# 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. The Author was, at the time of writing, a member of the Non-Destructive Testing team at the Admiralty Marine Engineering Establishment.

### 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<sup>2</sup>, 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 Log<sub>10</sub> 
$$\frac{d1}{d2}$$

Frequency Hertz	Velocity dB to	Acceleration $dB$ to Valority $dB$
20		+18
25	-16	+16
32		+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

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



where ar = the rms or peak measured acceleration in mm/s<sup>2</sup>

and  $a_2 =$  the predetermined reference level, normally  $OdB = 10^{-2} \text{ mm/s}^2$ 

(2) Velocity decibel (VdB) = 20 Log<sub>10</sub>  $\frac{V_{I}}{V_{2}}$ 

where VI = the rms or peak measured velocity in mm/s

and V<sub>2</sub> = 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  $2 \times 10^{-5}$  N/m<sup>2</sup>.

#### 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—

$$60 \text{ cpm} = 1 \text{ Hz}$$

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

$$\frac{F_{I}}{F_{2}} = 2^{N}$$
where  $F_{2} - F_{I} =$  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

Oct	ave
Pass Band Hz	Mid- Frequency Hz
	31.5
45	i
	63
90 -	1 
	125
180 -	
	250
355 -	500
710	500
	1000
1400	1000
1100 -	2000
-	4000
5600	
	8000
- 11200	

	1/3-0	ctave
	Pass Band Hz	Mid- Frequency Hz
	12 -	
	14	12.5
	18 .	16
	22.5	20
	22 5	25
	35.5	31.5
	45	40
	56 -	50
	71	63
	89	80
	121	100
	141	125
	178	160
	224	200
	224 -	250
	202 -	315
		400
	430 - 560	500
	360 -	630
	/10 -	800
	890 -	1000
	1120 -	1250
	1410 -	1600
	1/80 -	2000
	2240	2500
	2820 -	3150
	3000 -	4000
	4500 -	5000
	5600 -	6300
	/100 -	8000
	8900 -	10000
·	11200 -	



# 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  $\omega$  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  $\cos \omega t$ .

The velocity of Q towards  $O = \frac{d}{dt} r \cos \omega t$   $= -\omega r \sin \omega t$   $= \omega r \cos \left( \omega t + \frac{\pi^*}{2} \right)$   $= \omega x$ \*Velocity leads displacement by  $\frac{\pi}{2}$ Acceleration of  $Q = \frac{d^2}{dt^2} r \cos \omega t$   $= -\omega^2 r \cos \omega t$  $= -\omega^2 x$ 

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 = 0.001 mm rms Velocity  $\omega x$  = 0.001 × 2 $\pi$  × 250 = 1.571 mm/s rms Acceleration  $\omega^2 x$  = 0.001 × (2 $\pi$  × 250)<sup>2</sup> = 2477 mm/s<sup>2</sup> rms VdB rms = 20 Log<sub>10</sub>  $\frac{1.571 \text{ mm/s}}{10^{-5} \text{ mm/s}}$  = 84 AdB rms = 20 Log<sub>10</sub>  $\frac{2477 \text{ mm/s}^2}{10^{-2} \text{ mm/s}^2}$  = 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

$\binom{1}{2}$ Discrete frequency	x	<ul><li>6 Discrete low frequency</li><li>7 Total level indicator</li></ul>	0 0
3 Discrete frequency	0	(X—with velocity transducer,	
4 Octave	0	0-with accelerometer)	
5 $\frac{1}{3}$ -Octave	0		

*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(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



	 <u> </u>	L	L		L		
iOmm							
-	 -		1001	14	 	 +	-

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	$= \mathbf{R}$
Ball/roller radius	= r
Number of balls/rollers	= n
Speed of shaft	= N rev/sec
hen	

Then,

fr = N Hz.

fI 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.

f2 = 
$$\frac{(R-r)}{2R}$$
 N Hz

# TABLE IV—Discrete Frequency Identification ALL MACHINES

Cause	Frequency
Oil film whirl	$\frac{1}{2}$ x rotational speed
<ul><li>(a) Unbalance</li><li>(b) Eccentric journals</li></ul>	1 x rotational speed
<ul><li>(a) Misalignment</li><li>(b) Bent shaft</li></ul>	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 com- plete 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

<ul> <li>dc Machines:</li> <li>(a) Armature slots</li> <li>(b) Commutator Segments</li> </ul>	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	<ul> <li>2 x supply frequency</li> <li>(i) No. of slots x rotational speed</li> <li>(ii) No. of slots x rotational speed</li> <li>±2 x supply frequency</li> </ul>
All: Unbalanced magnetic pull	1 and 2 x rotational speed

f2 is due to the rotation of the rolling element train and indicates an irregu-larity (rough spot or indentation) of a rolling element or the cage. The spin frequency of a rolling element is:—

$$\frac{\mathbf{R}+\mathbf{r}}{\mathbf{r}}$$
 × f2 Hz

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

$$f_3 = 2\left(\frac{R+r}{r}\right)f_2$$
 Hz

because the irregularity strikes the inner and outer races alternatively.

 $f_4 = (f_1 - f_2) n$ Hz

f4 is due to an irregularity on the inner raceway.

 $f_{5} = f_{2} n$ Hz

f5 is due to an irregularity on the outer raceway or a variation in stiffness around the bearing housing.

f1-f5 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:— fi =  $\frac{3500}{60}$ = 58.3 Hz  $f_2 = \frac{(31 \cdot 75 - 6 \cdot 35)}{2 \times 31 \cdot 75} \times \frac{3500}{60} = 23 \cdot 3 \text{ Hz}$  $f_{3} = 2 \frac{(31 \cdot 75 + 6 \cdot 35)}{6 \cdot 35} \times 23 \cdot 3 = 280 \text{ Hz}$   $f_{4} = (58 \cdot 3 - 23 \cdot 3) \times 13 = 455 \text{ Hz}$   $f_{5} = 23 \cdot 3 \times 13 = 303 \text{ Hz}$ 

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.





B X MOTOR RUNNING COUPLED TO PUMP WITH DEFECTIVE BEARING

C A MOTOR RUNNING UNCOUPLED

ing to the end of the second second

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' vibration running the 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.



and the second second

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.

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Pick-up Bosition		Filter	Filter Out Filter In																
ros	1103	Disp.	Vel.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.	Vel.	Freq.
	н	0.08	5	1	30					1	250	4	500	;		0.7	900		
А	$\overline{\mathbf{v}}$	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
	н	0.05	15	1	30					6	``,	4	,,			3.5	900	5	560
в	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
	Н	0.01	7	0.7	30				· · · · · · · · · · · · · · · · · · ·	1	,,	2.5	,,	5	600	1	,,		
С	v	0.01	3	2	30	1		1		1	,,	2.5	,,		· · · · · · · · · · · · · · · · · · ·	0.6	,,		
	A	0.03	11	1.5	30					4	,,	3.5	.,	5	560	0.8	.,		
	Н	0.02	10	0.6	30		-/	1.7	83	0.7	,,	4	,,			0.8	910		
D	v		5	1.5	30			\		0.7	,,	3	560			1	900		
	A		6	1	30			1	91	2.5	,,,			2.5	800				
	Н	0.02	3					1	83	0.4	,,	2.5	580			0.5	900		
E	v	0.02	5	1.5	30	<u></u>				0.6	,,	2.5	500			0.5	900		
	A	0.02	6	1	30					1.7	.,			2.5	660	1.5	910	 ! !	
	н	0.02	4	1.5	30					0.4	,,	1.7	530					1	330
F	v	0.03	5	0.5	30					0.5	,,	0.7	500						i —
	A	0.02	4	1	30											0.2	900	0.5	400
	Н	0.01	10	0.5	23	0.7	30		-			5	500			1.5	,,	2	350
G	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	0.02	6	0.7	23	0.5	33			3	.,	1	510			0.8	910		
н	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		
		Displace	ment—n	nm	Veloc	ity—mn	n/s	Frequ	iency—H	z	H—Ho	orizontal		V-Vert	tical	A—	Axial		

# TABLE V—Diagnostic Vibration Sheet

a special contraction



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 for whitemetal

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.

VAM is a recent technological advance which provides the engineer with additional data for assessing the condition of his machinery. It can be used for defect detection, for maintenance purposes, for quality assurance in production of and as a check on design. It depends for its effectiveness on a number of basic factors:

- (a) Accuracy of measurement—offering repeatability and comparability
- (b) Understanding of the limitations of instrumentation and techniques
- (c) Knowledge of the dynamic characteristics of the machine under scrutiny
- (d) Knowledge of the association of vibration effects with dynamic characteristics
- (e) A technique which is administratively and technically as simple as possible, and cost- and time-effective.

Varied and versatile instruments are available and in use. However, few are ideal, lacking something in range, accuracy or discrimination, involving excessive user time or skill, weighing too much or being too large or too costly, etc.

A balance must be struck between instrument suitability and technical requirement while taking into account the necessary training task and the need for an appropriate administrative back-up. This involves complex assessment of the varying requirements, some apparently incompatible, and it is this study which is now going on with the support of ship trials and other investigations.

In the meantime the best use is being made of equipment and techniques currently available and basic training is given to that end. Such work is technically valid and produces useful results provided that the limitations of instruments are clearly understood and taken into account, and vibration findings are supported by other performance data and machine information.

## Present Practice of VAM in Naval Service

#### The Production and Refit Vibration Test—Ashore

Manufacturers/overseers and dockyards are involved in vibration testing a large number of machines of many types manufactured or refitted for naval service. The tests are carried out in particular environmental conditions in strict accordance with specified instructions (DGS G. 10008). Nine levels of vibration are measured covering eight frequency octaves for each machine. When a number of such vibration 'signatures' for any given type of equipment have been collated, a Vibration Acceptance Level (VAL) is derived statistically for that equipment which can be used either for strict contractual purposes or for guidance in accepting or rejecting it. By this means a further quality control is established over the production and refit of machines. It can be particularly effective in highlighting out-of-balance or misalignment, which are significant production faults. (It must be noted that VALs obtained in this way cannot be used for comparison with readings obtained in the ship because of the different environment, particularly the base supporting structure, which affects vibration levels.)

#### The Type Vibration Test

This is an extensive vibration survey of a machine at the prototype stage aimed at reducing excessive vibration and airborne noise which may derive from particular dynamic characteristics of the equipment. It also establishes useful machine vibration information at an early stage. This survey is more extensive than the 'production' test and includes investigations of dominant discrete frequencies measured at various parts of the machine. The relevant specification is DGS G.10009.

#### Dockyard Vibration Surveys

Dockyards have for some time, to a certain degree, made use of portable discrete vibration analysers to check the condition of particular machines, usually before a ship comes in for refit. By this means it has been possible to eliminate or reduce planned routine work which has been shown to be unnecessary. Vibration checks have also been carried out after refit to verify that a machine is in good working condition.

This assessment of machine condition from a single set of readings, the 'absolute' as opposed to 'comparative' method, demands from the users a formidable combination of diagnostic skill and confidence, and should be backed by clear supporting evidence from performance checks and machine history where possible.

#### Operational Use of VAM—Afloat

Some submarines have a semi-installed machinery vibration monitoring system which is used at stated intervals to take vibration measurements providing  $\frac{1}{3}$  octave 'signatures'. Successive 'signatures' are compared to show if any significant change has occurred in machine vibration levels. By this 'comparative' method a clear-cut indication of deterioration and its possible cause can be obtained. The system has on at least two occasions under operational conditions shown up faulty roller bearings in time for remedial action to be taken to avert catastrophic failure. The system is also capable of discrete vibration analysis if required.

Surface Fleet and Submarines. Commander-in-Chief, Western Fleet, and Submarine Squadron Staffs are equipped with portable discrete analysing equipment so that surveys of ships' equipments and trouble-shooting investigations of particular machines can be carried out as required. This work has produced useful results but, as with dockyard staff, this technique imposes considerable diagnostic burdens on the users.

# **MOD(N)** Support

### Vibration Analysis Techniques (VAT) Sub-Committee

This body considers the use of VAT in the Service, advising on uses, methods, instrumentation, units of measurement, and instigating trials and investigations. It is made up of representatives from DG Ships, CED, CFS, DGA, DGW, C.-in-C. WF, and SMA, with scientific and laboratory backing, and attempts the co-ordination of all the varying views and requirements.

# Admiralty Engineering Laboratory and Admiralty Marine Engineering Establishment

The AEL and AMEE support the work of the departments and the Fleet by scientific investigations into instrumentation and techniques, and by the collation of vibration information and its interpretation. AEL have sponsored the use of  $\frac{1}{3}$  octave vibration monitoring and analysis.

# Trials

#### Discrete Analysing Technique with IRD Instruments

A series of small-scale ship trials sponsored by DFM/SMA in Galatea, Scarborough, Tenby, Bacchante, London, Charybdis, Minerva and H.M.S. Fife has been underway for two years. Periodic vibration data has been collected on a number of machines and this, together with other information, has been used to assess the condition of these machines. These trials have substantiated the usefulness of this form of vibration monitoring while highlighting some of the difficulties that attend its use.

#### Overall Vibration Monitor—the PMCM

This instrument, the Point Machinery Condition Monitor, is a simple and coarse indicator of increased vibration level in a machine which was developed by DG Ships with Plessey, Limited, and has had successful trials in H.M.S. *Galatea.* It is now being used as the basic vibration monitor in the H.M.S. *Phoebe* trial. The PMCM will also be used in a submarine trial.

# Extensive VAM and Maintenance Trial in H.M.S. Phoebe

The *Phoebe* trial makes use of the instrumentation and techniques described in the previous two paragraphs. It is an elaborate and long term investigation, not only concerned with validation of techniques and the diagnosis of defects but mainly with an attempt to assess the rate of deterioration of a wide range of machines. If this can be done by vibration measurement while the machine continues to operate, much saving in time, cost and labour may be made by the elimination of unnecessary routine strip-down maintenance and routine overhaul.

Of its nature, this is a long-term trial which cannot properly be judged until its completion and when its results have been analysed. Early indications, however, are that it will in any event justify greater use of VAM in the Fleet\*.

# Instrumentation

Very broadly, instrumentation falls into two main categories:

- (a) The easily portable vibration measuring instrument with a hand-held probe incorporating a velocity transducer, usually of the discrete frequency analysing type, but sometimes reading octave or overall levels.
- (b) The less portable, often partially installed, equipment using  $\frac{1}{3}$  octave read-outs and incorporating the use of an attached accelerometer.

The former, because they are easily portable, readily available, have a capacity for probing at any external part of a machine, and can make use of a standard vibration 'severity' chart, have to date been the more generally used. However, the hand-held velocity transducer has a very limited frequency response and is subject to inaccuracies from contact resonance and magnetic field interference.

The latter has not the attractions listed above and indeed sometimes requires some installation involving wiring and attachment points on machines. However, the attached accelerometer is preferable to the hand-held velocity transducer, having a linear response up to and beyond 10kHz (10 000 cycles per second) and providing reliable and comparable measurements. For this reason, and because when used with a  $\frac{1}{3}$  octave read-out it offers simplified interpretation, this type of instrumentation is becoming more favoured where the rate of machine deterioration is the prime interest.

Some instruments, such as the Admiralty Pattern Instrument SPM2, have features from both of the above broad classifications but frequently are made for a limited purpose.

<sup>\*</sup>It is hoped to publish an account of these trials in the next issue of the Journal.

# Future Practice of VAM in Naval Service

The greatest need for really effective progress in VAM lies in the Fleet, so that ships can be operated as planned and without breakdown, and so that maintenance work and refits can be cut to a minimum. The best basis for overall advance appears to be the use of a technique which has a capacity for monitoring the rate of machine deterioration and for analysing its condition—what has been referred to as the 'comparative' method, using a  $\frac{1}{3}$  octave monitoring system.

Once a decision has been made about the main path of advance and the extent to which VAM should be used, arrangements must be made for the provision of suitable equipment, training of personnel and the correct administrative back-up.

None of these things will be easily or quickly decided. Until such time as there is a full working organization we must continue to make the best of current vibration analysis techniques, instrumentation and expertise while remaining aware of their limitations.