THE MEASUREMENT OF VIBRATION AS A DIAGNOSTIC TOOL

BY

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In this article the Author gives a brief introduction into the use of the vibration measurement technique as a machinery defect diagnostic tool as well as details of different types of measuring systems and methods of their application.

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Introduction

The measurement and analysis of machinery vibrations is fast becoming an accepted diagnostic tool, providing the maintenance engineer with an early

TABLE I—*List of machinery faults which it is considered may be discovered by vibration analysis*

1. Static or dynamic unbalance or eccentricity, broken impeller or rotor blades.
2. Uneven firing.
3. Worn or damaged gears.
4. Worn or damaged bearings or bearing housings. Effect of fretting corrosion.
5. Bent shafts.
6. Mechanical slackness or insecurity.
7. Onset of cavitation.
8. Shafts becoming misaligned.
9. Presence of solid bodies in a pump's fluid.
10. Incorrect re-assembly after maintenance.
11. Absence of lubricant.
12. Damaged or misaligned drive belts.

warning of failure. The techniques used today are primarily for detecting vibration which may be too slight for the operator to recognize. Relatively minor vibrations, if allowed to persist, could lead to more serious trouble and eventual failure. The technique makes use of the fact that all rotating machines vibrate to some degree or other. This is so because it is prohibitively expensive to design and build machinery which is free of vibration.

Two parameters of vibration are normally used for diagnostic work: the amplitude (i.e., displacement velocity or acceleration of the particle being measured) and the frequency at which it occurs; the amplitude of vibration giving an indication of the machine's condition, the frequency at which it occurs identifying its probable source.

Although vibration monitoring of machinery is a relatively new technique it is potentially one of the most cost-effective of the various non-destructive test techniques at present available. Correctly used as part of a controlled maintenance programme, together with other associated performance checks, including visual inspection, it can substantially reduce the number of man-hours spent on maintenance. The cost of machinery repairs can also be reduced as repairs can be planned to be carried out before catastrophic failure rather than on a time basis. Plant availability is also increased.

It can also be used as a quality control tool, reducing the number of defects built into new or refurbished machinery. A list of machinery faults which are most likely to be detected by the measurement of vibration is given in TABLE I. It will not usually detect worn or damaged plate valves in reciprocating air compressors or the fall-off in performance of pumping units, due to excessive sealing ring/impeller land clearances, or other similar defects.

Units of Measurement

Vibration

Although the amplitude of vibrations is measured in terms of various linear units, i.e., mm, mm/s, mm/s^2, etc., the range required to be covered is so wide that it has been internationally agreed to use a logarithmic representation of acceleration or velocity measurement. The system which serves as best is the 'Bel', which is the standard method of power level comparison.

As the 'Bel' is a large unit for convenience the decibel (abbreviation dB) is normally used and is defined as:—

(1) Acceleration decibel (AdB) $= 20 \text{ Log}_{10} \dfrac{a1}{a2}$

Table II—*Acceleration-decibel/velocity-decibel conversion chart (1/3-octave steps)*

Frequency Hertz	*Velocity dB to Acceleration dB*	*Acceleration dB to Velocity dB*
20	−18	+18
25	−16	+16
32	−14	+14
40	−12	+12
50	−10	+10
64	− 8	+ 8
80	− 6	+ 6
100	− 4	+ 4
125	− 2	+ 2
160	Acceleration dB	= Velocity dB
200	+ 2	− 2
250	+ 4	− 4
320	+ 6	− 6
400	+ 8	− 8
500	+10	−10
640	+12	−12
800	+14	−14
1000	+16	−16
1250	+18	−18
1600	+20	−20
2000	+22	−22
2500	+24	−24
3200	+26	−26
4000	+28	−28
5000	+30	−30
6400	+32	−32
8000	+34	−34
10000	+36	−36

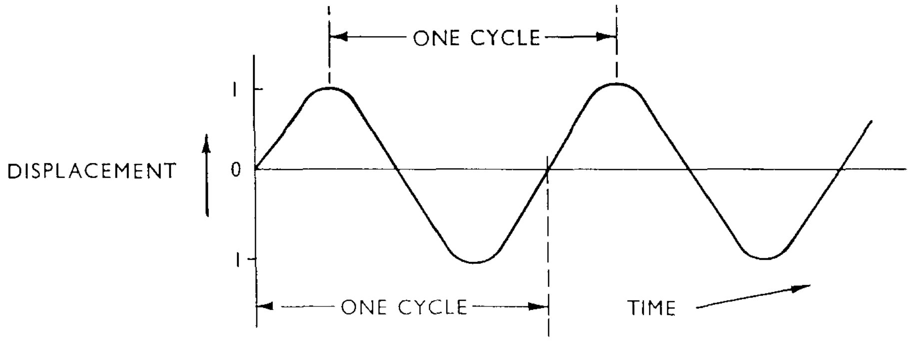


FIG. 1

where a_1 = the rms or peak measured acceleration in mm/s^2
and a_2 = the predetermined reference level, normally OdB = 10^{-2} mm/s^2

(2) Velocity decibel (VdB) = $20 \text{ Log}_{10} \frac{V_1}{V_2}$

where V_1 = the rms or peak measured velocity in mm/s
and V_2 = the predetermined reference level, normally OdB = 10^{-5} mm/s

To avoid misinterpretation of results the reference level on which the measurements are based should always be given.

AdB levels are easily coverted to VdB's and vice versa should it be so required. Conversions covering the frequency range 20–10 000 Hertz in 1/3 octave steps are given in TABLE II.

The decibel is also used in acoustics, for sound pressure measurement the reference level in air being 2×10^{-5} N/m^2.

Frequency

The standard unit of frequency is the Hertz (Hz), although cycles per minute which is directly equivalent to rpm is still used in some quarters—

60 cpm = 1 Hz

Frequency can be defined as 'the rate of repitition of a periodic phenomenon with respect to time'. Frequency (F) being the reciprocal of the period (T)—

$$F = \frac{1}{T}$$

A frequency spectrum may be sub-divided into a variety of sections termed discrete: octave or fractional octave bands, discrete frequency being at intervals of one complete cyclic function (FIG. 1).

The octave is a pitch interval of ratio 2:1

$$\frac{F_1}{F_2} = 2^N$$

where $F_2 - F_1$ = the Pass Band
and N = 1 (for octave intervals)
= 1/3 (for 1/3 octave intervals) etc.

A list of octave and 1/3 octave pass bands and mid-frequencies is given in TABLE III.

TABLE III—*Octave and 1/3-octave pass band and mid-frequencies*

Octave

Pass Band Hz	Mid-Frequency Hz
22·5	
	31·5
45	
	63
90	
	125
180	
	250
355	
	500
710	
	1000
1400	
	2000
2800	
	4000
5600	
	8000
11200	

1/3-Octave

Pass Band Hz	Mid-Frequency Hz
12	
	12·5
14	
	16
18	
	20
22·5	
	25
28	
	31·5
35·5	
	40
45	
	50
56	
	63
71	
	80
89	
	100
121	
	125
141	
	160
178	
	200
224	
	250
282	
	315
355	
	400
450	
	500
560	
	630
710	
	800
890	
	1000
1120	
	1250
1410	
	1600
1780	
	2000
2240	
	2500
2820	
	3150
3550	
	4000
4500	
	5000
5600	
	6300
7100	
	8000
8900	
	10000
11200	

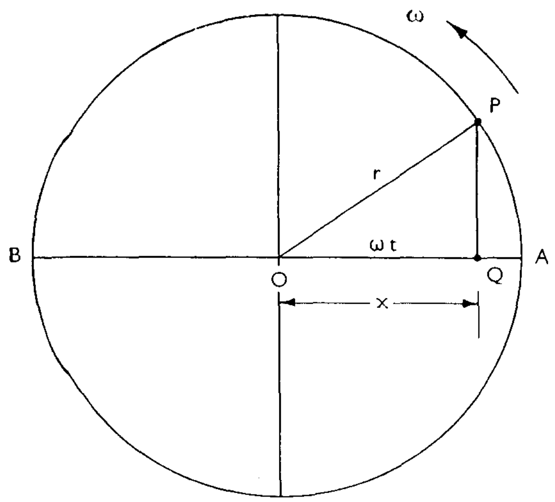


FIG. 2

Relationship between Displacement, Velocity and Acceleration

A particle displaced about a mean point has both a velocity and acceleration. The relationship between these is best illustrated in FIG. 2 as follows:—

If P rotates at radius r with uniform angular velocity ω then the projection of P on A O B (Q) is said to move with simple harmonic motion (SHM).

At any instant the horizontal component of vector r (displacement x) is equal to r Cosωt.

$$\text{The velocity of Q towards O} = \frac{d}{dt}\, r \operatorname{Cos}\omega t$$
$$= -\omega r \operatorname{Sin}\omega t$$
$$= \omega r \operatorname{Cos}\left(\omega t + \frac{\pi^*}{2}\right)$$
$$= \omega x$$

*Velocity leads displacement by $\frac{\pi}{2}$

$$\text{Acceleration of Q} = \frac{d^2}{dt^2}\, r \operatorname{Cos}\omega t$$
$$= -\omega^2 r \operatorname{Cos}\omega t$$
$$= -\omega^2 x$$

Acceleration is minimum when velocity is maximum.

Example:

A displacement of 0·001 mm rms is measured at a frequency 250 Hz (if vibration severity is measured in peak values multiply by 0·707).

displacement x $= 0{\cdot}001$ mm rms

Velocity ωx $= 0{\cdot}001 \times 2\pi \times 250 = 1{\cdot}571$ mm/s rms

Acceleration $\omega^2 x$ $= 0{\cdot}001 \times (2\pi \times 250)^2 = 2477$ mm/s² rms

$$\text{VdB rms} = 20 \operatorname{Log}_{10} \frac{1{\cdot}571 \text{ mm/s}}{10^{-5} \text{ mm/s}} = 84$$

$$\text{AdB rms} = 20 \operatorname{Log}_{10} \frac{2477 \text{ mm/s}^2}{10^{-2} \text{ mm/s}^2} = 88$$

Vibration Measuring Instruments

Each characteristic of a machine's vibration spectrum indicates some significant factor about the vibration. Amplitude, whether it be displacement, velocity or acceleration, tells us how severe the vibration is or, in terms of machinery condition, how good or bad it is. The frequency at which it occurs guides us to the possible cause. A typical machine comprises of a number of parts each rotating at a different frequency and contributing a different amount of vibration to the spectrum. Instruments for diagnostic work must therefore

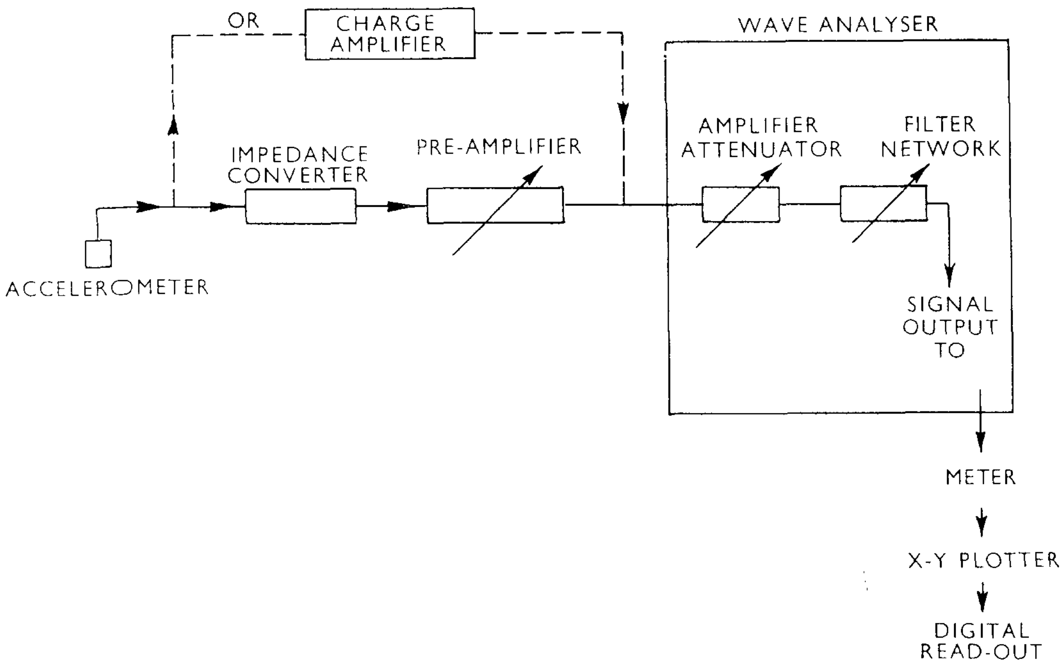


FIG. 3—BLOCK DIAGRAM OF VIBRATION MEASURING EQUIPMENT

be capable of measuring individual vibrations at their respective frequency as well as the total level. A typical block diagram of a vibration measuring system is shown in FIG. 3.

A variety of instruments are available ranging from simple total vibration level indicators to complicated custom built, computerized monitoring/recording systems. Although each has its merits the basic requirement for use in the marine or general maintenance field is a system incorporating the simplicity of the former with the accuracy of the latter.

Although not fully meeting this requirement, a number of useful instruments are available, see FIG. 4. These can be broken down into two categories:—

(*a*) Systems which measure vibration using an accelerometer as the transducer, together with an octave, 1/3 octave or discrete frequency analyser. The intensity of vibrations is normally measured in decibels.

(*b*) Systems which measure the displacement or velocity of a machine vibration in mm or mm/s respectively via a *seismic* pick-up linked to a discrete frequency analyser.

Some instruments are in fact capable of displaying the measured amplitude of vibration in each of the referred modes. The fundamental and most important difference is in the type of transducer used. The relative merits and disadvantages of using one or other of the transducers has for years been the basis of much argument.

Definition of Analysers

Discrete frequency—A tuneable wave analyser capable of measuring a vibration wave form over a specified frequency range by means of a filter network. Normally has the additional ability to measure total vibration levels.

Octave—1/3-octave frequency—A wave analyser capable of measuring vibration levels at predetermined points within the frequency range of the instrument filter network. Some instruments are also capable of measuring at discrete frequencies and obtaining total vibration levels.

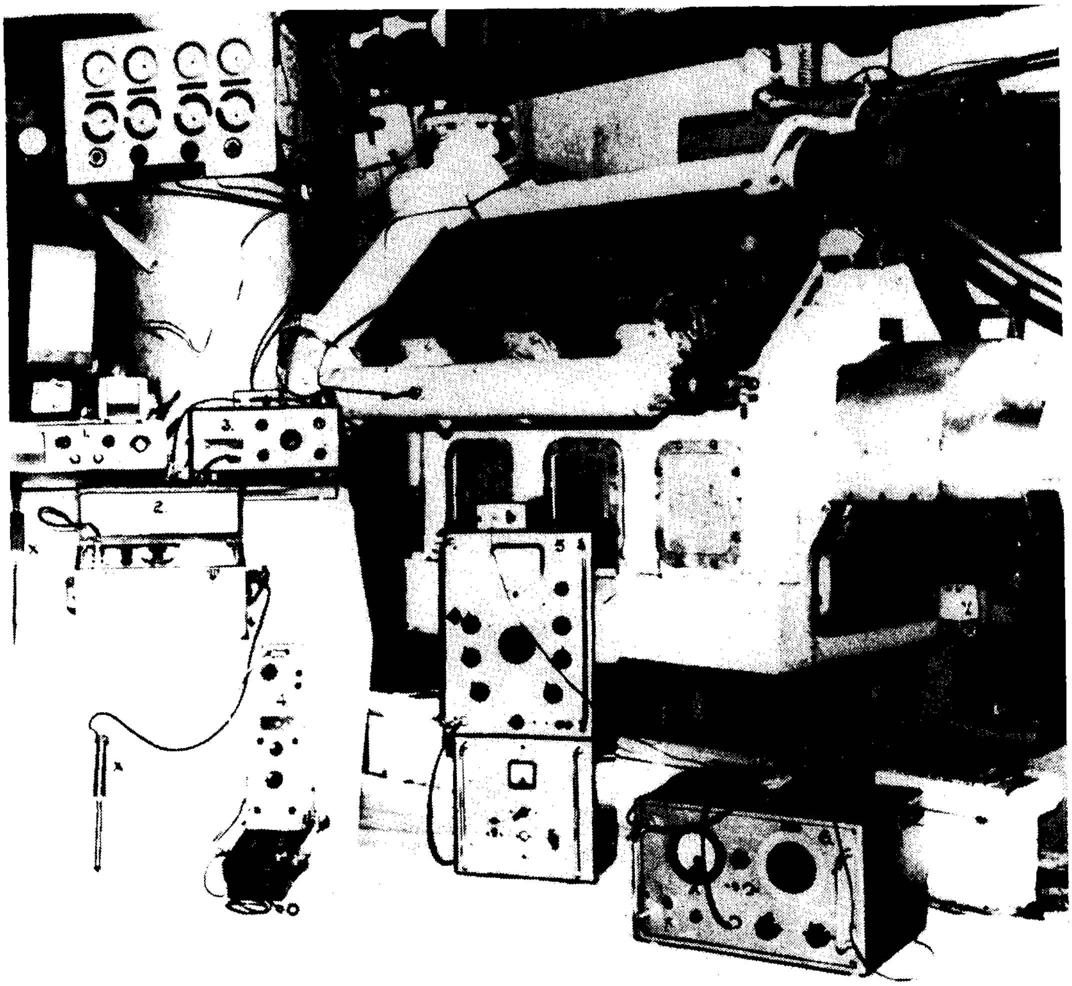


FIG. 4—VIBRATION MEASURING INSTRUMENTS

1, 2	Discrete frequency	X
3	Discrete frequency	0
4	Octave	0
5	$\frac{1}{3}$-Octave	0
6	Discrete low frequency	0
7	Total level indicator	0

(X—with velocity transducer, 0—with accelerometer)

Total level—Instruments that measure the total vibration level over a predetermined frequency range—20–10 kHz, 40–500 Hz, or some other specified range.

Velocity—Displacement

Portable analysers, together with velocity type transducer, FIG. 5(*a*), have been available for some years and have been used to great advantage as a machinery condition diagnostic tool. The transducer basically consists of a permanent magnet surrounded by a coil. The vibrations of a machine cause the magnet to move within the coil, thus converting the motions of the machine into an electrical signal. The signal produced is proportional to velocity; most instruments also indicate displacement which is achieved by feeding the signal through an integrating amplifier. The usable frequency range of this type of system is in the region of 10–1000 Hz. The range is, however, considerably influenced by the method adopted to attach the transducer to the machine. A typical advertised usable range is as follows:—

Method of attaching transducer	*Usable range-Hz*
Hand held with straight probe	10–260
Hand held without probe	10–2500*
Secured by means of a stud or bolt, etc.	10–2500*
Secured by vice type grips	10–120
Held in place by a magnet	10–620

*Upper limit may be high

The velocity transducer is in most cases used in conjunction with a discrete frequency analyser. An important fact to remember when considering the use of this combination is that regardless of the specified frequency range of the analyser, the limiting factor is the usable range of the transducer.

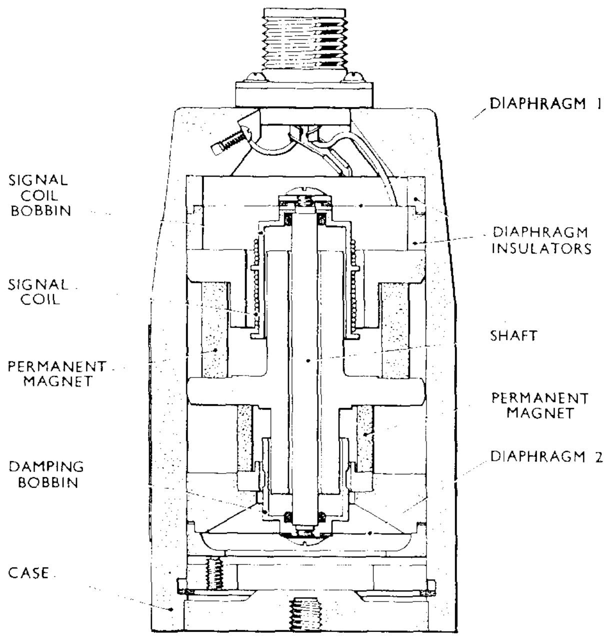


FIG. 5(*a*)—SECTION THROUGH VELOCITY TRANSDUCER

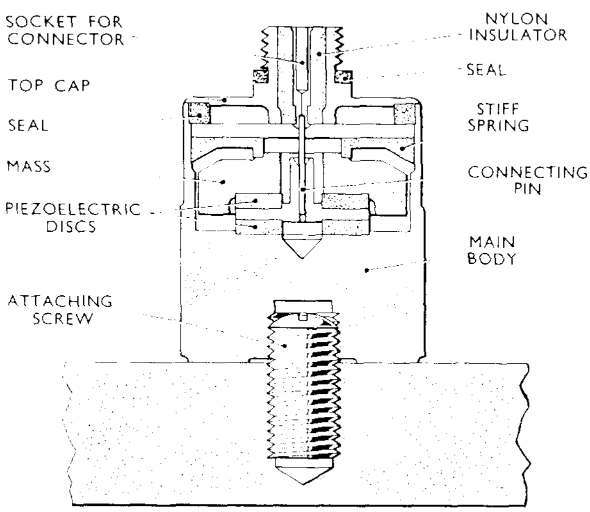


FIG. 5(*b*)—SECTION THROUGH ACCELEROMETER

Acceleration

During recent years the piezoelectric type of accelerometer, FIG. 5(*b*), has become the preferred type of vibration transducer. There are many different types of accelerometer available, each designed to meet some specific requirements. The type normally used in the field for machinery condition monitoring is termed 'compression type'. Basically they consist of a number of piezoelectric discs on which is placed a relatively heavy mass. The assembly is mounted on to a base preloaded by means of a stiff spring and sealed in a metal case.

On being subjected to vibration, the mass exerts a force onto the piezoelectric discs which generates an electrical signal directly protional to the force applied and therefore to the acceleration of the mass. The signal can also be displayed in velocity or displacement units by the use of integrating circuits which are built into some analysers.

The usable frequency range of many accelerometers is in the region of 10–40 kHz, although a 10–10 kHz range should be suitable for most machinery diagnostic work. As with the velocity type pick-up, the range is considerably influenced by the method of attachment. The most efficient

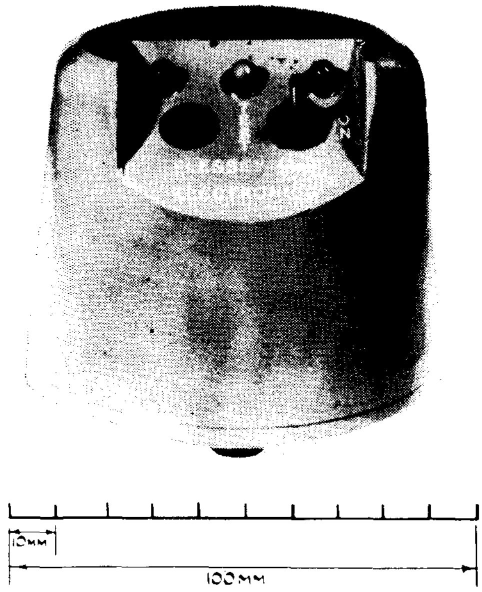


FIG. 6(*a*)—POINT MACHINERY CONDITION MONITOR

is attachment by a steel stud direct onto a rigid part of the machine.

Attachment by other methods such as special cement compounds, plasticine, magnet or hand-held probe lowers the upper frequency limit to between 2k–5k Hz.

Total Level Indicators

A recent innovation in this field is the introduction of point machinery condition monitors (PMCMs), FIG. 6. The generation of different levels of vibration resulting from mechanical deterioration is sensed by the PMCM via an accelerometer fitted to its base. The monitor sums all the vibrations in its frequency range and gives an indication of change in total level, by use of three coloured lights.

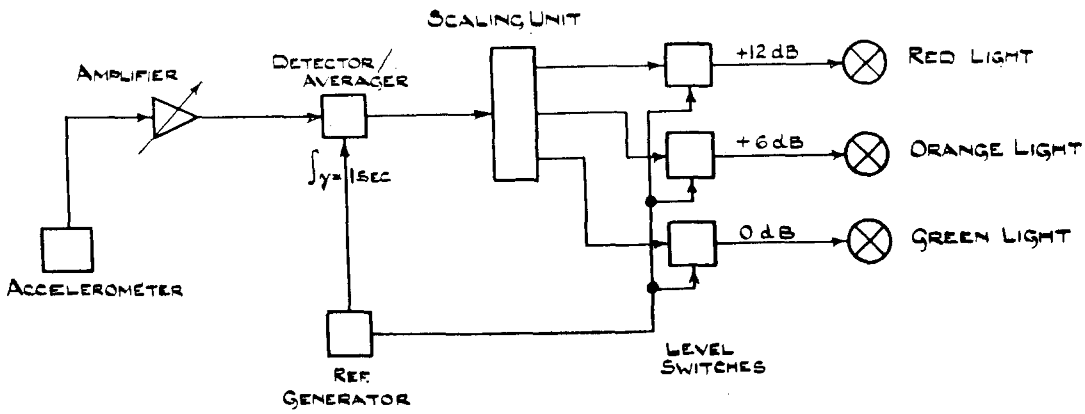


FIG. 6(*b*)—BLOCK SCHEMATIC

The total level of vibration of a machine under normal working conditions is termed the 'base level'. This level is equated to a green light condition on the PMCM. An increase in level of 6 AdB (g force doubled) causes both green and orange lamps to light. A further increase in level of 6 AdB (g force quadrupled) causes the green, orange and red lamps to light.

To establish the cause of increase in total vibration level, depending on the type of machine to which it was fitted, it would be necessary to carry out a full vibration survey after obtaining an orange or red lamp indication. Also the PMCM can only make valid comparisons of vibration if the machine operating conditions are broadly similar to those prevailing when the monitor is set up.

Vibration Measurement

Two basic methods of measurement have so far been adopted for monitoring machinery vibrations:

(*a*) In 3 planes (vertical, horizontal and axial) in the vicinity of each principle bearing and possibly other important rotating components. Readings are taken using portable analysers, and the results compared against

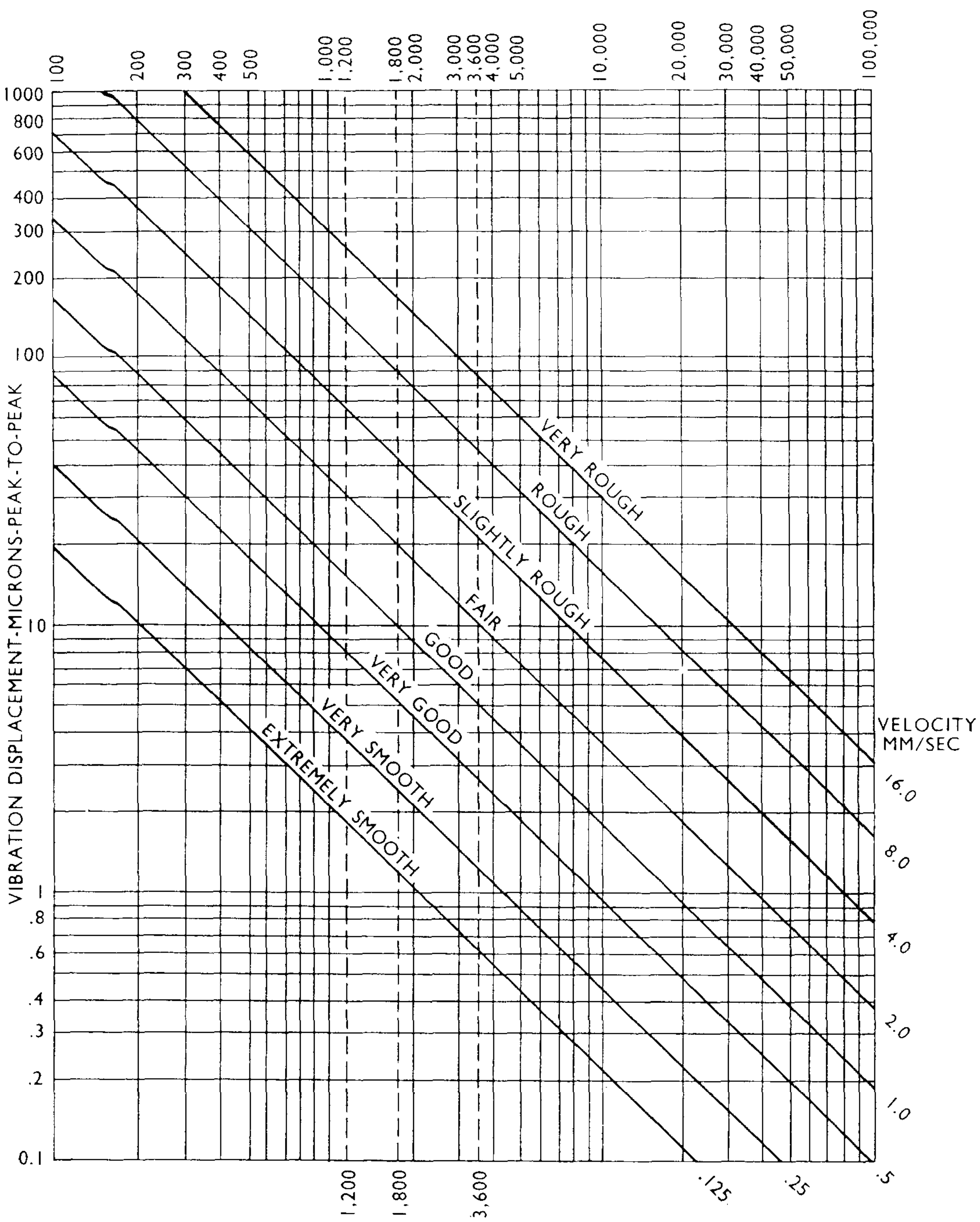


FIG. 7—GENERAL MACHINERY VIBRATION SEVERITY CHART

vibration severity charts, FIG. 7, which have been devised as a result of years of practical experience. This method is most suitable for assisting in the determination of the conditions of machinery for which no 'as new' vibration information is available, and for the diagnosis of suspected faults.

(*b*) At one or more positions on the base of a machine, above any resilient mounting arrangement that may be fitted. The principle adopted with this method is that measurement and acceptance of an 'as new' vibration signature based upon the average or maximum levels obtained from one or more machines of a particular type taken over a short period after commissioning. From this information a vibration severity envelope can

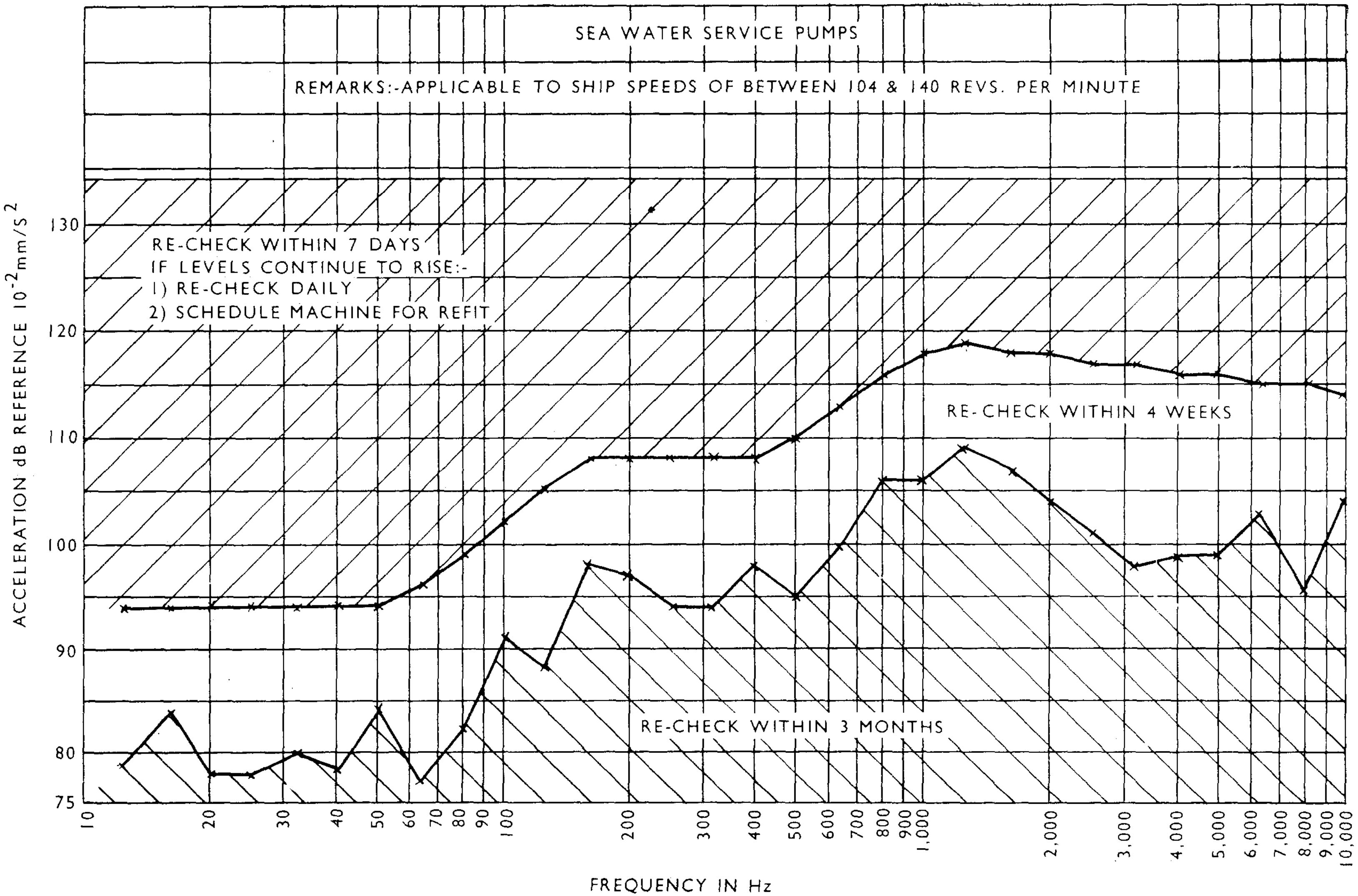


FIG. 8—MACHINERY VIBRATION SEVERITY ENVELOPE

be produced for the machine, taking into account any special knowledge of its mechanical or performance characteristics (FIG. 8).

This system has the advantage of being very flexible in as much that the envelope can be modified without affecting the vibration severity criteria of other machinery in the programme. Also the frequency at which specific faults occur, e.g., out of balance, bearing failure, etc., can be predetermined and identified, thereby reducing the analyst's work load.

Repeatability of Vibration Measurements

For vibration measurements to be meaningful and of any value in the pursuit of machinery condition analysis, it is essential that they are taken with the machine or system operating at steady conditions throughout the exercise. Similarly, if deterioration in mechanical state is to be identified by the recognition of change in the vibration spectrum, it is essential that subsequent measurements are taken with the equipment operating identical to, or as near as is practicable to, the original specified conditions.

This is no real problem with land-based machinery, where conditions can be set to meet almost any predetermined requirement and maintained thus for as long as is necessary without being unduly influenced by external sources.

The situation with ship-borne equipment is, however, somewhat different. Compressors and other motor driven auxiliary machinery can be set and maintained for reasonable periods at specified conditions. Other machinery, such as main feed pumps, forced draught blowers, extraction pumps, etc., are, however, considerably influenced by the particular operational requirements of the boiler at a given time. This problem can be minimized by maintaining as near as possible a set shaft speed while obtaining readings. The speed selected should preferably be the normal ship cruising speed, thus enabling the work to be carried out with negligible interferences with day-to-day machinery discipline.

Fault Diagnosis

Major changes in vibration levels will normally take place at frequencies associated with some mechanical or magnetic feature of the machine and its rotational speed. This being so, it is in general not too difficult for the trained analyst to interpret results and diagnose the possible cause of the change in spectrum. A list of important mechanical and electrical features which may cause vibration at various discrete frequencies is given in TABLE IV.

In the case of ball/roller bearings and gearing, the problem can be rather complicated. For example, the vibrational frequencies to be anticipated from ball/roller bearings are as follows:—

Consider a machine bearing having:

Pitch circle radius	= R
Ball/roller radius	= r
Number of balls/rollers	= n
Speed of shaft	= N rev/sec

Then,

$$f_1 = N \text{ Hz.}$$

f_1 is the shaft rotational frequency and appears at the slightest unbalance. In a normal machine the vibrational level at this frequency is due to unbalance of the rotating section to which bearing unbalance is almost always a very minor contribution.

$$f_2 = \frac{(R-r)}{2R} N \quad \text{Hz}$$

TABLE IV—*Discrete Frequency Identification*

ALL MACHINES

Cause	*Frequency*
Oil film whirl	$\frac{1}{2}$ x rotational speed
(*a*) Unbalance (*b*) Eccentric journals	1 x rotational speed
(*a*) Misalignment (*b*) Bent shaft	1 x rotational speed Sometimes 2 and 3 x rotational speed
Defective ball/roller bearings	Many times rotational speed
Defective gearing	Many times rotational speed—gear teeth x gear revolutions/second
Bad belt drives	1, 2, 3 and 4 x rotational speed of complete belt
Reciprocating forces	1, 2 and higher x rotational speed
Aerodynamic and hydrodynamic forces	1 x rotational speed or number of blades on fan or impeller x rotational speed
Mechanical looseness	2 x rotational speed

ELECTRICAL MACHINES

dc Machines: (*a*) Armature slots (*b*) Commutator Segments	 No. of slots x rotational speed No. of segments x rotational speed
Synchronous Machines: Magnetic field	 2 x supply frequency
Induction Motors: (*a*) Magnetic field (*b*) Rotor slots	 2 x supply frequency (*i*) No. of slots x rotational speed (*ii*) No. of slots x rotational speed ± 2 x supply frequency
All: Unbalanced magnetic pull	 1 and 2 x rotational speed

f2 is due to the rotation of the rolling element train and indicates an irregularity (rough spot or indentation) of a rolling element or the cage.

The spin frequency of a rolling element is:—

$$\frac{R + r}{r} \times f2 \quad \text{Hz}$$

and any irregularity of an element causes a vibrational frequency of:—

$$f3 = 2\left(\frac{R + r}{r}\right)f2 \quad \text{Hz}$$

because the irregularity strikes the inner and outer races alternatively.

$$f4 = (f1 - f2)\,n \quad \text{Hz}$$

f4 is due to an irregularity on the inner raceway.

$$f_5 = f_2 n \quad \text{Hz}$$

f_5 is due to an irregularity on the outer raceway or a variation in stiffness around the bearing housing.

f_1–f_5 are the fundamental frequencies due to the various causes and often these are accompanied by harmonics. In the case of irregularities, the more irregularities the more harmonics that are produced.

In the particular case of the upper bearing in a motor driven centrifugal pump, the bearing details were:—(*See Case History No. 1*)

Pitch circle radius	= 31·75 mm
Ball radius	= 6·35 mm
Number of balls	= 13
Speed of shaft	$= \frac{3500}{60}$ rev/sec

Thus:—

$$f_1 = \frac{3500}{60} \qquad = 58{\cdot}3 \text{ Hz}$$

$$f_2 = \frac{(31{\cdot}75 - 6{\cdot}35)}{2 \times 31{\cdot}75} \times \frac{3500}{60} = 23{\cdot}3 \text{ Hz}$$

$$f_3 = 2\frac{(31{\cdot}75 + 6{\cdot}35)}{6{\cdot}35} \times 23{\cdot}3 \qquad = 280 \text{ Hz}$$

$$f_4 = (58{\cdot}3 - 23{\cdot}3) \times 13 \qquad = 455 \text{ Hz}$$

$$f_5 = 23{\cdot}3 \times 13 \qquad = 303 \text{ Hz}$$

Also experience has shown that loss of lubricant from a bearing causes high vibrational amplitudes in the 2000–10 000 Hz frequency region.

Case Histories (Each example given concerns operational marine equipment)

Case 1

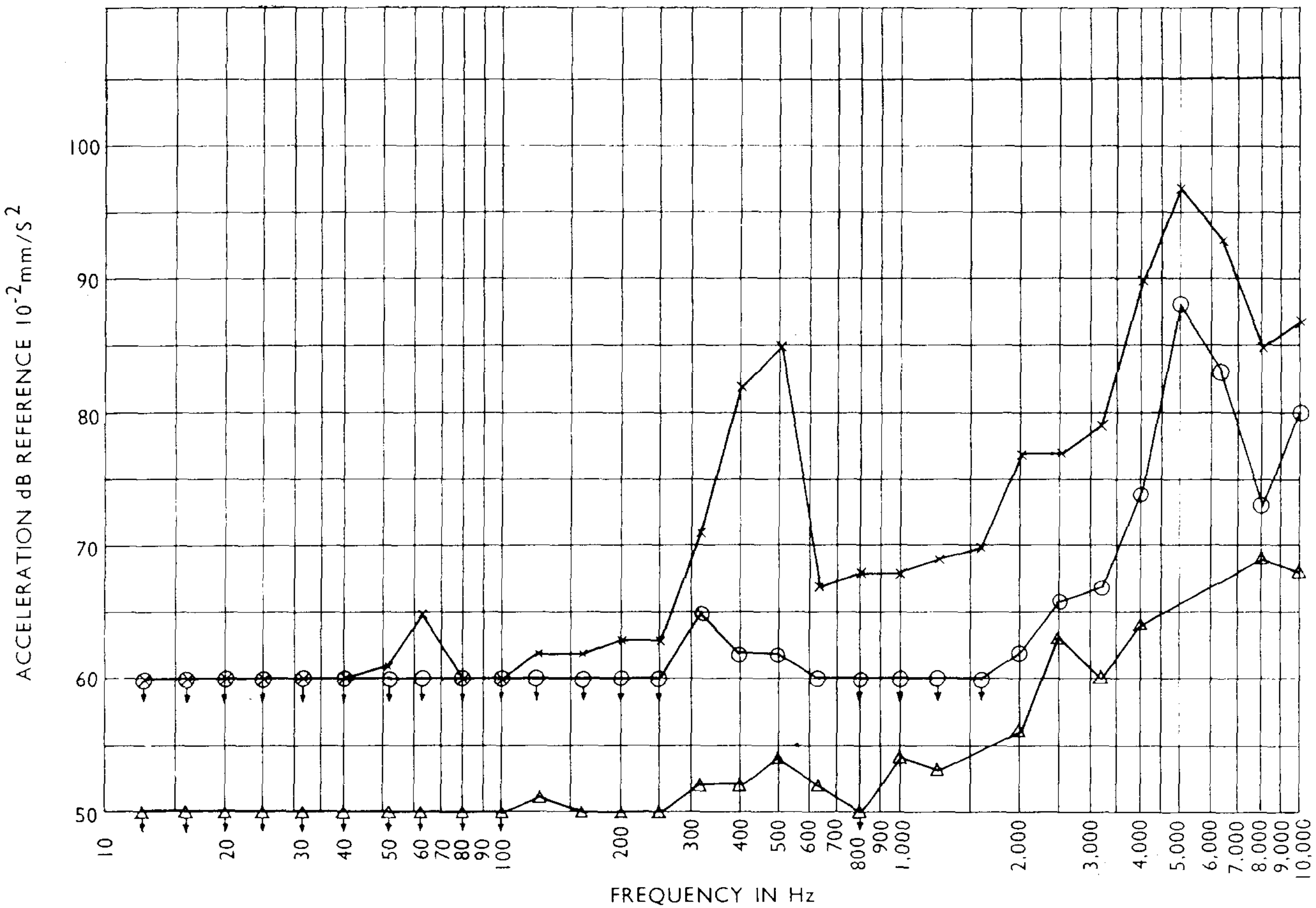
Type of machine—Centrifugal pumping unit driven by a 6·7 kW electric motor.

Vibration measurement equipment—Installed semi-automatic 1/3 octave analyser with x-y plot, print-out and using 50 mv/g compression type accelerometer.

Method of measurement—Measurements of vibration amplitude AdBs obtained at one position only on the base of the machine. The initial 'as new' readings being accepted as the datum. Measurements were taken at 2 to 3-monthly intervals.

History

After a period of running under normal operational conditions the vibration signature of the machine was observed to have changed significantly, FIG. 9A and B refer. The motor was uncoupled, run and a further signature obtained, FIG. 9C. In view of the reduction in dominant vibration levels it was concluded that the cause of the vibrational increase was in the pump end of the unit. Analysis of the dominant peaks in curve FIG. 9A indicated that the probable source of the pump vibrational frequency in the region of 450 Hz was an irregularity of the inner raceway of the pump upper bearing, the peak in the 60 Hz region indicating an increase in out of balance forces. Subsequent dismantling of the pump and examination of the bearing confirmed the diagnosis. The inner track had flaked away over approximately 1/3 of the ball path and the wear debris had indented both the inner and outer tracks, FIG. 10.

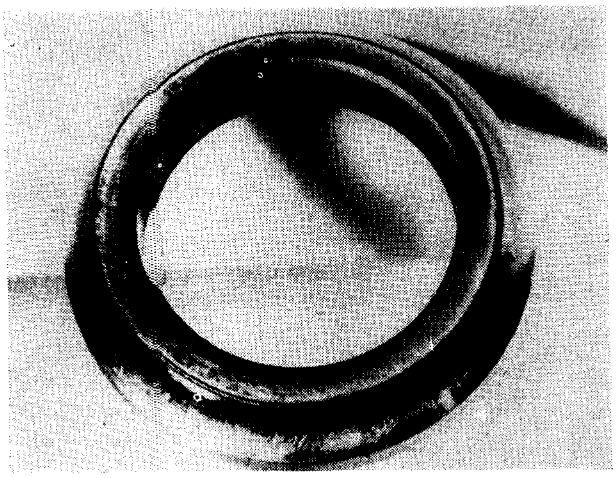


A MOTOR RUNNING COUPLED TO PUMP WITH GOOD BEARING

B MOTOR RUNNING COUPLED TO PUMP WITH DEFECTIVE BEARING

C MOTOR RUNNING UNCOUPLED

FIG. 9—DETECTION OF A DEFECTIVE BEARING IN A PUMPING UNIT



DAMAGED AREA ON TRACK OF INTERNAL RACE

PITTED BALL BEARINGS

FIG. 10—DEFECTIVE PUMP END THRUST BEARING

Case 2

Type of machine—7kW electric motor driving a lubricating oil transfer pump.

Vibration measurement equipment—Installed semi-automatic 1/3 octave analysis with x-y plot, print-out and using 50 mv/g compression type accelerometer.

Method measurement—Measurement of vibration amplitude AdBs obtained at one position only on the base of the machine, 'as new' readings being used as a datum. Measurements were taken at regular intervals.

History

After monitoring the machine's vibration signature over a period of 12 months, a significant increase in level was noted over the 2k–10k Hz frequency range, FIG. 11A and B. As the pump end of the unit was submerged in lubricating oil it was deduced that the cause of the vibrational increase was in the motor. Past experience indicating that as the increase was in the higher frequency range, the most probable cause was associated with lubrication of the motor ball bearings. Over a further period of 9 months' running the vibration levels recorded showed an increase over the whole spectrum, FIG. 11C. The pump was still operating satisfactorily and as no other indication of failure was apparent, it was decided to continue running. After a further 5 months a considerable increase in levels was recorded over the spectrum, FIG. 11D, and it was concluded that one or both of the motor end bearings was nearing failure. Subsequent investigation showed that the motor drive end bearing had suffered a loss of lubricant and that considerable damage to the inner and outer raceways and balls had occurred, FIG. 12. Severe fretting corrosion had also occurred on the outer diameter of the bearing and its locating face in the bearing housing.

Case 3

Type of machine—400 kW turbo-alternator.

Vibration measurement equipment—Portable discrete frequency analyser with velocity transducer.

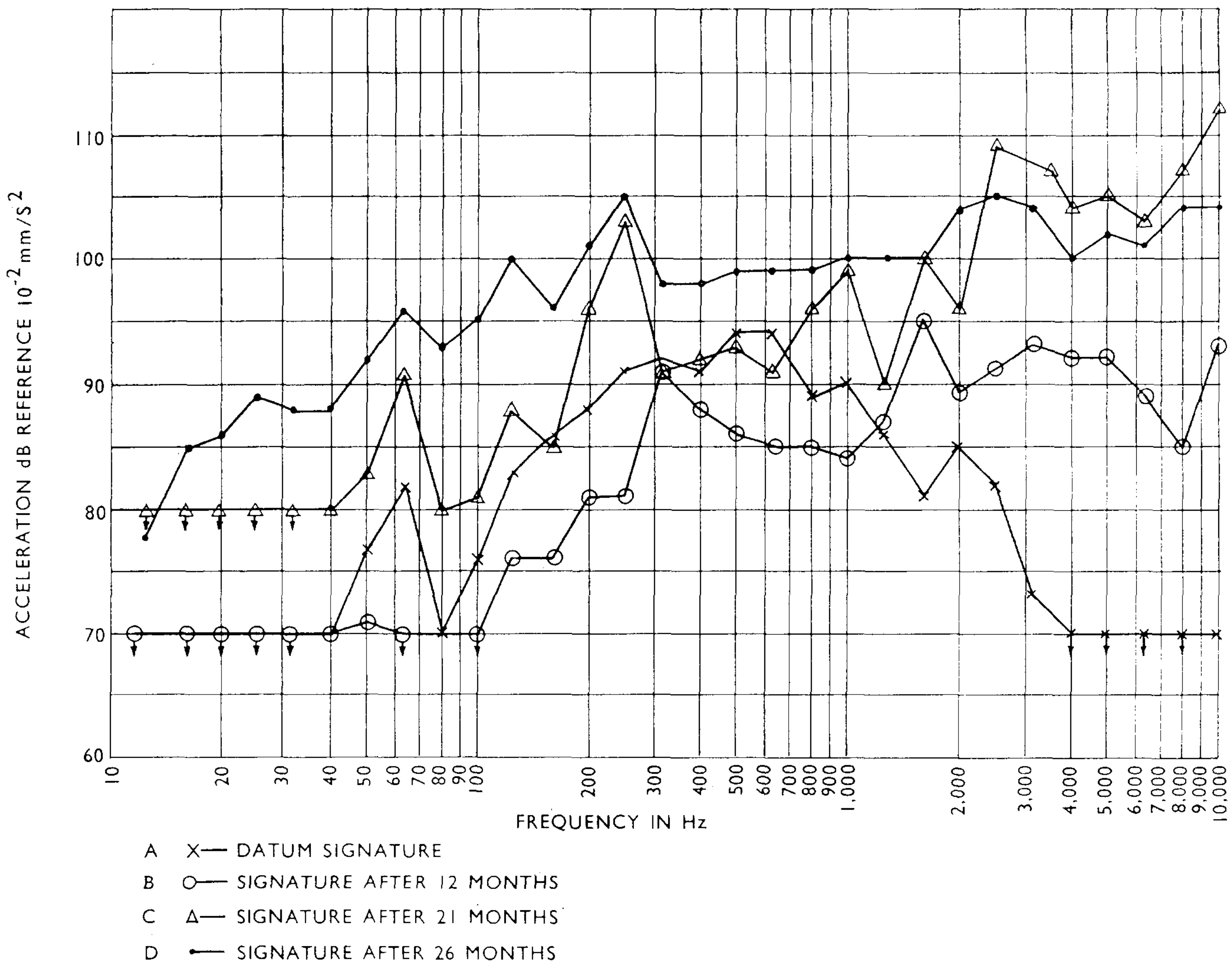


FIG. 11—DETECTION OF A DEFECTIVE BEARING IN A 7KW ELECTRIC MOTOR

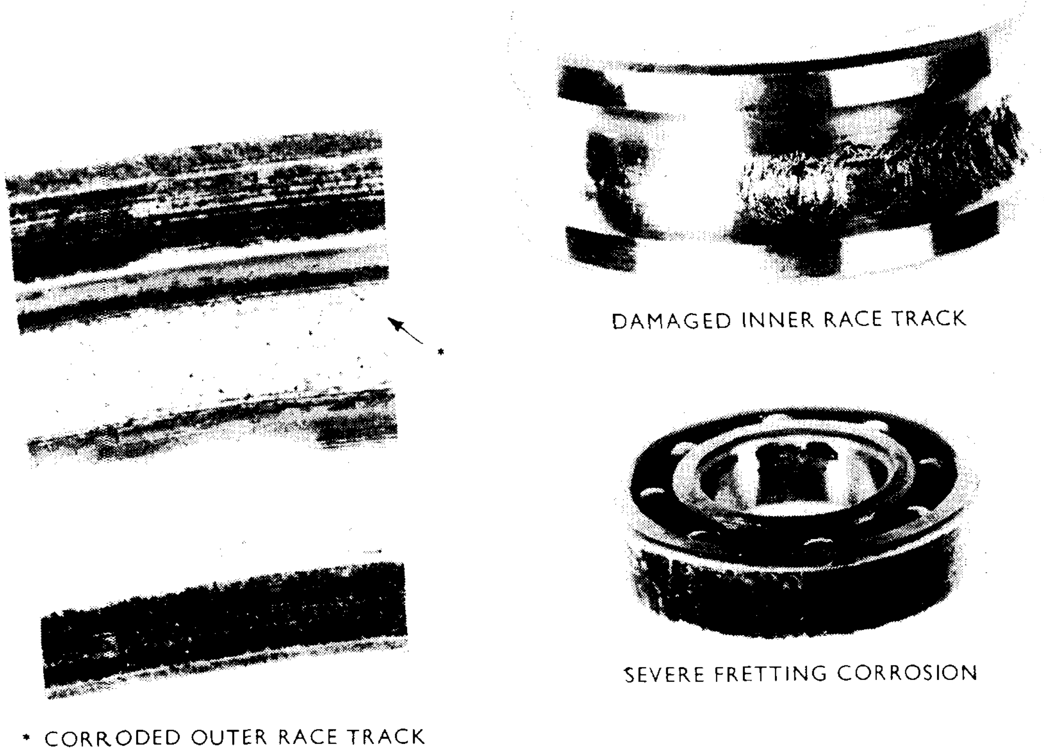


FIG. 12—DEFECTIVE MOTOR DRIVE END BEARING

Method of measurement—Vibration amplitude displacement (mm) or velocity (mm/s) measured in 3 planes in the vicinity of each bearing (Machine sketch TABLE V refers).

History

Subsequent to heavy priming of a main boiler, the boiler room turbo-alternator axial clearances were checked. Readings indicated that the rotor had moved 0·076 mm towards the steam end. After a further 21 hours' running the turbine shaft overheated in the vicinity of the exhaust end bearing. Axial clearances were again checked and indicated that the turbine shaft had moved 0·73 mm towards the exhaust end. New thrust pads were obtained and fitted and the TA run up. At 12 000 rpm there was slight noise and vibration evident, which became more pronounced as the turbine reached its normal running speed of 15 000 rpm. The source of vibration was difficult to trace and after an unsuccessful conventional examination of the machine's components, the assistance of a vibration analyst was requested.

The results of the analysis indicated that the fault lay in the turbine shaft and rotor, the worst readings being obtained at the exhaust end bearing, TABLE V. Further indications were a bent or misaligned shaft. The analyst recommended that the turbine rotor be lifted at the earliest convenient opportunity, and checked for truth, and that alignment of the turbine bearing housing be checked. It was also stated that accelerated bearing wear was to be expected until the defect was rectified.

The condition of the turbine on opening up was as follows:—

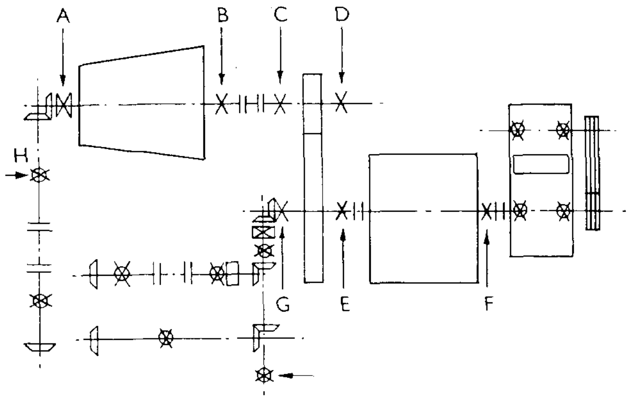
(*a*) Turbine Rotor: several sections of shrouding on the first row of moving blades missing.

(*b*) Hammering marks on the leading and trailing edges of the first and second stages of the curtis wheel.

TABLE V—*Diagnostic Vibration Sheet*

Pick-up Position		Filter Out		Filter In													
		Disp.	*Vel.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*	*Vel.*	*Freq.*
A	H	0·08	5	1	30					1	250	4	500			0·7	900		
	V	0·03	5	2·5	30					1·7	,,	2·5	,,			1	910		
	A	0·07	7	0·5	30	1	46			1·5	,,	1	,,			0·4	900	4	330
B	H	0·05	15	1	30					6	,,	4	,,			3·5	900	5	560
	V	0·02	12	1	30					4	,,	6	,,			1	,,		
	A	0·02	12	0·7	30	1	46	1·5	91	4	,,			2·5	660	1·5	,,	8	400
C	H	0·01	7	0·7	30					1	,,	2·5	,,	5	600	1	,,		
	V	0·01	3	2	30					1	,,	2·5	,,			0·6	,,		
	A	0·03	11	1·5	30					4	,,	3·5	,,	5	560	0·8	,,		
D	H	0·02	10	0·6	30			1·7	83	0·7	,,	4	,,			0·8	910		
	V		5	1·5	30					0·7	,,	3	560			1	900		
	A		6	1	30			1	91	2·5	,,			2·5	800				
E	H	0·02	3	—	—			1	83	0·4	,,	2·5	580			0·5	900		
	V	0·02	5	1·5	30					0·6	,,	2·5	500			0·5	900		
	A	0·02	6	1	30					1·7	,,			2·5	660	1·5	910		
F	H	0·02	4	1·5	30					0·4	,,	1·7	530					1	330
	V	0·03	5	0·5	30					0·5	,,	0·7	500						
	A	0·02	4	1	30						—					0·2	900	0·5	400
G	H	0·01	10	0·5	23	0·7	30				—	5	500			1·5	,,	2	350
	V	0·02	12	1	30	1	48			1·7	,,	4	500	6	660	1·4	,,		
	A	0·02	6	1	30						—			2·5	610	2	,,	1·7	330
H	H	0·02	6	0·7	23	0·5	33			3	,,	1	510			0·8	910		
	V	0·03	7	1	23					4	,,	1·7	510			0·4	900	0·6	160
	A	0·02	11	0·2	23			1·7	83	6	,,	2·5	510			0·3	900		

Displacement—mm Velocity—mm/s Frequency—Hz H—Horizontal V—Vertical A—Axial



HMS	
MACHINE LOCATION AND TYPE	
TA	BOILER RM
SERIAL No.	
SCHEDULE No.	
DATE INSTALLED	
OPERATOR	

SYMBOL	IDENTIFIES
——→	PICKUP POINT
X	PLAIN BRG
⊗	ANTI -FRICTION BRG
X̄	THRUST BRG
—⊣ ⊢—	COUPLING

DATE	1.10.70
HOURS RUN	
MAIN ENG SHAFT RPM	80
SUB-ASSEMBLY RPM	15,000/1,800

KEY TO TABLE V

(*c*) Scoring in way of gland and diaphragm labyrinths.

(*d*) Guide Blades: leading and trailing edges of blades damaged with wear on the side facing towards the steam end.

(*e*) Steam End Bearing, Exhaust End Bearing — Some wiping of white-metal

In this particular case vibration analysis identified the problem area and assisted maintenance staff in the provision of spares and the planning of the repair operation. It is considered, however, that had vibration measurements been taken sooner the expense of having a new rotor fitted to the turbo-alternator and disruption of the ship's programme may well have been avoided.

Conclusions

Early detection and diagnosis of mechanical deterioration in rotating machinery can be achieved by the measurement and comparison of changes in a machine's vibration signature. Considerable laboratory and field effort is still required if the full potential of the technique is to be realized.

A suitable data bank will aid easy and speedy diagnosis of faults, as well as provide the designer with critical component data, to enable him to improve reliability of equipment and increase mean time between failure.

Acknowledgement:

The author is grateful to the many colleagues who have contributed to or assisted in the production of this paper.

* * *

Comment by D. G. Ships

As so ably stated in Mr Carmody's paper, the correct interpretation of the vibrations of a rotating machine can, within certain limits, give good indication of its condition. Periodic vibration measurements will demonstrate changes in a machine's condition and its rate of change of condition or deterioration.

This comment is added to put this work into a naval context by discussing the present practice and the potential of Vibration Analysis and Monitoring (VAM) to meet Service requirements.