ARL-PROP-TM-468 AR-006-058

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MELBOURNE, VICTORIA

Propulsion Technical Memorandum 468

COMPLEX DEMODULATION FOR BEARING FAULT DETECTION

by

I. M. HOWARD

Approved for public release

OCTOBER 1989

AR-006-058

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SUMMARY

Vibration analysis using the high frequency resonance technique has been used successfully to detect incipient failure in rolling element bearings. This memo outlines a new method of obtaining the demodulated narrow band envelope of a bearing vibration signal from the high frequency resonant response. Previous techniques have used an analogue envelope where the raw analogue signal is bandpass filtered and enveloped before digitising. The method outlined in this document first digitises the vibration signal, performs a complex demodulation operation, then computes the envelope followed by its spectrum. Fault repetition frequencies can be identified in the demodulated spectrum of the high frequency resonant response.

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CONTENTS

1. INTRODUCTION

Incipient faults in rotating machines can be detected and diagnosed by the measurement and analysis of the vibration from the machine structure. A wide range of techniques, simple and complex, have been used in the past to detect changes in the operating condition of the machine from the vibration signature. For the detection of early failure in rolling element bearings, these have included spike energy, shock pulse, kurtosis and crest factor [1-5], and the high frequency resonance technique [6-11].

Some of these techniques have proved to be unsuccessful in certain instances. The difficulty is due to the fact that bearing faults often produce vibrations of low level which are hidden in the vibrations from other machine components, such as pumps and gears. The successful diagnostic techniques have concentrated on detecting the periodic impulses which occur when the damaged rolling elements come into contact with the other bearing elements.

In the case of defects on the bearing race elements, impacts will be generated each time a rolling element rotates past the defect. The repetition frequency of the impacts can be determined from the characteristic bearing frequency equations [4, 9]. In this regard, the high frequency resonance technique has shown significant promise in being able to discern not only the bearing condition, but the bearing element which contains the initial damage.

In the early stages of bearing failure, the impacts have a very high frequency content, >200 kHz being reported [7]. These impacts excite the various resonances of the bearing elements. By analysing the modulation about these structural resonances, the bearing impact frequencies caused by the repetitive rotation of the bearing elements over the damage can be determined.

In the past [6-11], the high frequency resonance technique has been implemented by first obtaining the analogue envelope of the band pass filtered vibration signal. The pass band is centred around a structural resonance. Some of the difficulties which have been reported in using this approach have concerned the design of the envelope detector [11]. Often a half or full wave rectifier is used followed by a smoothing circuit. The selection of these components, especially the time constant to achieve the correct decay rate of the envelope signal, is critical to the success of this technique.

In this memorandum, a digital process is presented for demodulating the vibration signal around a structural resonance, to obtain the envelope of the bearing vibration.

2. COMPLEX DEMODULATION

The demodulation of the bearing vibration signal can be performed digitally, giving a number of advantages compared with the analogue approach [11]. The major advantage is that the signal is demodulated by operations in the time domain only; no Fourier transform into the frequency domain is needed. Another advantage occurs in the filtering operation, where a simple low pass digital filter is required as opposed to a bandpass filter for the analogue approach. The flexibility in specifying the centre frequency of the structural resonance to be demodulated is also an advantage. For the digital process this is achieved by simply specifying the carrier frequency in software. For the analogue approach, the hardware filter characteristics require changing, although this could also be achieved in a software controlled system using the more expensive programmable filters.

2.1 Digital Bandpass Filtering

Complex demodulation of the structural resonance can be achieved digitally using heterodyning in the time domain [12-16]. The complete operation can be expressed as outlined below.

A vibration signal U from a rolling element bearing, and containing a response from at least one structural resonant frequency, can be assumed to have a frequency spectrum as shown in Figure 1. The measured vibration signal can be digitised by an analogue to digital converter at a fixed sampling frequency f_S. The Nyquist criterion requires that the maximum frequency content of the analogue signal should be limited to less than $f_S/2$.

Once sampled, the signal is no longer continuous and is a function of sample number rather than time. With N data points the sampled function U can be expressed as

$$\{ U \} = u(i\,T_S), \qquad i=1...N \tag{1}$$

where T_S is the sample interval. Demodulation of the vibration signal around the structural resonance shown in Figure 1 is obtained by heterodyning or frequency shifting the digitised vibration time signal. This is achieved by multiplying the signal by the complex exponential function

$$e^{-j2\pi f_o i T_S},$$

which produces a time domain signal shifted by frequency f_o. The resulting heterodyned signal is then given by

$$v\ (iT_s) = u\ (iT_s)\ e^{-j2\pi f_o iT_s},\ i=1...N \qquad (2)$$

which can be expanded to give,

$$= u(iT_s)\mathrm{Cos}(2\pi f_o iT_s)\ -ju(iT_s)\mathrm{Sin}(2\pi f_o iT_s).$$

This can be rewritten in terms of real and imaginary components to be,

$$\{\ V\ \} = v\ (iT_s) = a(iT_s)\ +jb(iT_s) \qquad (3)$$

where $j = \sqrt{-1}$. In the frequency domain, the frequency shift is shown in Figure 2.

This signal is now complex, containing both real and imaginary parts. To obtain the complex demodulated vibration, this signal is now low pass filtered. Both real and imaginary components are filtered, using a digital, zero phase filter. A frequency domain representation of the filtered signal is shown in Figure 3.

Digital filtering can be seen as the convolution of the heterodyned signal with the impulse response of the zero phase filter. The resulting function {W} is then given by,

$$\{\ W\ \} = \{\ V\ \} * \{\ H\ \}\ , \qquad (4)$$

where * denotes convolution and {H} is the impulse response of the zero phase filter. This can be rewritten using the convolution integral,

$$\{\ W\ \} = \int_{-\infty}^{\infty} v(\tau)\ H(t-\tau)d\tau, \qquad (5)$$

or in discrete terms

$$\{\ W\ \} = \sum_{k=1}^{N} v(kT_s)\ H\{(i-k)T_s\}, \qquad (6)$$

The resulting function can then be expressed as,

$$w\ (iT_s) = c(iT_s)+jd(iT_s),\ i=1...N \qquad (7)$$

separating out the real and imaginary components.

The complex signal can now be decimated in time by resampling at low frequency as the resulting signal {W} contains only frequency components between $-f_a < f < f_a$. Decimation by a factor of say ten, can be used where the bandwidth of the filtered signal is less than one tenth of the bandwidth of the original time signal. The decimation of the time signal can be achieved by multiplying the complex time signal by a unit sampling sequence {G}. This can be defined using a unit sampling function g(n) defined as

$$g(n) = \begin{cases} 0 & -\infty < n < \infty, \text{ except } n = 0 \\ 1 & n = 0 \end{cases}$$

The unit sample sequence {G} for resampling the analytic signal {W} with a sampling period T_d, can now be defined as,

$$\{ G \} = \sum_{k=-\infty}^{\infty} g(iT_s - kT_d) \tag{8}$$

or for a finite length sequence, as shown in Figure 4,

$$\{ G \} = \sum_{k=1}^{M} g(iT_s - kT_d) \tag{9}$$

The resulting function {X} after resampling is equal to the low pass filtered function {W}, multiplied by the resampling sequence {G}, shown in Figure 4, which can be expressed as,

$$\{X\} = \{W\} \quad \{G\} = \sum_{k=1}^{M} w(iT_s) \; g(iT_s - kT_d), \tag{10}$$

which results in a sequence,

$$x(kT_d) = c(kT_d) + jd(kT_d), \; k=1...M \tag{11}$$

The new sampling frequency is f_d, with a period between samples of T_d. To avoid the need for interpolation, M should be an integer factor of N. The analytic signal {X} can now be enveloped to obtain the desired function where,

$$|x(kT_d)| = \sqrt{c^2(kT_d) + d^2(kT_d)}, \; k=1...M \tag{12}$$

The transform of the demodulated signal {X} can be reshifted in frequency to obtain the high resolution frequency response if desired. However, for the analysis of the high frequency resonant modulation this is not required. The reshifting of the frequency of the signal in the time domain does not affect the envelope of the demodulate {X} but only the phase.

The frequency spectrum of the envelope function $|x(kT_d)|$ can now be computed to determine the presence of any characteristic bearing frequency components as shown in Figure 5.

The complete process of computing the envelope of the modulated bearing vibration signal is shown in Figure 6 and can be compared to the standard analogue method of computing the envelope of a band pass filtered signal shown in Figure 7.

3. EXPERIMENTAL VERIFICATION

The above method of generating the envelope of the modulation of the bearing vibration signal has been successfully implemented and used to analyse the vibration signature measured from a rolling element bearing with an implanted fault in the outer race.

3.1 Bearing Vibration Test Rig.

The experimental bearing vibration test arrangement is shown in Figure 8.

The tested rolling element bearing was located on a chuck, flexibly coupled to a 40mm diameter shaft supported by a single row spherical roller bearing and a double row ball bearing 180mm apart. The shaft was also flexibly coupled to a 0.5 HP DC motor and controlled by a Servomex controller. The calibrated thrust load was applied to the test bearing through a cantilever weight system as shown in the diagram. The principal dimensions of the taper roller bearing (NSK/NTN 30204) used in the test, are shown in Figure 9. The outer race was pressed into a bearing housing which contained a small Bruel and Kjaer accelerometer type 8309, mounted in the radial direction. The fault implanted into the outer race of the rolling element bearing consisted of a fine scribed line, as shown in Figure 10, scratched across the full width of the race surface. The width of the damage was measured to be 0.17mm.

The test bearing with the implanted fault was run in the bearing test rig at 2000 rpm with a thrust load of 295 N. The measured vibration signature was signal conditioned using a Bruel and Kjaer selectable gain and anti-aliasing low pass charge amplifier type 2635. With the low pass filter cut-off set to 30 kHz, the vibration signature was digitised at 116.2 kHz using a Metrabyte data acquisition card in an IBM compatible personal computer.

3.2 Experimental Results

A short section (10 msec) of the measured vibration signal with the bearing fault is shown in Figure 11. The two large bursts of vibration are the result of the rollers rotating past the defective outer race, the time spacing corresponding to the rolling element pass frequency in the outer race, REPFO. The frequency response of this signal is shown in Figure 12, where the first four structural resonant modes of the bearing are evident.

The fourth resonant mode occurring at 32.5 kHz, was chosen for the verification of the new complex demodulation technique. Twenty thousand data points were digitised at the 116.2 kHz sampling rate (0.172 sec, 5.7 shaft revolutions) using the IBM pc. The data were then frequency shifted in the time domain so the resonant frequency was centred on the origin. A low pass digital filter with a cut-off at 1.1 kHz was then used to remove the unwanted frequency components which creates the analytic signal. These data were then decimated in time by resampling at 10 kHz. The envelope of the bearing vibration around the 32.5 kHz resonant mode was then computed and is shown in Figure 13. The repetitive impacts can be clearly seen in the demodulated envelope from the fourth structural mode, the repetition rate being equal to the REPFO.

The frequency spectrum of the envelope was then computed using a fast Fourier transform and the result is shown in Figure 14 with the first seven harmonics of the rolling element pass frequency of the outer race clearly visible, confirming the defect position as being on the outer race. The other resonant modes of the bearing were also found to give similar results to those given above.

3.3 Discussion

This memorandum has developed a theoretical basis for computing the high frequency resonant response, in the digital domain, using complex demodulation of a structural resonance. This technique is notable in that the envelope of the narrow frequency band around the structural resonance, is obtained without transforming the time record to the frequency domain.

The experimental results verify the effectiveness of the digital demodulation technique. The characteristic defect frequency was visible both in the envelope and the resulting spectrum. The results confirm that the repetitive impacts which occur when defects appear on bearing elements will cause modulation around a number of structural resonances and over a wide frequency range.

4. CONCLUSIONS

This paper has detailed a new method of obtaining the narrow band envelope modulation of bearing resonances. Formerly, the envelope modulation has been obtained by using an analogue band pass filter, centred on a structural resonance, and enveloping before digitising and computing the frequency spectrum. The method presented and verified in this paper involves demodulating the bearing resonant response by first digitising, then frequency shifting and low pass filtering, all in the time domain, to obtain an analytic signal from whose envelope the frequency spectrum is obtained.

The notable points regarding this technique are as follows;

1. The method operates on the digital signal, analogue enveloping is not used.

2. The demodulation is performed wholly in the time domain, no Fourier transform is required to perform the demodulation.

3. A simple low pass filter operation is used instead of the previous bandpass filter.

4. Any structural resonance can be selected by a software change of the frequency shift operation.

5. ACKNOWLEDGEMENTS

The author acknowledges the provision and use of the bearing vibration test rig in the Department of Mechanical Engineering, University of Western Australia.

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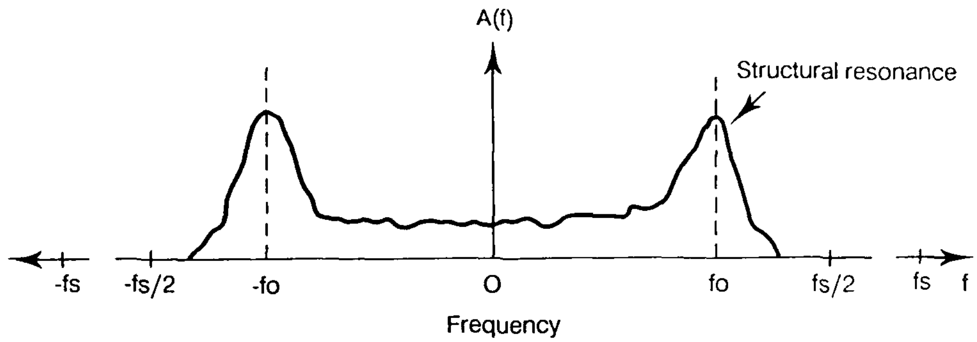


FIGURE 1. TWO SIDED SPECTRUM FROM A ROLLING ELEMENT BEARING.

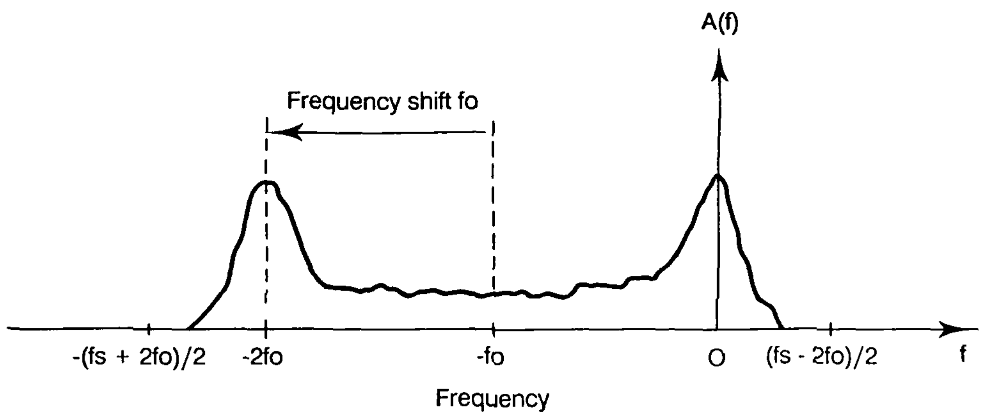


FIGURE 2. TWO SIDED SPECTRUM, FREQUENCY SHIFTED BY -fo, TO CENTRE THE STRUCTURAL RESONANCE AROUND THE ORIGIN.

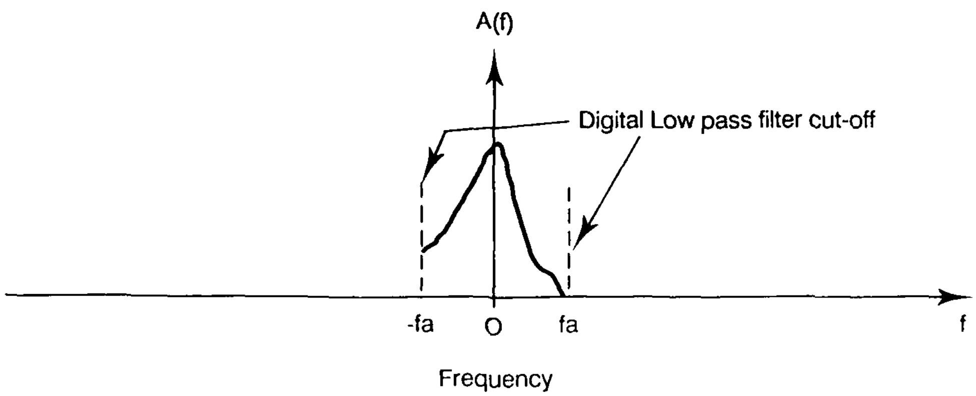


FIGURE 3. RESULTING SPECTRUM AFTER LOW PASS DIGITAL FILTERING THE TIME SIGNAL AT FREQUENCY $\pm$ fa.

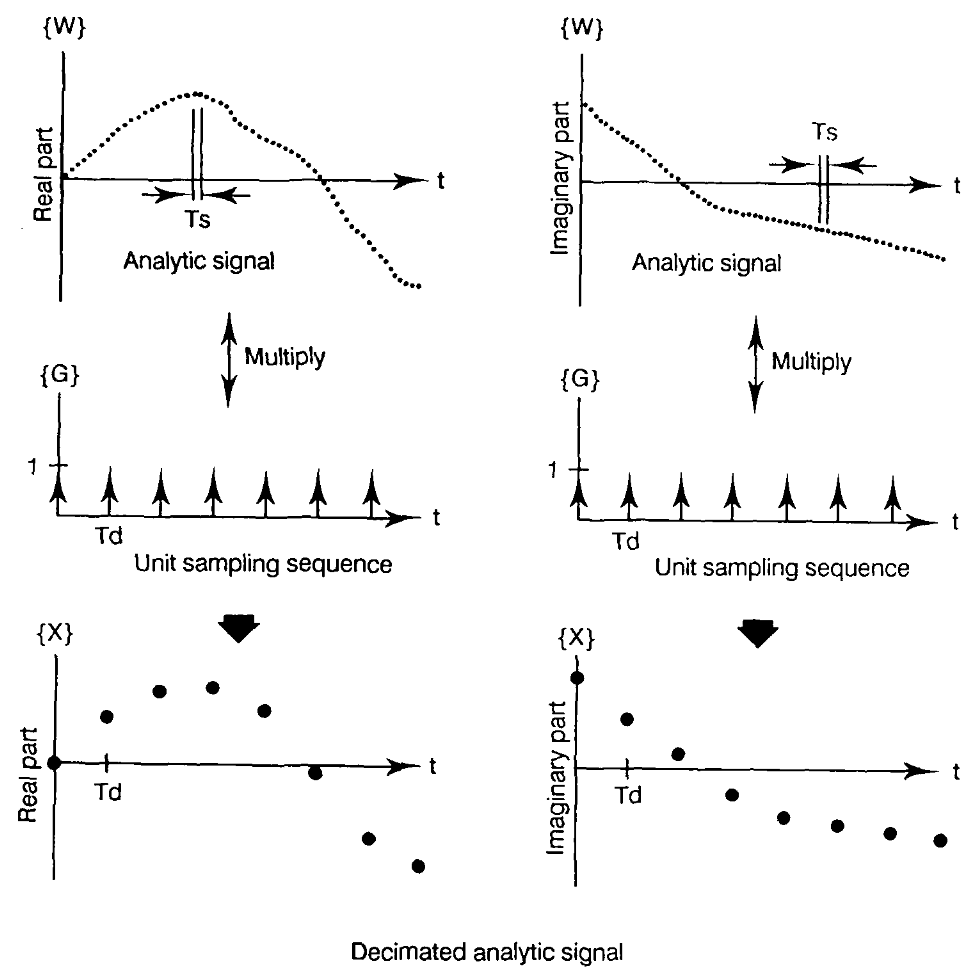
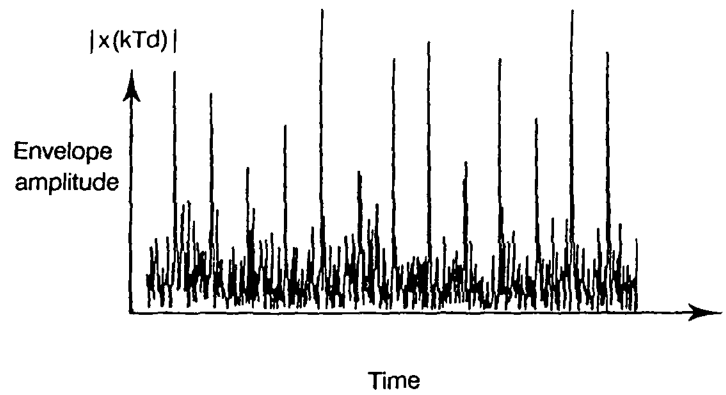


FIGURE 4. DECIMATION OF ANALYTIC SIGNAL WITH UNIT SAMPLING SEQUENCE



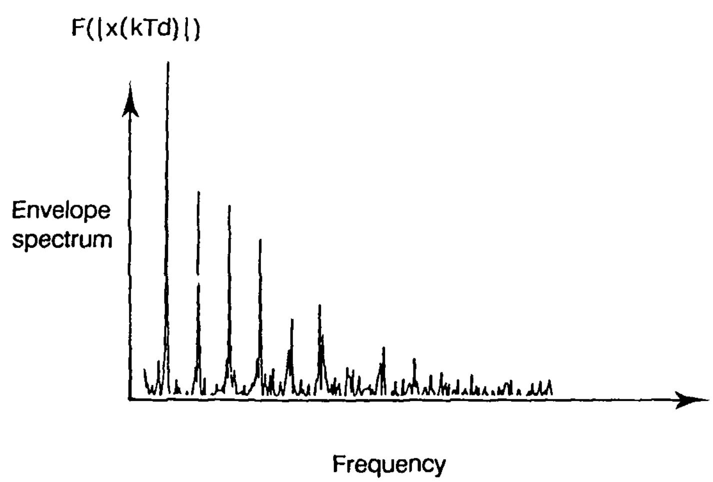


FIGURE 5. THE ENVELOPE OF THE BANDPASS FILTERED ANALYTIC TIME SIGNAL AND THE RESULTING FREQUENCY SPECTRUM.

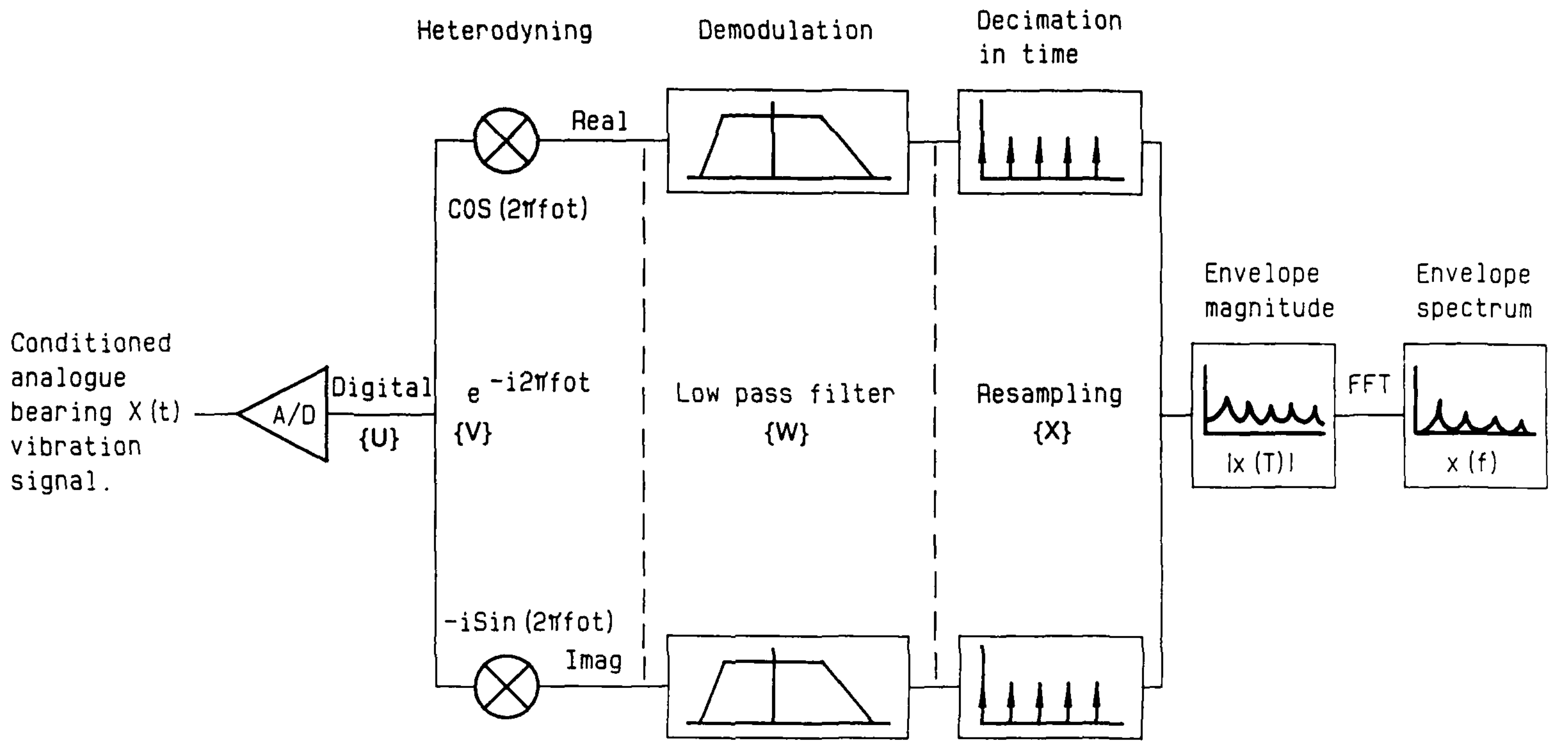


FIG 6. COMPLEX DEMODULATION TO COMPUTE THE ENVELOPE OF THE BEARING VIBRATION SIGNAL

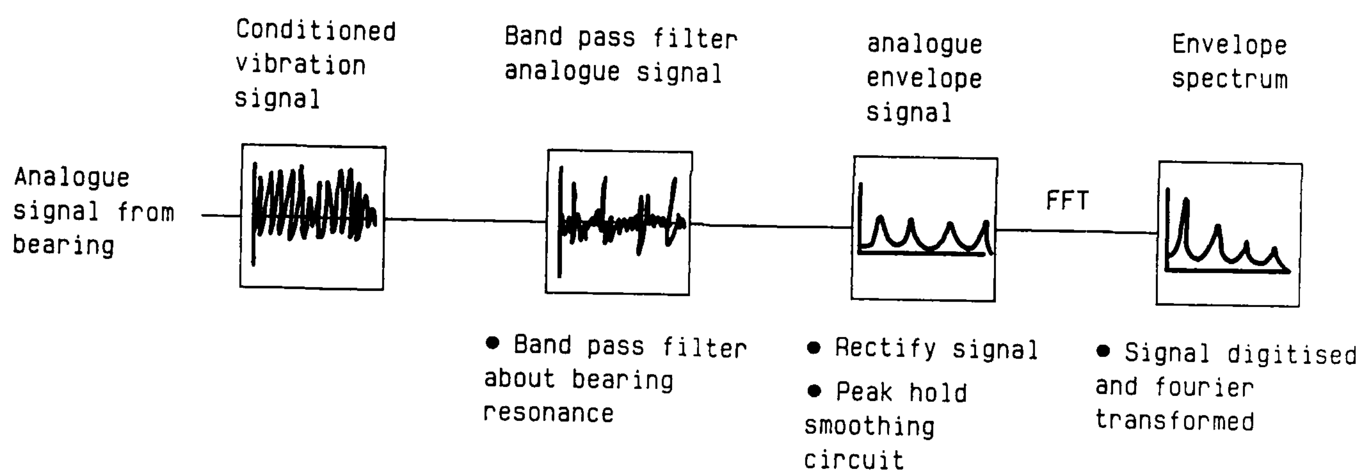


FIG 7. THE HIGH FREQUENCY RESONANCE TECHNIQUE USING AN ANALOGUE ENVELOPE CIRCUIT

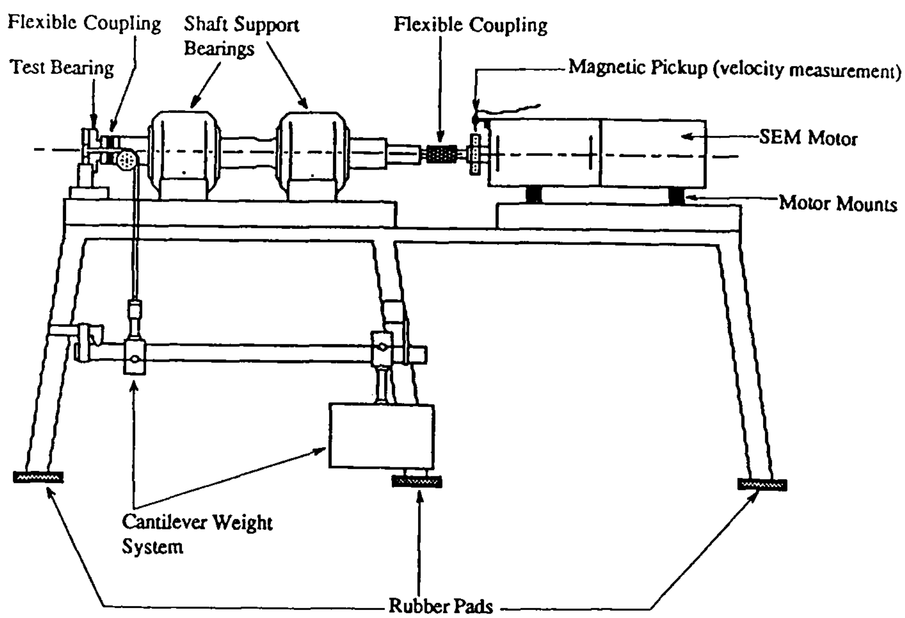
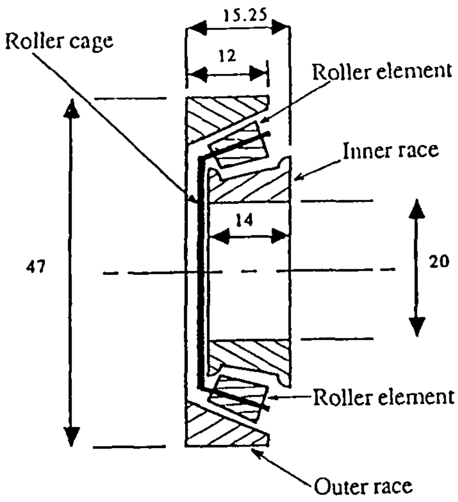


FIGURE 8. SCHEMATIC DIAGRAM OF THE BEARING RIG SHOWING THE TEST BEARING ARRANGEMENT



Pitch diameter 34 mm

Roller diameter 6 mm

Contact angle 12.96°

Number of rollers 15

All dimensions in mm

FIGURE 9. SCHEMATIC DIAGRAM OF THE TESTED ROLLING ELEMENT BEARING.

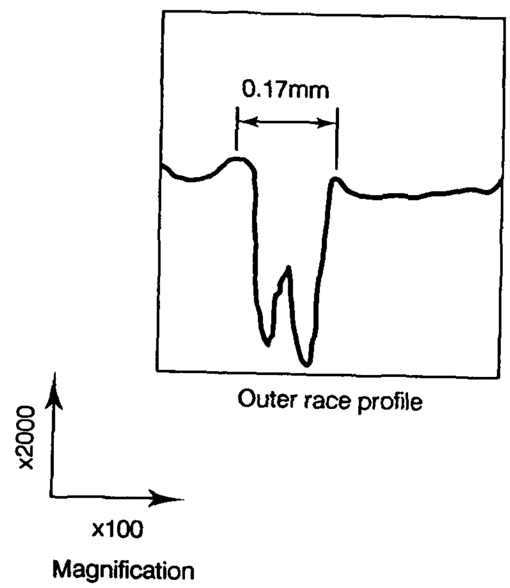


FIGURE 10. SURFACE PROFILE OF THE TEST BEARING ACROSS THE LINE DEFECT

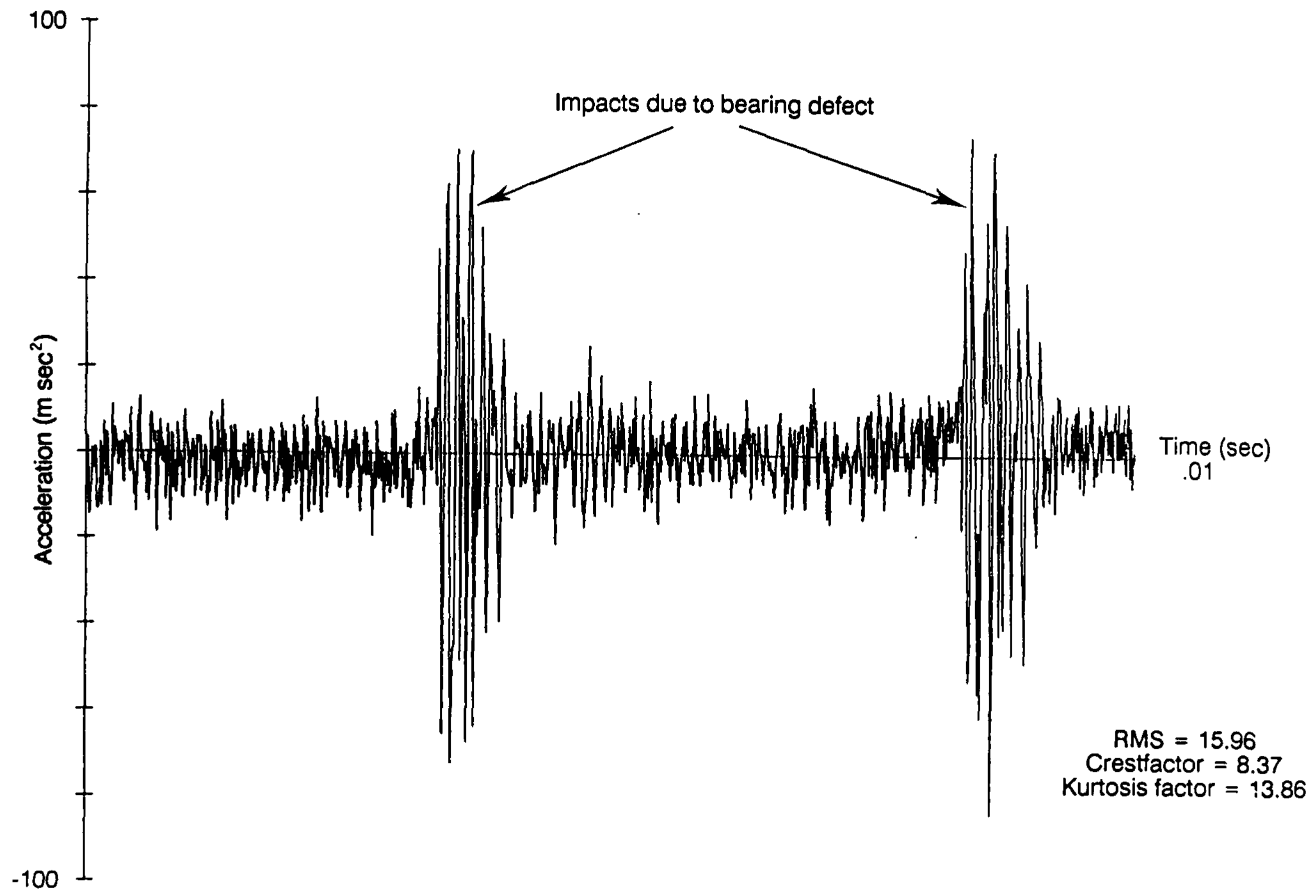


FIGURE 11. ANALOGUE BEARING VIBRATION SIGNAL WITH A LINE DEFECT IN THE OUTER RACE.

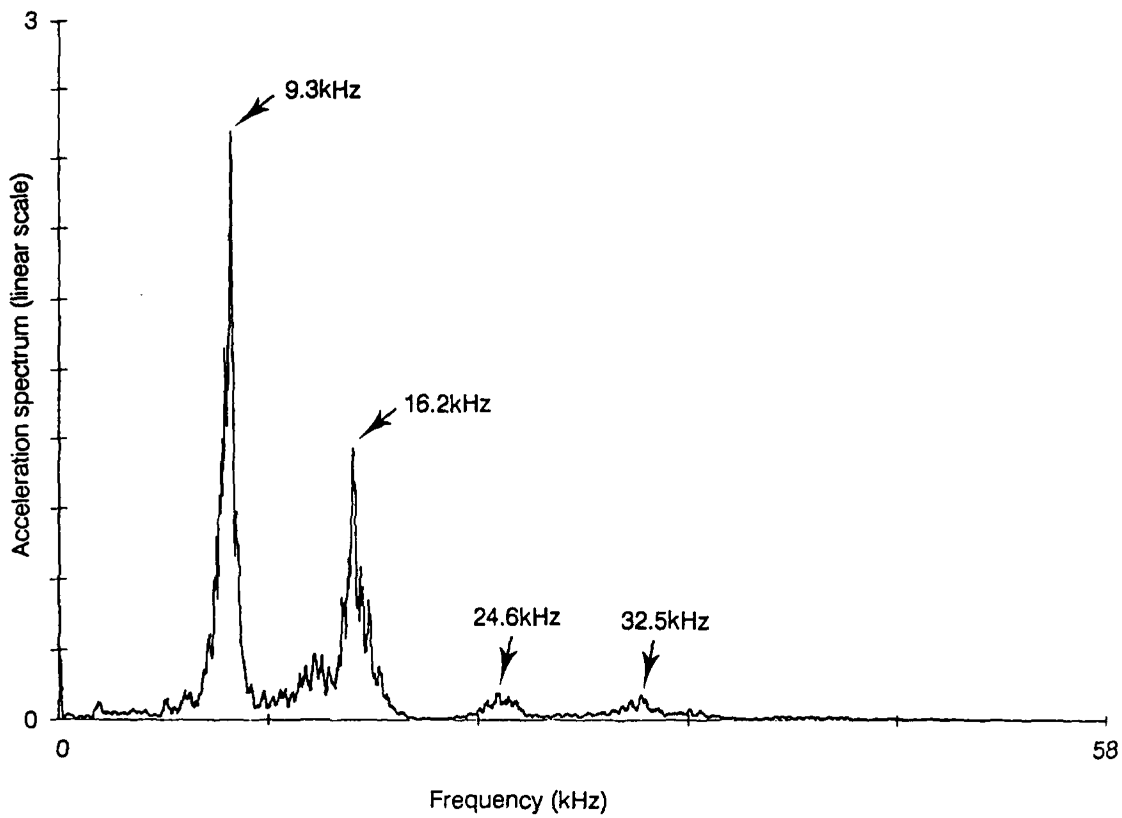


FIGURE 12. HIGH FREQUENCY STRUCTURAL RESPONSE MEASURED FROM THE TEST BEARING SHOWING FOUR BEARING RESONANCES.

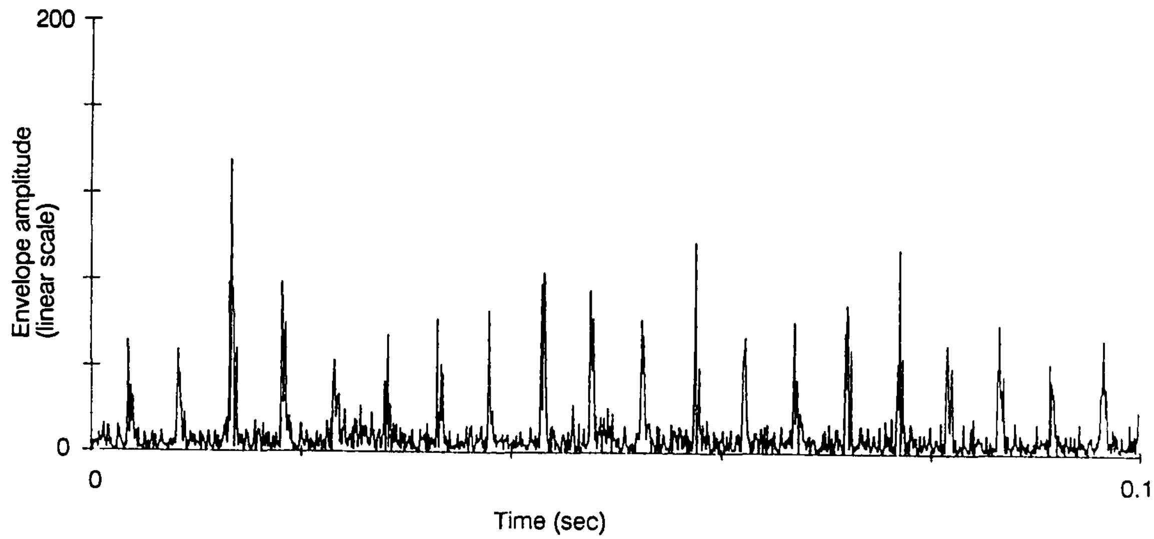


FIGURE 13. ENVELOPE OF THE DEMODULATED BEARING VIBRATION SIGNAL FROM THE 32.5 kHz RESONANCE.

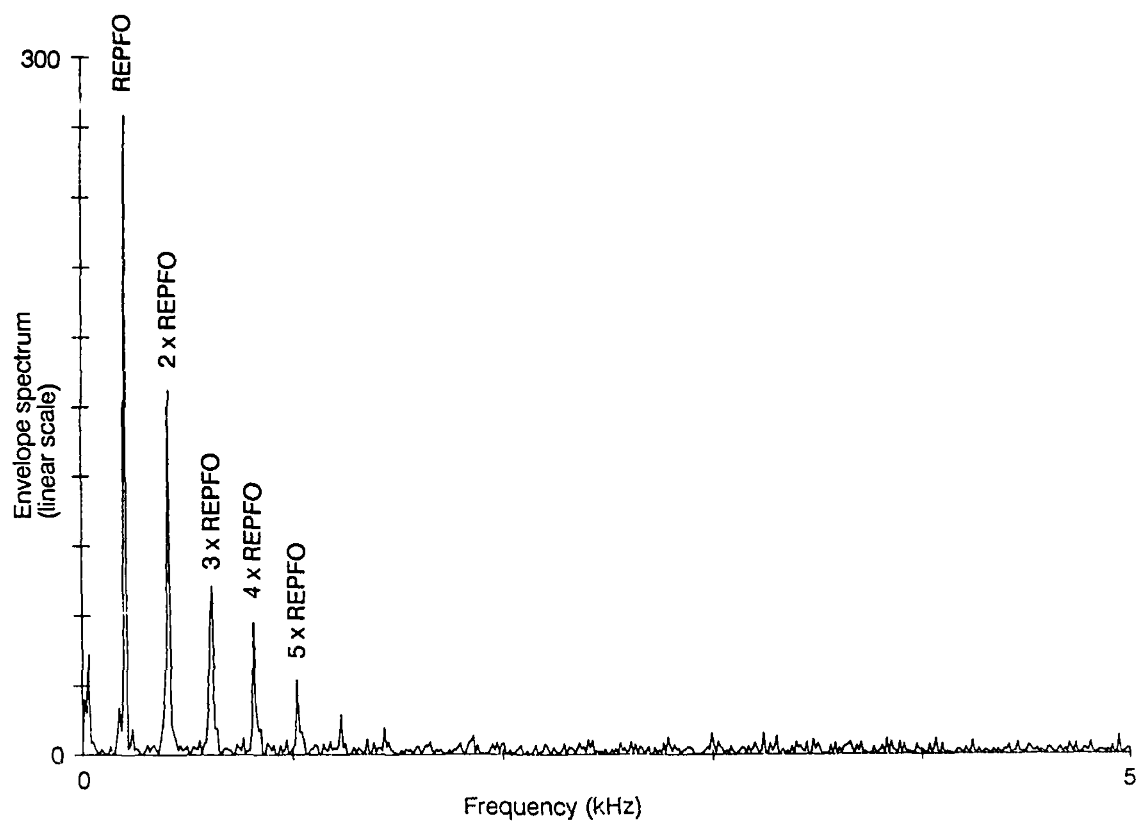


FIGURE 14. ENVELOPE SPECTRUM FROM THE 32.5 kHz RESONANCE MODE SHOWING THE ROLLING ELEMENT PASS FREQUENCY ON THE OUTER RACE.

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4. TITLE COMPLEX DEMODULATION FOR BEARING FAULT DETECTION	5. SECURITY CLASSIFICATION (PLACE APPROPRIATE CLASSIFICATION IN BOX(S) IE. SECRET (S), CONF. (C) RESTRICTED (R), UNCLASSIFIED (U)). U DOCUMENT · U TITLE · U ABSTRACT	6. NO. PAGES 22 7. NO. REFS. 16

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21. COMPUTER PROGRAMS USED		
22. ESTABLISHMENT FILE REF.(S)		
23. ADDITIONAL INFORMATION (AS REQUIRED)		