Vibration monitoring as a predictive maintenance tool for reciprocating engines

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Abstract: The vibration signature of a four-stroke, four-cylinder carburetted spark ignition engine has been analysed. The significance and contribution of the signature components with respect to the overall information about the engine health have been explored. The engine block vibrations were recorded at four different locations, two in the vicinity of the rear crankshaft bearing and two at opposing sides of the engine block. The vibrations were measured along the three principal axes. It was found that the engine block side is the most sensitive location for collecting data and the direction transverse to the pistons' movement plane is the most informative one. For the sake of simplicity in possible practical applications in the future the tests were conducted under idle conditions. The measured vibration waveform has been synchronized with the crankshaft position by using the primary coil signal and then transformed to the frequency domain by a fast Fourier transform procedure. Frequencies below 1 Hz and above 10 kHz were filtered out.

A reliable data persistence was obtained by averaging eight spectra. Spectral analyses of several common faults, such as disconnected spark plug, early spark timing, late spark timing, worn spark plug, fouled spark plug and loose support, have been carried out. Reflections of these faults have been revealed in the vibration signature. General evaluation criteria for the engine's health have been proposed. For example, the appearance of vibration components with unusual frequencies can provide an early warning that the engine is operating at abnormal conditions. The appearance of peaks at one-half or three-halves of the fundamental frequency indicates that a severe malfunction has developed in the engine. The appearance of a peak at the fundamental frequency implies that a mechanical looseness enables the engine to vibrate at the crankshaft frequency. In general terms, dispersion of the energy at subharmonic frequencies, or at six to eight times the fundamental frequency, indicates a developing malfunction.

Keywords: vibrations monitoring of IC engines, predictive maintenance of IC engines, IC engine fault detection

1 INTRODUCTION

During the operational life of mechanical systems maintenance costs constitute a major portion of their total expense. Reciprocating engines are no exception: since the maintenance costs usually increase with the complexity of the equipment, it is expected that the maintenance of a reciprocating engine will be far more expensive than that of common industrial machinery. Accordingly, an adequate maintenance programme can reduce the maintenance costs and, consequently, the overall costs over the life cycle of the engine. In recent years, predictive maintenance programmes have been implemented in various plants worldwide replacing older maintenance philosophies and realizing the task of reducing maintenance costs. The purpose for applying a predictive maintenance programme is four-fold: improving productivity, product quality, profitability and overall effectiveness of the system **[1]**. In contrast to the run-to-failure and preventive maintenance philosophies, which are the more common maintenance programmes for reciprocating engines, predictive maintenance is a condition-driven programme. Thus, according to this method, the mechanical condition of the equipment is monitored regularly and a decision on a repair is made on grounds of the data accumulated before a serious problem occurs.

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To achieve its goals, a complete predictive maintenance programme involves several different techniques. One important method is based on lubricating oil analysis, where the condition of the lubricating film is examined. To enable a conclusive analysis of the engine condition, this method requires a complicated set of tests. In addition, the results are rather limited owing to their substantial sensitivity to the specific oil sample and to the environmental conditions (temperature for example). In spite of the limitations of this technique, in an effort to shift the maintenance routine for reciprocating engines to predictive maintenance preliminary attempts to employ on-line analysis with a limited amount of data were recently reported by DeGaspari **[2]** and deBotton *et al.* [3]. Notwithstanding the ability to apply the oil analysis technique, for machines and other equipment with moving parts the vibration monitoring method is undoubtedly the commonest and most reliable technique (see, for example, Mitchell **[4]**, Eshleman **[5]** and Davies **[6]**.

The concept behind the vibration analysis method is that any machine with moving parts vibrates in response to the excitations employed on its components. Variations in the excitation forces, the machine's components or its integrity will affect the vibration pattern. Accordingly, periodic monitoring of the machine's vibration signature provides useful information regarding its components and the excitations acting on them. With a sufficient database about the machine's operating order, its components and history of malfunctions, the vibration monitoring technique can provide early information about progressing malfunctions, the sources of these malfunctions and in some cases an estimation of the time period before the problem becomes serious. In the field of industrial rotating machines, this technique is well acknowledged and reliable correlations between the pattern of the vibrations and corresponding sources for abnormal operating conditions were established (see, for example, Eisenmann and Eisenmann **[7]** and Michel **[8]**).

Despite the common and wide usage of vibration analysis methods for rotating and other equipment, these methods have not been practised widely as a diagnostic tool for reciprocating machines. This is probably due to the complex nature of the reciprocating machines, which incorporate a large number of moving parts. Methods for determining the rotational vibrations of crankshafts can be found in the literature (see, for example, Eshleman and Lewis **[9]**). However, these studies address the design issue rather than the application of the vibration response of the crankshaft as a maintenance tool. General guidelines for vibration analysis and treatment of reciprocating compressors can be found in a technical report by Engineering Dynamics Incorporated **[10]**. Nurhadi *et al.* **[11]** studied the correlation between the measured vibrations of an engine and its components, as a source for the excitations, and concluded that the source of the vibrations can definitely be identified by employing the vibration analysis method. Autar **[12]** introduced an autonomic diagnostic system for diesel engines. The primary component of the proposed assemblage, which is based on the vibration analysis method, enables malfunctions such as cylinder compression and combustion faults, valve-related problems and piston slap to be detected. Macian *et al.* **[13]** have measured the vibrations of an engine block and demonstrated that these vibrations are related to the instantaneous crankshaft torque. deBotton *et al.* [3] and Ben-Ari *et al.* **[14]** performed a series of experiments and measured the vibrations of an engine block while the engine was running under normal and abnormal conditions. They found that the vibration level of the engine provides reliable information regarding the engine condition and demonstrated that in many cases the vibration signature can be used to identify the source of the malfunction.

The present work stems from the work of deBotton *et al.* [3]. Here the significance and the contribution of the measured parameters to the overall information related to the engine health is explored. The engine block vibrations were recorded at four different locations, two in the vicinity of the rear crankshaft bearing and two at opposing sides of the engine block. In addition, the vibrations were measured along three principal axes, in the vertical direction, in the axial direction along the engine crankshaft and in the horizontal plane transverse to the crankshaft axis. Conclusions regarding the optimal measuring point and direction are drawn and the consistency of the proposed method is demonstrated. General guidelines for distinction between normal and abnormal operations are also proposed.

2 EXPERIMENTAL SET-UP

The experiments were performed with a four-stroke, four-cylinder in-line, carburetted spark ignition, Volkswagen-Polo type engine with a total displacement volume of 1272 cm^3. The vibrations were measured with a uniaxial Bruel and Kjaer 4371 accelerometer transducer. The accelerometer transducer had to be mounted with a stud since other mounting techniques were found to be inadequate owing to the significant vibrations of the engine. Measurements were taken at four different points (see Fig. 1). Two points were on opposing sides of the engine-block between cylinders 2 and 3 a few centimetres above the crankcase. The other two points were in the vicinity of the rear crankshaft bearing at 70° and 240° relative to each cylinder axis. At each point measurements were taken in three principal directions, the vertical direction along the cylinder axis, the axial direction along the crankshaft axis and the transverse direction in the horizontal plane and normal to the crankshaft axis (see Fig. 1). The signal

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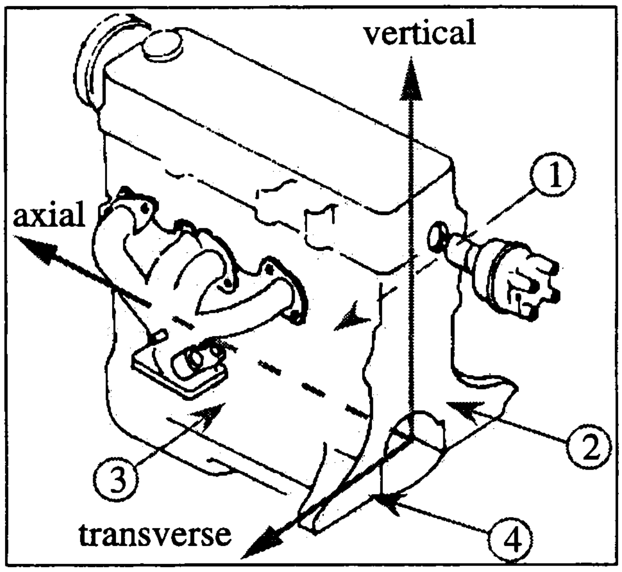

Fig. 1 Locations and directions of vibration measurements

from the transducer was amplified with a Kistler 5001 amplifier. Frequencies below 1 Hz and above 10 kHz were filtered out with a Disa 55D26 bandpass filter (see Fig. 2). The analogue signal was then digitized with a 12-bit National Instrument AT-Mio16E1 board and was stored on a hard drive of a 586-compatible host computer. At the same time, the magnitude of the electric potential at the primary coil line was recorded through the second channel of the analogue-to-digital (A/D) converter as a reference phase signal for the crank angle. The A/D converter has a dynamic range of 66 dB, allowing a maximal amplitude resolution of 2000 to 1. At a data acquisition rate of 12.5 kHz per channel, the National Instrument AT-Mio16E1 A/D card enables more than

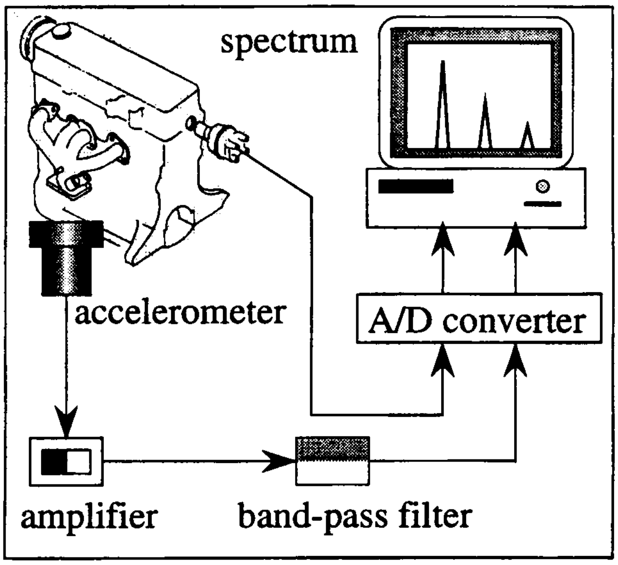

Fig. 2 Experimental set-up

30 000 data points per channel to be acquired. Thus, at an average engine speed of 1700 r/min (28 Hz), more than 64 crankshaft revolutions can be recorded. These are equivalent to 32 thermodynamic cycles of the engine.

3 RESULTS ANALYSIS

The transformation of the measured vibration waveform to the frequency domain was performed on the host computer with the aid of a fast Fourier transform (FFT) subroutine. However, it should be noted that, unlike the situation with common rotating machines, the angular velocity of a reciprocating engine may vary during the completion of a single crankshaft revolution, and certainly between two consecutive revolutions **[15]**. Owing to the engine flywheel, angle-to-angle variations are less than 1/1000, but cyclic variations in the angular velocity of the crankshaft, of more than 5 per cent were detected. Because the roots of the oscillatory motions of the engine block are mechanical, and hence the fundamental period of the vibrations is one crankshaft revolution, these cyclic variations must be accounted for. Thus, to perform an accurate FFT analysis, the sampling rate needs to be synchronized with the position of the crankshaft, rather than being dictated by the clock of the A/D board. To achieve this requirement without the use of complicated auxiliary equipment which can hardly be used in the field (for example, crankshaft angle pick-up devices), data from the primary coil line have been acquired to the second channel of the A/D converter. Once the data from the accelerometer and the primary coil line were acquired, the time intervals between the peaks of the ignition sparks were divided into a fixed number of subintervals and the vibration measurements were interpolated to obtain the corresponding vibrations at those time intervals. A detailed discussion concerning this signal analysis procedure can be found in deBotton *et al.* **[3]**.

A second aspect that needs to be addressed is the persistence of the measurements. Thus, because of the complex nature of reciprocating engines and the large number of moving parts, it should be verified that, for a given state of the engine, the vibration signature is indeed characteristic. Hence, the variations of the vibration signature between two different measurements should be small enough to enable a firm correlation to be made between the condition of the engine and the measured signature. A common method to improve the persistence of the measurements while eliminating random disturbances and noise is to average a sequence of spectra **[16]**. In the course of the present work it was found that an average of eight spectra is sufficient to obtain a persistent signature, and henceforth all the presented results are averages of eight distinct measurements.

The vibration signature of the engine under normal operating conditions is presented in Fig. 3 in terms of

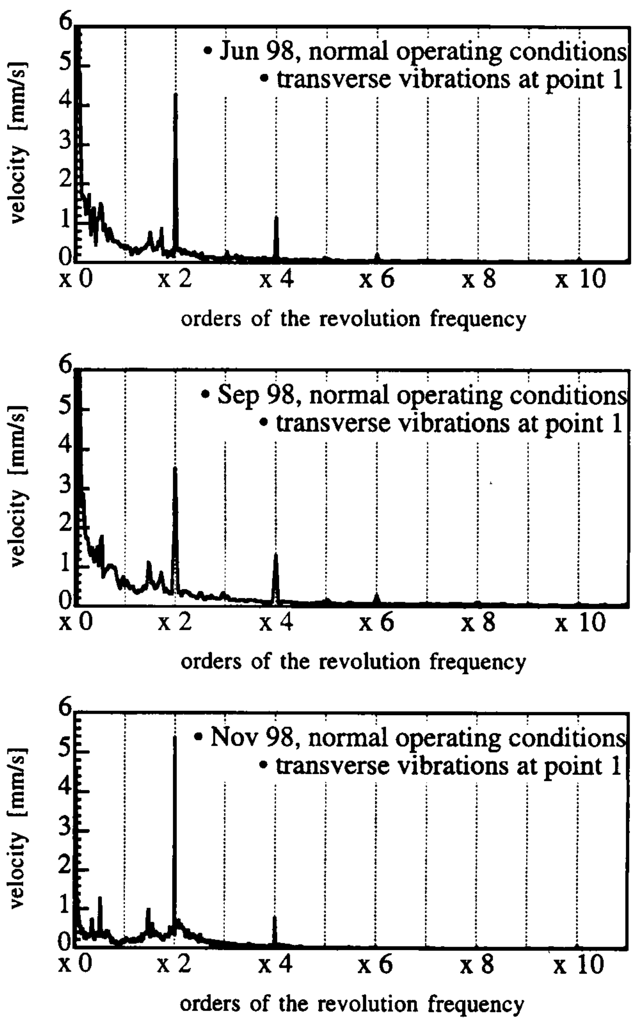

Fig. 3 The vibration signatures of the engine under normal operating conditions as measured at three different times

magnitude spectrum plots. These show the amplitudes of the harmonic components of the vibration waveform versus the frequency which is normalized by the revolution frequency of the engine. The integration from the measured acceleration to velocity was performed in the frequency domain. Here, as well as in the following figures, the engine is running at an average angular speed of 1700 r/min (28 Hz) with no external load. The measurements presented in Fig. 3 were taken at point 1 on the side of the engine block along the transverse direction (see Fig. 1).

Figure 3 depicts the vibration signatures as measured at three different times, approximately 3 months apart. It can be seen that the three signatures are analogous, and in all figures the major peak is attained at twice the revolution or the fundamental frequency (56 Hz). This peak corresponds to the combustion process that occurs every second crankshaft revolution in each of the four cylinders, and hence two combustion processes during each revolution. A second noticeable peak, at four times the revolution frequency, also results from the combustion process and it corresponds to the second term of the Fourier expansion series. Thus, since the cyclic transverse motion of the engine block due to the combustion process is not a pure harmonic motion, this higher-order term is required to enable the characterization of the motion by a sum of harmonic functions. An additional smaller correction peak is observed at six times the fundamental frequency. The noticeable differences between the plots are in the low frequency range. The variations in the low frequency range result from the fact that the first two data sets were obtained during 32 cycles of the crankshaft, or 1.15 s, while the latter data set was obtained during twice this time interval. In addition an adjustment was made to the high-pass frequency filter, allowing the cut-off frequency to be set to lower levels.

It should be emphasized that, during the time period between the three measurements, numerous experiments, at various speeds and loading conditions and with different malfunctions, were performed. The same engine was also used for instructional purposes and was run for long periods by inexperienced students. Also, during this time interval, the engine was shifted to another location and was mounted on an all-new base. The resemblance of the three spectra suggests that the vibration signature of the engine under normal operating conditions is indeed characteristic. These typical spectra will provide the reference pattern of relative amplitudes that reflects the system vibrations under normal operating conditions.

The persistence of the vibration signature under abnormal operating conditions was also verified. For example, three signatures of the engine vibrations while running with one disconnected spark plug are presented in Fig. 4. These measurements were taken together with those shown in Fig. 3, at the same point and in the same direction. It is evident that the pattern of the vibration signature is very different from the one obtained during normal operating conditions. Nonetheless, except for the differences in the low frequency range, the three spectra in Fig. 4 are analogous. It can be seen that while the peaks at twice and four times the fundamental frequency still exist, the primary peaks appear at 0.5 and 1.5 times this frequency (42 Hz). This is attributed to the fact that, since only three of the four cylinders undergo combustion, the primary component of the oscillatory motion is no longer harmonic. In fact, the periodicity of the motion is twice the fundamental period (two revolutions of the crankshaft). This uneven state of operation can easily explain the evident presence of subharmonic frequencies. Additional peaks can be distinguished at frequencies which are multiples of half the revolution frequency (that is 0.5, 1, 1.5, ...). Similar persistence verifications were made for all the examined malfunctions.

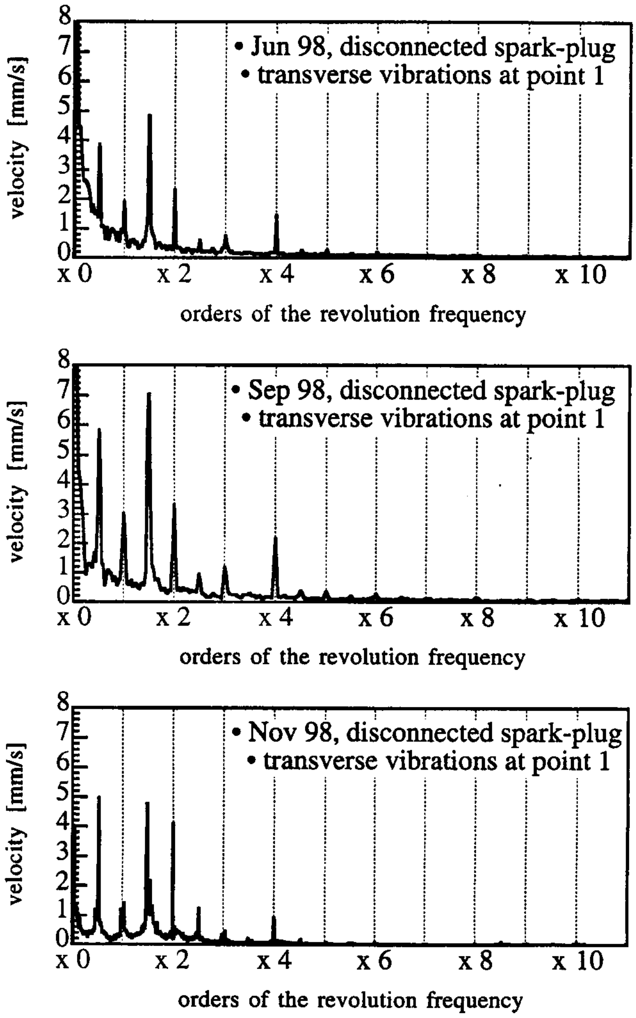

Fig. 4 The vibration signatures of the engine when one of the spark plugs is disconnected, as measured at three different times

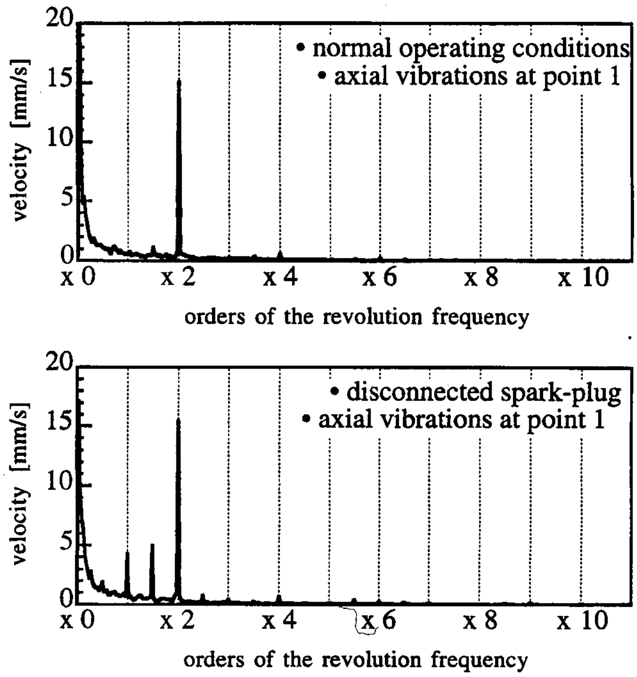

Fig. 5 The vibration signatures of the engine under normal operating conditions and when one of the spark plugs is disconnected, as measured in the axial direction at point 1

One of the important aspects in vibration monitoring is the location and the direction of the vibration transducer. In industrial applications the transducer is usually located as close as possible to the load zone, normally on the housing of the bearing along the direction of the applied excitation [5]. To determine the best location for the vibration transducer, four measuring points were examined, where at each point measurements were taken in the three principal directions (see Fig. 1). The aim of these measurements was to explore at which point and in which direction the variations between the vibration signatures that correspond to the different operating conditions are most noticeable. As an example, the effect of a disconnected spark plug on the measured vibration signature is illustrated in the following figures.

The measured vibration signature at point 1 in the axial direction, at normal operating conditions and with a disconnected spark plug is shown in Fig. 5. In both figures the major peak in the spectrum plot appears at twice the fundamental frequency. The two peaks at one-half and three-halves of the fundamental frequency that appear when the engine is running with a disconnected spark plug constitute the main difference between the two cases. A comparison of the variations between these two plots with the corresponding variations between Figs 3 and 4 that correspond to the measurements in the transverse direction, demonstrates that the sensitivity of the measurements in the transverse direction is higher. The measurements in the vertical direction at this point (not shown here) were found to be even less sensitive than those measured in the axial direction.

The vibration signatures at point 2, near the rear crankshaft bearing at an angle of 70° from the cylinder axis, are shown in Fig. 6. Once again, the top plot corresponds to the signature for normal operating conditions and the bottom plot for the one obtained with a disconnected spark plug. These measurements were taken in the transverse direction. Under normal conditions the major peak appears at the combustion frequency. More peaks, at one-half and at three-halves of the fundamental frequency, appear when one of the spark plugs is disconnected. We note that the variations in the vibration signature at point 1 in the transverse direction are larger than those obtained at point 2 along the same direction. The situation is similar to that for the vibrations that were measured along the

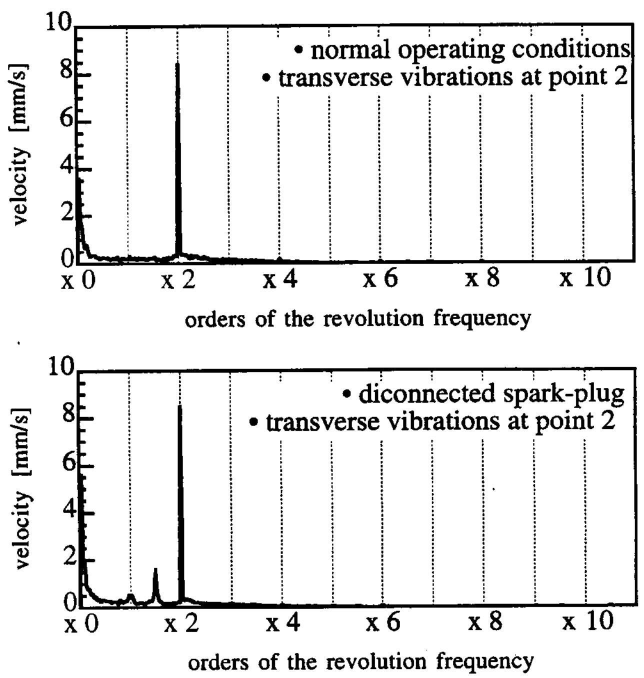

Fig. 6 The vibration signatures of the engine under normal operating conditions and when one of the spark plugs is disconnected, as measured in the transverse direction at point 2

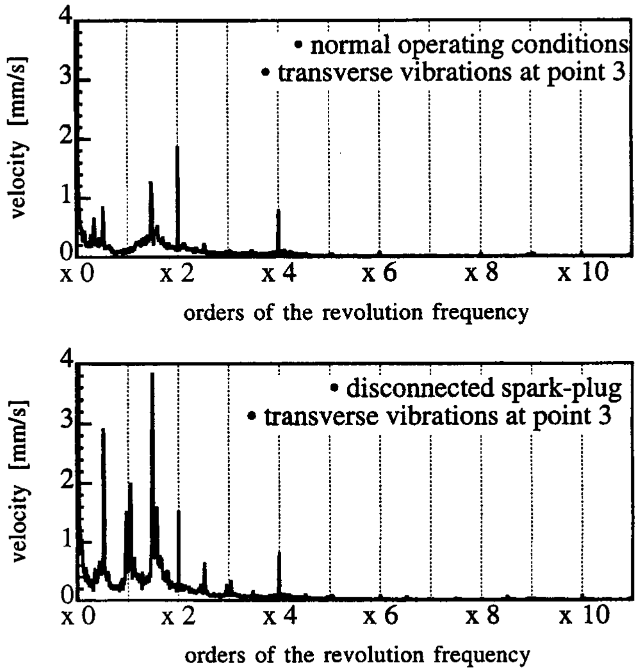

Fig. 7 The vibration signatures of the engine under normal operating conditions and when one of the spark plugs is disconnected, as measured in the transverse direction at point 3

axial and the vertical directions at this point. It was found that, for the entire set of examined malfunctions, the signatures that were measured along the transverse direction at point 1 are more sensitive to the engine condition than those measured at point 2 in the three principal directions.

Point 3 is located on the side of the engine block opposing point 1. The vibration signatures that were measured at this point along the transverse direction during normal operating conditions and with a disconnected spark plug are shown in Fig. 7. In the topmost plot, for the normal engine condition, the peak at the combustion frequency is the dominating one; however, a peak at three-halves of the revolution frequency can also be seen. This peak probably results from a small leakage of ambient air into the inlet manifold, a problem that was detected as a consequence of the vibration analysis that was carried out on this data set. After the problem was solved, the peak at three-halves of the revolution frequency diminished. The signature measured with a disconnected spark plug is significantly different from the one for the normal condition. Thus, two large peaks at one-half and three-halves of the fundamental frequency, and two smaller peaks at five-halves and four times this frequency, can be easily identified in the spectrum plot. The sensitivity of the measurements along the axial and the vertical directions was also examined and found to be lower than the one experienced along the transverse direction.

Analogous analyses of the vibration signature at this point while the engine is running with other malfunctions revealed that the quality of the information available from this point is comparable with that of point 1. However, since point 3 is located in the hot region between the exhaust manifold and the engine block, practical circumstances did not allow further measurements at this point.

Spectra of the vibration signature at point 4, which is located near the housing of the rear bearing 240° relative to the cylinder axis, are shown in Fig. 8. These measurements were taken along the transverse direction. Under normal conditions only one peak appears at the combustion frequency with no noticeable peaks for the higher-order correction terms. When a spark plug is disconnected, a significant peak at one-half of the revolution frequency appears.

The above measurements, together with additional measurements that were carried out during other operating conditions [**14**], indicate that the most informative signature is obtained at point 1 along the transverse direction. Accordingly, in the following section, while dealing with the diagnosis of the vibration signature, the main emphasis is given to the measurements that were taken at this point in the transverse direction. Nonetheless, it is stressed that, in order to establish the condition of the engine, in a manner similar to the one practised in industrial applications, it is possible that measurements will have to be taken at several points and directions.

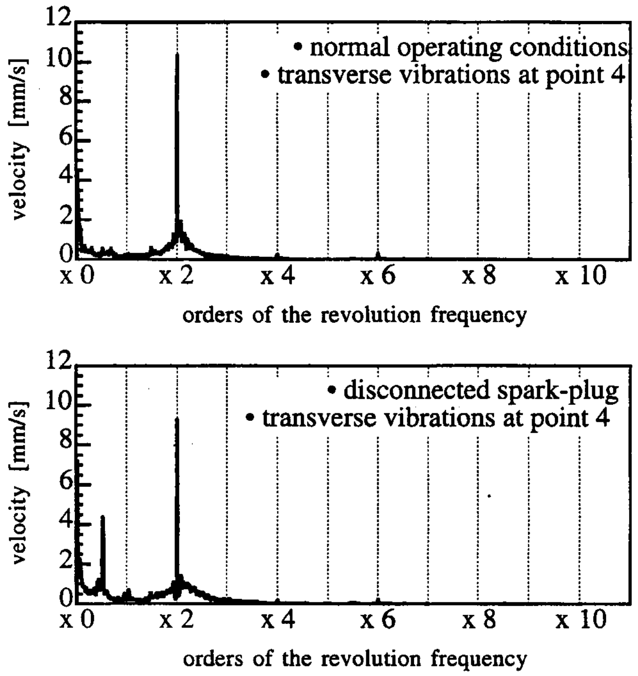

Fig. 8 The vibration signatures of the engine under normal operating conditions and when one of the spark plugs is disconnected, as measured in the transverse direction at point 4

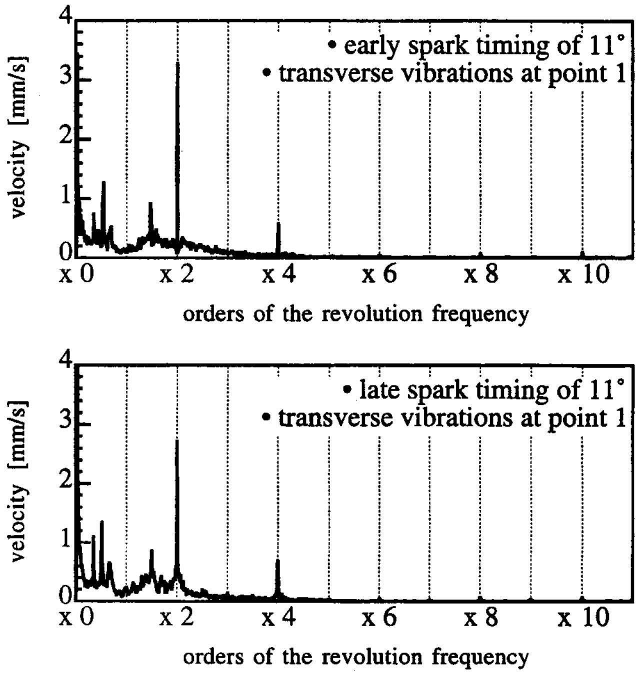

Fig. 9 The vibration signatures of the engine with an early spark timing of 11° (top) and with a late spark timing of 11° (bottom), as measured at point 1 along the transverse direction

4 VIBRATION SIGNATURE DIAGNOSIS

Various malfunctions were applied to the engine and the corresponding vibrations were measured while the engine was running under these abnormal conditions. The reference signatures with which the abnormal signature is compared are the ones for the normal operating conditions which are depicted in Fig. 3. The vibration signature of the engine while running with a disconnected spark plug was analysed in the previous section. The vibration signature of the engine with off-optimal spark timing is shown in Fig. 9. The top spectrum plot corresponds to an early spark timing of 11° and the second plot to a late spark timing of 11°. In both figures there is a noticeable decrease in the magnitude of the combustion peaks at two and at four times the fundamental frequency. A dispersion of the peaks in the subharmonic range of frequencies can also be observed, particularly with a late spark timing. It can be seen that a malfunction involving an off-optimal spark timing is an 'even' type of malfunction in the sense that all the cylinders are going through the same variations. Accordingly, it is expected that the presentation in the frequency domain will be relatively insensitive to this family of malfunctions since it is expected that no appreciable peaks will appear at abnormal frequencies (such as orders of one-half the fundamental frequency or at asynchronous frequencies). In fact, the vibration signatures of other even malfunctions, such as applications of rich or poor fuel–air mixtures to the engine, exhibited similar indifference.

Further, it is emphasized that, in the present case, the engine is running unloaded. Consequently, in comparison with normal operating conditions, the temporal variations of the pressure in the cylinders due to off-optimal spark timings are relatively small. Since the pressure in the cylinders is the primary excitation of the system, small variations in the pressure will give rise to small variations in the vibration signatures. On the other hand, the variations of the pressure due to off-optimal spark timings will be larger while the engine is loaded; it is expected that the corresponding variations in the vibration signature will be more substantial under these circumstances too.

The vibration signature of the engine operating with an incorrect spark plug gap is shown in Fig. 10. The top spectrum plot corresponds to operation with a spark plug gap widened in one cylinder from the optimal value of 0.7 to 1.2 mm. This malfunction simulates a worn spark plug. The only noticeable variation of this signature from the normal one is a decrease in the magnitudes of the combustion component and its correction term. This is attributable to the longer ignition delay in the impaired cylinder because in this cylinder, the cylinder charge breaks down to allow spark discharge only at a later stage, when the pressure and temperature

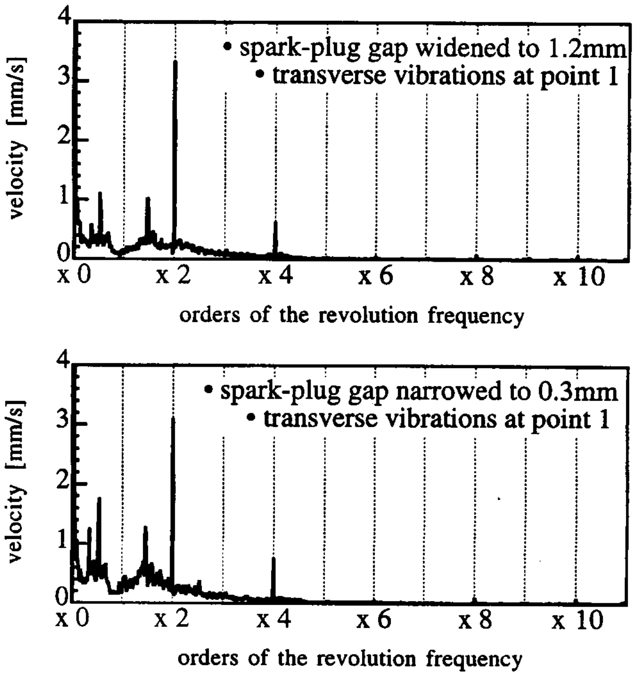

Fig. 10 The vibration signatures of the engine with a worn spark plug whose gap was widened to 1.2 mm (top) and a fouled spark plug whose gap was narrowed to 0.3 mm from the optimal gap width of 0.7 mm. Measurements were at point 1 along the transverse direction

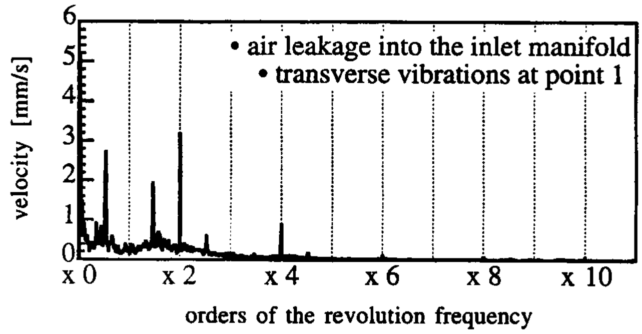

Fig. 11 The vibration signature of the engine with a leakage of ambient air into the inlet manifold, as measured at point 1 along the transverse direction

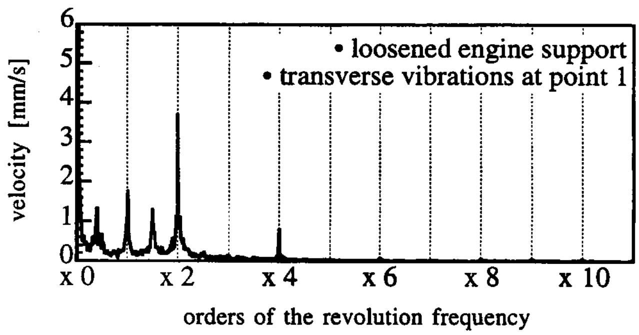

Fig. 12 The vibration signature of the engine with a loosened engine support, as measured at point 1 along the transverse direction

reach higher values than those needed with a correctly set spark plug gap.

A simulation of spark plug fouling was accomplished by narrowing the spark plug gap to 0.3 mm. The corresponding vibration signature is shown in the lower spectrum plot of Fig. 10. While the intensity of the combustion component decreased in a similar manner to that in the previous case, there is a dispersion of the energy in the low frequency range, primarily around one-half of the fundamental frequency. This is due to the occasional misfires that occurred in the impaired cylinder.

Interestingly, an abnormal vibration signature of the engine was measured immediately after it was shifted to a new location. Analysis of the source of the problem revealed a small leakage of air into the inlet manifold. After the problem was solved, a similar malfunction was intentionally applied to the engine and the corresponding vibration signature is depicted in Fig. 11. The harmonic components at the combustion frequency and the correction term are present; however, noticeable peaks can be distinguished at one-half, three-halves and five-halves of the fundamental frequency. There is also a severe dispersion of the energy in the subharmonic range of frequencies and some dispersion in the high frequency range of eight to ten times the revolution frequency.

While various faults in the combustion process can be easily detected when the exhaust gas is analysed for its composition, mechanical malfunctions such as looseness and worn or broken components are very hard to detect by the application of regular maintenance routines. In this respect, the vibration analysis technique can provide a powerful tool for early detection of these crucial malfunctions. In Fig. 12 the vibration signature of the engine running with a loosened support is shown. As expected, there are no noticeable changes in the combustion peaks; however, the second highest peak is at the revolution frequency. This peak, that results from the cyclic motion of the loosened engine block in response to the revolution of the crankshaft, provides a clear indication of the mechanical disruption of the engine. Peaks at one-half and three-halves of the revolution frequency also appear in the plot, probably resulting from the non-pure harmonic motion of the engine.

5 SUMMARY

A vibration analysis method has been performed for a reciprocating engine. The measuring device does not require any custom-made components, no special adjustments of the engine, such as holes for oil monitoring transducers, are needed and there is no direct contact of the transducer with the internal components of the engine. Accordingly, the influence of the measuring transducer on the functionality of the engine is practically zero. The required FFT analysis can be accomplished either with an on-line or an off-line computing element while the engine is running. Because of the cycle-to-cycle variations of the engine speed, it was found that the wavelet measurements must be synchronized with the crankshaft angle in order to perform an accurate transform to the frequency domain. A straightforward mathematical procedure, which requires an additional triggering from the primary coil line, was developed for this purpose. This procedure eliminated the need to mount a complicated and expensive encoder on the crankshaft.

The persistence of the vibration signature, under normal and abnormal conditions, was examined and verified by comparing different sets of measurements that were taken during a time interval of more than 6 months. The optimal measuring location and direction were found to be on the side of the engine block perpendicular to the crankshaft. The engine was tested with a number of artificial malfunctions and the corresponding vibration signatures were measured. It was found that the vibration signatures are indicative and provide useful information about the corresponding malfunctions. It was further found that the vibration signature is more sensitive to the family of uneven malfunctions (where the process in one cylinder differs from that occurring in the others), to which most mechanical problems belong.

General evaluation criteria for the health of the engine can be established. The appearance of peaks at one-half or three-halves of the fundamental frequency, or, alternatively, dispersion of the energy at sub-harmonic frequencies or at six to eight times the fundamental frequency, clearly indicates that a severe malfunction had developed in the engine. The appearance of a peak at the fundamental frequency implies that a mechanical looseness enabled the engine to vibrate at the crankshaft frequency. In contrast with other predictive maintenance techniques, the vibration analysis method provides information about a wide variety of problems, from various process disorders to mechanical failures and degradations. The method also provides the capability for root cause analyses and diagnosis of worn units or components.

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