THE EFFECTIVENESS OF VIBRATION ANALYSIS AS A MAINTENANCE TOOL

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The Canadian Forces use the portable octave band analyser as a machinery health monitor as part of their preventative maintenance programme on ships and aircraft.

Velocity measurements are taken near each principal bearing on a machine, on an overall and octave band basis. These readings enable the operator to diagnose rapidly any serious deterioration due to unbalance, misalignment, bent shafts, defective gears or bearings in the machine, because the overall velocity level defines whether a machine is in good condition, or not, and the octave band analysis enables the nature of the fault to be determined.

To take machinery health measurements, monitoring points are welded at standardized positions on each machine and vibration measurements are taken periodically at specified operating conditions. The state of the machine is determined by comparing these measurements with the machinery norms and with the previous readings at each point. These operations are computerized.

A cost effectiveness study shows that unscheduled maintenance during refits and machinery breakdowns during operational periods, have been reduced by 45 per cent. The reduction of repair costs and the improved ship availability are worth £800 000 per year over a fleet of 20 destroyers

INTRODUCTION

The Canadian Forces have developed a machinery health monitoring programme in ships and on aircraft, based upon the use of the portable octave band noise and vibration analyser as a vibration monitoring tool. In ships, this non-destructive testing technique is in general use prior to refit to help determine the machinery overhaul requirements and, after refit, to help determine the acceptability of repair work. As a result, repair costs and ships' outage costs because of breakdown at sea have been almost halved. Unscheduled maintenance work on machines in the vibration analysis (VA) programme, has been similarly reduced. Total savings effected are over $100 000 (£40 000) per ship per year.

At specified overhaul shops, VA is now being applied during the test runs after machinery overhaul with marked success, and a programme for periodic VA monitoring of machinery during the ships' operational periods has undergone sea trials and will shortly be brought into general use. On aircraft, the tool is presently used for specialized tasks such as checks on the alignment of engine drive trains and gearbox vibration levels. Fig. 1 illustrates a survey in progress on shipboard equipment.

The present use of the instrument has developed from an evaluation of the potential of vibration analysis as a machinery defect indicator, conducted by The Naval Engineering Test Establishment (NETE) for the Sea element of the Canadian Forces by Glew & Watson[1].

In this paper the factors involved in implementing octave band analysis as a machinery health monitor are summarized. Then follows a review of the programmes in use or about to be implemented in the services, and finally a cost and value effectiveness analysis of these programmes is presented.

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FACTORS IN IMPLEMENTING OCTAVE BAND ANALYSIS AS A MACHINERY HEALTH MONITOR

During the experimental surveys carried out for the CF(S) it was realized that successful diagnosis of machinery faults at

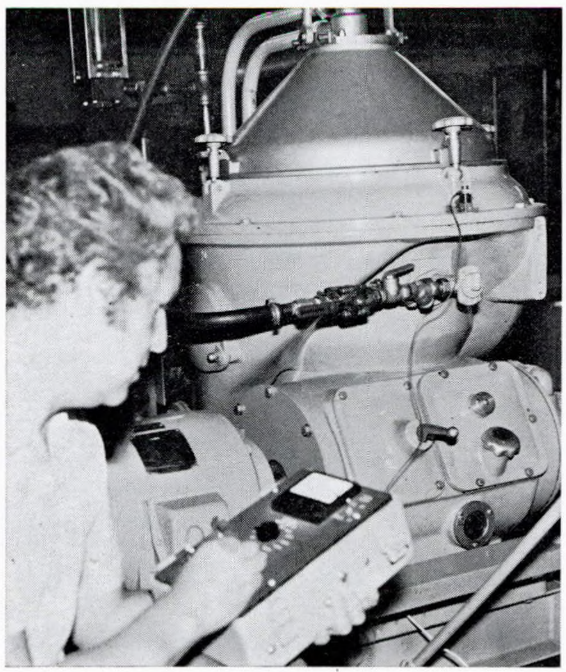


FIG. 1—*Octave band analyser in use*

sea depends upon the following seven factors, which have very wide applications.

The Velocity Parameter is a Good Indicator of Machinery Health

Empirically it has been found that the destructive power of a vibration is related to its r.m.s. velocity, at both high and low frequencies. Fig. 2 is reproduced by courtesy of the IRD Mechanalysis Corporation. It gives the relationship between vibration (displacement and frequency) and machinery condition which has been found generally applicable for electrical and mechanical rotary machines. The lines defining machinery condition are lines of constant velocity.

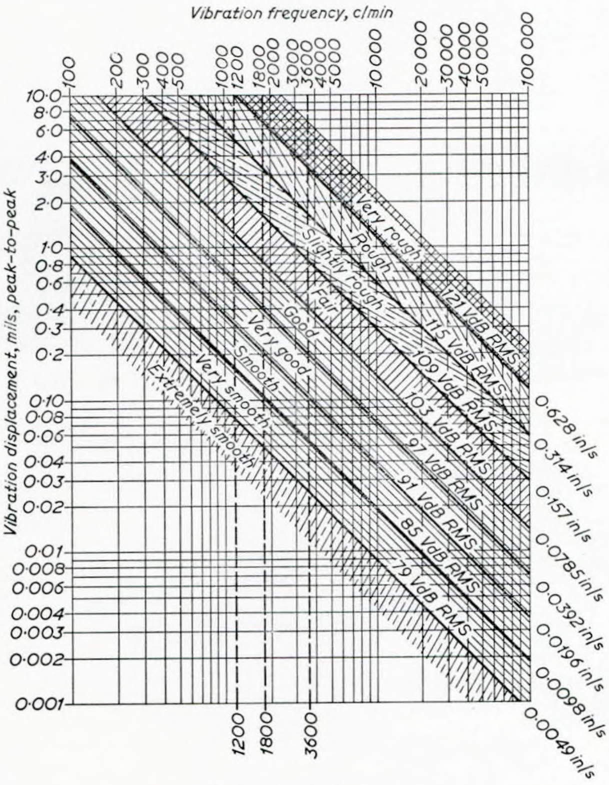


FIG. 2—*General machinery vibration severity chart*

The Canadian Forces use the velocity decibel as a machinery health measurement parameter; this is directly analogous to the decibel sound pressure measurement scale, and enables a good correlation between noise and vibration to be obtained in acoustical problems. The relationship between velocity decibels and inches per second has been defined on Fig. 2.

The Octave Band Vibration Analyser has a Definite Diagnostic Capability

When a high overall vibration level is encountered, it has been found that an octave band analysis of the vibration is a convenient way of defining the nature of the problem which has arisen on the machine.

This is because most faults lie within the following simple grouping:

Type of fault	Axis of dominant vibration	Frequency of vibration
Unbalance	Radial	Shaft frequency
Misalignment	Axial	Shaft frequency and low multiples
Bent shaft	Axial	Shaft frequency and low multiples
Gears	Can be either axis	Gear teeth $\pm$ shaft frequency and multiples thereof
Bearings	Can be either axis	High frequency 250 Hz to 5 KHz

As an example Fig. 3 shows the relationship between the octave band readings and the discrete frequency readings taken on a typical marine pump. It can be seen that the octave band reading is generally only a little higher than the largest discrete frequency reading in that octave band and, in this case, the

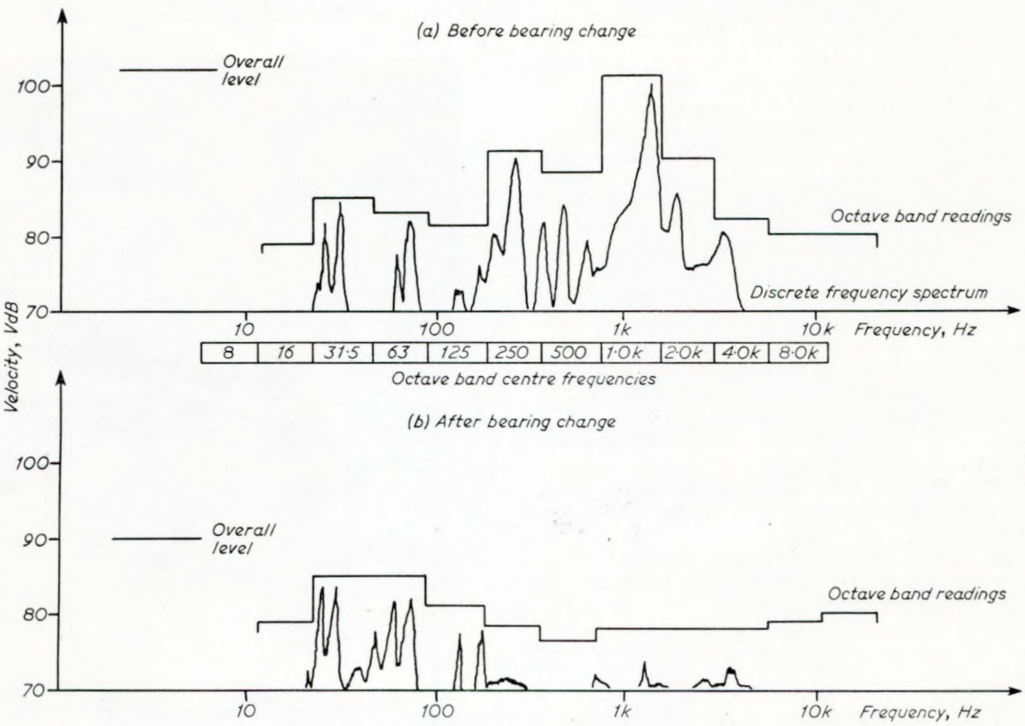


FIG. 3—*Vibration record of pump at lower bearing*

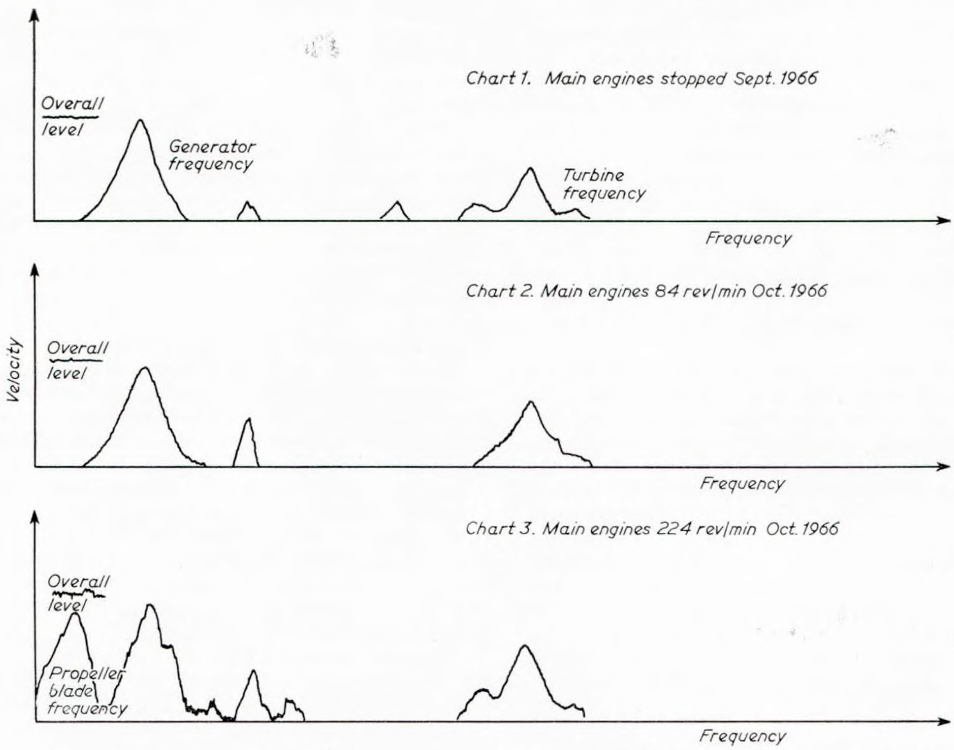


FIG. 4—*Vibration record of generator measured vertically on generator bearing*

relatively high octave band reading in the 1 KHz band gave an indication of bearing problems.

When using the octave band analyser on a complex machine, it is not always possible to define which gear or bearing in an assembly is giving trouble, but the indication is generally sufficient to enable the correct preventive action to be taken.

Vibration Readings are Repeatable at Normal Operating Conditions

Repeatable vibration readings can be obtained from ships' machinery at sea, or from industrial equipment, provided that the machinery is operated at normal cruising or any other standardized datum conditions.

To illustrate this point, Fig. 4 shows narrow band frequency analyses of the vibration of a ship's generator, as measured on the main bearing cap. Chart 1 was taken in September with the main engines stopped. Chart 2 was taken in October with the main engines at economical cruising speed and Chart 3 was taken with the main engines at full power.

It can be seen that there was no significant change in the vibration pattern on the bearing between the readings with main engines stopped in September and those at the economical cruising condition in October, but a considerable amount of interference from the main engines is evident in the chart which was taken at full power.

Standardized Vibration Monitoring Points are Required at Each Principal Bearing

It is essential that the vibration readings be taken close to each principal bearing of a machine. An example which occurred on the first shipboard survey carried out by NETE illustrates this in Fig. 5.

Measurements were taken at the lower bearing on two pumps. No. 1 unit was considered satisfactory. The reading on No. 2 unit indicated trouble and further diagnostic measurement pinpointed the trouble as being at the upper bearing location of this pump. The distance between the two bearings was 230 mm (9 in) but in that distance the vibration had attenuated 10 Vdb. An octave band analysis showed that the 1 KHz component of vibration was dominant. As already shown, this is typical of a defective bearing. The unit was stripped and it was found that the upper bearing was badly corroded.

Machinery Deterioration is Indicated by an Increase of Vibration Levels with Time

Since periodic measurements of vibration under controlled conditions are repeatable and since deterioration of machinery is indicated by a high vibration reading, it follows that deterioration of machinery can be monitored by taking periodic measurements of the overall vibration level.

	Vibration level VdB					
	Point 17			Point 16		
	A	V	L	A	V	L
No. 1 unit	97	88	92			
No. 2 unit	108	104	106	118	106	110

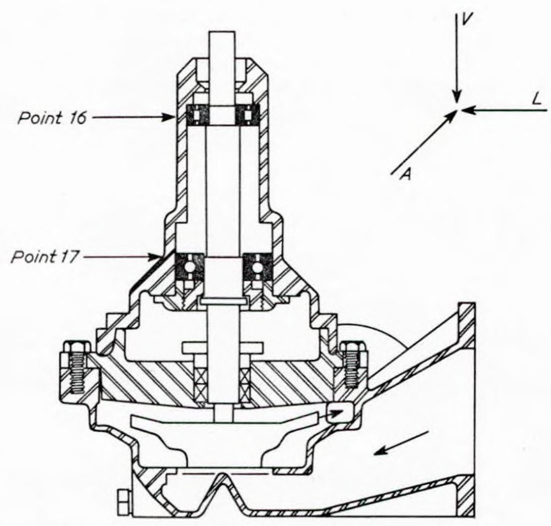


FIG. 5—*Vibration of two pumps*

When the vibration reaches a certain level, octave band or discrete frequency analysis can be employed to measure the frequency of the components of the vibration and thus assist in pinpointing the cause. This is illustrated in the following table of overall and discrete frequency measurements, taken from two diesel generators on the generator bearing (Table I).

Unit B was shut down and stripped. It was found that the flexible coupling between the diesel motor and the generator was worn out and a complete overhaul of the coupling was required. Had the unit been allowed to run much longer, serious damage to the generator or motor would have resulted.

Each Type of Machine has a Characteristic Vibration Signature when in Good Order

In the initial stages of the programme, it was found that the IRD Mechanalysis Corporation "Machinery Vibration Severity Chart" (Fig. 2) gave a good first approximation of the serviceability of steam driven and electrical rotating machinery. However, a significant improvement in diagnostic capability was obtained by establishing norms for the vibration levels at each monitoring point on the various machines in the VA programme. These norms consist of the tabulated values of the highest, average and lowest Vdb levels recorded, both for the overall and the octave band vibration levels, at every point. As is discussed in the next main section, graphs of these norms have been prepared for the dockyard and shipboard maintainers, and it has been found that the average value plus 6 Vdb is a realistic cautionary level for all units, above which machines should be monitored closely.

Machinery Health is Established by Comparison of Data

The good maintainer has a keen ear and generally a stethoscope. When he listens to machinery noise and is alerted to developing problems by unusual sounds, he is subconsciously comparing the noise of the machine with what it used to be and with the normal noise patterns generated by that type of machine.

To monitor machinery health by vibration analysis requires exactly the same logical process. The octave band analyser can be regarded as a very sensitive stethoscope (or a screwdriver) in the hands of the experienced maintainer. It is slightly more complex to handle, but it gives much better results.

A machinery monitoring programme is started by selecting external monitoring points on the machines. These should be permanently established (in the CF 19 mm (0·75 in) cubic mild steel blocks are attached at each selected point). If the machines have been recently overhauled and are known to be in good condition, their vibration levels will be normal and will constitute a set of norms with which future data can be compared.

With the machinery operating normally, an overall reading is taken and an octave band survey is conducted triaxially at each point on every machine in the programme. If no norms are available the data should be compared initially with the IRD chart. It should be noted, however, that the levels on this chart apply to industrial rotating machinery, and that on reciprocating machinery and aircraft type gas turbines, normal levels are usually 10 Vdb higher than those indicated on the chart.

When there are two or more machines of the same type, it is surprising how quickly the norms for that machine can be established. For example, Fig. 6 gives the octave band readings from two sea water circulating pumps in a new ship. The high readings in the 63, 125 and 250 Hz bands on the No. 2 unit clearly indicated bad misalignment between the pump and the motor. When this was rectified the readings of the two units were similar and thus defined the initial norms for this type of machine.

Upon completion of the initial survey and the eradication of faults which have been diagnosed, monthly checks of the overall vibration level should be incorporated into the planned maintenance routines for the machines. Generally one axis of measurement at each monitoring point is sufficient for this check. Whenever a significant increase (say 6 Vdb) in the overall level occurs, and prior to major planned maintenance inspections, complete octave band surveys on the affected machines are recommended. When these are compared with the original readings, any signs of deterioration will be evident. The maintainers can then plan the necessary shutdowns for rectification, and also eliminate good machines from "Open and Inspect" types of examinations.

In medium sized industrial operations, manual analysis of the vibration data is adequate, but in large organizations, which operate numbers of similar machines, it is considered economic to automate the data analysis process. This has certainly proven to be the case in the Canadian Forces.

COMPUTERIZED VIBRATION DATA ANALYSIS IN THE CANADIAN FORCES

The principal CF(S) VA programmes are very large and it was realized that analysis of the data could be considerably expedited, and reduced in cost, by computerizing the operation where possible. In fact the last three of the four named programmes discussed in this section perform operations which could not be considered practical without the computer. The data collected during the Dockyard pre and post refit surveys, the post overhaul test programmes and the machinery acceptance surveys is transcribed onto data cards for the following operations.

The principal program is VIBANAL, which checks new data for errors and compares the new data with the average values for the machine. The program output consists of the machine vibration levels and the listing of each point that is more than 6 Vdb over the average value, in the format shown later in Figs. 10, 11 and 12. As can be seen this format enables the analyst to determine quickly the faults in the machine and to recommend the best refurbishment. This program saves four engineering man-days per ship's survey and costs one day for a punch card operator, plus about $10.00 computer charges.

Another program, VIBLIST, is usually run in conjunction

TABLE I

Date	Type of reading	Overall vibration level Vdb					
		D.G. "A"			D.G. "B"		
		X	Y	Z	X	Y	Z
September	Overall	107	104	104	116	102	99
	Component at 20 Hz	107	100	99	116	102	89
18 Dec.	Overall (Main Component was at 20 Hz)	119	110	112	125	113	99
22 Dec.	Overall (Main Component was at 20 Hz)	—	—	—	141 unit shut down.		

VIBRATION ANALYSIS OCTAVE BAND HISTORY RECORD
(VALUES IN Vdb re 10^{-6} cm/sec.)

INITIALS / INSTRUMENT NO.	DATE	MAIN ENG. SPEED	UNIT SPEED OR LOAD	HRS. RUN	POINT & PLANE	O.A.	8	16	31	63	125	250	500	1K	2K	4K	8K
								NO.	2 MACHINE								
					1A	100			86	98	88	86	83	76	74	84	78
					1V	105			89	99	94	93	81	75/80	71	77	79
					2A	99			85	98	88	88	87	73	73	79	75
					2L	114			85	102	110	110	79	72	76	82	80
					2V	115			87	88	111	112	82	72	72	78	77
					3A	101			76	91	98	95	90	83	80	85	77
					3L	106			84	103	93	91	83	77	79	84	81
					3V	114			80	100	110	110	78	74	82	88	76
								NO.	3 MACHINE								
					1A	97			77	89	83	73	74	71	72	85	76
					1V	99			75	91	85	74	75	72	72	80	77
					2A	98			78	97	82	73	78	70	73	83	77
					2L	99			82	92	90	80	74	76	82	88	80
					2V	98			75	83	87	84	74	71	77	86	75
					3A	107			84	105	100	84	85	82	85	83	86
					3L	103			82	101	93	79	76	76	80	80	87
					3V	101/105			80	99	87	84	72	74	83	84	85

SERIAL NO.	H.M.C.S.	UNIT	LOCATION	GUIDE LIST NO.
		COOLING WATER PUMP	MAIN MACHINERY ROOM	

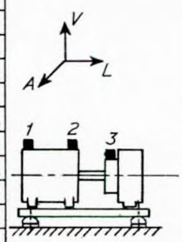


FIG. 6

with VIBANAL, and lists and compares the results, at each point, of the last three vibration surveys, thus expediting the historical comparison of the machinery vibration.

IBM utility programs enter the data into the master storage tape, from which it can be recalled when required by VIBLIST or other comparative work. The data is stored on the master tape so that all the readings from a given point on any machine type are together. This enables the machinery norms to be updated when required.

The program AVLIST takes all the readings between specified time periods and gives an updated set of the norms or of the average values alone, for use by the shipboard operator. For each point the norms consist of:

a) the maximum values recorded in service;
b) the minimum values recorded in service, but with a low limit of 70 Vdb (any reading lower than 70 Vdb indicates an instrument defect);
c) the average of all the readings, but ignoring readings greater than 130 Vdb (which would indicate a seriously defective machine).

The averages for a main extraction pump are shown in Fig. 7. It has been found that the norms are relatively stable. Over a period of five years, the average values have reduced by only 2 or 3 Vdb. Some high levels which remain suggest that very considerable improvements can still be made to our machinery overhaul standards. At present AVLIST is run every two years.

A graphical plotting program, VIBPLOT, takes the output from AVLIST and draws a new set of norms, as in Fig. 8.

MAIN EXTRACTION PUMP

VIBRATION OCTAVE AVERAGES FROM JAN.68 -DEC.1970

MACHINE	POINT	O.A.	8C	16C	31C	63C	125	250	500	1KC	2KC	4KC	8KC
E05020	01A	110	105	100	92	89	92	84	88	99	94	91	87
E05020	01L	110	100	98	97	93	95	90	92	103	99	96	86
E05020	01V	111	108	103	96	86	87	87	85	94	92	90	86
E05020	02A	107	97	95	92	90	95	90	90	99	92	87	84
E05020	02L	108	97	94	93	90	95	91	91	99	99	92	87
E05020	02V	106	99	97	93	83	85	88	87	97	94	87	84
E05020	03A	107	98	98	93	90	95	86	88	98	94	91	87
E05020	03L	108	98	97	94	87	89	88	90	99	98	93	86
E05020	03V	107	96	95	93	85	90	89	90	99	96	90	86
E05020	04A	106	96	96	91	88	93	87	89	98	95	88	85
E05020	04L	107	95	95	92	87	92	89	93	99	97	90	85
E05020	04V	106	95	94	92	84	86	89	90	99	97	89	84
E05020	05A	110	100	100	95	88	95	89	89	103	95	91	87
E05020	05L	109	99	101	98	94	95	93	97	100	94	91	86
E05020	05V	107	97	96	93	86	90	91	92	98	97	96	86
E05020	06A	108	99	99	93	89	93	86	89	101	97	94	85
E05020	06L	107	94	98	95	93	90	91	93	99	98	92	85
E05020	06V	106	96	95	94	86	90	89	90	97	96	92	85
E05020	07A	106	95	96	91	88	93	88	90	99	93	90	87
E05020	07L	107	95	96	94	92	92	91	93	99	94	91	87
E05020	07V	107	95	95	95	86	89	89	91	101	95	91	88
E05020	08A	107	95	95	93	88	94	92	98	102	93	83	81
E05020	08L	109	96	97	94	89	94	92	98	102	96	86	81
E05020	08V	107	95	96	93	85	90	92	98	100	93	88	81
E05020	09A	105	91	95	95	89	89	89	89	96	90	86	84
E05020	09L	106	89	97	96	87	90	90	87	96	95	86	82
E05020	09V	106	93	96	94	87	92	94	92	96	89	85	83
E05020	10L	106	82	91	98	87	90	88	86	101	90	86	83
E05020	10V	105	91	94	93	88	92	94	93	97	90	87	80
E05020	11L	94	81	80	85	82	83	84	80	84	80	76	78
E05020	11V	96	85	83	85	85	85	83	82	85	79	77	78

FIG. 7

THE CANADIAN FORCES MACHINERY HEALTH VIBRATION MONITORING PROGRAMMES

Of the five VA programmes discussed in this section, the Dockyard VA Programme and the Helicopter Drive Train Alignment Programme are fully operational. The Shipboard Monitoring Programme, the Quality Control VA Programme and the New Machinery Acceptance Programmes are currently operating on an exploratory basis.

The Dockyard Pre and Post Refit Vibration Surveys

The programme was set up by NETE for the 44 principal machines on each ship but the dockyards expanded it to include over 100 mechanical and electrical machines. For each type of machine in the programme, standardized monitoring points were defined adjacent to each principal bearing. The machines were fitted with 19 mm (0·75 in) cubic mild steel blocks at each vibration monitoring point. A list of normal vibration modes and a table of machinery faults and their vibrational effects were prepared. Graphs of the vibration norms showing the maximum, average and minimum levels experienced in the fleet at each point on each type of machine were drawn (by computer) and standard reporting documentation was prepared. As an example, the documentation pertaining to the extraction pump, including a sample of the graphical norms prepared for shipboard use, is given in Figs. 8 and 9, Tables II and III.

Between one and four months prior to a ship entering refit, dockyard personnel carry out complete octave band vibration surveys at all points on the machines. These are supplemented by

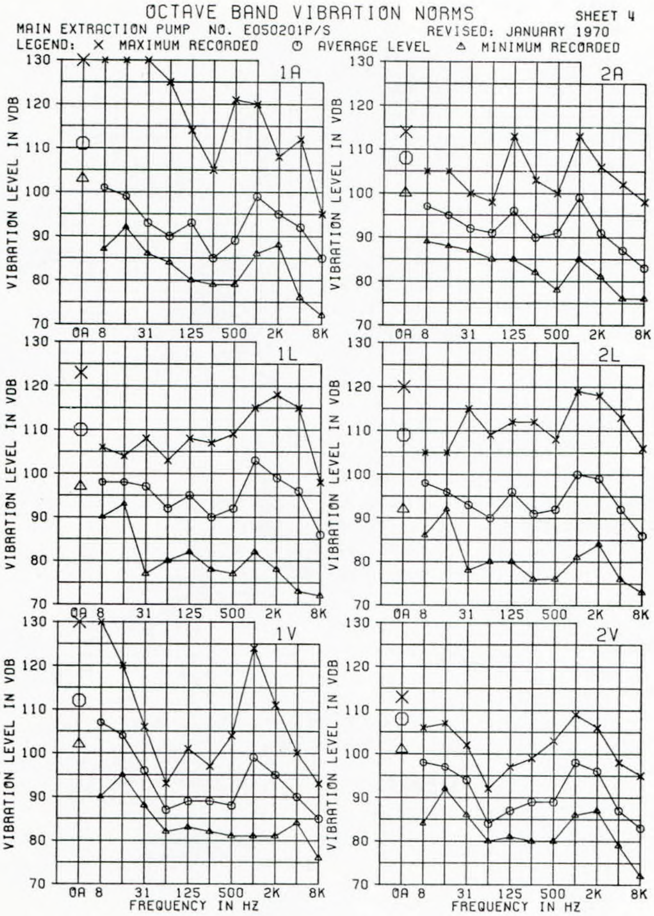
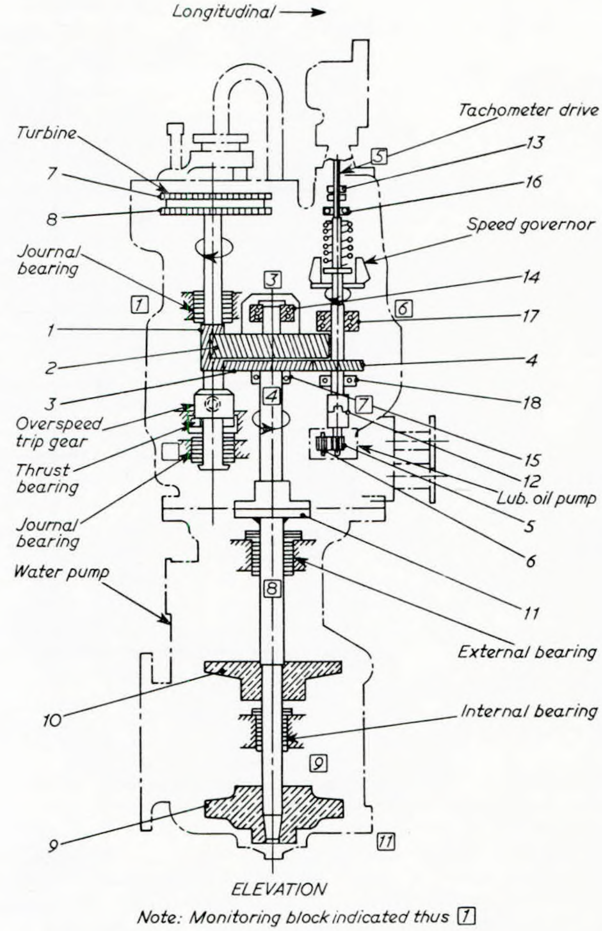


FIG. 8



Item No.	Gears	No. of teeth	Tooth engagement per second at 6500 rev/min turbine	Rev/min	Rev/sec
1.	Primary pinion (turbine	22	2382·6	6500	108·3
2.	Primary gear	124		1172·1	19·5
3.	Secondary wheel	99	1930·5	1172·1	19·5
4.	Secondary pinion	47	1931·7	2468·8	41·1
5.	Lub. oil pump driver	12	493·2	2468·8	41·1
6.	Lub. oil pump follower	12		2648·8	41·1
	Turbine	No. of blades			
7.	First stage	67		6500	108·3
8.	Second stage	69		6500	108·3
	Impellers	No. of vanes			
9.	First stage			1172·1	19·5
10.	Second stage			1172·1	19·5
	Couplings	Type	Mfr.		
11.	Gearwheel to pump	Bolted		1172·1	19·5
12.	Sec'y pinion to L.O. pump	Slotted		2468·8	41·5
13.	Pinion to tach. drive	Slotted		2468·8	41·5
	Ball bearings	No.			
14.	Double row journal	5309	S.K.F.	1172·1	19·5
15.	Single row ..	6309	..	1172·1	19·5
16.	.. .. ..	6303	..	2468·8	41·5
17.	Double row ..	5306	..	2468·8	41·5
18.	Single row ..	6306	..	2468·8	41·5

FIG. 9

discrete frequency surveys whenever a problem has been indicated, but not adequately defined, by octave band measurements.

At this stage, the ship's maintenance and repair list is reviewed in the light of the analysis of the surveys and other available performance data. Machines which are in good condition are not opened for routine shafting, coupling, gear or bearing inspections, and machines not scheduled for planned maintenance are included on the list if the vibration readings suggest deterioration.

Post refit vibration surveys are conducted as a means of quality control to establish that maintenance work has been carried out satisfactorily and to establish the datum levels of the shipboard periodic vibration monitoring programme.

Shipboard Periodic Machinery Health Vibration Surveys

A two year trial programme was carried out during which machinery vibration levels were measured monthly by the ships' staffs on a number of destroyers. Overall vibration measurements were taken and compared with the post refit levels. Where levels had risen by a factor of two (6 Vdb) it was recommended that ships' staffs take octave band measurements to determine the cause of problems and take appropriate action. The programme was remarkable in that few machinery failures occurred in the ships concerned. For a time it appeared that the dockyard programme was so successful that the need for the shipboard monitoring programme on the destroyers was eliminated. However, as the effectiveness analysis discussed later in this paper has shown, there is much to be gained from a periodic shipboard monitoring programme, and steps are presently being undertaken to reintroduce this in the ships.

At present the large Operational Support Ships are using vibration analysers as a diagnostic tool at sea and favourable reports are being received.

Quality Control of Overhauled Machinery by VA

A programme of acceptance vibration analysis of machinery during test after overhaul has been introduced at one repair facility and is gaining momentum. The dockyard VA norms are being used very successfully, whenever the ships' refit schedules allow time for defects discovered during tests to be rectified. This is not always possible at present, but the most serious faults are corrected and the ships warned of the remaining defects, which are to be repaired as soon as possible.

TABLE II—NORMAL VIBRATION FREQUENCIES

Machine: Main extraction pump — Dwg. No.: IT 211/C, Sheet 2

Guide List No.: E–05–02–01 — Date: 20 January, 1969

Operating condition: 6400 rev/min (turbine)

Discrete frequency c/s	Vibration description	Points at which dominant	Octave band
	Hull propeller interference	all	8
	Propeller blade	all	16
18·9	Pump shaft	3, 4, 8	16
38	Pump shaft × 2	3, 4, 8	31
40	Lubricating oil pump and governor drive shaft	5, 6, 7	63
57	Pump shaft × 3	3, 4, 8	63
79·5	L.O. pump and governor drive shaft × 2	5, 6, 7	63
106	Turbine shaft	1, 2	125
152	Pump vanes	9, 10	125
160	L.O.P. and governor drive shaft × 3	5, 6, 7	125
211	Turbine shaft × 2	1, 2	250
304	Pump vanes	9, 10	250
360	L.O. pump gears		250
423	Bearings—various		500
1268	Turbine pinion and primary gear wheel × 1/2		1K
1860	Secondary gear wheel and governor drive pinion	5, 6, 7	2K
2340	Turbine pinion and primary gear wheel	2, 3, 4	2K
7120	Turbine blade 1st stage	1, 2	8K
7340	Turbine blades 2nd stage	1, 2	8K

Note: When machine rotational speed differs from the desired rotational speed shown above, then discrete frequencies must be adjusted using the following formula:

$$\text{New Discrete Freq.} = \frac{\text{Indicated Frequency} \times \text{Actual Rotational Speed}}{\text{Desired Operating Rotational Speed}}$$

TABLE III

IT 211/C, Sheet 3

Diagnosis of faults for main extraction pump

Defects to be expected	Symptoms
Turbine unbalance	Points 1 and 2 large radial vibrations in the 125 octave band.
Pump unbalance	Points 3, 4, 8 and 9 large radial vibrations in the 16 octave band.
Governor and l.o. pump unbalance	Points 5, 6, 7 large radial vibrations in the 63 octave band.
Pump misalignment	Points 4 and 8 longitudinal vibrations in the 31, 63 and 125 octave bands.
Worn gear teeth or bearings on primary gear wheel	Points 1, 2, 3 and 4 high readings in the 1K, 2K and 4K bands.
Worn gear teeth or bearing in governor drive shaft	Points 5, 6 and 7 high readings in the 500, 2K and 4K bands.
Defective internal pump bearing	Point 9 high readings in the 250 and 500 octave bands.

As an example of this, Fig. 10 shows the vibration levels on an extraction pump during its post overhaul tests. A high turbine unbalance was diagnosed and the unit was returned to the overhaul facility for rework. Fig. 11 shows the vibration level when it was tested a second time. It can be seen that the turbine unbalance had dropped considerably, but that the vibration in the adjacent frequency, associated with the governor, had not diminished. Time being short, the test shop balanced the governor; the results are shown in Fig. 12 and the unit was shipped (express) to meet the ship's schedule, still containing some residual turbine unbalance.

In this context it should be noted that NETE has prepared a set of "Overhauled Machinery Norms". These norms are the average, maximum and minimum vibration levels obtained from

THE USE OF VIBANAL AS A QUALITY CONTROL TOOL

① EXTRACTION PUMP
AS RECEIVED FROM OVERHAUL FACILITY

PAGE 1

VIBRATION READINGS TAKEN ON MAIN EXTR. PUMP OF N.E.T.E. AT SURVEY OF FEB. 7/72

SHIP NO. 0 | MACHINE NO E05020 | SURVEY DATE 7 2 72 | MACHINE SPEED 6500 | SHAFT RPM 0 | SERIAL NO. 49819

COMPARISON WITH AVERAGES READINGS OF 6 VDB OR MORE THAN THE AVERAGES ARE SHOWN

PNT	OA	8	16	31	63	125	250	500	1K	2K	4K	8K	-	OA	8	16	31	63	125	250	500	1K	2K	4K	8K
01A	107			92	92	105	82	80	85	90	89	83	-						13						
01L	113			97	99	113	85	82	85	90	89	93	-					6	18						7
01V	104			90	88	100	85	81	78	94	94	82	-						12						
02A	113			97	99	112	86	86	85	90	84	76	-	6				9	17						
02L	112			92	97	112	85	84	86	89	82	83	-					7	16						
02V	104			90	88	101	84	85	86	93	98	77	-						16					11	
03A	111			94	97	110	82	82	84	88	96	81	-					7	15						
03L	101			92	88	94	81	81	83	87	92	91	-												
03V	106			87	93	104	85	84	86	91	95	88	-					8	14						
04A	110			94	96	109	83	82	88	91	86	84	-					8	16						
04L	108			89	93	107	84	86	84	85	83	94	-						15						8
04V	105			87	90	104	84	88	92	91	89	82	-					6	18						
05A	106			93	94	104	87	84	89	93	96	90	-					6	9						
05L	113			100	101	112	92	95	88	91	96	87	-					7	17						
05V	104			85	89	101	86	86	82	88	96	92	-						11						6
06A	110			93	97	109	83	86	91	95	86	87	-					8	16						
06L	107			98	97	103	84	85	88	100	98	82	-						12					6	
06V	108			85	93	106	87	88	89	96	102	82	-					7	16					10	
07A	108			93	93	107	83	85	88	88	91	94	-						14						7
07L	108			96	96	106	87	84	86	85	89	102	-						14						15
07V	106			90	93	104	85	92	90	89	92	91	-					7	15						
08A	109			92	96	108	90	91	95	86	85	82	-					8	14						
08L	113			90	97	113	88	86	86	89	86	79	-					8	19						
08V	105			86	88	100	90	94	87	89	93	77	-						9						
09A	100			86	89	95	79	78	85	84	85	89	-												
09L	106			96	95	102	84	78	86	91	85	86	-					8	13						
09V	104			85	90	103	87	82	90	83	83	89	-						11						7
10A	105			94	94	103	85	82	83	90	84	85	-												
10L	107			98	96	104	83	82	88	89	82	85	-					9	14						
10V	114			92	88	103	92	86	93	85	83	86	-						11						6

← VIBRATION MEASUREMENTS → ← VIBANALYSIS →

GOVERNOR UNBALANCE

TURBINE ROTOR UNBALANCE

FIG. 10

THE USE OF VIBANAL AS A QUALITY CONTROL TOOL

② EXTRACTION PUMP
AFTER BALANCING OF TURBINE BY OVERHAULER

PAGE 1

VIBRATION READINGS TAKEN ON MAIN EXTR. PUMP OF N.E.T.E. AT SURVEY OF FEB. 72

SHIP NO. 0 | MACHINE NO E05020 | IP | SURVEY DATE 10 2 72 | MACHINE SPEED 6500 | SHAFT RPM 0 | SERIAL NO. 49819

COMPARISON WITH AVERAGES READINGS OF 6 VDB OR MORE THAN THE AVERAGES ARE SHOWN

PNT	OA	8	16	31	63	125	250	500	1K	2K	4K	8K	-	OA	8	16	31	63	125	250	500	1K	2K	4K	8K
01A	105			94	92	96	82	81	87	101	92	88	-										7		
01L	107			100	98	103	83	86	87	90	88	90	-						8						
01V	104			93	92	96	85	84	88	90	91	85	-						8						
02A	106			98	100	103	87	86	87	89	84	80	-				6	10	8						
02L	106			96	96	102	86	84	82	87	80	85	-					6	6						
02V	103			89	90	94	83	88	86	96	96	81	-					7	9					9	
03A	105			98	96	100	83	83	86	90	94	83	-					6							
03L	104			94	91	86	83	84	84	88	87	89	-												
03V	103			92	93	94	89	85	85	90	95	85	-					8							
04A	105			96	95	100	85	84	87	91	89	86	-					7	7						
04L	106			91	95	96	88	88	85	93	95	93	-					6							7
04V	104			93	91	97	86	90	94	95	88	84	-					7	11						
05A	106			100	96	97	89	86	92	95	88	89	-					8							
05L	110			105	103	104	93	94	94	92	93	86	-				7	9	9						
05V	104			94	92	96	88	88	84	88	91	94	-						6						8
06A	109			99	98	100	85	88	96	96	105	89	-				6	9	7					11	
06L	108			101	100	95	84	87	92	101	99	86	-				6	7						7	
06V	106			94	95	98	90	90	97	96	94	84	-					9	8						
07A	104			96	97	100	85	86	89	92	90	92	-					9	7						
07L	107			99	98	97	88	87	83	86	91	97	-				6	7							10
07V	105			94	92	97	88	91	93	91	95	94	-					6	8						6
08A	108			99	99	100	90	92	93	89	86	85	-				6	11	6						
08L	107			94	95	102	88	88	88	89	85	84	-					6	8						
08V	102			93	90	91	90	96	88	91	89	85	-												
09A	102			95	90	88	80	80	83	89	90	94	-												10
09L	106			99	96	95	86	80	88	94	86	91	-					9	6						8
09V	105			90	92	93	88	84	86	87	94	94	-											10	12
ERROR10A	104	0	0	99	97	92	84	84	90	90	91	93													
10L	108			100	97	93	84	85	93	92	88	91	-					10							8
10V	107			92	95	104	91	90	88	91	90	92	-					7	12						12

← VIBRATION MEASUREMENTS → ← VIBANALYSIS →

GOVERNOR UNBALANCE HASN'T CHANGED

TURBINE-ROTOR UNBALANCE HAS BEEN REDUCED BUT IS STILL PRESENT

FIG. 11

THE USE OF VIBANAL AS A QUALITY CONTROL TOOL

③ EXTRACTION PUMP
FINAL VA AFTER BALANCING OF GOVERNOR BY TEST FACILITY

PAGE 2

VIBRATION READINGS TAKEN ON MAIN EXTR. PUMP OF N.E.T.E. AT SURVEY OF FEB. 72

SHIP NO.	MACHINE NO		SURVEY DATE	MACHINE SPEED	SHAFT RPM	SERIAL NO.
0	E05020	IP	23 2 72	6500	0	49819

COMPARISON WITH AVERAGES
READINGS OF 6 VDB OR MORE THAN THE AVERAGES ARE SHOWN

PNT	OA	8	16	31	63	125	250	500	1K	2K	4K	8K	-	OA	8	16	31	63	125	250	500	1K	2K	4K	8K
01A	102			88	83	95	78	82	88	91	83	83	-												
01L	106			98	89	102	82	85	86	94	87	87	-						7						
01V	110			90	83	88	81	83	85	90	83	84	-												
02A	105			94	90	102	82	86	88	90	83	80	-						7						
02L	105			92	89	103	84	86	82	83	78	84	-						7						
02V	102			86	84	84	81	89	86	88	83	80	-												
03A	103			94	86	99	78	83	85	91	87	84	-												
03L	102			91	83	84	82	85	86	88	84	89	-												
03V	102			86	86	95	85	87	87	95	85	85	-												
04A	103			91	86	98	81	83	88	89	87	87	-												
04L	104			90	89	99	86	89	86	90	84	93	-						7						7
04V	104			86	85	98	83	92	95	94	87	86	-						12						
05A	104			92	88	92	87	94	99	91	88	88	-												
05L	108			98	94	104	94	98	94	94	87	86	-						9						
05V	103			90	86	97	86	91	84	90	83	93	-						7						7
06A	104			93	87	98	81	90	96	96	87	86	-												
06L	106			96	90	95	84	88	92	102	87	84	-												
06V	105			89	89	100	90	96	96	96	85	83	-						10		6				
07A	102			92	87	97	82	87	92	91	86	89	-												
07L	104			94	89	96	88	89	86	88	87	96	-												9
07V	104			90	86	99	87	93	96	88	94	92	-						10						
08A	104			92	90	100	87	93	94	90	78	80	-						6						
08L	104			92	89	92	86	88	86	87	79	80	-												
08V	102			88	84	86	88	95	87	89	80	80	-												
09A	98			89	85	84	76	82	78	83	81	83	-												
09L	104			97	89	95	84	80	83	88	79	83	-						6						
09V	99			90	86	89	83	84	78	86	80	84	-												
10L	105			99	91	88	82	85	88	86	79	81	-												
10V	106			94	90	104	89	90	88	87	82	82	-						12						

←VIBRATION MEASUREMENTS→ ←VIBANALYSIS→

GOVERNOR UNBALANCE HAS DISAPPEARED

THIS SHOWS OVERHAULER HAD NOT ADEQUATELY BALANCED THE TURBINE

FIG. 12

a number of known good units during testing of overhauled machines. They run several Vdb lower than the shipboard norms presently in use, and they may become the yardstick for repair facilities in the future.

Acceptance Inspection of Machinery

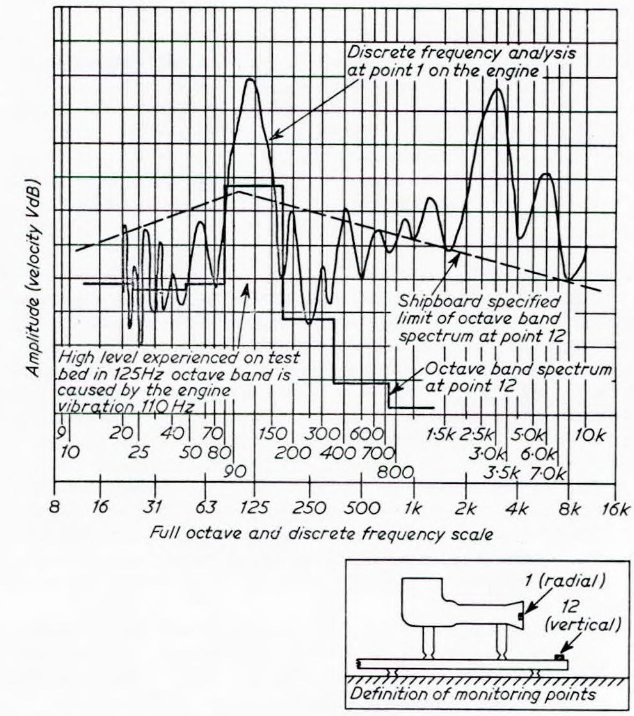
CF(S) vibration specifications for new types of machines presently define acceptable octave band vibration levels when measured at the base of the unit. These levels are not necessarily related to the machinery health condition. During the vibration tests in the new ships, the vibration characteristics at the bearings are also being measured, generally in chart record form. This enables the characteristics of any component of the vibration spectrum, which exceeds the specification, to be defined and remedial action to be taken.

As an example of this, during shore test bed runs, the vibration levels adjacent to the engine support of a gas turbine appeared somewhat high in the 125 Hz octave band as shown in Fig. 13. Above this octave band curve is shown the discrete frequency vibration spectrum taken at the front of the engine where the vibration level was at a maximum. The peak vibration was observed to emanate at the low pressure compressor frequency, and the manufacturer has proposed to field balance the engine during sea trials.

CF Helicopter Engine Alignment Check Procedures

The helicopter squadrons of the Land Element of the Canadian Forces [CF(L)] use the standard octave band vibration analyser to carry out routine vibration checks on the CH113 helicopter.

At engine installation, the manufacturer specifies a maximum vibration level, at power turbine frequency (325 Hz), of 16 g peak. With an octave band analyser this is denoted by a reading of 135 Vdb in the 250 Hz octave band. Remedial action consists of step by step alignment of the power turbine drive train components.



Machine type: Maine engine. Machine condition: Full power.

Monitoring points: 1(R) and 12(V). Date: December 1970.

FIG. 13—*Octave band and discrete frequency vibration analysis*

TABLE IV

Action taken	Overall level	Octave band readings (Vdbs) 31	63	125	250	500	1K	2K	4K	8K
Engine installed	136	106	102	125	135	130	128	113	111	99
Curvic coupling bolts retorqued	134	107	101	123	130	127	129	113	104	98
Flexible coupling checked. Three broken plates replaced	133	106	97	116	132	126	124	110	102	92
Mix box adapter rotated 120° to drive shaft	127	109	95	114	120	121	121	108	100	91

The following results (Table IV), taken during some experimental surveys, show that it is usually possible to improve on the manufacturer's maximum permitted level by a factor of two (6 Vdb) or more. Points worthy of note are that the repair of the flexible coupling reduced the vibration in the 125 Hz and 8 KHz bands very considerably, and that the subsequent alignment of the mix box adapter reduced the vibration at power turbine frequency by 12 Vdb, the final level being 3 g.

EFFECTIVENESS OF THE DOCKYARD VA PROGRAMME

Vice-Admiral Raper[2] pointed out the difficulties in obtaining a complete cost or value effectiveness analysis on warships and recommended that such an analysis be limited to showing the order of improvement obtained by effecting a given change. The following study analyses the cost effectiveness of VA on the main engines and gearboxes, and 11 of the principal auxiliary machines in six of a class of twenty CF ships, and whilst quantitatively it is possible that the costs and savings have been under-estimated by as much as 20 per cent, the comparison ratios are considered to be valid. These twelve types of machines represent 50 per cent of the VA monitoring effort. For all machinery on the 20 ships it is shown that the repair costs, due to breakdowns at sea and unscheduled maintenance during ship's refit and minor maintenance periods, have been reduced from \$514 000 (£205 000) a year to \$287 000 (£115 000) a year. Thus, on this class of 20 ships, when the costs of the VA programme are included, direct machinery repair costs are reduced by \$200 000 (£80 000). Perhaps more significantly, the losses due to ships' outages are reduced by almost \$2 000 000 (£800 000) a year.

Method of Analysis

The method chosen to examine the effects of the vibration analysis programme on the overall serviceability of machines in the six ships was to compare the reliability of twelve types of machines in the VA programme, over a two year period before the programme started, and over a recent two year period since the programme has been in operation. The sources of information available to the writer for this analysis were the ships' quarterly engineering reports and the machinery failure reports.

A ship's operational cycle consists of three phases, the refit phase, the sea trials and work-up phase, and the operational phase, during which there are several minor maintenance periods. It was not practical to calculate total maintenance costs. Therefore, for this study the following categories of failures have been examined:

i) unscheduled maintenance: this covers all major defects discovered during ships' refit and minor maintenance periods, and not anticipated by the ships' "Maintenance and Repair" Planning Lists;

ii) sea trials: all major defects discovered during the ships' post refit trials and work-up phase;

iii) operational: all major defects occurring at sea when the ships were operational.

These defects have been examined in two ways. First, to determine the cost effectiveness of VA the direct dollar costs of the machinery repairs have been estimated separately for the three above-mentioned classifications, and the repair costs incurred by each ship in the "pre-VA" days have been compared with those incurred in the later "VA-oriented" period. The second comparison has been made on a "Value Effectiveness" basis. In this case the time that each machine was out of action as a result of its failure (in days) has been multiplied by a "Capability Loss" percentage. This loss is an estimate of the reduction of the ship's capability to meet its objectives because of the failure.

Thus, if a ship's main feed pump failed and took 4 days to repair, costing say \$1500 (£600), the ship's effectiveness would be reduced by 50 per cent during that time and the "capability loss" of the ship has been rated as 200 points in the value effectiveness comparison. Using methods similar to those postulated in reference 3 the life cycle average daily value of a destroyer has been rated as \$10 000 (£4000).Thus a loss of 100 points can be regarded as equivalent to a loss of \$10 000 to the navy.

To illustrate the method, Fig. 14 depicts, in chronological form, the cost rating and the value rating of the major repairs during the two periods studied, on one of the six ships used in the analysis.

Cost and Value Effectiveness

In Fig. 15, for each of the six ships studied, the "ships' repair costs" and the "ships' capability losses" have been compared for the two periods.

Fig. 16 compares the totals of repair costs and capability losses for the 12 machines studied on the six ships.

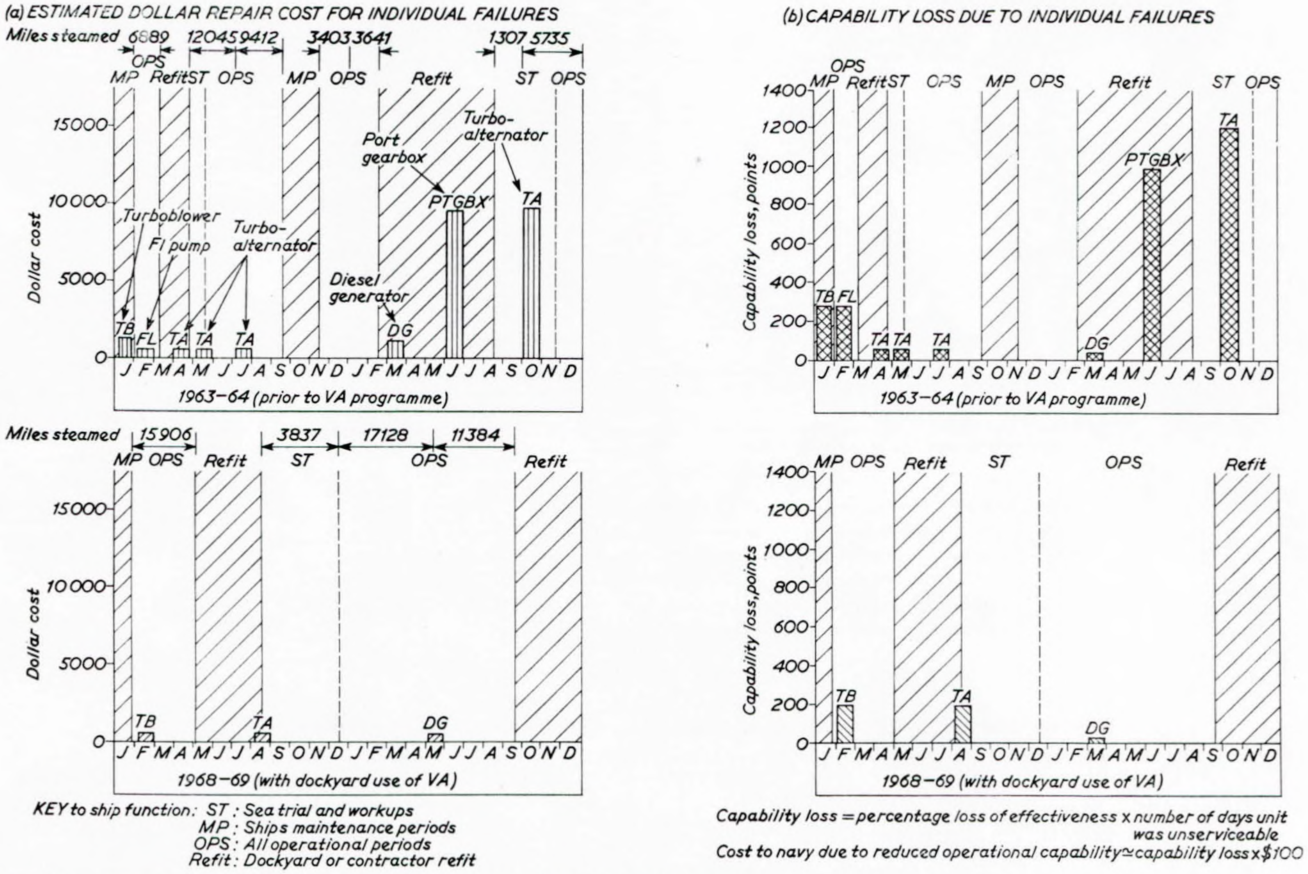
Table V lists the above values, and extends them to cover all the machines in the programme for the class of 20 ships, giving total costs and savings per year, by multiplying by the factors, $2 \times 20/6 \times 1/2$, i.e. 3·33.

From Figs. 15 and 16 and Table V, it can be seen that the number of repairs has reduced only slightly, but the repair costs and the ships' capability losses have been almost halved. They show that the amount of unscheduled maintenance, during refits and ships' minor maintenance periods, has been reduced by two thirds, and that there has been an increase of repair costs during the sea trials and work-up periods. This is because a significant number of faults are being caught in the post refit trials, with the benefit of a 45 per cent reduction of repair costs and capability losses during the operational period.

The significance of these figures is clearly shown by a comparison which has been made with data obtained from CFHQ of ships' overall repair manhour costs. Over the last ten years the ships' overall repair costs incurred between refits have almost doubled, whereas the last line in Table V shows that, between refits, repair costs on the machines in the VA programme, have been reduced by 15 per cent.

The Effectiveness of VA as a Quality Control Tool

To examine the potential for the use of VA as a quality control tool at repair facilities, the average running hours of machines which were rated "good" on the post refit surveys have



Note: This survey covers main engines and the 11 auxiliary engines used on the survey.

FIG. 14—*The effectiveness of VA as a maintenance tool on Ship A—Comparison of repair cost (cost effectiveness) over a two-year period prior to and since the introduction of the dockyard pre- and post-refit VA programme*

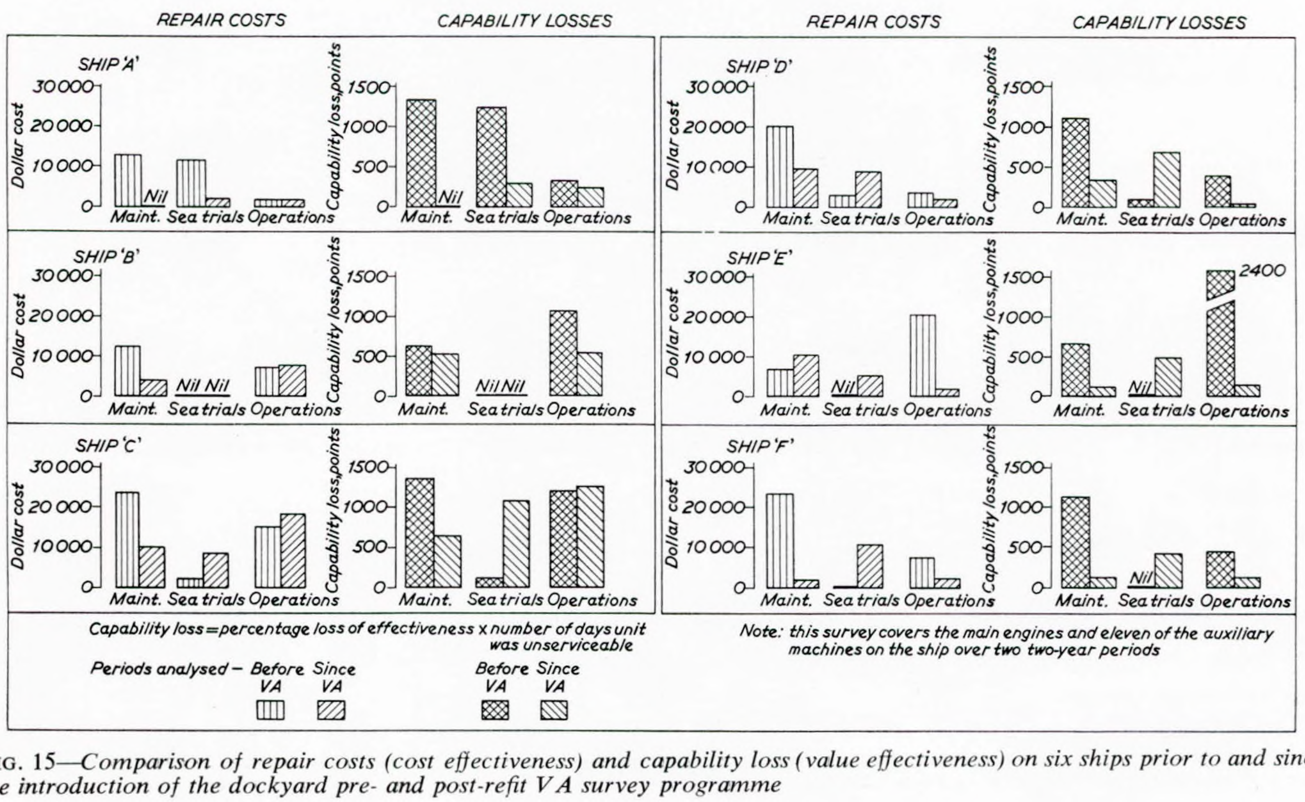


FIG. 15—*Comparison of repair costs (cost effectiveness) and capability loss (value effectiveness) on six ships prior to and since the introduction of the dockyard pre- and post-refit VA survey programme*

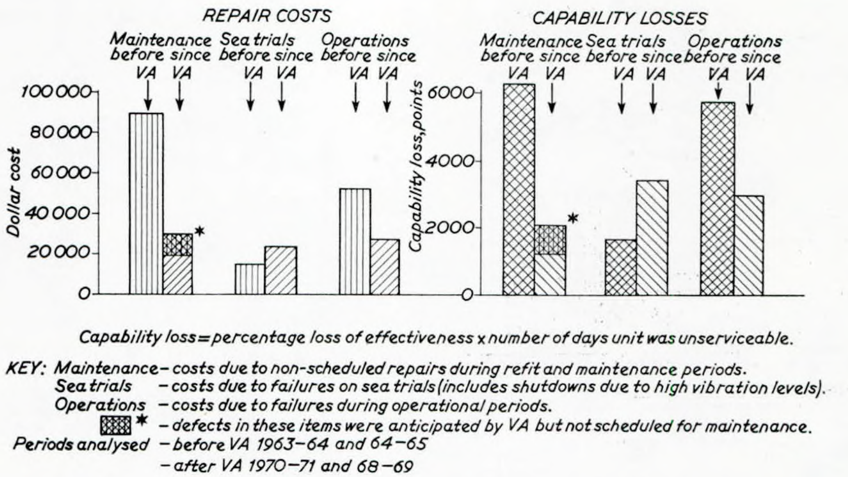


FIG. 16—*Comparison of repair cost (cost effectiveness)—and capability loss (value effectiveness)—Survey covers the main engines and eleven of the auxiliary engines in the ships over two two-year periods—Summary of totals for all six ships prior to and since the introduction of dockyard pre- and post-refit VA analysis programme*

TABLE V—THE EFFECTIVENESS OF VA

A (Sample size 6 ships and 12 types of machine over two two-year periods)

		No. of failures		Cost effectiveness, $		Value effectiveness (points)	
		Before VA	With VA	Repair costs before VA	Repair costs since VA	Capability loss before VA	Capability loss since VA
Sea trials	(a)	5	12	11 900	26 800	1 360	3160
Operational	(b)	20	19	52 300	27 450	5 535	2625
Unscheduled maintenance	(c)	25	14	90 000	32 000	6 510	1680
Total				154 200	86 250	13 405	7465

B (Totals for all machines on the class of 20 ships per year)

		Cost effectiveness, $			Value effectiveness, equivalent $*		
		Repair costs before VA	Repair costs since VA	Savings	Capability loss before VA	Capability loss since VA	Savings
Sea trials	a × 3·33	39 700	89 300	—49 600	453 000	1 053 000	—600 000
Operational	b × 3·33	174 300	91 500	82 800	1 845 000	875 000	950 000
Unscheduled maintenance	c × 3·33	300 000	106 700	193 300	2 170 000	593 000	1 577 000
Total per year		514 000	287 500	226 500	4 468 000	2 521 000	1 947 000
Sea trials and operational costs per year	(a + b) × 3·33	214 000	180 800	33 200			

* For this evaluation 100 points is rated at $10 000.

been compared with the running hours of machines which were rated "moderate".

The significant factor which emerged from this analysis was that machines which were rated good by the post overhaul VA have an operating life about twice that of the machines which were rated moderate after overhaul. The results for three types of machine are given in Fig. 17. Similar results have been obtained in the exploratory quality control programme discussed earlier where, because of ship scheduling commitments, some machines were put back into service before the repair facility was able to get them into good condition.

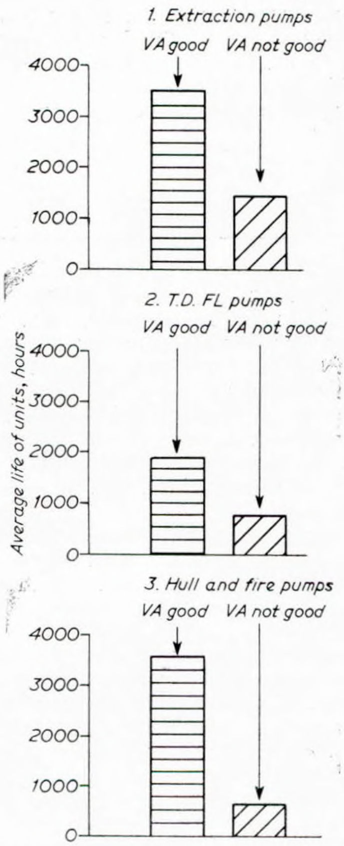


FIG. 17—*The effectiveness of vibration analysis as a quality control tool—Operating life of dockyard overhauled units as a function of VA condition at post refit survey*

DISCUSSION OF THE ANALYSES

Unscheduled Maintenance during Ships' Refit and Minor Maintenance Periods

The most pronounced effect of the dockyard VA programme has been the 66 per cent estimated cost reduction in unscheduled maintenance from \$300 000 (£120 000) a year to \$106 700 (£42 400) a year. Fig. 16 includes a note, resulting from a detailed check by the author, which shows that about 40 per cent of this remaining unscheduled maintenance work had in fact been anticipated by VA. In some cases these repair jobs were not scheduled at the appropriate period because of maintenance manpower shortages. In other cases, rapid deterioration of the machines, subsequent to the VA survey and prior to the scheduled maintenance period, resulted in emergency maintenance procedures.

The increased availability of the ships has been rated as worth \$1 947 000 (£778 000) a year, of which \$1 577 000 (£628 000) has been the result of the reduction of unscheduled maintenance. It has been argued that, since the ships are on extended steaming notice during refit, this latter figure overrates the value effectiveness of VA. However, it is considered that the reduction of unscheduled maintenance will have resulted in better-controlled maintenance planning and, in the long run, a reduction in the time allocated to this phase. Therefore the inclusion of this figure in the value effectiveness study is considered valid.

Machinery Repairs During the Sea Trials Phase

The use of VA as a machinery health monitor during sea trials has enabled a large number of defects to be diagnosed and rectified. Thus machinery repair costs during this phase have more than doubled since VA was introduced, with a consequent reduction of 45 per cent of repair and ships' outage costs during the operational period. From another point of view, however, it can be said that failures during the sea trials phase are largely the result of inadequate quality control of machinery repair during the overhaul period. With this in mind, the CF have introduced the programme of testing the principal ships' auxiliary machines, with the performance parameters including the use of VA, prior to their installation in the ships. This programme is rated by the CF as being successful, but there have been some machinery failures because the time has not always been available to correct the defects diagnosed by VA.

Machinery Failures during the Operational Phase

The gratifying reduction of 45 per cent in the cost of machinery failures during the operational period is a direct result of the dockyard programme. It presents a strong case for the re-introduction of the shipboard periodic VA monitoring programmes. Clearly, if the remaining failures could be anticipated, and provided that staff be made available to effect repairs at suitable times in the ships' schedules, then breakdowns during operational missions could be eliminated.

Improvement of the VA Programmes

The limitations to the present dockyard VA programme arise from two sources. First, the pre-refit survey is often carried out several months before refit, and some of the machines designated for repair fail before the refit starts. Secondly, perusal of the data has shown that VA predictions are not fully integrated into the machinery overhaul planning function. Many machines which are clearly indicated to be in good condition by VA surveys are still removed because of planned maintenance routines and, conversely, some machines which clearly needed attention have not received it. It can be seen that the introduction of shipboard monitoring of machinery, coupled with a more flexible approach to machinery repair, will enable the naval engineering authorities to run their machines to a more consistent "well worn" level and thus to deploy their limited maintenance manpower in a more efficient manner.

Testing of Overhauled Machinery

It has been shown that VA is an effective quality control tool during post overhaul testing. It can be readily appreciated that it is easier to rectify machinery defects in a test bay than in the bowels of a ship. Thus, there is no doubt that when adequate time for machinery rectification is allowed in the testing schedules, the programme for performance testing of overhauled machines in the test bays, utilizing VA as a quality control parameter, will significantly improve shipboard machinery reliability.

Establishment of VA Norms on New Types of Machines

When a new type of machine is purchased, it is recommended that provision to determine the VA norms be written into the contract. This requires that the first three units, known to be in good condition, have VA surveys run during acceptance testing. The average levels from these units adequately define the vibration norms for this type of machine for operational and future overhaul purposes.

THE WAY AHEAD

In conclusion, then, the CF have developed a system of machinery health monitoring based upon the use of the portable octave band vibration analyser. A 45 per cent reduction in the cost of rotating and reciprocating machinery breakdowns has been achieved. Ships' outages due to machinery failures have been similarly reduced. There are very considerable gains still to be made by more efficient utilization of the present machinery health information system.

Other areas of development, which the author expects to see implemented, are the combined use of vibration analysis and spectrometric oil analysis or filter particle examination, together with the measurements of the flow, pressure, temperature and work parameters of the concerned machines to give a complete record of machinery condition on a periodic or real time basis. Maintenance routines will be based on the analysis of these records.

The author is convinced, for example, that in ships the cost of maintenance for aircraft type gas turbines will be considerably reduced when this type of monitoring is introduced as a control tool.

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2) RAPER, R. G. 1970–71. Parsons Memorial Lecture—"Designing Warships for a Cost Effective Life." *Proc.I.Mech.E.*, Vol. 185, Part 1, p. 159.
3) KASSAN, K. 1972. "A Proposed Simplified Steam Plant for Marine Propulsion." *Jo. SNAME*, April.

ACKNOWLEDGEMENTS

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Discussion

Mr. D. H. CAMERON said that the Ship Department at Bath had had liaison with NETE, and had followed the programme with great interest.

Papers on vibration analysis were usually an account of successful applications of the technique. However, laboratories and offices which administered NDT in the field usually contained at least one cynic who had little respect for the method. Perhaps the truth lay somewhere in the centre, and everyone would welcome greater emphasis not on the successes of VA but on a critical examination of some failures. Most practitioners could show a success, as in the details he gave of a bearing monitored over a period of years, showing a progressive deterioration, but perhaps his next example should be set against it. Here a new pump impeller was shown to give higher vibration levels than a worn one. This was just one example which showed that VA was a tool in machinery condition assessment, but not one which could stand alone. He asked if Mr. Glew could give an indication of the percentage of defects which could be diagnosed successfully by VA, and how many of these showed other more readily discernible symptoms. Did the diesel generator of Table 1 alarm the operator to request a second survey?

The problem of vibration levels standing alone was complex. The use of a computer processor made the technique appear a soundly based science but the experience of MOD(N) had left them questioning whether or not it was, in large part, an operator art needing some sympathetic vibration between the operator and the machine as well as Vdb on the meter. He wondered if Mr. Glew could expand on the operator training. How successful would the trained person be in diagnosing from a set of numbers on a page without personal observation of what the output was, what the suction was, and so forth?

Turning to the costing of benefits, he said that while he did not necessarily agree with the basis of the figures he accepted wholeheartedly the pressures to produce them. Genuine savings from a new maintenance technique must, in the final assessment, come from less total output to achieve the same, or better, availability. The case of warship capability was a complicating factor, but in a pre-refit survey, a decision to leave an equipment which was scheduled for overhaul must be based on the expectation of running on till the next scheduled refit. This was particularly true of major equipment. The savings thus arose from identification of areas where existing maintenance schedules were too strict. To use VA alone in this respect was, he submitted, an act of considerable faith. However, producing figures of cost benefit always aroused great interest, and he was sure that many present would pursue this further.

With regard to the establishing of VA norms on new machines, Mr. Cameron said that naval equipment, and particularly warship equipment, was designed to withstand environmental conditions not found ashore. Some of these influenced the design and, in consequence, the vibration signature. Was low machine vibration an indication of reliability, and did one encounter a law of diminishing returns? It seemed rather odd to ask a manufacturer to build three machines from which a level would be set and which would then be contractually binding. This was no criticism; it was, in fact, the method MOD(N) used, but was there any way that an acceptance criterion could be set on ordering a new equipment? This, after all, was what was required.

Finally, he asked Mr. Glew what progress had been made in extending the periods between refits of Canadian warships following the success of his work, and, if such a step had been taken, what had been the results?

Mr. A. R. HINSON said that he had found the paper interesting in view of discussions which had recently taken place with manufacturers of diesel engines and consultants who were investigating new methods to monitor diesel engines in merchant ships.

It seemed feasible to monitor and analyse vibration signals by means of a computing device which would initiate an alarm when the frequencies varied from predetermined limits.

It was generally accepted that for pressure, temperature, flow and liquid level, sensors were used to monitor plant. One of the few places where it was usual to monitor vibrations was on the steam turbine. Vibration pick-ups were also fitted to the turbo-blowers on large slow-running diesels. On both these applications a filter was included in the circuit to eliminate low frequency vibration from the hull. A beginning had therefore been made in merchant ships in the use of vibration monitoring where, as on turbines, vibration was usually the first sign of trouble.

Mr. Glew had indicated that vibration analysis might be extended to other parts of the plant. He shared this view (a correct one) with Mr. Carmody, who had read a paper recently on the measurement of vibration as a diagnostic tool.

The author used linear pick-ups in his work, and had shown that vibration analysis might give early warning of failures, but Mr. Hinson had the impression that the technique the author described was more suitable to ball and roller bearings than to plain bearings. This seemed reasonable because he thought a pitted ball race would give a better signal than a wiped plain bearing. If the plain bearing failed a good signal would be generated only after substantial damage had occurred. Would Mr. Glew comment on this aspect?

He also wondered if Mr. Glew had any comment to make on the use of strain gauges as load cells to measure the cyclic load on the supports of plain bearings. It was the stiffness of the support which really determined the sensitivity of the alarm system—if the support was very stiff the signal would be quite small.

The author had made no reference to the use of torsional pick-ups and diesel engines and Mr. Hinson wondered if he had used them.

Torsional vibration was more difficult to monitor continuously. Probably strain gauges were the most reliable sensors but they could only measure torque and therefore were usually located near a node. For torsional amplitude a pick-up was usually attached to the forward end of the crankshaft. These pick-ups seemed to be unreliable when used over long periods.

Excessive torsional vibration could suddenly develop in service in a multicylinder oil engine due to misfiring of one cylinder, or due to appreciable unbalance of combustion although the latter was generally a gradual process.

In some cases loss of one cylinder firing could cause sufficient increase in vibratory torque to damage a highly loaded portion of the transmission, e.g. the rubber element of a flexible coupling.

Where calculations showed that one cylinder mis-firing could be serious, Lloyd's Register of Shipping called for measurement of the vibration and for suitable monitoring to be provided in service. Exhaust gas temperature monitoring was the only practical method at present.

To give some idea of the sort of overloading possible, the calculated vibratory torque for a 4000 HP geared propulsion installation was shown in Fig. 18.

The measured vibratory torque for a 10 800 HP directly coupled installation is shown in Fig. 19.

It will be seen that in both cases misfiring on one cylinder produced excessively high vibratory torques.

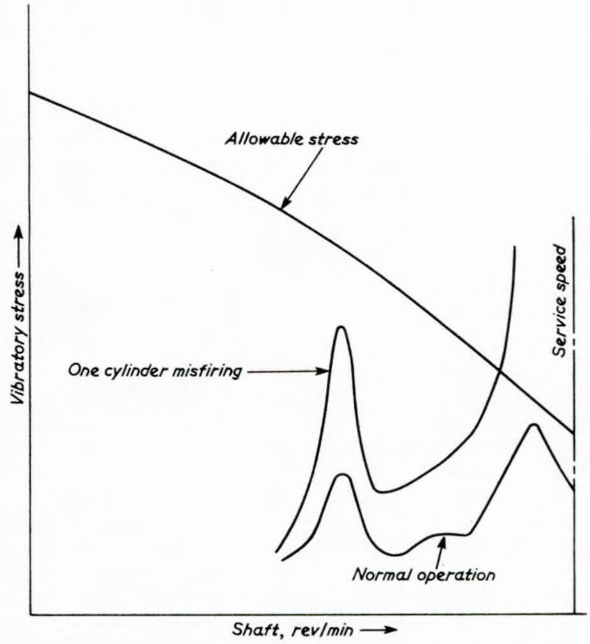

FIG. 19—*The measured vibratory torque for a 10 000 hp directly coupled installation*

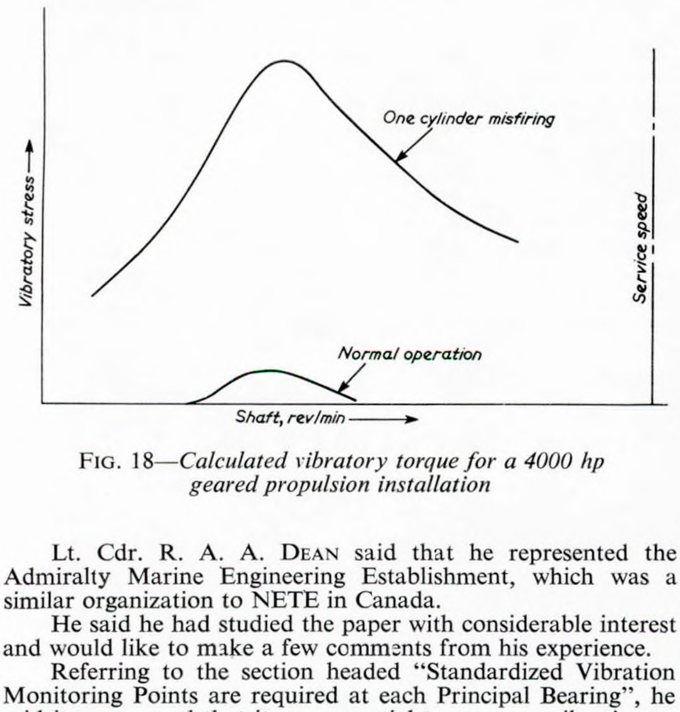

FIG. 18—*Calculated vibratory torque for a 4000 hp geared propulsion installation*

Lt. Cdr. R. A. A. DEAN said that he represented the Admiralty Marine Engineering Establishment, which was a similar organization to NETE in Canada.

He said he had studied the paper with considerable interest and would like to make a few comments from his experience.

Referring to the section headed "Standardized Vibration Monitoring Points are required at each Principal Bearing", he said it was agreed that it was essential to measure vibration at the bearing areas. Other measuring points, e.g. above the mounts, were likely to give misleading information. Measurements taken on one machine at AMEE had shown that a change of vibration of 12 dBs at the bearing was attenuated to a change of 5 dBs above the mount. It was also agreed that vibration transducers must be positively attached to the machine for taking readings. Hand held probes depended on the operator's "touch", and were less acceptable.

There was a limitation to any direct relationship between the machinery state and the vibration readings taken. In the introduction at the beginning of the paper Mr. Glew said, in the third paragraph: "The state of the machine is determined by comparing these (V/A) measurements with the machinery norms and with the previous readings at each point". A considerable amount of wear could be experienced in some machines without an identifiable change in vibration levels taking place (see Table VI). The air compressor ran for eighteen months deliberately without an oil change, and although wear obviously took place the V/A readings showed no significant alteration.

Many defects were not detectable by V/A. In measurements taken over two years on a number of machines only 50 per cent of the defects that had occurred were predicted by a significant change in vibration signature (see Table VII).

With reference to the VIBANAL system, it was stated in the paper: "the format enables the analyst to determine quickly the faults in the machine and to recommend the best refurbishment". It was necessary to differentiate between analysis and diagnosis. The aim of analysis was to highlight significant trends from the mass of data recorded. The aim of diagnosis was to interpret these trends in terms of machine condition. Analysis was a reasonably straightforward process well suited to computer techniques, but in the experience of AMEE, diagnosis was more difficult than was suggested. For example, at frequencies above 1K Hz there could be several answers. In Fig. 3, which showed the vibration record of a typical marine pump, the high readings above 1K Hz were assumed to be an indication of bearing problems. But on a feed pump at AMEE, where similar high readings in the same frequency band had been recorded, the cause was cavitation of fluid through a valve adjacent to the pump (see Table VIII). Also, at lower frequencies high vibration level could be recorded for which there was apparently no obvious cause in terms of component frequencies within the machine (see Table IX).

The author had mentioned the use, in conjunction with V/A, of other quantitative methods of fault diagnosis, e.g. spectrometric oil analysis. AMEE considered that a combined approach of this sort, using all the tools available, was essential. V/A was no panacea. It was a very significant advance over subjective methods of vibration measurement, but not all the faults on a machine could be directly related to change in vibration signature, and where they were related considerable experience was required to interpret the information effectively.

Mr. L. F. MOORE, F.I.Mar.E., said that the company with which he was connected manufactured "old-fashioned" types of machines which used pistons—in this case in refrigeration

TABLE VI—EXAMPLE OF LIMITED INCREASE OF READINGS OVER A LONG PERIOD OF SURVEILLANCE
AIR COMPRESSOR—NO. ONE (1) VERTICAL

1/3 Oct Mid Freq C/Sec	1	2	3	4	5	6	7	8	9	10	11	12	13
	28 Apr 1970	8 Jun 1970	8 Sept 1970	14 Oct 1970	14 Dec 1970	18 Mar 1971	29 Apr 1971	27 May 1971	22 July 1971	8 Aug 1971	18 Oct 1971	18 Nov 1971	19 Nov 1971
20	59	58	58	57	58	63	64	63	55	69	57	72	66
25	70	67	67	68	66	70	71	69	65	71	61	73	65
32	73	71	71	72	72	73	75	71	68	76	67	76	70
40	81	82	82	84	83	83	84	82	79	82	81	82	85
50	81	80	82	82	81	80	80	79	78	81	77	79	82
64	88	80	81	80	78	78	78	76	76	77	75	77	80
80	92	87	88	89	88	88	88	86	87	87	86	87	89
100	88	84	85	86	85	86	86	82	83	84	83	84	87
125	87	83	83	84	82	80	82	80	80	81	82	83	82
160	92	95	89	89	89	89	88	85	86	89	90	90	95
200	92	94	87	89	90	91	90	87	87	89	89	90	95
250	91	95	91	90	91	91	92	91	93	91	93	95	93
320	93	91	89	89	90	90	89	87	88	88	89	90	91
400	95	94	93	94	94	94	93	92	90	92	92	93	91
500	99	100	99	99	98	99	100	99	97	99	98	98	96
640	92	93	91	93	92	92	94	92	90	92	91	91	90
800	86	87	87	89	88	89	89	86	85	87	86	87	86
1000	92	92	91	91	91	94	93	92	89	92	91	90	91
1250	94	94	92	91	93	96	94	91	91	93	90	92	92
1600	100	100	99	99	99	102	101	99	95	99	100	100	104
2000	102	102	102	101	103	103	101	99	100	103	103	104	103
2500	99	98	98	99	99	98	97	95	99	99	99	99	100
3200	102	101	102	103	103	104	105	103	102	104	103	105	103
4000	102	100	100	102	102	104	103	100	100	102	100	103	104
5000	101	99	99	100	102	107	103	100	101	101	99	100	100
6400	103	102	100	102	105	111	105	101	105	106	102	105	106
8000	96	95	94	96	98	108	100	96	102	99	95	97	98
10 000	98	94	93	96	95	98	98	95	99	97	93	96	95
Total	112	111	111	112	112	115	114	110	111	114	114	113	113

compressors. Most of the comments, so far, had been related to purely rotating machinery.

He asked if Mr. Glew would enlighten the audience on whether all octave bands from 8 Hz to 8 kHz were used in his measurements, as it appeared from the figures shown in the paper that a lot of the readings only went down to the 31 Hz octave band. He wondered if it was normal practice to take measurements in the 8 Hz and 16 Hz octave bands. The British Standard on vibration measurement for machinery[(1)] and also the draft ISO standard[(2)] recommended that the frequency range

TABLE VII—SUMMARY OF DEFECTS
(OCTOBER 1971 TO JUNE 1973)

Machine	Number of Machines Involved	Method of V/A Measurement	Defect Number	Defect or Occurrence	Whether V/A gave warning of trouble	Whether V/A accurately diagnosed defect
Air Compressors Belt Driven	3	Total level with hand-held probe at bearing area	1	Misalignment	Yes	Yes
Vertical Pumps (Motor Driven)	3	1/3 Octave measured above mounts	2	Deterioration in performance	No	No
			3	Fluid cavitation in discharge control valve	Yes	No
			4	Failure of motor bearing (Non-drive end)	No	No
			5	Pump end interstage sealing ring broken	No	No
			6	Circumferential movement of impeller on shaft	Yes	No
Horizontal Pumps (Motor Driven)	3	Total level with hand held probe	7	Misalignment	Yes	No
		also 1/3 Octave at bearing end	8	Worn coupling bolts and bushes	Yes	No

should be 10 Hz to 1 kHz. If measurements were being taken on slow speed machines, such as refrigeration compressors, running at 750 rev/min, one had a fundamental frequency at 12·5 Hz and a second order component at 25 Hz, both of which would be liable to be "lost" if one started at the 31 Hz octave band.

He also asked Mr. Glew if at the top end he found much useful information in the 2 kHz, 4 kHz and 8 kHz octave bands, and especially when taking overall levels as, when looking at Fig. 13, there seemed to be a strong peak round about 3 kHz. When taking overall levels one was liable to get a little bit mixed up with the true vibrations at these higher frequencies and transducer resonances.

He said that he had found, when taking measurements on the reciprocating machines mentioned earlier, that the second order unbalanced forces and couples round about 25 Hz were liable to predominate in the vibration spectrum. If one then took an overall acceleration level one was liable to lose track of any increases in vibration at higher frequencies, possibly due to valve damage. He asked Mr. Glew if he would comment on this.

A lot of people used Fig. 2, the I.R.D. curves for assessing machinery vibration severity. In the last nine years there had been two standards produced[1, 3] suggesting that instead of having one vibration severity chart covering all classes of machinery, that machinery should be divided in to six different classes ranging from, say, small electric motors on rigid foundations to large unbalanced reciprocating machines on resilient mountings. He did not think it was fair to use the same criterion for all classes.

REFERENCES

1) British Standards Institution B.S.4675: 1971 "Recommendations for a basis for comparative evaluation of vibration in machinery" (London).
2) International Organization for Standardization ISO/DIS.2954 (1973) "Mechanical vibration of rotating and reciprocating machinery—requirements for instruments for measuring vibration severity" (Switzerland).
3) Verein Deutscher Ingenicure VDI.2056 (1964) "Criteria for assessing mechanical vibrations of machines".

Mr. G. MOYES noticed that on several of Mr. Glew's machines there were between 11 and 15 monitoring points, with three directions each, which was 33 directions of analysis per machine times eight octave bands. This was a great number of numbers to record. Coupled with moving around a machine in perhaps a difficult ship-board environment, this represented a great deal of work. He had not seen a costing of this large and difficult amount of work in Mr. Glew's cost effectiveness, and wondered if he could comment on this.

The scatter which Mr. Glew showed on machines was quite large, but he used a small departure from norm to signify the alarm light. Recent experience at MOD had been that the very detailed way in which one attached the transducer to a machine might scatter the vibration readings by at least as much again as the scatter used as an alarm. He said he would be interested to know whether Mr. Glew's organization saw this as a problem.

Lt. Cdr. D. P. KEOHANE, R.N. thought that Commander Dean had poured rather a lot of cold water on what had been done in the Navy in the past couple of years. They might usefully have heard, perhaps, something of the distinction which should be made between vibration monitoring and analysis. The former was the taking of a series of readings to monitor the state of the machine over a period of time. Once one detected some change one had then to decide what significance to attach to it by means of analysis—a separate problem altogether. He did not think this had been made clear. It was just possible in the case mentioned that the monitoring was very effective but that the analysis was bad.

In Fig. 10 rises in vibration seemed to have occurred at every measuring point, both with the governor unbalance and the turbine rotor unbalance. This did rather support his view that measurements could be made on the base of a machine remote from defective parts, and still be significant, although, of course, they might not necessarily be of the same magnitude as at the precise point. Again, it was a matter of deciding what significance to attach to the changes in vibration.

Mr. NEWTON asked Mr. Glew if he got sufficient information from an octave system so that he could virtually discard a discrete analyser.

He said that he found that he was fixing blocks in the various bearing areas of his machinery to speed up this work. In other words, he did not want a welder following him around on the job.

TABLE VIII—EXAMPLE OF READINGS
FEED PUMP

1/3 Oct Mid Freq C/Sec	No. 1V 17/11/71	No. 1V 2/12/71	± AdB	No. 1R 17/11/71	No. 1R 2/12/71	± AdB	No. 2V 17/11/71	No. 2V 2/12/71	± AdB	No. 2R 17/11/71	No. 2R 2/12/71	± AdB
20	60	51		50	49		65	57		57	51	
25	62	54		51	50		66	60		57	54	
32	64	57		56	55		65	60		57	56	
40	65	63		62	64		63	58		61	61	
50	72	63		75	76		63	58		74	74	
64	77	77		80	82		68	64		78	74	
80	66	65		65	68		63	69		74	75	
100	76	75		65	65		73	66		74	73	
125	84	87		66	73		80	77		70	77	
160	78	83		75	77		83	83		76	81	
200	71	73		77	77		83	83		78	78	
250	71	70		77	77		73	75		79	78	
320	73	67		82	78		75	72		81	79	
400	74	70		80	78		74	73		80	80	
500	75	70		80	76		78	78		81	77	
640	89	85		94	86		82	82		95	86	
800	94	93		94	87		90	92		95	90	
1000	103	95		91	84		96	95		96	88	
1250	102	94		93	84		99	94		95	88	
1600	101	90	−11	98	86	−12	101	89	−12	102	93	
2000	102	87	−15	93	84		99	89	−10	110	88	−22
2500	96	84	−12	87	87		95	89		105	88	−17
3200	87	78		82	80		88	84		89	81	
4000	89	79	−10	82	81		84	85		86	81	
5000	87	83		83	84		83	85		86	84	
6400	84	77		83	83		81	83		88	87	
8000	87	80		88	89		80	87		101	101	
10 000	90	83		89	88		91	91		100	98	
Total	109	102		105	100		109	104		111	104	

He said it had unfortunately been his experience that defects did not conveniently follow a trend. In other words, there was no gradual build up of defects. A defect, when it occurred, was invariably catastrophic, and happened just before he was able to go along and take a measurement. He wondered if this had also been Mr. Glew's experience.

Mr. T. CARMODY considered that the whole basis for the paper was the effectiveness of vibration analysis; in other words, could you save money by it?

Over the past three weeks he had been involved in tests on three equipments required for shipboard use. One, a 750 Kw turbo driven generator which, if it had not been for vibration

TABLE IX—EXAMPLE OF LIMITED RELATIONSHIP BETWEEN ACTUAL AND CALCULATED FREQUENCIES
FEED PUMP—NO. ONE (1) RADIAL

Peak Frequencies 23/10/72	Level dB	Discrete Frequency	Level dB	Previously Calculated Frequencies
64	82	60	82	Motor and Pump speed = 59·4 c/s
		185	83	No related component
200	78	221	57	
		222	57	
		225	60	
		315	61	Pump 2nd–7th Stage Impellers = 350 c/s
320	80	325	57	
		351	74	
				Pump Diffusers = 415 c/s
				Pump 1st Stage Impeller = 475 c/s
				Top and Bottom Motor Bearings = 594 c/s
		750	81	Plateau of equal levels. No related component
		758	81	
		774	82	Plateau of equal levels. No related component
		777	82	
800	87	831	76	Plateau of equal levels. No related component
		837	76	
1600	92	1430	81	No related component
		1610	80	No related component
		1710	79	Plateau of equal levels. No related component
		1760	79	
				Motor Rotor = 2380 c/s
				Motor Rotor = 2850 c/s
		9300	84	No related component
10 000	100	9600	84	
		10 500	89	
		10 700	89	

analysis highlighting a problem area, would have eventually failed in service. Fortunately, the checks carried out by the firm with a very simple vibration measuring equipment had shown that something was obviously wrong. On stripping, it was found that thermal distortion had taken place causing considerable out of balance. Money had been saved.

Secondly, of two blowers destined for a vessel, which was required to run quietly, it had been suggested to the manufacturer of the diesel engines that it would be as well to have vibration checks carried out on the turbo-blowers prior to their despatch. On one blower it was found that the vibration level was double that specified by the design authority. Although the design authority considered this acceptable, it was decided to investigate.

On stripping the two blowers, the build standard on one was considered questionable and on the second there had been a labyrinth rub. Again, money had been saved.

The position at which vibration measurements are taken is to a degree entirely up to the person involved. Obviously, measurements taken at a bearing would differ from those obtained on the base of the machines, but, however, both were useful and this was the most important factor.

Mr. Carmody also said that what had to be looked for, in his opinion, was the significant change in levels. One could initially take a set of readings and then decide whether they were acceptable as the norm.

Consider a bad machine which had to be kept in service for as long as possible. In this case what needed to be known was whether the machine was deteriorating further or not, and when it was near to failure. Again a significant change in level was being looked for. An increase of 6 dB seemed to be the magic figure used at present to indicate a degree of deterioration requiring investigation. However, it was really up to the engineer to specify his known maxim, taking into account his knowledge of the machine, and any past vibration records.

Vibration analysis in itself would not cover the whole machinery failure spectrum. Spectrometric oil analysis, magnetic chip detection, etc., were required, depending on the machine.

In reply to the gentleman from AMEE, having spent 4½ happy years at the establishment, one of the things discovered after looking for hours at readings from compressors, was that if you wanted to prove the suitability of vibration measurement as a maintenance tool, you should keep away from reciprocating machinery. When measuring compressor vibrations very peculiar things happened. On checking at the motor, one picked up dominating frequencies of the compressor, and on the compressor dominant frequencies from the motor were picked up; most confusing. He had come to the conclusion that the measuring of vibrations on reciprocating machinery was something of an art that one could spend years trying to sort it out.

In conclusion, he considered that at present, vibration analysis was more suited to, and would save most money, in the rotating machinery field.

Examinations had been carried out on machines time and time again. On the blowers they had specified a figure but this had been picked out of the air and the designers were prepared to double this, or possibly treble it, provided nobody shouted. It was a way of getting machines out of the factory.

Correspondence

Capt. R. M. Inches, R.N., F.I.Mar.E., wrote that he was happy to note the further progress in the application of VA to machinery maintenance made in the Canadian Forces Sea Element whom he regarded as being amongst the leaders in this field.

Since cost effectiveness was the current criterion, it was especially satisfying to see how they had managed to apply this to VA as a machinery maintenance aid, and with what impressive results.

Apart from further direct gains to be expected there should also be some indirect ones. For example the light shed on "inadequate quality control of machinery repair during the overhaul period" will presumably lead to separate curative action which should save more money.

The plans for further development were also very encouraging. In particular the continuous and careful monitoring of running temperatures for sudden variations (as distinct from the reading at intervals, in poor light, of a bulb thermometer removed from an oil, or air, pocket) would pay good dividends. Spectrametric oil analysis on the other hand would be recommended for keeping in reserve as a laboratory back-up because of the cost of the equipment and the difficulties associated with using it properly.

There was one additional benefit from the successful application of a new diagnastic tool much as VA to the maintenance of ships machinery. That was, increased job satisfaction and raised morale first of the ER Department and then of the whole ship's company. There was no direct measure for this yet but there were some indirect ones and Capt. Inches wondered if the author could say anything on this point.

Author's Reply

Mr. Glew agreed with Mr. Cameron that if one wanted to improve one's technique one should study one's failures in detail.

As regarded the new impeller which had higher vibration than a worn one, this was typical of pump impellers. He had generally found that as impellers wore, the vibrations dropped, because they were not doing so much work. On the other hand, if there were bearing defects or wear ring rubs, etc., the vibration would increase by at least 10 VdB. The practice at NETE was to conduct a VA survey and a performance check to see if the flow pressure curve had dropped away from the rated point. This combination gave a good indication of eroded pump internals as well as the defects normally identifiable by VA.

With regard to the technical effectiveness of VA, Lt. Cdr. May, who had been in charge of the VA programme at Halifax Dockyard before he had come to NETE as OIC, reckoned that the dockyard VA predictions were correct in 85% of all cases. He used to have people taking performance and VA readings and could do a ship's analysis without going near the ship. On the other hand, on the spot analysis was invaluable. It had been alleged on one occasion that there was something wrong with the Diesel Generator referred to in Table 1 because, on shipboard visit, a bottle of beer kept moving across the mess table.

Mr. Glew thought the key to successful vibration analysis on machinery was to train the engineering staff to use the VA tool and for them to use it in the same way as they used a screwdriver. But this seemed to be quite a difficult thing to achieve. Maybe this was because the octave band analyser required that the concept of frequency analysis be understood. As regarded the establishment of the vibration norms, this was very easy. NETE simply took all the readings from all the machines that were being measured, totted them up and averaged them, and said: "This average is our norm". This meant that the vibration readings from each machine were compared with the average levels in the fleet for that machine. One could argue that the norms would deteriorate over the years because one usually did not worry about a machine unless it was at least 6 dB above the norm. However, in practice the system worked well, as had the computerized readout, and one could usually pick up the defects from the "VIBANAL" programme. This was indicated in Figs. 10, 11 and 12. If one looked at the individual levels, in, say, the 63 Hz band, they did not look at all high, they were all below 100 Vdb. Likewise, the turbine unbalance readings in the 125 Hz band were all below 120 dB. However, the VIBANAL presentation clearly indicated the faults noted.

As regarded the determination of VA norms on new types of machine, he thought that every machine had its own distinctive patterns, and the only way to determine these was to get machines which were known to be in good condition, run them up on the initial acceptance test, check the performance and the VA levels. This method would inevitably cause additional initial costs, but he was sure that if this was taken into account contractually the manufacturer would be glad to accept it. In the long run he was going to know that his machines were good by using this as a monitoring tool in his own quality control branch. In fact, the aircraft people were doing that already.

In a written contribution, Mr. Glew added that the ships' period between refit had been extended from 16 months to 20 months. VA had enabled machinery life to be extended, NDT examination methods had improved the reliability of boiler systems, and improved paints had reduced hull corrosion problems.

A study of failure reports on the machines in the VA programme showed that 40% of all failures on these machines were predictable by VA. Faults which could not be anticipated included cooler failures, and fatigue failures of static components.

Mr. Glew said with conviction that Mr. Moyes was absolutely right. There was a lot of work in the octave band analysis. However the analyst's costs were included in the cost effectiveness study. These amounted to $26,000 per year. The analysts took overall readings triaxially or biaxially, and followed this up with an octave band survey in one plane. They had their own pet theories as to which was the best direction to monitor, and they had developed short cuts from the NETE plan, as they gained know-how. These short cuts had never been formalized.

There were indeed only one or two key octave bands for each point on each machine, and NETE had not as yet endeavoured to identify these. It was an area that could be developed. Mr. Moore and Mr. Hinson had raised questions which were relevant to this subject. For example, in the main engine turbines, if one got a badly wiped white-metal bearing, one got high vibration levels in the 4 and 8 kHz octave bands. Similarly in the main engine gearboxes, worn main bearings caused high vibrations at gear teeth harmonics. However, there were not too many cases where these octave band readings were useful, and there was a case for eliminating them from many machines, so that the VA crews did not have so much to do.

With regard to the question about the 6 dB indicator used in the VIBANAL program, he said he agreed that it could be 10 and not 6. Sometimes the readings would swing as much as 10 dB and the analyst then visually averaged a reading in the middle. Thus VIBANAL often showed a few scattered readings which were more than 6 Vdb above the norms, but as long as one did not get a definite trend they were cheerfully ignored. Figures 10, 11 and 12 clearly showed the indications resulting from two faults in the machine.

With regard to Mr. Keohane's comments about readings at the base, he thought a bit more work could be done in this area. If one could get away with readings at the base one would save a lot of time. In fact Figs. 10, 11 and 12 supported this view. Point 10 was at the base, above the resilient mounting (unfortunately it had been omitted in Fig. 9). This point clearly indicated the high 63 and 125 Hz vibration levels, but the VIBANAL presentation gave a feeling of confidence when the complete string of figures were available. One could then dig one's heels in and say "This machine is bad".

He agreed with the points raised by Lt. Cdr. Dean. He said NETE found that there was a greater spread in the vibration levels of individual new compressors, than in other machines, and often these readings remained constant over the two year life of the machine. The Canadian Navy had recently adopted a policy of performance testing compressors after their two year life, including VA and spectrometric analysis, and if the indications were satisfactory, the machines commenced a second cycle. They had started this procedure about nine months ago and five units out of about fifteen had been given a clean bill of health and their behaviour was being monitored during the second cycle.

As Mr. Hinson and Lt. Cdr. Dean had observed, if vibration levels dropped in pumps or compressors this could be an indication of reduced compression ratio and hence of performance deterioration. For example the U.S. Army Frankfurt Arsenal, as one of their diesel engine test facility performance monitors, put accelerometers on each cylinder head, and used the vibration levels on the cylinder head as an indication of the firing efficiency. This was certainly worth investigating.

Mr. Hinson and Lt. Cdr. Dean had concluded that monitoring of unattended machinery spaces was slowly gaining acceptance. The General Electric Corporation used VA in a very sophisticated way on their gas turbines and they were able to say "Yes, we can tell our gas turbine bearings are deteriorating before they actually fail". The Pratt and Whitney Airborne Integrated Data System (AIDS) was good, but it was weak in the area of bearing failure prediction. The monitoring of overall vibration levels between certain pass bands was not as effective as the GE method which, although it was shrouded in secrecy, seemed to be similar to the SKF bearing monitor method.

Mr. Hinson had mentioned the potential use of torsional vibration measurement as a machinery health monitor. NETE had not done any work in this area. He agreed with Mr. Hinson's conclusion that the NETE system gave an excellent prediction of bearing deterioration on anti-friction bearings, but did not predict plain bearing deterioration until quite a lot of wear had occurred.

In a written contribution, Mr. Glew added that à propos of torsional vibration, it was found that a misfiring cylinder caused a rocking motion on the crankcase on the resiliently mounted Naval engines, and crankcase mounted accelerometers clearly picked this up. None the less it appeared worthwhile investigating the availability and reliability of a cheap torsion meter for routine checks on a diesel engine. With regard to strain gauged pedestal bearings, NETE had not done any investigational work in this area; however, this was considered a potentially fruitful area of investigation.

With regard to the use of VA as a plain bearing fault indicator, on the main engine gearboxes the CAF policy was to use temperature monitors on each gear train bearing as a primary health monitor, with VA indications acting as a back-up.

In reply to Mr. Newton, Mr. Glew said the programme was set up for the dockyard team to use the IRD 330 narrow band analyser whenever they could not analyse a defect by the octave band analyser. In fact, they very seldom used it on their pre-refit or post refit surveys. One could usually get sufficient information from the octave band analysis, as illustrated in the paper (see Fig. 20). In fact, they had recently applied the octave band analyser to the Solar Saturn gas turbine, driving the 750 kW generators on the new ships.

There had been several compressor thrust bearing failures on these units after the ships were commissioned. Redesign of the bearing area was in progress, but in the meantime the ships' staffs were satisfactorily trend monitoring these engines. The measurements in Fig. 21, below, were typical of this engine.

Mr. Glew had found trend monitoring an excellent machinery health monitor. On shipboard machinery, deterioration usually took place over months and could easily be monitored. On aircraft gas turbines, the deterioration period could be minutes and so they required on-line monitoring systems, with suitable warning indicators.

He said he often used adhesives to secure the mild steel blocks to machines on investigational work. This was adequate for most purposes. Sometimes, however, (particularly on aircraft gas turbines) it was necessary to mount the accelerometers directly to the machines, or to mount them from special brackets to get satisfactory results.

Mr. Newton had asked if he had found the accelerometers affected by the temperature levels.

Mr. Glew said standard accelerometers were good to 280°C, and the high temperature ones were good to 555°C. The leads had the same temperature limitations but the connectors were the weak link in the system and often gave trouble. It appeared that their problems had been overcome in the aircraft field, and, as already mentioned, GE put permanent systems right inside the gas turbines to monitor the main bearing conditions.

Mr. Glew said the essence of his recommendation to the Canadian Navy had been that they should have their inspectors at the manufacturer plant at the beginning of each contract,

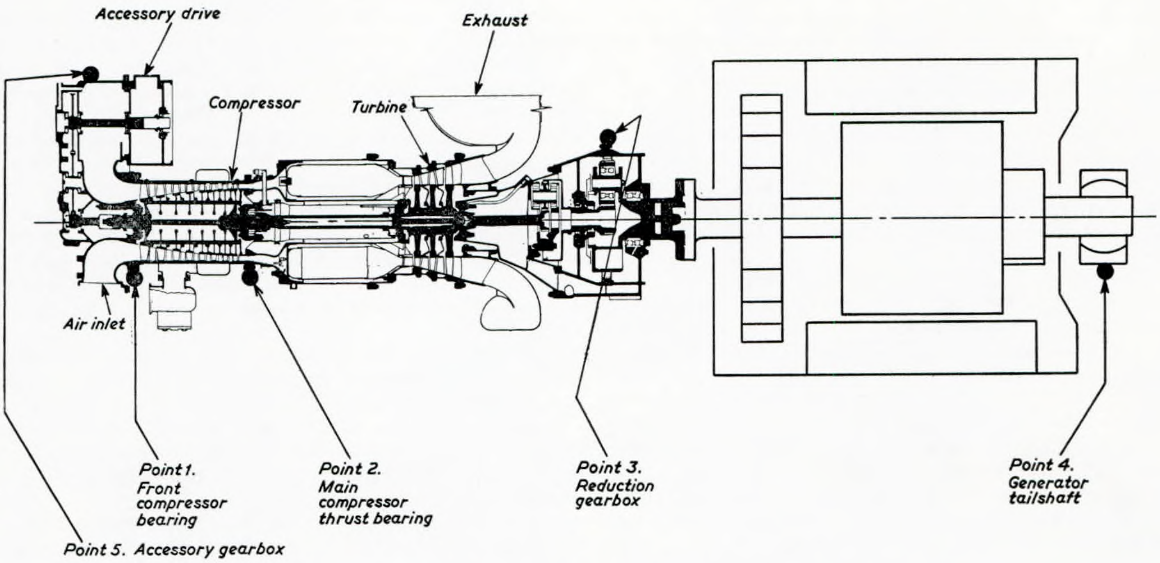

FIG. 20—*Solar Saturn 750 kW generator set C.A.F. vibration monitoring points*

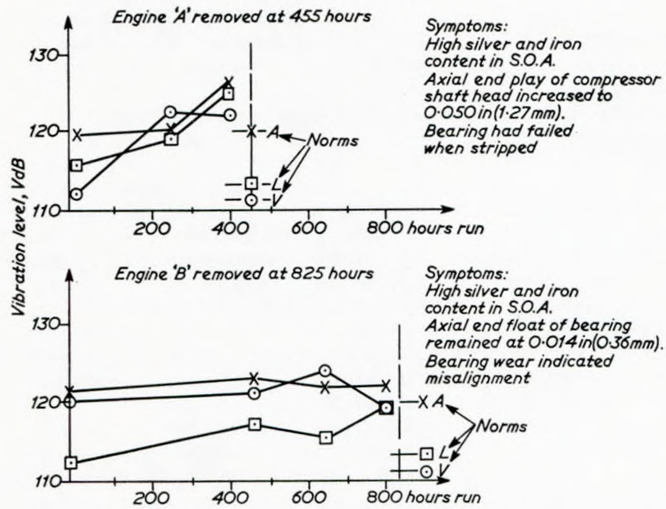

FIG. 21—*Examples of trend monitoring Solar Saturn engine*

to see that the first three machines were good, and to have VA taken from those machines.

He thought the wide variation one got in the levels of apparently good compressors was because there were big variations in the initial unbalance with which these things were assembled. In the past, a good balance on reciprocating compressors had not been necessary, but if one wanted to use VA as a quality control tool then a system of balancing (or half balancing) the rotating and reciprocating masses for the units that were being dealt with was essential. For example, aircraft piston engine crank-shaft assemblies were balanced, and these machines could be relied on to exhibit specific vibration patterns.

Captain Inches had emphasized the concept of the integrated machinery health monitoring program. Mr. Glew said that he had perhaps not emphasized sufficiently in the paper and discussion that the CAF engineering consensus was that for maintenance planning

a) VA is usually the most useful single parameter of a machine's health, but
b) it should always be read in conjunction with the relevant performance measurements and oil condition analysis of the machine.

NETE was presently engaged in a project to determine how best to measure these performance parameters accurately on board ships.

NETE was also studying the latest reported developments of VA as a bearing health monitor. These developments included mounting the accelerometers directly on the bearing housings inside the machines, and the investigation of high frequency shock pulses generated by ball bearings.

With regard to the last point raised by Captain Inches, Mr. Glew said that for a long time the dockyard programme was viewed with suspicion by many ships' crews, because the ships' staff were left out of the action. The ships' crews disliked the concept of the specialist teams coming on board, taking measurements, disappearing into the bright blue yonder, and issuing a report a month or so later.

However, now that the shipboard programme had begun in earnest, and the ships' machinery trend analysis was becoming basic to the health programme, the ships' staff were reacting in a very positive manner. He was sure this augured well for the future of the service.

Related Abstracts

Whirling of line shafting

The long standing problem of whirling due to first order (unbalance) and propeller blade order excitations is discussed. The source of the blade-order harmonic components and the close correlation with alignments and the resulting bearing loadings are traced. The possible damages caused by the phenomenon are illustrated by examples. The problems associated with frequency calculations are outlined and a computer program is described.—*Toms, A. E. and Martyn, D. K.: Trans.I.Mar.E., 1972, Vol. 84, pp. 176–191.*

An examination of the effects of variable inertia on the torsional vibrations of marine engine systems

The analysis of torsional vibrations in the running gear of reciprocating engine systems is normally carried out by neglecting the variation in inertia torques of the system arising from the motion of the reciprocating parts. When the variable inertia effect is allowed for the equation of motion taking into account the effect is non-linear. Assuming small displacements, the equation can be linearized to predict important characteristics of the motion. Such an equation when solved by numerical methods using a digital computer predicts the regions of instability and the manner in which the amplitude and frequency vary with the speed of rotation of the engine. The responses of the system show a modulation of amplitude and frequency at definite rotational speeds. The occurrence of such a modulation in amplitude and frequency is established by use of the process given by Wentzel, Kramers, Brillouin and Jeffreys generally known as the WKBJ approximation. The first order term in the equation and the forcing term which represents the outer impulse from the reciprocating parts, are investigated for their effect on the waveforms of the responses of the system. Further investigations of the effect of lower order external excitations on the characteristics of the motion are given. Theoretical results are compared with solutions of the equation obtained from an analogue computer. A discussion of some actual cases from engines in service is included.—*Carnegie, W. and Pasricha, M. S.: Trans.I.Mar.E., 1972, Vol. 84, pp. 160–167.*

Balancing of large turbine rotors

As the size and flexibility of turbine rotors increase, the problems associated with their smooth running becomes more complex. There is a growing awareness that the function of low speed balancing machines is quite inadequate for many types of flexible rotor. Both the rigid and flexible modal behaviour of such rotors is therefore discussed at some length and balancing machines are considered in this context. Modal balancing techniques are described, comparisons are drawn between this and low speed methods and the limitations and advantages of each are demonstrated. It is shown that modal balancing can be carried out by the application of a comparatively simple set of rules and standard graphical constructions. Although available elsewhere, these are included here for convenience of reference. Their use is amplified by discussion of the practical application of the rules to particular examples of balancing problems.—*Moore, L. S.: Trans.I.Mar.E., 1969, Vol. 81, pp. 105–115.*

Axial shaft vibration in large turbine-powered merchant ships

The paper mainly deals with axial shaft vibration in large turbine-powered passenger liners, although some aspects of the problem which relate to oil tankers and cargo vessels are considered. The methods employed in the evaluation of thrust block seating stiffness are discussed and values are given for a number of widely differing structures. The factors which influence the fore and aft stiffness of thrust seatings are examined, as also is the propriety of making radical changes in positioning the seating. Analysis of the shafting axial characteristics at the design stage indicate whether a maximum or a controlled degree of seating stiffness is required; passenger liners, with high shaft speeds, and tankers are shown to be subject to the latter stipulation. Suggested levels of acceptability for both vibration amplitude and acceleration are applied when discussing the amplitude-frequency response data, presented for five ships. The dynamic magnification applicable to merchant ship shaft systems, the thrust variation levels, which exist in practice at the thrust block, and the propeller thrust variation are subjects which are discussed at length and values are presented as basic design data.—*Couchman, A. A. I.: Trans.I.Mar.E., 1965, Vol. 77, pp. 53–83.*

The relationship between machinery vibration levels and machinery deterioration and failures

The paper proposes the use of periodic vibration monitoring as a diagnostic maintenance tool. For a small initial investment, the monitoring program will provide in most cases an early warning of impending equipment failure. Vibratory frequency is used to determine the nature of a malfunction, while the amplitude history indicates the seriousness of the problem. This information is useful in determining when to overhaul a machinery item and how extensive the overhaul should be; the approach is useful for all machinery installations, but it is particularly advantageous for shipboard equipment where unscheduled repairs may cause costly delays. The need for specifying maximum vibration levels initially is stressed, as also for a vibration survey after installation, to establish a basis for subsequent comparisons. A programme for periodic vibration monitoring has to be organized, with a threefold purpose: to be able to prevent deterioration from reaching failure point, to determine when overhaul is necessary, and to act as a guide to the extent of the overhaul. There are notes on setting up such a programme, and on the instrumentation available for measuring and recording vibration.—*Lundgaard, B.: Marine Technology, Jan. 1973, Vol. 10, No. 1, pp. 22–29.*