EFFECT OF PULSE PARAMETERS ON CRANKSHAFT BEARING VIBRATIONS - SPECTRAL ANALYSIS

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ABSTRACT

In finite element modeling of engine skirt vibration and radiated noise, it is important to simulate the bearing vibration signal, as the crankshaft main bearing vibration is the main exciting source of the engine outer surface. The aim of this investigation is to look at the parameters affecting the bearing vibration signal, Such as pulse amplitude, duration and repetition. A square pulse (or pulses), digitally generated on the computer to simulate the bearing vibration, are analyzed and the different parameters are altered and the consequent differences are discussed. The spectral analysis is carried out by calculating the Fourier coefficients. It is found that the variations in pulse amplitude, width and pulse repetition in one complete cycle of the engine revolutions are those shape parameters affecting the harmonic contents of the crankshaft bearing vibration. The harmonic level is proportional to the logarithm of amplitude of pulse.

Keywords : Crankshaft vibration, Bearing vibration, Spectral analysis, Fourier analysis

INTRODUCTION

The engine main bearing forces are of impulsive characteristics. These main bearing forces are the main source of excitation of the engine structure and have a great influence on the engine block and engine skirt vibration. This, in turn, affects to a large scale the engine emitted noise. There are many factors affecting the spectral analysis of the engine main bearing forces, and consequently the engine main bearing vibration. The engine structure itself influences the engine radiated vibration and noise [1-3]. The operating cycle is the drive parameter for the vibration and noise generation mechanism of the engine block [4,5]. Many researchers investigated those parameters affecting engine radiated vibration and noise [6-8]. Anderton, [9] investigated combustion as a source of diesel engine noise, he compared between the spectral level of combustion in two and four stroke engines (single pulse in one revolution and single pulse in two revolutions).

The aim of this research is to investigate the pulse shape parameters affecting the spectral analysis of the engine main bearing vibration signals.

THEORETICAL BACKGROUND

Fourier Analysis

As the vibration signals of the crankshaft bearings are almost periodical, Fourier analysis can adequately be applied. The periodic signal f(t), of period 2L, is represented by Fourier series as:

$$f(t)=a_o/2 + \sum_{n=1}^{\infty} a_n \cos w_n t + b_n \sin w_n t \qquad (1)$$

where : $w_n = n\pi/L$, at the corresponding frequency $f_n = w_n/2\pi$

and

$$a_n = 1/L \int_0^{2L} f(t) \cos w_n t dt \qquad (2)$$

$$b_n = 1/L \int_0^{2L} f(t) \sin w_n t dt \qquad (3)$$

The calculated amplitudes of the Fourier series $\{ \sqrt{(a_n)^2 + (b_n)^2} \}$ are plotted against frequency f_n as a series of discrete lines to give the Fourier spectrum.

The vibration signals of the end bearings, of in-line engines, have only one predominant pulse. In a 4-stroke, six-cylinder, seven-bearing, in-line engine there is a cylinder firing each 120°. If the firing order is 1-5-3-6-2-4, one will find that bearings Number 2 and 6 (intermediate bearings) have two predominant pulses of 240° crank angles separation interval. The other two intermediate bearings (Number 3 and Number 5) have two predominant pulses of 480° crank angles separation interval. The middle bearing vibration signal also contains two predominant pulses but it is however different since the separation interval is 360°.

The pulses in the three different bearings mentioned in the previous paragraph, are represented by square pulses of duration Δ and amplitude A. The signals representing vibration of end, intermediate and middle bearings are shown schematically in Figure1-a, 1-b and 1-c in order. These signals could be represented mathematically using the unit step function u(t) as follows :

Figure1-a

$$f(t) = A [u(t)-u(t-\Delta)] \qquad (4)$$

Figure1-b

$$f(t) = A[\{u(t)-u(t-\Delta_1)\}+\{u(t-4\pi/3) -u(t-4\pi/3-\Delta_2)\}] \qquad (5)$$

Figure1-c

$$f(t) = A [\{u(t)-u(t-\Delta_1)\} + \{u(t-2\pi) -u (t-2\pi-\Delta_2)\}] \qquad (6)$$

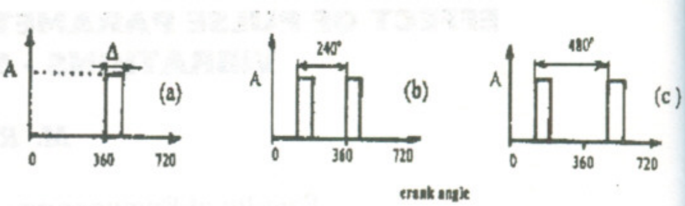

Figure 1 Signals representing bearings vibration of a 4-stroke, six-cylinder, seven-bearing, in-line engine.
a) signal representing end bearing vibration,
b) signal representing intermediate bearing vibration,
c) Signal representing middle bearing vibration.

For any of the previously mentioned functions representing the vibration signals of the different crankshaft bearings, the Fourier series can be mathematically calculated. For the functions representing bearing vibration signals Equations 4,5 and 6, the Fourier series coefficients a_n and b_n are calculated using Equations 2 and 3.

An algorithm is constructed for the calculation of Fourier series coefficients of a periodic function represented in a digital form with any number of digits (points). The program is written in FORTRAN for use with personal computers operating under DOS or Unix systems.

Practical measurements of the spectral analysis of the vibration acceleration of the different main bearings for an actual running engine show clear variations from one bearing to another [10]. So, it is of prime interest to theoretically investigate the effect of each parameter affecting this spectral analysis. Parameters that mainly affect the spectral analysis of the bearing vibration are: (1)the number of adjacent cylinders to any particular bearing, (2) the separation time interval between firing of the adjacent cylinders[11], (3) the variation between the pressure development in the adjacent cylinders [12], and (4)the pulse width (pulse duration) or the pulse width to cycle ratio.

In in-line engines there is either one or two main pulses in the vibration signals resulting from firing of the adjacent cylinder(s). The separation interval between the pulses; in terms of crank angle; depends on the number of cylinders.

For an in-line, six-cylinder seven-bearing engine, considering firing of the adjacent cylinders only, there are three possible cases namely: (1)one pulse in the whole cycle representing end bearings namely Number 1 and Number 7 bearings, (2)two pulses with 240° crank-angle degrees separation (or 480°), representing the intermediate bearings namely Number 2, 3, 5 and 6, and, (3) two pluses of 360° crank-angle degrees separation interval representing the middle main bearing (Number 4 bearing).

FACTORS AFFECTING THE SPECTRAL ANALYSIS

Number of Pulses

For a six-cylinder, in-line engine running at speed of 1500 rpm (fundamental frequency is 12.5 Hz), if pulse width is 60° crank-angle degrees, Figure 2 shows the harmonic spectra of (a) a single pulse representing end bearing vibration, (b) two pulses representing the intermediate bearing vibration with 240° (or 480°) crank-angle separation interval and (c) two pluses of 360° separation interval representing the middle bearing vibration signal.

The spectra show typical harmonic envelop humps. These humps are decreasing in level with a slope of -20 dB/decade with increasing frequency. The spectra are nearly flat up to 50 Hz and are 6 dB lower at 100 Hz.

The harmonic spectra of the digital signal containing a single pulse, representing vibration of end bearings (Number 1 and Number 7) have all harmonic existing Figure 2-a. The harmonic spectra of the signal representing the middle bearing vibration is shown in Figure 2-b. It is seen that it has half the harmonic density compared to the harmonic spectra of the signal when only one pulse exists. Also, it is 6 dB higher in level over the whole range of frequency. It should be noted that the harmonic spectra of Figure 2-c are not shifted to the right (humps occur at the same frequencies), but all the odd harmonics namely f_o , $3f_o$, $5f_o$, $7f_o$....are of zero value, where f_o is the cycle fundamental frequency (f_o = rpm/120).

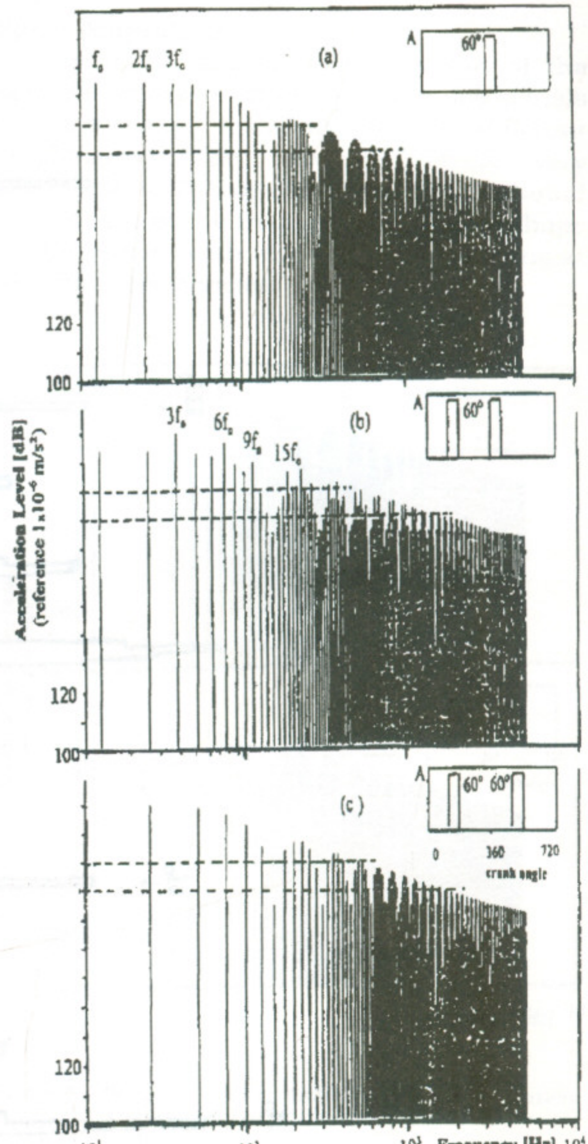

Figure 2 Harmonic spectra of signals representing bearing vibration of a 4-stroke, six-cyliner, seven-bearing, in-line engine.

a) Harmonic spectra of end bearing vibration signal,
b) Harmonic spectra of intermediate bearing vibration signal,
c) Harmonic spectra of middle bearing vibration signal,

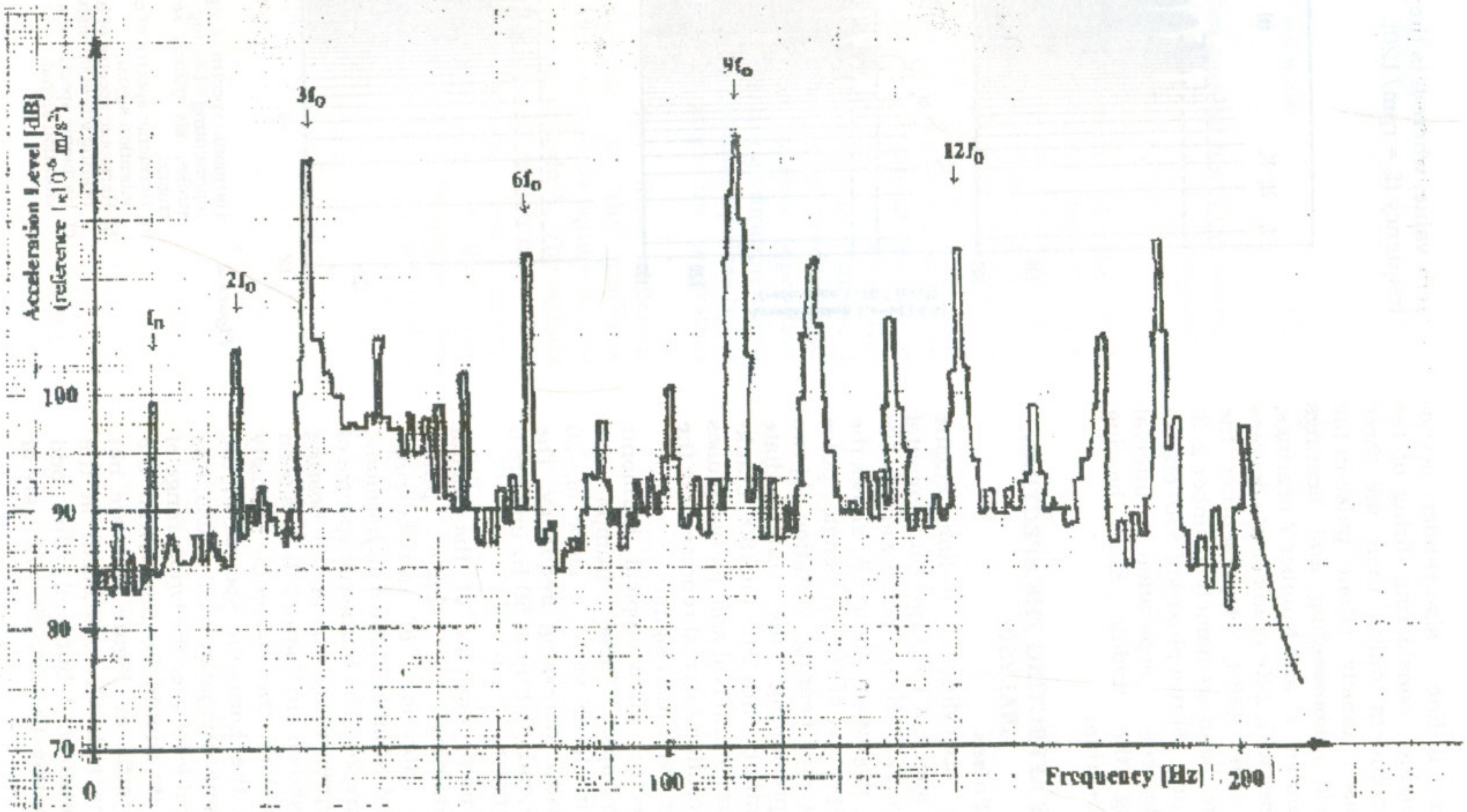

Figure 3 Measured narrow band frequency of an intermediate bearing vibration at 1500 rpm

The reason for the disappearance of odd harmonics could be explained in two different ways:

(i) when there exists two equi-spaced pulses in a signal analyzed covering a complete cycle of 720° crank angles, there exists a mirror symmetry and hence the existing Fourier Coefficients are a_o, a_{2n} and b_{2n}, where a_{2n+1} and b_{2n+1} are of zero values. Therefore, the odd harmonics are zeros. Also, because there exists two identical pulses, the amplitude of the coefficients, when calculating the integration, are twice that resulting when calculating those in case of a single pulse signal. So, on taking the logarithm of the amplitude it will be higher by 6 dB (20*log2).

or

(ii) in this case (two equi-spaced identical pulses) the analyzed signal could be considered as if it were a case of a single pulse signal but over only half the period (360° crank angle period). This would be corresponding to the effect of doubling the engine speed, thus the fundamental frequency will be doubled as well. In other words if the engine speed is 1500 rpm, then f_o equals to 12.5 Hz for the case of a single pulse over 720° crank angle. While f_o is 25 Hz for the case of a single pulse over 360° crank angle (two identical equi-spaced pulses over 720° crank angle).

The harmonic spectra of the signal having two pulses separated by 240° crank angle; simulating the intermediate; namely Number 1, Number 6 bearing vibrations is shown in Figure 2-b. It resembles that of the single pulse signal (Figure 1-a) in level, slope and the existence of all harmonic. But due to the existence of two pluses separated by 240° interval, it is found that every third harmonic; namely 3rd, 6th, 9th, 12th,...etc, is 6 dB higher compared with the amplitude of the corresponding harmonic when the single pulse is analyzed (Figure 2-a). This phenomenon is observed during a narrow band frequency analysis session of the vibration signals of intermediate bearings. A sample of these narrow band measurements is shown in Figure 3.

Pulse Amplitude

Figure 4 shows a comparison of the spectral analysis of two single pulse signals of the same duration but of different amplitudes. The spectral levels vary according to 20*Log (AMPLITUDE). It should be noted that the frequency spectrum shape is the same, with the humps occurring at the same frequencies.

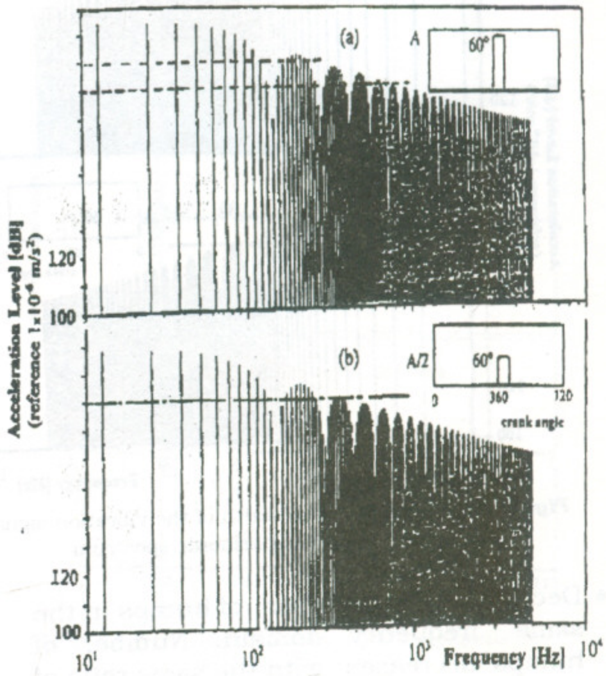

Figure 4 Effect of pulse amplitude of the vibration signal on the harmonic spectrum

Pulse Width Variation

a- Signals with single pulse

Figure 5 shows the harmonic spectra of two single pulse signals. First signal has a pulse width of 30° crank angle, and the second has a pulse width 60° of crank angle. The spectral shape of the two signals are of the same pattern, but not of the same shape. Decreased pulse width has the following effects:

Increases the flat region of the spectral shape; i.e the frequency up to which the level is nearly flat (f_f). This frequency is given by $f_f = 0.333/\Delta T$, where ΔT is the pulse duration in seconds

ΔT= pulse duration in degrees * T/720, where T is the cyclic time = 120/rpm and f_f (for 30° pulse) = 100 Hz and f_f (for 60° pulse)=50 Hz).

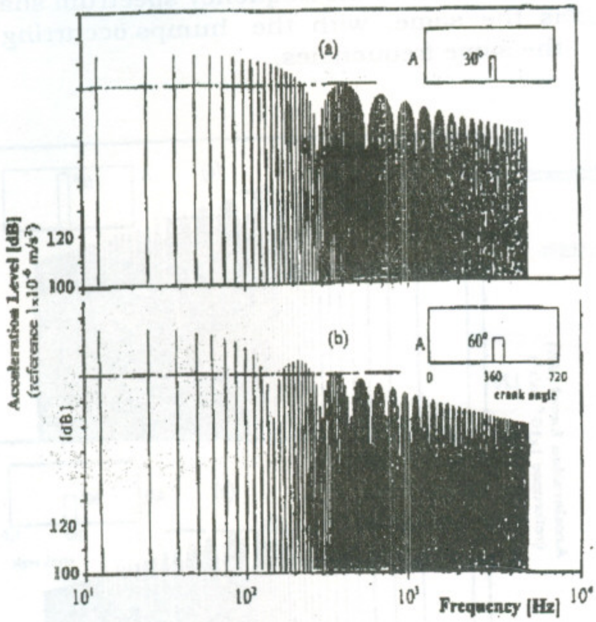

Figure 5 Effect of pulse width of the vibration signal on the harmonic spectrum

- Decreases the number of humps in the same frequency domain. Number of humps decreases with the same ratio of increasing pulse width. In the shown comparison the number is halved.

and the most important effect is that,

- The low frequency (the first four or five harmonics) level is decreased according to 20*log(ratio of increased pulse width). In the shown comparison, the level in the low frequency region decreases by 6 dB (20*log2).

b-Signals with two pulses

Figures 6 (a,b,c and d) illustrate the effect of varying the duration of one pulse on the spectral analysis. The figure shows the spectral analysis of two-pulse signals separated by 360° crank angle, the duration of the first pulse is maintained at 30° while the duration of the second pulse is altered as following; (i) 30° Figure 6-a, (ii) 28° Figure 6-b, (iii) 25° Figure 6-c and (iv) 20° Figure 6-d. The cycle fundamental frequency is 12.5 Hz (assuming 1500 rpm engine speed).

The analyzed signals in Figures 6-b, c and d could be treated as the summation of two signals (i)a signal having two identical pulses each of width equals to the smallest pulse width and (ii)a signal with one pulse of width equals to the difference between the pulse width of the two pulses. That is to say, for example, the analyzed signal in Figure 6-b could be treated as the summation of two signals (i)a signal containing two identical pulses of 28° width and (ii)a signal having a single pulse of 2° width. For all the signals analyzed in Figure 6, the one considered with two identical pulses (360° separation) would generate only even harmonics. While the superimposed single pulse signal would have all the harmonics. Then the output is the logarithmic summation of the levels of the corresponding even harmonics. The level of the odd harmonics will be of the same level out from analyzing the single pulse signal. The level of the odd harmonics will increase as the pulse width increases from 2° (Figure 6-b) to 5° (Figure 6-c) to 10° (Figure 6-d) (please refer to Figure 5 and the corresponding text).

The spectrum criterion in Figures 6-b,c,d are approaching actual measured spectra of the bearing vibration of a six-cylinder engine; where the pulses are never identical; as indicated in the following section.

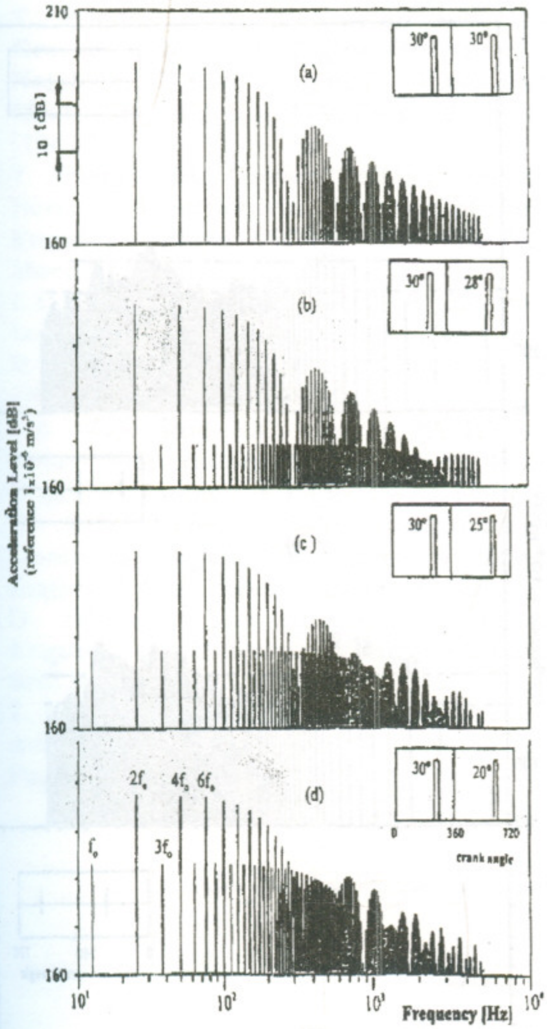

Figure 6 Effect of varying pulse width of a vibration signal containing two pulses on the harmonic spectrum
a)two identical pulses,
b)(b.c.d) two pulses of different width

BEARING VIBRATION FOR A RUNNING ENGINE

The time domain vibration signals for different bearing caps, an end bearing, an intermediate bearing and the middle bearing for a six-cylinder, in line T/C diesel engine, are shown in Figure 7.

The end bearing (Number 7 bearing) vibration signal has only one predominant pulse induced by firing of Number 6 cylinder. The intermediate (Number 6 bearing) vibration signal has two predominant pulses induced by firing of the two adjacent cylinders; namely Number 6 and Number 5; with 240° separation interval. The middle (central) bearing (Number 4) vibration signal has also two predominant pluses induced by firing of the two adjacent (Number 4 and Number 3) cylinders, in this instant the separation interval is 360° crank angle.

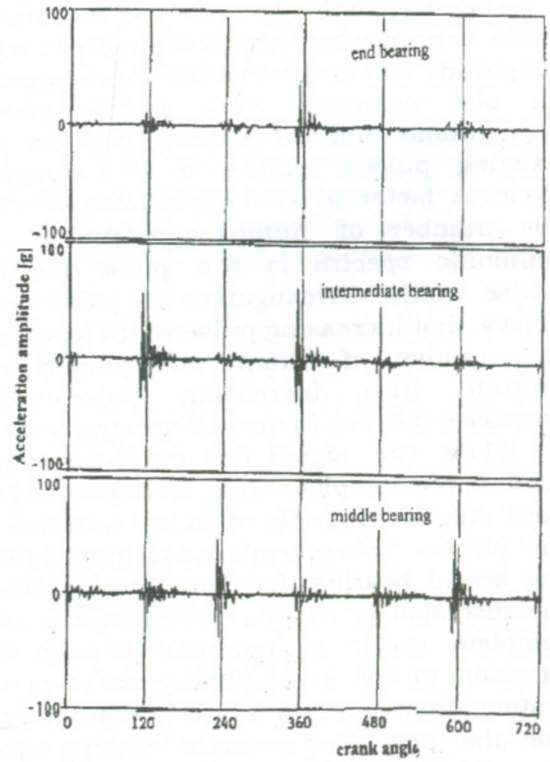

Figure 7 Time domain vibration signals of different bearing caps

The effect of pulse separation interval of actual bearing vibration on spectral analysis can be seen in Figure 8. The figure shows the frequency spectra of the vibration signal when all other pulses rather than the adjacent ones are eliminated. The spectral level of Number 6 bearing is by 6 dB higher than that of the end bearing at each third harmonic namely, 3rd, 6th, 9th, 12th,.. etc. This phenomenon is theoretically illustrated in Figure 2-c(the case of a signal having two pulses with 240° separation interval). The figure also shows that the level of the odd harmonics namely 3rd, 5th, 7th, ..etc. of Number 4 bearing are 6 to 10(and more) dB lower in level than those of the neighboring even harmonic. The odd harmonics do not completely disappear due to the fact that the two pluses, being analyzed, are not completely identical, Figure 6-d illustrates the case when the two pulses in the signal are not identical).

CONCLUSIONS

The harmonic content of the engine main bearing vibrations is dependent on the amplitude and number of transients existing in one complete cycle of the engine revolutions and the spacing between the existing pulses relative to the complete cycle. A factor of vital importance affecting the number of humps appearing in the harmonic spectra is the pulse duration (pulse width). Investigation of pulse width shows that increasing pulse width increases the number of humps in a proportional attitude. Also, increasing pulse width increases the low frequency spectral level.

When the signal has multi-transients exist in the complete cycle, all harmonics do have values when there is any variation in the pulses. This resembles to a great extent the actual bearing force or vibration where the discrepancy in the pulse shape; in one complete cycle; is imperative due to the variation in the crank shaft-connecting rod system from one cylinder to another and that the developed pressure (exciting force) in all the engine cylinders are never identical.

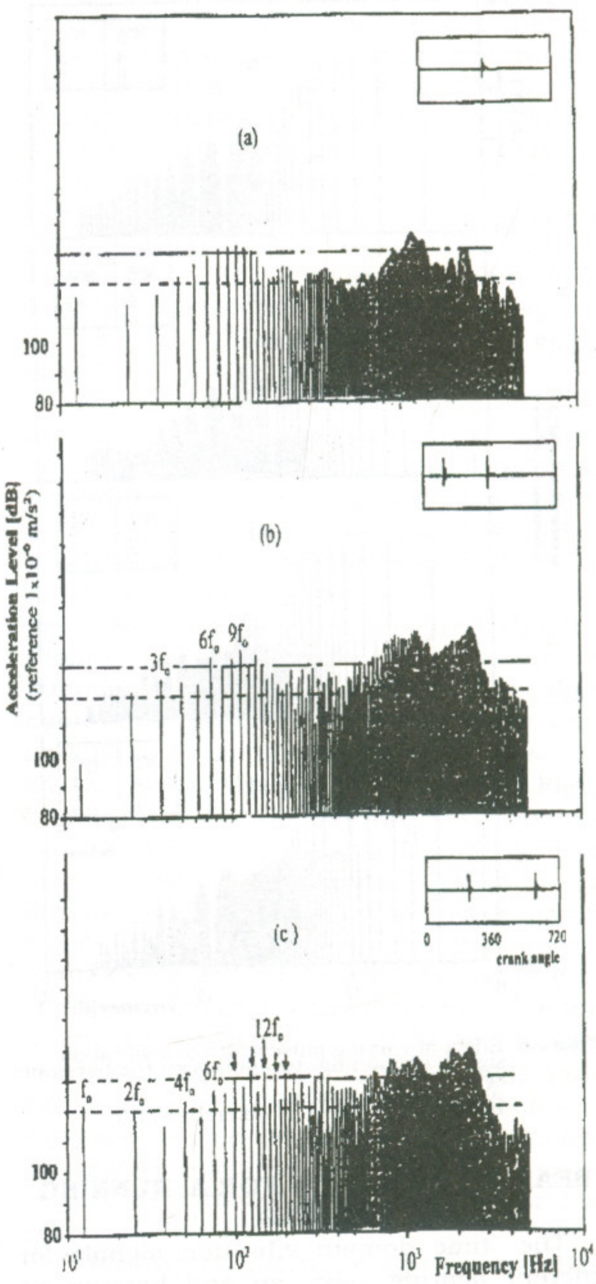

Figure 8 Effect of pulse separation interval; of actual bearing vibration, on spectral analysis

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العوامل المؤثره على التحليل الطيفى لاهتزاز كراسى عمود الكرنك

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ملخص البحث

عند عمل نموذج لإهتزاز جسم المحرك والضوضاء الصادره عنه بإستخدام طريقة العناصر المحدودة، فانه من المهم بمكان تمثيل إشارة اهتزاز كراسى عمود الكرنك حيث أنها المصدر الأساسى لأثارة الجسم الخارجى للمحرك.

لهذا كان هدف هذه الدراسة هو التعرف على العوامل التى تؤثر على إشارة اهتزاز كراسى عمود الكرنك من ناحية شكل النبضات وزمن تأثيرها وعدد مرات تكرارها.

وقد تم تمثيل النبضات التى تحتويها إشارة الاهتزاز لكراسى عمود الكرنك بنبضات رقمية مربعة وتم تحليلها مع تغير العوامل المختلفة وتمت مناقشة النواتج المختلفة.

وقد وجد أن التفاوت فى عرض النبضة وتكرارها خلال دورة كاملة من دوران المحرك هى تلك العوامل التى تؤثر على المحتوى الطيفى لكراسى عمود الكرنك. وقد وجد ان المستوى الطيفى يتناسب مع القيمة اللوغاريتمية للنبضة.