A VIBRATION ANALYSIS OF A DIESEL GENERATOR

BY

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Introduction

This article is an account of an unprogrammed vibration analysis survey of a Paxman 8YJCAZ 8 cylinder diesel generator in a LEANDER Class frigate. It does not cover all the causes of vibration in diesel engines and the frequencies at which they occur but is intended to show the results which may be achieved using vibration monitoring equipment held on board and the train of thought leading to the analysis and final diagnosis of the defect.

The ship was due to arrive in Hull for a week-end visit. High vibrations emanating from G2 diesel generator had prompted the ship to signal C-in-C Fleet in accordance with Fleet Engineering Order (FEO) 0305 requesting a vibration analysis survey of the engine.

A member of the Fleet Vibration Analysis Unit was in the area undertaking a vibration survey on an inshore survey vessel based at Lowestoft. On returning to Lowestoft he was contacted by telephone and instructed to proceed to Hull and to meet the frigate in question on arrival at Hull docks.

Equipment

The equipment used was an IRD Mechanalysis 820 Vibration Analyser, a 970 accelerometer transducer, and a 544 magnet. An IRD Mechanalysis 880 vibration analyser was available as a back-up in case of instrument failure. Spare transducers, cables, paper rolls, battery charger, strobe light, and probe were also carried.

The Survey

The engine was run up and loaded to 500 kW on ship's load. Using the IRD 820 Vibration Analyser overall vibration levels (filter out)[1] and Sharp filter, vibration spectra were recorded at points A, B and C (FIG. 1), and Broad filter spectra were recorded at points X and Y using the velocity mode (mm/s peak) in the horizontal vertical and axial planes.

Overall readings were taken above and below the resilient mounts on both the engine and the generator in the vertical plane only. In order to assist in the analysis of vibration spectra, exhaust pyrometer temperatures were also recorded. High vibration levels recorded throughout the machine gave cause for immediate concern regarding the machine's health and the safety of the operators. In order to pin-point the defect accurately it was necessary to run the machine for a second survey under 'no load' conditions. On this occasion it was considered prudent to reduce the number of survey points to a minimum. Only points A, B, and C, and above and below mounts were therefore surveyed. On completion of this survey the engine was shut down.

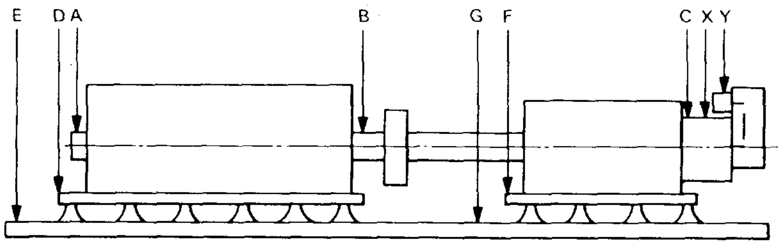

FIG. 1—ENGINE VIBRATION SURVEY POINTS

Quick Look Diagnosis

A quick look diagnosis was made and a verbal report rendered to the MEO that the engine had suffered damage-failure to the secondary balance weights and/or associated gearing. It was recommended that the engine should not be run except in an operational emergency.

POSSIBLE CAUSES OF VIBRATION ON PAXMAN 8YJ ENGINES

Engine and Generator

Half Order Vibration

Vibration at a frequency of half engine speed (600 c.p.m.) should be negligible, i.e. less than 2.00 mm/s, and is caused by faulty combustion. This can usually be confirmed by exhaust pyrometer temperature readings being out of specification, either high, low, or with a large variation between cylinders. Many defects can cause combustion problems and a comprehensive list is contained in BR 2000(27) Article 0306.

A limit of 5 mm/s half order vibration (i.e. vibrations at half running speed) is set by Paxmans but corrective action to the engine's combustion must be taken at vibration levels less than this if exhaust temperatures are out of specification.

1st Order Vibration

Vibration at a frequency of 1 × engine speed (1200 c.p.m.) is caused by unbalance or misalignment. Unbalance should not occur on service engines unless new components have been fitted. Static unbalance is usually indicated by high vibration levels in the radial planes but dynamic unbalance also shows an axial component. Misalignment is indicated by high vibration levels in the axial plane, usually accompanied by a second harmonic, and can be caused by mechanical misalignment between the engine and generator or defective resilient mountings causing the engine raft to twist.

The limit for first order vibration is 7 mm/s.

2nd Order Vibration

Vibration at a frequency of 2 × engine speed (2.4 K c.p.m.) may be caused by:

(*a*) *Mechanical looseness*—usually indicated by high vibration levels in the vertical plane.

(*b*) *Torsional vibration*—indicated by high vibration levels in the radial planes with smaller components at $2\frac{1}{2}$, $3\frac{1}{2}$, 4, $4\frac{1}{2}$ and 6 times engine speeds, caused by defective torsional vibration dampers. The amplitudes of this vibration are load-related and will attenuate considerably with a reduction in electrical load.

(*c*) *Secondary unbalance*—Most Paxman engines are fully balanced for external primary and secondary forces and couples. Paxman 8YJ engines are fitted with secondary balance weights driven from the crankshaft to overcome these secondary forces. Vibration due to unbalance is proportional to speed and is not load-related but in certain circumstances load may affect vibration amplitudes. Secondary balance problems are indicated by high vibration levels in all radial planes whether the machine is on load or running under no load conditions.

The limit for 2nd order vibration is 12 mm/s.

3rd Order Vibration

Vibration at a frequency of 3 × engine speed is rare unless poor combustion is present. However, 3.6K c.p.m. is equivalent to the electrical line frequency (60 Hz) and therefore may indicate problems within the electrical generator.

Possible Causes of Vibration of the Exciter

Due to the position of the exciter, which is mounted on a bedplate affixed to the top of the overhung generator end casing, vibration amplitude levels tend to be higher than those experienced on resiliently mounted machines. The running speed of the exciter is 2400 r.p.m. (2 × engine speed). Problems encountered on the exciter are:

(*a*) Unsatisfactory rolling element bearings indicated by Spike Energy (S/E) amplitudes above 0·8 g S/E.

(*b*) Misalignment between the generator and the exciter. Indicated by high amplitudes in the axial plane at a frequency of 1 × exciter speed.

(*c*) Damaged belt. Belt defects cause vibration at frequencies which are multiples of belt speed. Belt speed may be measured using a strobe light or calculated if the length of the belt, the pitch diameter of one pulley and the pulley speed are known.

$$\text{Belt r.p.m.} = \frac{\text{Pulley Pitch dia.} \times \pi \times \text{Pulley r.p.m.}}{\text{Belt length}}$$

(*d*) Problems causing vibration on the engine or generator accentuate vibration levels on the exciter.

RESULTS

The Analysis

On returning to C-in-C Fleet Portsmouth a detailed analysis of the vibration spectra obtained was undertaken.

The predominant amplitudes and the frequencies at which they occurred were transcribed from the IRD 820 print-outs (as in FIGS. 2 and 3) to pre-printed forms (FIGS. 4 and 5). Tabulating the data in this manner showed that the highest vibration amplitudes recorded, 35 mm/s at point B horizontal with the engine on load and 30 mm/s with the engine on no load, occurred at a frequency of 2.4K c.p.m. (2×engine speed). As the high vibration amplitudes occurred both on and off load and after comparing this data with previous vibration surveys and data from similar engines on other ships, it was diagnosed that the engine had suffered damage-failure to the secondary balance equipment, confirming the quick diagnosis. Experience has shown that similar vibration levels may be encountered at 2×fundamental frequency on this type of engine after rebuild if the secondary balance weight driven gear is out of phase with the driving gear wheel. A telephone call from Hull to Portsmouth had eliminated this possibility prior to the quick look diagnosis as the engine had been surveyed some 3 months before and all vibration levels had been satisfactory.

The Report

The findings of the survey were reported formally to the ship with vibration data sheets, feedback form, and analysis sheets containing analysis and recommendations. The report was copied to Flag Officer Plymouth with a summary of recommendations for repair action.

The report recommended that the engine should not be run except in an emergency. It was also recommended that at the first opportunity (the next period in harbour):

(*a*) The secondary weights and associated gearing be inspected for looseness, wear or damage.

(*b*) The secondary balance weight brackets be examined for damage/cracking.

(*c*) The sump be inspected for debris.

Feedback

Feedback forms are forwarded to ships whenever a vibration analysis report contains a recommendation. The purpose of the form is to confirm the diagnosis when repair action has been completed. This is an important aspect of vibration analysis and any other information, such as strip reports or photographs which can be of assistance to Vibration Analysis Units, is always appreciated and will benefit other ships with similar defects in the future. In this particular case, on close inspection of the free end of the engine it was found that:

(*a*) All split gear wheel fastenings had sheared.

(*b*) Half the split gear wheel secondary balance weight drive had fouled the gear train (FIG. 6).

(*c*) All vibration damper fastenings had sheared (FIG. 6), allowing the torsional vibration damper to float on the crankshaft and causing damage-wear to the crankshaft end cover (FIG. 7).

Further feedback after metallurgical examination of the failed damper and secondary balance gear securing bolts recommended that the bolts be replaced using quality assured items.

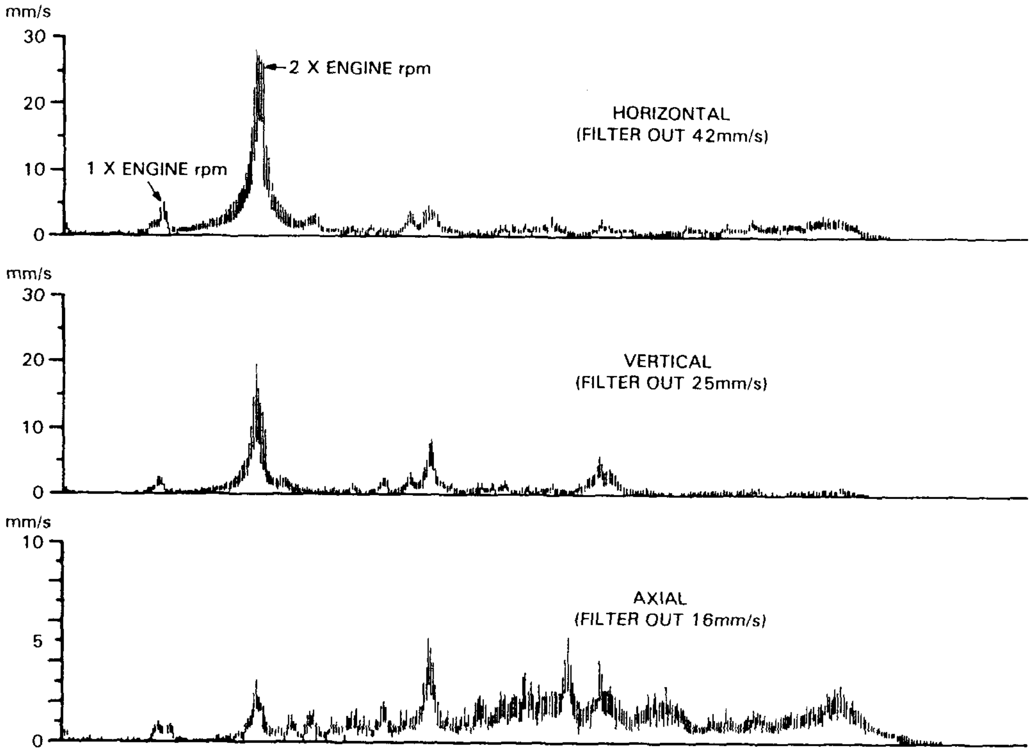

FIG. 2—VIBRATION TRACES AT POSITION B: LOAD 500 KW

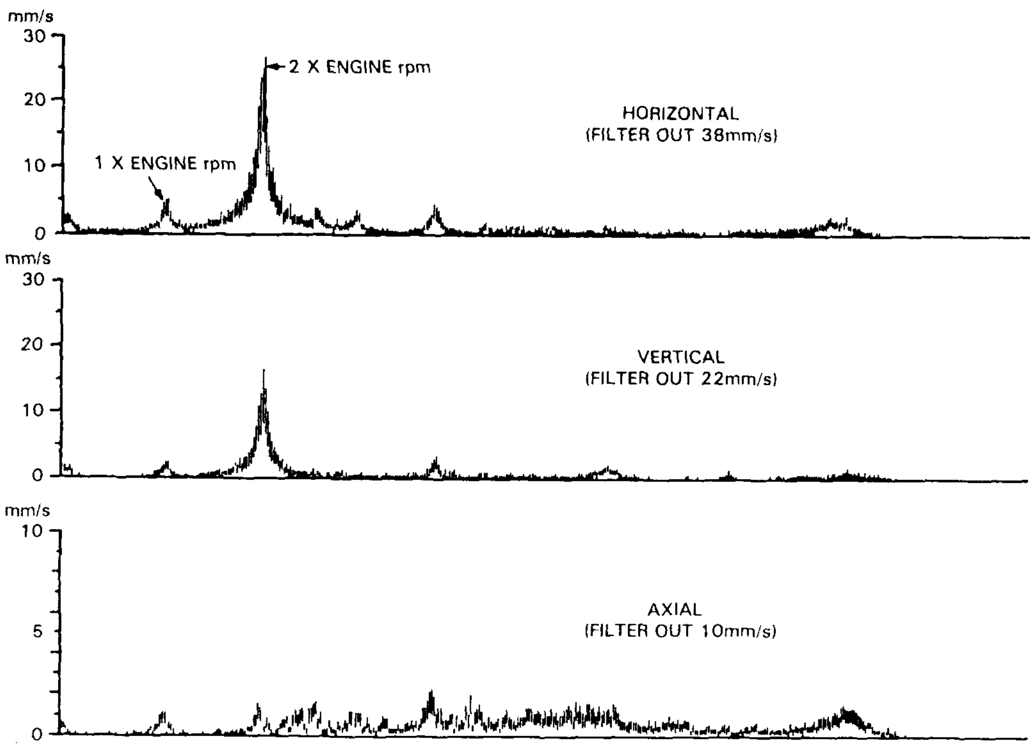

FIG. 3—VIBRATION TRACES AT POSITION B: NO LOAD

PICK-UP POSITION		FILTER OUT	FILTER IN		
			½ rpm	1 rpm	2 rpm
A	H	26	2	5.4	11
	V	26		4	12
	Ax	15		0.5	6.2
B	H	42		5.5	35
	V	25		2.5	17
	Ax	16	1	1.2	3.1
C	H	32		2	26
	V	19			10
	Ax	22			12
X	H	80			42
	V	54			28
	Ax	36			15
Y	H	95			60
	V	46			28
	Ax	32			15

RESILIENT MOUNTS

D	48/60
E	1.1
F	25
G	0.6

°F EXHAUST TEMPERATURE

CYLINDER	1	2	3	4
'A' BANK	760	760	700	740
'B' BANK	790	720	700	820

FIG. 4—VIBRATION LEVELS (MM/S) AND OTHER DATA: LOAD 500 KW

PICK-UP POSITION		FILTER OUT	FILTER IN		
			½ rpm	1 rpm	2 rpm
A	H	15	1.8	6.2	7
	V	22		0.9	16
	Ax	9.2			5
B	H	38	0.6	5	30
	V	22	2	2.5	17
	Ax	10	0.8	1	1.8
C	H	28	1	1.5	26
	V	16		1	10
	Ax	16			10.5

RESILIENT MOUNTS

D	32/78
E	0.7
F	18
G	0.5

FIG. 5—VIBRATION LEVELS: NO LOAD

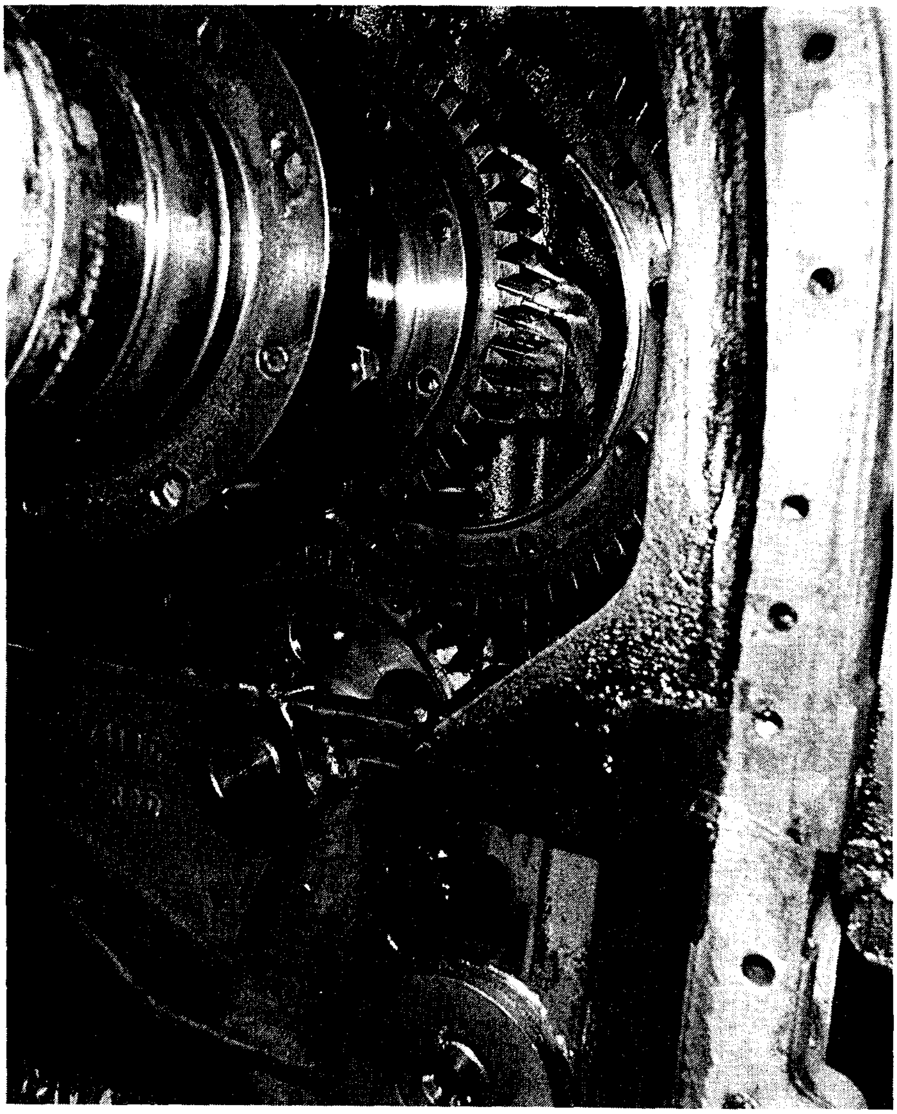

FIG. 6—SHOWING FAILED FASTENINGS ON FREE END SPLIT GEAR WHEEL, THE SECONDARY BALANCE WEIGHT DRIVE, AND FAILED FASTENINGS ON TORSIONAL VIBRATION DAMPER. THE SPLIT GEAR WHEEL HAS FOULED THE SECONDARY BALANCE WEIGHT DRIVE TRAIN. THE DAMPER HAS BECOME DETACHED AND MOVED FORWARD ON THE CRANKSHAFT

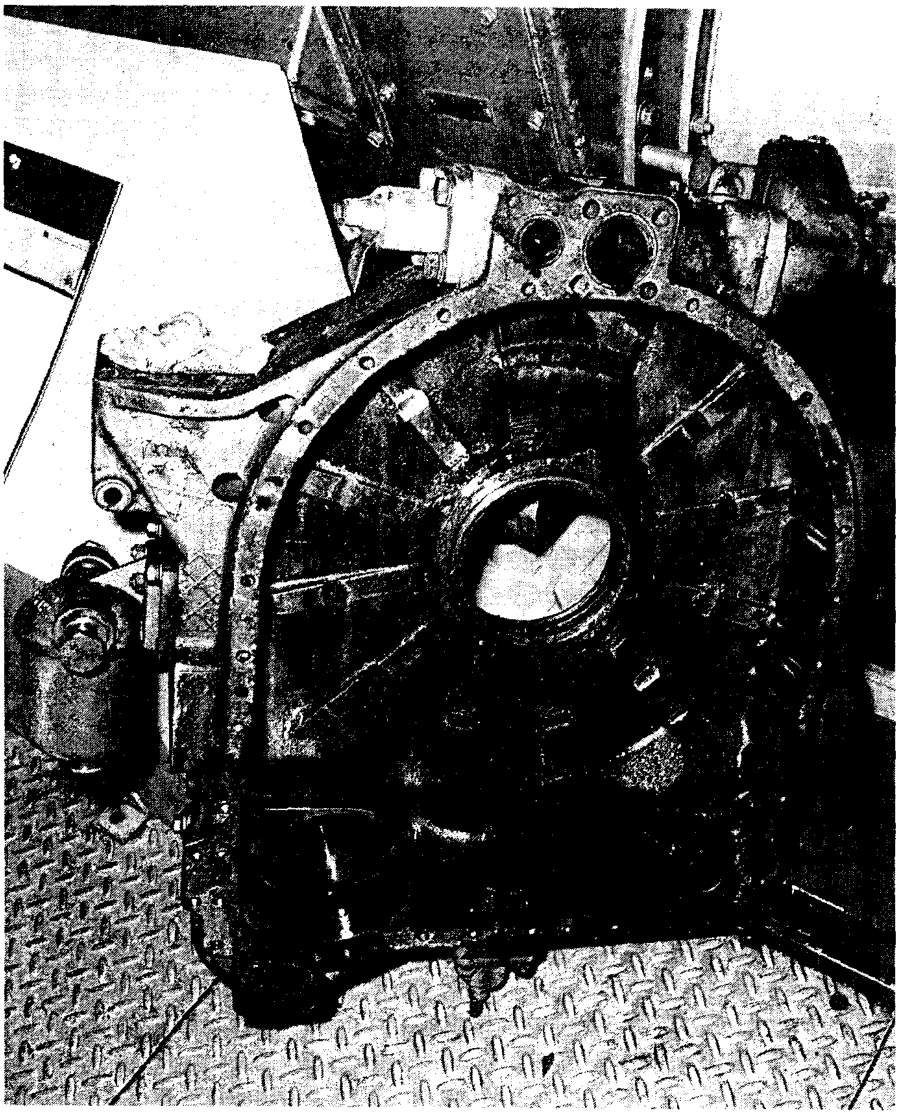

FIG. 7—SHOWING DAMAGE TO END COVER CAUSED BY DAMPER BECOMING DETACHED FROM THE CRANKSHAFT

PICK-UP POSITION		FILTER OUT	FILTER IN		
			½rpm	1rpm	2rpm
A	H	18	1.3	1	2
	V	18	1.0	0.6	5.8
	Ax	11.5		0.6	0.9
B	H	8.8	1.7	1.8	2.3
	V	7.8	0.7	2.3	1.7
	Ax	6.7		0.3	0.8
C	H	8	0.6	2.4	1.1
	V	5.9		2.3	0.7
	Ax	6.4		0.6	1.1
X	H	-			
	V	-			
	Ax	-			
Y	H	25			
	V	12			
	Ax	15			

RESILIENT MOUNTS

D	23
E	0.5
F	10
G	0.4

°F EXHAUST TEMPERATURE

CYLINDER	1	2	3	4
'A' BANK	690	695	680	690
'B' BANK	740	720	720	710

FIG. 8—BASELINE VIBRATION DATA ON REPLACEMENT ENGINE: LOAD 470 KW. 'VIBRATION LEVELS SATISFACTORY.'

Conclusion

The defective engine was changed and the new engine was surveyed for baseline vibration data. Vibration levels then proved to be satisfactory, as in FIG. 8.

Reference

1. Thorne, R. W.: Good vibrations—bad vibrations; *Journal of Naval Engineering*, vol. 29, no. 1, June 1985, pp. 147–153.