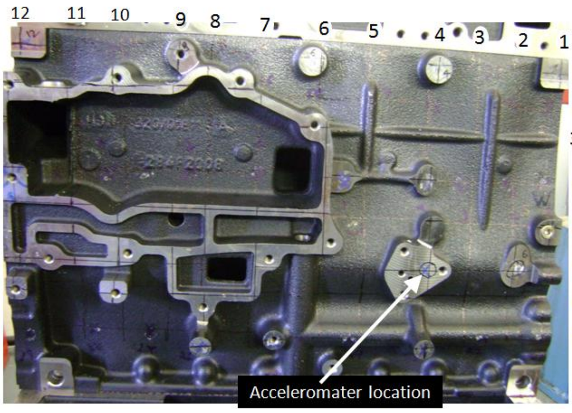

Figure 1: Impact hammer excitation locations and the response accelerometer location on cylinder block surface

One of the locations, point 64 (as shown in figure 2), is used for the accelerometer mounting while the remaining points are used as an excitation location with the impact hammer.

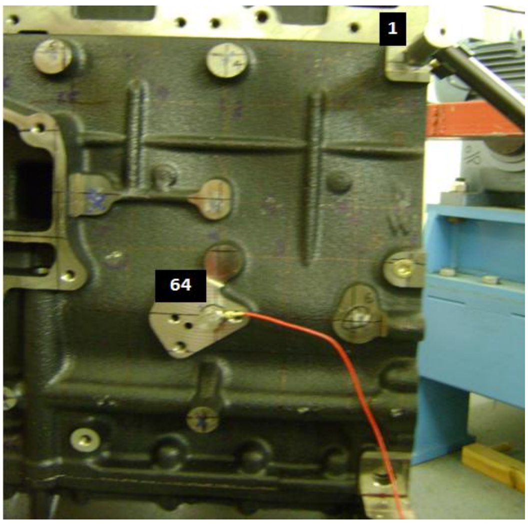

Figure 2: Photograph of impact hammer testing of the cylinder block

FRF measurements

At each of the points, structure is excited with impact hammer. For each FRF measurement, an impact test is carried out 10 times and an average measurement, thus, formed. After observing the coherence function for all 96 measurement locations together with the relevant power spectral density of force excitation a frequency range of 500 to 5500 Hz is chosen for modal analysis.

Modal parameters

The modal parameters of the structure are determined using a proprietary software package. Modal peak function of software indicates many modes on FRF data corresponding to different peaks within the frequency range of interest. Eight modes occur between 500 and 2010 Hz. The modal frequencies are similar to those of the modes identified previously using the electromagnetic-exciter testing. The advantage of impact hammer testing compared to using an electromagnetic exciter is the shorter time required to produce a full set of FRF measurements. A total of 49 modes of vibration lie within the selected frequency range of 500 to 5500 Hz. The modal frequencies and modal damping are listed in Table 1. High structural damping is indicated for a number of modes.

Mode	Frequency (Hz)	Damping (Hz)	Damping (%)
1	577	8.55	1.480
2	1030	21.1	2.050
3	1410	10.7	0.760
4	1540	18.3	1.190
5	1640	28.2	1.720
6	1700	26.7	1.570
7	1920	23.1	1.200
8	2010	28.1	1.400
9	2050	16.9	0.823
10	2170	27.4	1.260
11	2390	18.6	0.778
12	2440	35.4	1.450
13	2530	11.8	0.465
14	2630	13.6	0.516
15	2710	15.5	0.572
16	2830	24.9	0.881
17	2920	26	0.892
18	3030	39.5	1.300
19	3070	12.9	0.422
20	3120	7.28	0.233
21	3160	15.9	0.503
22	3210	17.7	0.552
23	3310	17.5	0.531
24	3440	29.8	0.868
25	3520	32.8	0.933
26	3580	30.9	0.863
27	3670	41.8	1.140
28	3760	19	0.504
29	3830	21.1	0.552
30	3900	16.7	0.428
31	3970	22.8	0.575
32	4050	20.4	0.503
33	4140	14.8	0.358
34	4190	33.6	0.801
35	4260	30.6	0.720
36	4320	22.5	0.521
37	4390	25.3	0.576
38	4450	18.2	0.408
39	4530	14.2	0.313
40	4550	13.3	0.293
41	4610	18.6	0.403
42	4690	23	0.491
43	4710	13.2	0.279
44	4770	8.35	0.175
45	4850	5.09	0.105
46	5010	24.3	0.485
47	5090	18.3	0.360
48	5360	16.7	0.311
49	5550	17.1	0.142

Table 1: Modal data for the cylinder block – impact hammer testing

The 1st mode of vibration occurs at 577 Hz. The corresponding mode shape is shown in Figure 3. A nodal line can be identified running horizontally across the structure and its location is shown approximately in Figure 3. From the deflection pattern the structure can be seen to be twisting in the vertical plane. Hence, the 1st mode of vibration is a torsional mode.

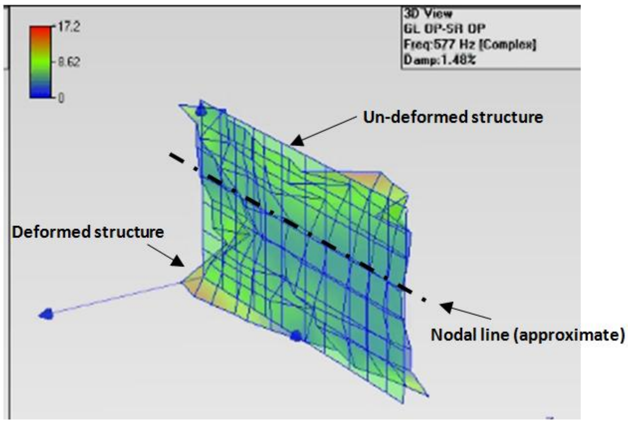

Figure 3: First mode of vibration at 577 Hz (torsional mode)

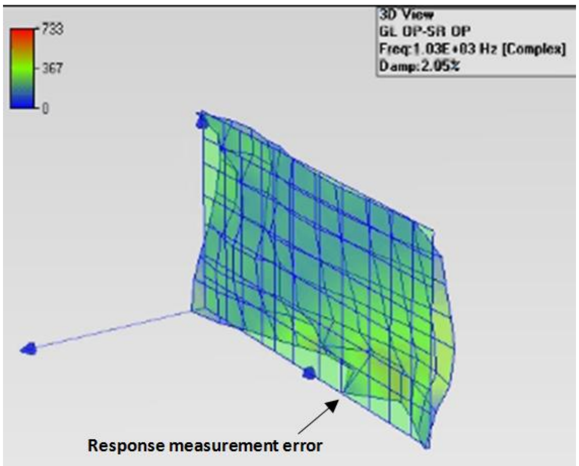

Figure 4: 2nd mode of vibration at 1030 Hz (bending mode)

The second mode of vibration occurs at 1030 Hz. Figure 4 shows the corresponding mode shape. One of the points in the bottom row does not appear to show any deflection. This is possibly due to an error in the response measurement at this location. The deflection pattern shown in Figure 4 indicates that the structure is bending in both the vertical and horizontal directions with approximately 1/2 wavelength bending in both directions. The deflection of upper half of the structure is relatively low compared to the lower half of the structure. Figure 5 shows the deflection pattern of the 3rd mode shape occurring at 1410 Hz. This mode shape is similar in character to the 2nd mode shape at 1030 Hz. Hence, the 2nd and 3rd modes of vibration can be considered to be bending modes of vibration.

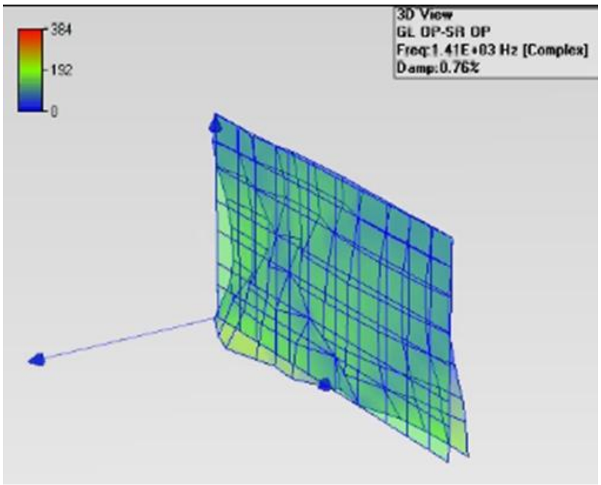

Figure 5: Third mode of vibration at 1410 Hz

The mode shapes occurring at high frequencies exhibit very complex deflection pattern. Abnormal deflection of some points (highly different from neighboring points) at some particular frequencies indicates noise in the measurements (poor coherence) at that instant.

The complex pattern of mode shapes at higher frequencies can be attributed to the highly rigid and complex structure of the cylinder block. For example, at high frequencies each panel of the block structure has a propensity to vibrate independently [1].

IDENTIFICATION OF THE OPTIMUM KNOCK SENSOR LOCATION

Knock frequency calculation

A knock sensor is typically a unidirectional accelerometer mounted onto the surface of the engine structure. It is traditionally a narrow band device tuned to the lowest modal frequency of the chamber. The typical frequency range is from 5 kHz to 15 kHz.

The fundamental knock frequency is a function of the distance it travels across the bore diameter and the local speed of sound as given by the average bulk temperature of the combustion chamber. This relationship can be expressed by following the equation [4]:

$$fn = \frac{kc*c}{B} = \frac{kt}{B}\sqrt{T} \qquad (1)$$

Where fn is the fundamental knock frequency, kc and kt are constants, c is the local speed of sound, B is the cylinder bore diameter and T is the bulk combustion chamber temperature. Equation (1) can be multiplied by a mode shape factor to identify the frequencies of the higher vibrational modes. For disc-shaped chambers the multiplication factors for mode shapes two, three and four have been documented as approximately 1.65, 2.07, and 2.26, respectively [4].

Because of its temperature dependence, the actual knock frequency is subject to the same statistical variation as the combustion process. Deviations from the mean fundamental knock frequency as high as +/- 400 hertz have been observed for a given engine model. The engine under test has a bore diameter of 110 mm. Hence, from equation (1) the fundamental knock frequency for the engine is assumed to occur between 5000 to 5500 Hz.

Optimum knock sensor location

Assuming that the knock sensor is tuned for the lowest modal frequency of the chamber then the optimum location for the knock sensor can be estimated by considering the mode shape patterns for modal frequencies between 5000 to 5500 Hz. A region on the cylinder block where high levels of deflection occur can be considered as the region most sensitive to the excitation force at these frequencies and, hence, an optimum location for knock sensor mounting.

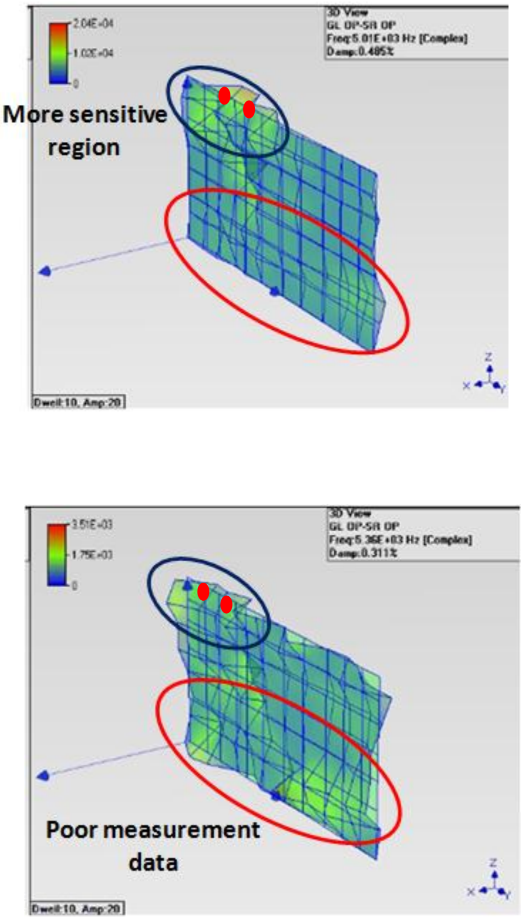

Figure 6: Mode shapes of the cylinder block at 5010 and 5360 Hz

Mode shapes of the cylinder block reference surface are shown in Figure 6 for modal frequencies at 5010 Hz and 5360 Hz. The mode shapes appear very complex, however, regions of maximum deflection can be identified from the deflection patterns. The measurements made at the bottom rows, shown in figure 6, are not necessarily accurate as the coherence function at these points for this particular frequency range was poor. Furthermore, this region of the cylinder block is not normally considered for the placement of a knock sensor. Knocking takes place approximately at Top Dead Center (TDC) because of the higher pressures and temperatures in the combustion chamber. It is known that the knocking signal amplitude progressively diminishes as a function of the distance from the source. Hence, knock sensor location within the upper portion of the cylinder block, close to TDC, is desirable.

Considering the upper portion of the cylinder block surface, it can be seen that the region at the top left-hand side, shown by two solid dots in Figure 6, is a very sensitive region for the two given modes. A similar observation can be made for the other two modes of vibration occurring at 5090 Hz and 5550 Hz, respectively. Hence, this region can be identified as an optimum location for the knock sensor. This region on cylinder block surface is indicated with two solid dots and a circle in Figure 7. The optimum location for the knock sensor determined by Bengisu in reference [9] using computational techniques was also in the region of the top of the cylinder block, close to the cylinder head.

Of course, an advantage of experimental modal analysis is the shorter time and hence lower cost of analysis, compared to developing a full FE model of the structure. The method provides baseline data for accurate FE analysis. The outcome of this method will also help in faster diagnosis and analysis of engine knock problems. Another improvement point of this method is improved accuracy of knock sensing i.e. less noise in recorded knock data as the location of sensors are near sensitive regions.

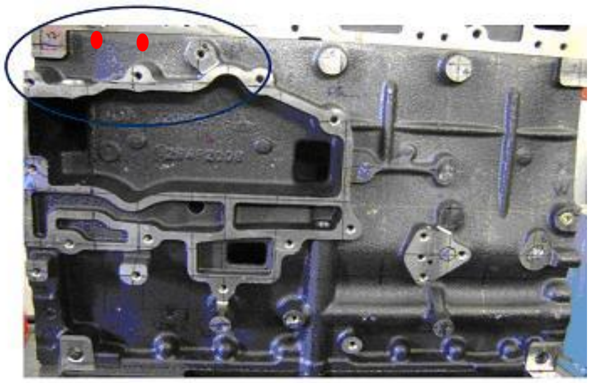

Figure 7: Optimum location for knock sensor mounting on cylinder block

For this particular engine, the knock sensor location design change was not applied. Thus, for future work it would be useful to implement the method on other types of engine and, hence, compare recorded knock sensor output from both optimum and non-optimum locations.

CONCLUSION

An experimental modal analysis of a cylinder block was undertaken and the dynamic characteristics of the structure were derived in the form of modal frequencies, modal damping and corresponding mode shapes. The fundamental knock frequency for the engine under test was established using empirical relations. The modal information and knock frequency calculation were then combined to identify the optimum location for the knock sensor. A systematic design methodology is derived to find out optimum location for the knock sensor placement to correctly assess the engine knocking condition and same methodology can be used in future to design robust knock control systems.

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