Practical Examples of Vibration Analysis Onboard Ships and on Offshore Structures

B. H. Thomas MSc, PhD
Vibration Consultants and Instrumentation Ltd

SYNOPSIS

Most practical engineers have, at some time or other, experienced difficulty in determining the cause of a problem on a machine or structure. Consequently the machines or structures have been dismantled for a more detailed examination to try and establish the reasons for the trouble. However, sometimes even detailed examination of the component parts has not revealed the nature of the problem. Vibration analysis techniques present an alternative method to stripping machines or structures unnecessarily, and the methods and instrumentation have been developed to allow the majority of machine and structural problems to be identified. This has led to on-site trouble-shooting and in-position balancing for both the marine and offshore industries. Over the years, numerous specific problems on rotating machines, structures and pipework have been examined and a selection of case histories, showing the measuring and analysing techniques which lead to the solution of a problem, are discussed in this paper.

INTRODUCTION

Unpredictable failure of machinery or structures of essential and critical items on ships or oil platforms usually results in large additional costs. These costs accumulate as a result of non-productivity because of the down time, manpower charges and replacement parts costs. The inconvenience is further exacerbated when new or replacement parts are not readily available, in which case the delay periods, with the accompanying loss of revenue, are extended. Clearly this is an invidious position for any shipowner or oil-platform operator.

Standard maintenance procedures will not normally highlight unusual or fundamental problems, with the result that operators find that for certain machinery and structures problems become unpredictable and recurrent. In the past, most engineers have been faced with a suspect machine or structure which resulted in faults such as fractures of pipes, frames and bases, etc. Where this has been the case, the normal process has been to repair the fractures and tighten everything solidly, assuming that this action is the answer. In many cases such action will usually improve the overall conditions, but it will not get to the root of the problem, which will re-appear later with similar consequences.

Ideally, one would like to forecast when a machine or structure is likely to fail and to establish the reason for the failure. Today this can be achieved by using vibration analysis techniques. The range of vibration equipment is extensive and comprehensive, with prices from £1000 to over £100 000, depending on the degree of sophistication required.

However, it should be noted that the instrumentation alone, no matter how advanced or sophisticated, will not give the direct answer to a problem. Even the most expensive equipment in the wrong hands could be virtually impotent. It is important to understand that for the results of a difficult vibration problem to be meaningful, they have to be unravelled and analysed by someone with experience. Most engine room staff with some basic training can achieve a good standard of analysis. It should be appreciated that an experienced engineer with some training in the vibration field, using the most basic equipment, will be able to make intelligible the measurements taken and convert them into a practical solution. It is only when an exceptionally difficult problem is encountered that one has to resort to the more comprehensive range of expensive analysing instrumentation. In real terms, probably only 5% of such problems may be considered to be in this category.

Vibration techniques can therefore predict changes and impending failures before they become serious or even disastrous. Furthermore, subsequent analysis of the vibration enables its source(s) to be located without dismantling a machine or structure. Vibration analysis has now developed into a precise science in which the acceptable magnitudes for different machines and structures are well defined.

Although vibration analysis is not the panacea for all engineering problems, it can be an exceptionally useful tool and aid for solving and removing most of them. This paper reviews a selection of typical examples of problems encountered by V.C.I. on ships and oil platforms over recent years. For each of these case histories, the method of vibration measurement and the analysing techniques leading to a solution are discussed.

Brian H. Thomas served a five year 'sandwich course' apprenticeship starting in 1955 with Metropolitan Vickers Electrical Company. This was followed by postgraduate research work on electrical machines at Manchester University for an MSc. Then, in 1963, as a turbo-generator design engineer, he became involved with problems relating to vibrations and high-speed balancing techniques. This led to him being sponsored by A.E.I. (latterley G.E.C.) to undertake a PhD degree on 'Modal balancing of turbo generator rotors'. In 1971 Dr Thomas started his company 'Vibration Consultants & Instrumentation Limited' which, over the years, has specialised in consultancy services and instrumentation for the marine and offshore industries.

DESCRIPTION OF TESTS

Although the tests described for each of the following case histories vary depending upon the nature of the investigation, certain procedures are standard and will be outlined below.

Magnitude

The type of measurement selected during each test is dependent upon the nature of the problem, but the three forms of magnitude used fall into the following categories:

Low-frequency vibrations (0–100 Hz)
Displacement (peak to peak) (μm)
Mid-range/frequencies (5 Hz–2 kHz)
Velocity (RMS) (mm/s)
High frequencies (500 Hz+)
Acceleration (gravitational units peak) (g)

Direction

Initially, vibrations are normally measured on the bearings and sometimes the framework of each machine in three directions:

1. Vertical (V).
2. Horizontal or transverse (T).
3. Axial (A) (parallel with the machine shaft).

Frequency

Where the vibration levels are found to be high, a harmonic analysis is carried out. A complex vibration waveform may comprise several harmonics, in which case an analyser is used to determine the magnitude of vibration at each frequency. The harmonic analyser used is usually an infinitely variable unit in which each frequency is examined progressively. For more complicated problems, a real-time or FFT analyser and X–Y plotter are used to give the vibration trace of all the harmonics at a single instant of time.

Vibration monitoring by marine engineers

Most marine engineers, with some training, can use basic vibration instrumentation so that, in the first instance, they can determine whether or not machinery is operating within an acceptable International Standard. There are many standards relating to the level of acceptable vibration in ships' structures. In the case of machinery, the most significant is VDI 2056, which is long established. The Bibliography at the end of this paper gives details of this and other standards.

Next, if a problem does exist, they should be capable of resolving and identifying where it stems from, such as:

1. When a unit is suspected of being unbalanced, first the order of vibration must be determined. It will be understood that in normal circumstances the first order points to an unbalanced condition which can be readily corrected.
2. Misalignment between units, which can result in high levels of vibration.
3. Damaged or worn bearings, whether sleeve or rolling element type.
4. Damaged or worn gearing.
5. Unsatisfactory foundation/support arrangements on all types of machinery.
6. Whether any unit is influenced by external forces, propeller, hull or other machinery.

CASE HISTORY 1: MAIN DIESEL ENGINE

During acceptance trials, measurements were taken on the main engine under two different conditions. Initial measurements were carried out during speed trials when large vibrations occurred. Subsequently, an inspection was carried out and the holding-down bolts and the forward drive chain were tightened. A second set of measurements was then taken during the endurance run.

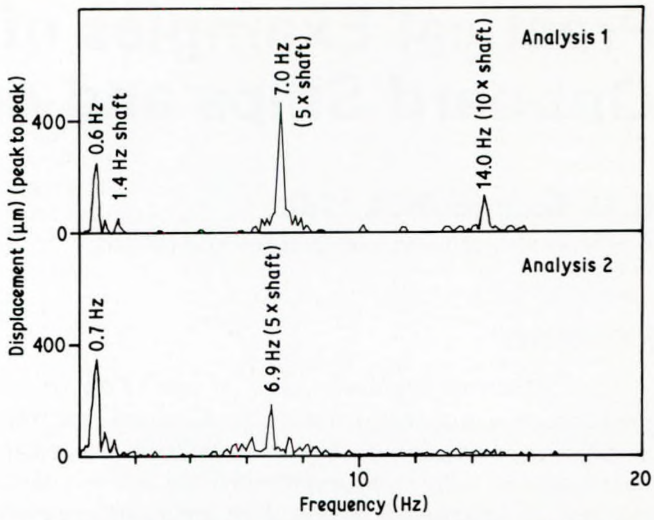


FIG. 1: Frequency spectra for main engine forward end at bottom grating level in axial direction. Upper: analysis no. 1 before inspection. Lower: analysis no. 2 after inspection and adjustment

Test 1

Initially, very high levels of vibration were measured on the main engine, particularly in the axial and vertical directions at the forward end (grating level). The predominant frequency was at 7 Hz (see analysis 1 in Fig. 1) and which corresponded to: the engine firing rate (number of cylinders × shaft speed) and the camshaft rotational frequency.

It was found that the vibration at the propeller blade rate (5.5 Hz) was very small. Finally, the degree of mechanical unbalance was shown by the 1.4 Hz component, which was most prominent in the horizontal direction.

The major problem identified was associated with vibration at 7 Hz, occurring mainly at the forward end. Consequently, the owner asked for a mechanical inspection of the forward drive mechanism. It was found that the chain driving the second moment balance shaft was slack and that the weights were out of adjustment. The chain and weights were adjusted and, in addition, the main holding-down bolts were tightened.

Test 2

The vibration levels were greatly reduced after the adjustments described above. At the bottom grating level the axial vibration velocity had reduced (see analysis 2 in Fig. 1) and the vertical velocity was down to 60%. The main reduction was at 7 Hz. The shaft rotational vibration was also slightly reduced in the horizontal direction. However, because the frequency was so low, such a small change was not considered to be significant, as in velocity terms the reduction was only by 0.4 mm/s (the displacement change at 7 Hz in the axial direction was 300 μm). Note that the peak below 1 Hz was most probably caused by hull girder vibration. This was identified as a frequency occurring at about 0.6–0.7 Hz.

Engine stays

One other problem at the top of the engine was that relative movement occurred across the stays (of the order of 650 μm), ie between the engine and the ship structure. The horizontal and axial movements at the engine end of the stays were always much greater than at the ship's structure end. Also, on the two forward/aft stays, very large vertical vibrations occurred at the structure (forward) end. These were much greater than engine vertical vibrations, suggesting that a resonant condition of the 'box' structure to which they were attached was present. No attempt was made to improve this aspect during these tests.

Conclusions

The adjustments made at the forward end of the main engine and the tightening of the holding-down bolts resulted in a significant reduction in vibration. The final levels in the fully loaded condition were within the manufacturer's specifications but were still on the high side. The remaining vibration was mainly at 7 Hz (main engine firing order) and 1.4 Hz (shaft speed). The propeller-induced vibration also remained low.

The structure to which the forward/aft engine stays were attached was clearly too flexible and needed stiffening in the vertical direction to avoid future problems.

CASE HISTORY 2: RADAR SCANNER

It was reported by the owner of a bulk carrier that a number of failures had occurred on the radar scanners when operating at a propeller speed of 95 rev/min. At this speed, severe vibration occurred which resulted in cracking of the support structure and thus a constant need for maintenance and repairs. A comprehensive range of measurements were requested:

1. To verify that a vibration problem existed.
2. To establish what was causing the problem.
3. To advise how the problem could be solved.

In practice, radar scanners and masts are often subject to failure caused by vibration and this is a fairly common problem on a variety of different vessels. For the specific tests, a number of measuring positions were selected, as shown in Fig. 2. The vibration levels shown were taken first forward to aft, secondly port to starboard and thirdly vertically.

A graph of vibration against position up the mast was then plotted for the mast and radar in the direction of maximum vibration, ie forward to aft. Finally, a harmonic analysis was carried out on the vibration waveform to determine the frequency and hence the exciting source of the vibration.

From the results and graph in Fig. 2 it was found that the vibration was excessive on the radar. Usually the vibration level at the top of the radar would not exceed about 60 mm/s but in this case the level is above this figure.

A harmonic analysis of the waveform revealed that the main component of vibration was at 15.8 Hz, ie twice the propeller blade passing frequency (where there were five propeller blades):

$$\frac{2 \times 5 \times 95}{60} \text{ Hz} = 15.8 \text{ Hz}.$$

Therefore, the propeller blade passing frequency was the exciting force causing the vibration.

The problem became more readily understood by examining the graph in Fig. 2. It will be seen that the vibration in the forward to aft direction on the port radar scanner was about four times higher than it should be, so this was the reason for the constant failures. It can also be seen that the vibration in the port to starboard direction was less than in the forward to aft direction and in the vertical direction the levels were negligible.

From the graph Fig. 2 it can be seen that in the vicinity of the radar support the rate of change of vibration was very rapid and, in fact, the rate of increase was even larger for the scanner vibration. This led to the conclusion that the radar support frame required substantial stiffening, mainly forward to aft but also partially in the port to starboard direction. Therefore, the radar support was fitted with two extra substantial stiffening tubes in the forward to aft direction, inclined at about 45° to one another, as shown in Fig. 3.

Position on mast	Vibration (mm/s)		
	Fwd/Aft	Pt/Stbd	Vertical
14	225	62.5	75
13	125	42.5	3.0
12	100	37.5	3.25
11	87.5	32.5	3.0
10	75.0	30.0	2.75
9	52.5	25.0	3.5
8	17.5	16.25	3.0
7	16.25	13.75	3.0
6	15.0	12.5	2.5
5	12.5	8.75	2.5
4	11.25	7.5	2.5
3	10.0	6.25	3.0
2	8.0	2.75	3.0
1	5.5	2.5	3.0

Propeller speed = 95 rev/min.

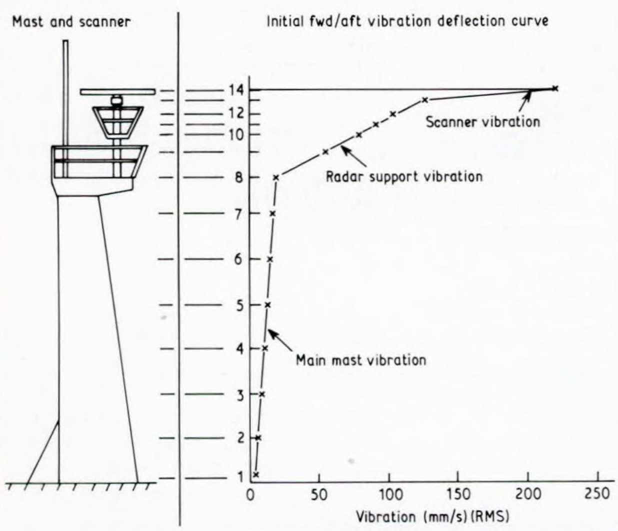


FIG. 2: Initial vibration measurements for mast and radar scanner

Position on mast	Vibration (mm/s)		
	Fwd/Aft	Pt/Stbd	Vertical
14	55.0	30.0	15.0
13	27.0	25.0	3.5
12	25.0	23.0	3.0
11	23.0	22.0	3.5
10	21.5	19.5	3.0
9	19.0	17.5	3.0
8	17.0	16.5	2.5
7	16.0	13.5	2.5
6	14.5	11.5	2.5
5	12.0	8.5	2.5
4	11.0	7.5	3.0
3	9.5	6.5	2.5
2	8.0	3.0	2.5
1	5.0	2.5	2.5

Propeller speed = 95 rev/min

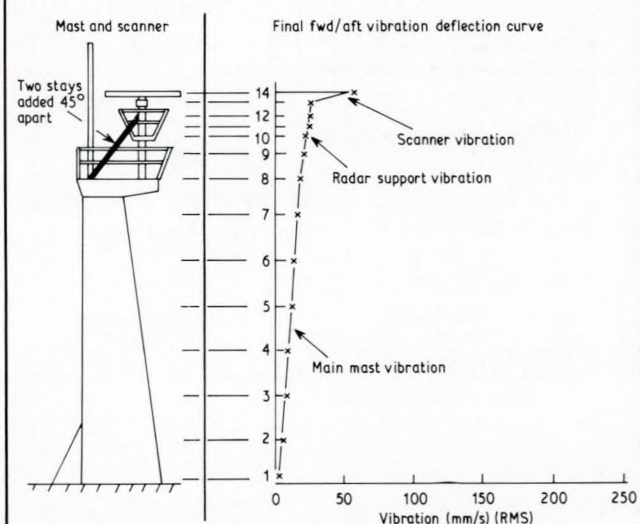
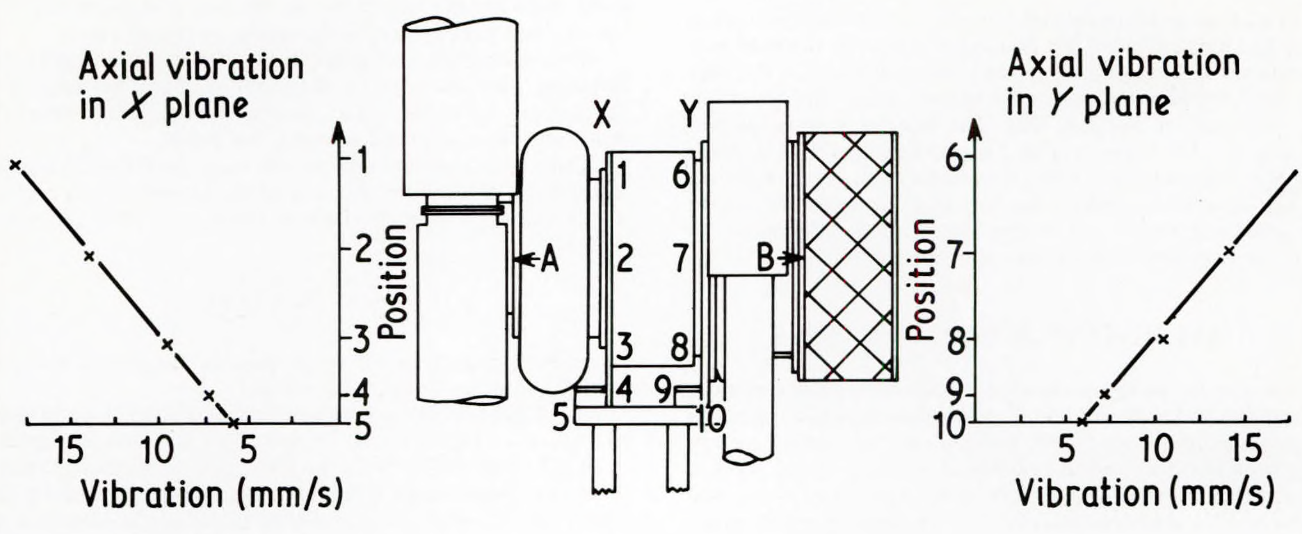
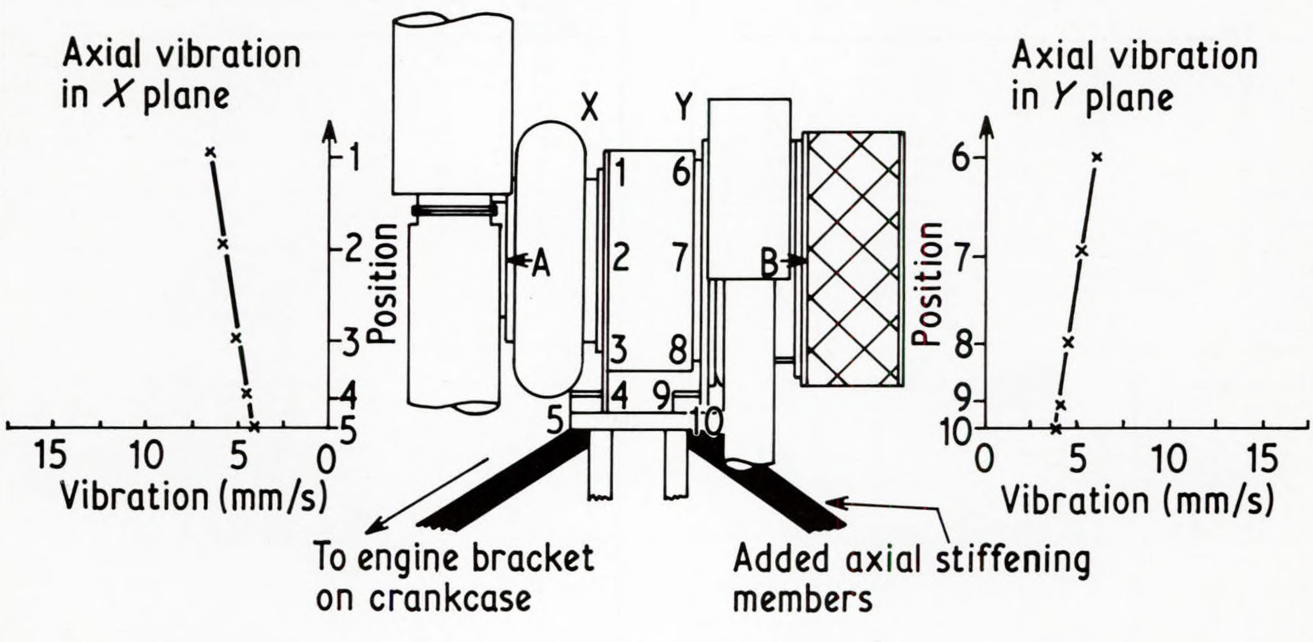


FIG. 3: Final vibration measurements for mast and radar scanner



Vibration velocity (mm/s) in X plane				Vibration in vicinity of bearing A				Vibration in vicinity of bearing B				Vibration velocity (mm/s) in Y plane			
Posn.	Vertical	Pt/Stbd	Fwd/Aft		Vertical	Pt/Stbd	Fwd/Aft		Vertical	Pt/Stbd	Fwd/Aft	Posn.	Vertical	Pt/Stbd	Fwd/Aft
1	4.0	5.5	18.2	Disp (min)	230	345	800[1]	Disp (min)	205	265	715[3]	6	4.2	6.0	19.0
2	4.5	5.5	14.0	Vel (mm/s)	4.4	6.5	15.0^2	Vel (mm/s)	4.1	5.0	13.5	7	4.5	5.8	14.1
3	4.3	5.3	9.8	Accn (*g*)	0.8	1.9	1.8	Accn (*g*)	2.2	1.5	1.8^4	8	4.5	5.6	10.5
4	4.3	5.1	7.4	(1) 750 @ 8.5 Hz, 100 @ 17 Hz				(2) 14.5 @ 8.5 Hz, 3 @ 17 Hz				9	4.4	5.4	7.1
5	4.2	5.0	6.2	(3) 680 @ 8.5 Hz, 85 @ 17 Hz				(4) 1.1 @ 8.6 Hz, 1.4 @ 115 Hz				10	4.4	5.4	6.0

FIG. 4: First vibration measurement on turbocharger during sea trials



Vibration velocity (mm/s) in X plane				Vibration in vicinity of bearing A				Vibration in vicinity of bearing B				Vibration velocity (mm/s) in Y plane			
Posn.	Vertical	Pt/Stbd	Fwd/Aft		Vertical	Pt/Stbd	Fwd/Aft		Vertical	Pt/Stbd	Fwd/Aft	Posn.	Vertical	Pt/Stbd	Fwd/Aft
1	3.8	4.4	6.3	Disp (mm)	180	240	370[1]	Disp (mm)	175	240	350[3]	6	3.6	4.7	6.1
2	3.9	4.6	6.0	Vel (mm/s)	3.8	4.8	6.9^2	Vel (mm/s)	3.7	4.6	6.8	7	3.7	4.7	5.2
3	3.8	4.4	5.0	Accn (*g*)	0.7	1.3	1.0	Accn (*g*)	1.6	1.0	1.1^4	8	3.6	4.6	4.5
4	3.6	4.2	4.5	(1) 300 @ 8.5 Hz, 50 @ 17 Hz				(2) 6.0 @ 8.5 Hz, 1.0 @ 17 Hz				9	3.4	4.4	4.0
5	3.7	4.1	4.2	(3) 320 @ 8.5 Hz, 40 @ 17 Hz				(4) 0.5 @ 8.6 Hz, 1.0 @ 115 Hz				10	3.3	4.2	3.8

FIG. 5: Vibration measurements on turbocharger at sea after modifications to base

It will be seen that when the measurements were repeated, the effect of the supports was to reduce considerably the vibration on the radar frame and scanner. In fact, the overall picture can be best appreicated by comparing the deflection curves in Figs 2 and 3.

Before adding supports, one has to determine in which direction the vibration is most severe and what is causing the vibration. In this case, the cause of the vibration could not easily be altered and so remedial action was successfully taken to minimise the effects.

CASE HISTORY 3: TURBOCHARGERS

Turbochargers used for main engines, diesel generators and other auxiliary machines are subject to vibration transmitted as a result of the diesel forces and from unbalance of the turboblower rotors, which could be caused by either wear or build-up of deposits. Such forces, when they are excessive, can cause failure of the blower bearings, the support frame, the holding-down bolts, the pipework and/or any flexible bolts.

Further, the problems occur because many turboblowers are mounted in an unsatisfactory manner. For example, some are supported by brackets or a framework which is often cantilevered from the diesel. Where this is the case, the vibration levels increase towards the free end of the cantilever and the turbocharger. In other cases, it will be found that the support frame is sufficiently rigid in the vertical and horizontal directions with respect to the blower shaft, but it is far too flexible in the direction axially to the blower.

Consequently, it is the forces specifically resulting from the diesel, which include firing sequence, reciprocating action and unbalance, which cause excessive levels of vibration to occur on the blowers and the supporting brackets. Clearly this is an unacceptable situation, but it often exists at the outset.

One typical example concerned the turboblowers for a main engine. This engine had six cylinders with a maximum speed of about 85 rev/min. Tests were carried out during the initial commissioning stages and it was found that the vibration on the turboblowers increased rapidly with an increase of the main engine speed. At 85 rev/min the initial vibration levels on both blowers was unacceptable. The measurements were taken at a main engine speed of 85 rev/min in the vicinity of the turbocharger bearings at positions A and B for one unit, as shown in Fig. 4. From the measurements of vibration, displacement, velocity and acceleration it will be seen that in the vertical and port to starboard directions the vibration levels were satisfactory. However, in the forward to aft direction (ie in line with the blower shaft) the vibrations were excessive.

A harmonic analysis revealed that there was virtually negligible vibrations being generated by the blower itself but most was coming from the main engine as a result of the forces from the firing sequence, causing vibration at 8.5 Hz.

Five vertical measuring positions were selected at specific increments either side of the blower supports in planes *X* and *Y* in Fig. 4. It will be seen from the results that the main vibration was in the forward to aft direction, at a frequency of 8.5 HZ. From the vibration deflection in the axial direction plotted for each end it can be seen that very large and unacceptable levels were measured.

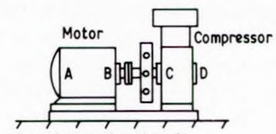
It was deduced that additional axial stiffening was required and this was installed by adding two substantial RSJs from brackets on the main engine to the underframe of the turboblower, as shown in Fig. 5. These supports resulted in the vibration levels in the axial direction being reduced to less than half their initial levels. The overall reduction can be seen by comparing the vibration measurements and graphs in Figs 4 and 5.

It should be noted that it was initially difficult to persuade the shipyard engineers to carry out the additional stiffening, but the eventual improvement satisfied the shipowner and all the other parties concerned.

Vibration velocity (mm/s)

Position		*Total*	14.7 Hz 1/rev	29.4 Hz 2/rev	*Motor only*
A	V	3.75	2.0	1.5	2.0
	H	11.0	8.5	3.75	2.5
	A	8.75	6.75	2.5	1.5
B	V	6.0	4.25	1.75	2.5
	H	11.25	10.0	3.75	3.0
	A	6.25	5.25	2.25	1.5
C	V	13.75	12.5	4.25	
	H	27.50	25.5	4.0	
	A	16.25	13.25	3.75	
D	V	12.5	11.75	3.75	
	H	25.0	24.0	4.0	
	A	15.0	13.75	3.5	

Details



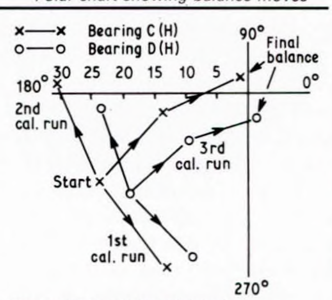
Base bolted directly onto ship's structure
Compressor speed = 880 rev/min

FIG. 6: Initial vibrations on instrument air compressor

Vibration velocity (mm/s)

Position		*Total*	14.7 Hz 1/rev	29.4 Hz 2/rev
A	V	2.5	1.5	2.0
	H	4.0	3.5	3.25
	A	2.5	2.5	2.0
B	V	3.0	2.5	1.5
	H	4.25	3.75	2.75
	A	3.25	3.25	2.0
C	V	3.75	3.5	3.0
	H	3.75	3.0	3.5
	A	3.0	2.75	3.25
D	V	4.25	3.0	3.5
	H	6.0	4.5	3.5
	A	3.5	2.75	3.0
Change:		Large	Large	Small

Polar chart showing balance moves



Final weights: 3.1 kg mild steel plugs inserted into flywheel

FIG. 7: Vibrations on instrument air compressor after balancing

CASE HISTORY 4: INSTRUMENT AIR COMPRESSOR

Vibration problems associated with reciprocating compressors are caused mainly by too flexible bases, excessive wear and mechanical unbalance. In practice, it is surprising to find how many compressors, particularly single-cylinder types, have large mechanical unbalance from new. According to various manufacturers, these units are made to such high tolerances that they should not need mechanically balancing. However, although this may appear to be correct in theory, in real terms this is often not the case.

One shipping company operating container vessels requested a series of vibration investigations on a number of single-cylinder instrument air compressors on several of their ships. They had found that they were faced with bearing and crankshaft failures, which were a cause of embarrassment and resulted in excessive repair costs.

Prior to the vibration investigations a series of tests had been carried out to determine that the correct lubrication was taking place and the under-structure had been strengthened to no avail.

The measurements were taken at the positions shown in Fig. 6, which was typical of the others. It will be seen that vibrations on the compressor in the horizontal direction (H) were particularly large, and from the harmonic analysis the main component of vibration was at a frequency of 14.7 Hz. This corresponded to the once per revolution of the machine where the speed was 880 rev/min. This indicated that the machine was out of balance.

Normally the largest levels of vibration tend to indicate the area where a problem is occurring, but as this is not always the case the motor was uncoupled and tested separately. From the measurements for the 'motor only', as shown in the right-hand column of Fig. 6, it can be seen that the vibration levels were within International Standards. Therefore, this clearly pointed to the unbalance being on the compressor and flywheel.

The unit was re-assembled and an in-position high speed dynamic balance was carried out. The holes for turning the flywheel were drilled and tapped to take screwed mild steel plugs. These plugs, which weighed from about 0.5 to 1 kg each, were used for balance weights.

First a calibration run was carried out to find the correct position and the magnitude of the final balance weights. In Fig. 7 it can be seen that each move was plotted on a polar chart of vibration against phase angle. The largest levels on either side of the compressor were used, ie for bearings C and D in the horizontal direction.

Once the vibration levels in these horizontal directions had been reduced to a satisfactory level, measurements were taken in the vertical, horizontal and axial directions on all the bearings throughout the unit to ensure that they had reduced as a result of the balancing. This is an important step because sometimes, as one improves the vibration levels in one plane, they can increase in the other directions. However, from the results in Fig. 7 it will be seen that after the dynamic balance all the vibrations reduced to an acceptable standard.

Note that the twice per revolution component at 29.4 Hz as shown in the tables of Figs 6 and 7 did not alter very much as a result of balancing. This is to be expected as this component is caused by the reciprocal action of the compressor.

One of the main points of this example is to show that all machines, including reciprocal and rotating, should be checked for unbalance and excessive vibration, preferably from new. Another useful point to note is that most machines, where they are out of balance, can be given a high-speed in-position dynamic balance onboard without dismantling the unit. Such a balance keeps down-time to a minimum and also ensures that the overall vibration levels are reduced to an optimum level.

CASE HISTORY 5: CARGO PUMPS

Problems on cargo pumps in oil tankers are not uncommon. These are expensive and critical pieces of machinery which tend to operate out of site for most of the time in the depths of the pump rooms. Consequently, as problems build up, they tend to go unnoticed.

A typical vertical unit is shown in Fig. 8, where an overhung turbine drives the main output shaft via a gearbox. Problems with such machines can arise in almost any area and typically they may be categorised as follows:

Cargo pump problem area

Turbine

This unit can become unbalanced as a result of erosion of the blades or build-up of deposits. Also, over a long period, bearing failures can occur.

Gearbox

Normally gearbox vibrations are associated with wear and poor mesh of the gear teeth. Sometimes, where the mesh is insufficient, this indicates that the bearings are worn.

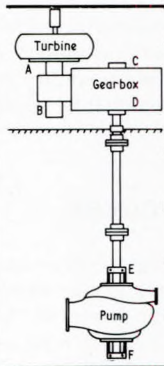
Drive shaft

The drive shaft alignment is critical to ensure that vibration on the bulkhead seal, under gearbox bearing D, is not too large and the pump bearings are not being subjected to unnecessary additional loads. Clearly, the condition of the flexible couplings does have a significant overall influence if the couplings become siezed, say because of lack of lubrication, and the flexibility of the shaft is then lost.

Cargo pumps

In our experience, the cargo pump is the main area where one can most expect failures. The pumps are designed to achieve the best compromise between pumping water and a variety of oils of different specific gravities. The main problems associated with these machines are that they are readily prone to cavitation and turbulence if not operated under the correct conditions. Such forces lead to severe bearing and seal failures. In addition, the impeller tends to become badly eroded and the wear rings worn. The overall levels of vibration on the pump, which are usually largest in the axial plane, can be controlled by adjusting the suction pressure but the specific magnitude is more readily changed by altering the discharge pressure.

Sketch showing vibration measuring positions



Position		Vibration veolocity (mm/s) Total	Harmonic Analysis (Hz) 20	100	117	1.2 k	Broad 1 k–3.5 k
A	V	9.25	1.5		9.2	0.5	
	H	10.0	2.5		9.75	0.4	
	A	2.0	0.5		1.5	0.6	
B	V	8.5	1.0		7.0	0.8	
	H	9.25	2.0		8.5	0.7	
	A	2.25	0.5		1.0	0.8	
C	V	3.5	2.0		1.5	1.2	
	H	3.25	3.5		1.0	0.5	
	A	2.5	1.5		0.5	1.0	
D	V	3.5	3.5		1.0	1.5	
	H	2.75	2.5		0.8	0.75	
	A	2.0	1.5		0.5	0.5	
E	V	15.75	4.2	15.25			
	H	12.5	5.5	11.75			
	A	20.0	6.0	19.5			
F	V	14.25	5.5	12.75			
	H	13.75	6.75	13.0			
	A	18.5	6.25	17.25			
		Acceleration (*g*)					
E	H	7.5		2.1			5.5
F	H	8.2		2.5			6.0
Source of vibration			LS shaft	Pump blade passive frequency	Turbine shaft	Gearbox teeth	Worn rolling element bearing

Turbine Speed = 7000 rev/min
Pump speed = 1200 rev/min
Discharge pressure = 7 kg/cm^2
Note: five vanes on pump impeller

FIG. 8: Cargo pump vibration levels prior to drydocking

Discharge pressure (kg/cm^2)	Pump speed (rev/min) 900	1000	1100	1200	1350
2	5	13.75	17.5	19.75	22.5
4	2.5	9.0	14.0	16.5	19.0
6	2.5	5.5	10.25	12.0	13.75
8	3.0	4.0	7.0	8.5	9.75
10		6.0	6.5	6.0	6.5
12		9.0	11.5	5.5	4.5
14				7.5	4.5

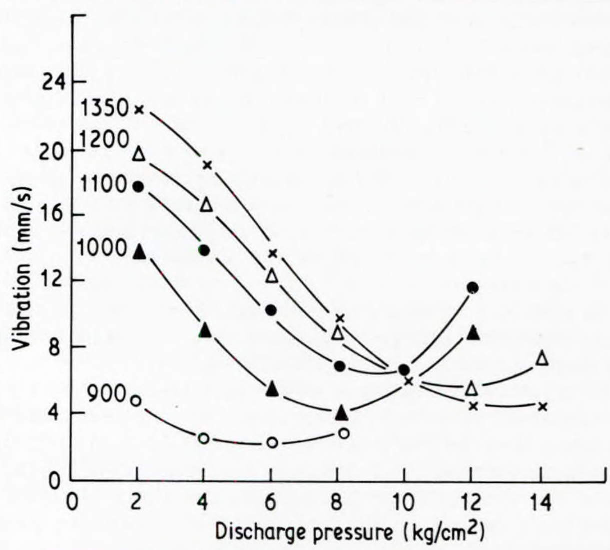


FIG. 9: Typical variation of cargo pump vibrations with change of speed and discharge pressure

Typical vibration levels when pumping water are shown and plotted against varying discharge pressures in Fig. 9, where each curve is drawn with the machine at constant speed. From these graphs it will be seen that there is an optimum discharge pressure at each speed. This optimum range is indicated by the values within the lines in the table. In general terms, as the pump speed is increased, so does the most suitable discharge pressure. Unfortunately, in practice, there is a tendency to operate the pumps outside the optimum ranges with the result that large vibrations occur and then cause failure.

Many companies today fit permanent vibration transducers to the pumps so that the vibrations can be monitored continuously in the cargo pump control room. Therefore, if high levels of vibration occur, these can be reduced by altering the combination of speed, discharge and suction pressure.

Example

The vibration measurements prior to drydocking for such a cargo pump, which had been subjected to much use, are shown in the table of Fig. 8.

For the turbine, which was operating at 7000 rev/min, an acceptable level of vibration may be taken as 3 mm/s (RMS) but it will be seen that this level is exceeded on both bearings A and B in the vertical (V = port to starboard) and horizontal (H = forward to aft) directions. From the harmonic analysis it is concluded that most vibration occurs at 117 Hz, which corresponds to the rotational speed of the turbine. This indicates that there is unbalance and possibly some bearing wear.

The gearbox vibrations for bearing C and D are generally satisfactory. The harmonic analysis shows that the gear teeth vibration at 1.2 kHz is small, indicating that there is little wear and that the mesh is good.

Normally, for a cargo pump, one would expect the vibration velocity not to exceed 6.0 mm/s (RMS) in the radial direction, and the vibration accelerations to be below 3*g* peak. However, the vibration levels on the pump bearings E and F were excessive in all directions, particularly in the axial plane parallel to the shaft. The main vibration is at 100 Hz, which is produced by the blade passing frequency of the five-vaned pump impeller. Also, from the acceleration measurements, which are in terms of peak gravitational units, the levels are high and at a broad-band frequency of between 1 and 3.5 kHz. These high acceleration levels, in this frequency range, indicate that the bearings are failing.

During these tests, a relatively large leak was occurring past the seals on the lower pump bearing. It was therefore recommended that the turbine spindle and bearings should be examined and rectified and that the cargo pump should be given a major overhaul.

During the drydocking, the turbine was balanced and the cargo pump fitted with a new impeller, wear rings, seals and bearings. At the completion of the work, the vibration tests were repeated under similar conditions to the previous tests. It will be seen from Table I that a considerable reduction in vibration was achieved throughout the set, resulting in a satisfactory conclusion to this investigation.

Table I: Cargo pump vibration levels (in mm/s) after drydocking

		Harmonic analysis (Hz)					
Position		20	100	117	1.2K	Broad 1k–3.5k	*Total*
A	V	0.4		2.0	0.3		2.5
	H	0.5		2.0	0.5		3.0
	A	0.1		0.3	0.4		0.3
B	V	0.5		1.5	0.4		2.2
	H	0.3		2.2	0.5		2.5
	A	0.3		0.4	0.5		0.4
C	V	1.5		1.0	0.8		3.0
	H	1.6		1.5	0.6		2.5
	A	0.2		0.4	0.6		0.5
D	V	1.8		1.0	0.7		2.5
	H	1.9		0.8	0.8		2.8
	A	1.0		0.3	0.6		1.6
E	V	1.5	3.0				4.5
	H	2.0	3.25				4.5
	A	1.6	4.25				5.5
F	V	1.0	4.25				5.0
	H	2.0	3.0				4.5
	A	1.5	4.25				6.0
		Acceleration (*g*)					
E	H		0.8			0.8	1.9
F	H		1.0			0.9	2.0

Unit		*Power turbine*				*Generator*
Speed		7900 rev/min				1800 rev/min
Power (MW)		0	1.0	1.2	1.4	
Compressor Speed		72%	85%	89%	91%	
A	V	7.3	7.1	7.0	8.3	
	H	8.5	8.4	8.6	9.5	
	A	4.0	4.0	4.2	4.5	
B	V	3.8	4.0	4.0	4.4	
	H	3.9	4.5	5.0	5.5	
	A	4.1	4.2	4.3	4.4	
C	V	1.2	1.3	1.4	1.4	
	H	1.7	1.8	1.7	1.6	
	A	3.5	3.6	3.7	3.9	
D	V	2.6	2.4	2.5	2.7	
	H	2.0	2.3	2.6	2.6	
	A	3.5	3.2	3.4	3.3	
E	V	1.4	1.3	1.4	1.6	
	H	2.3	2.4	2.6	2.5	
	A	3.2	2.8	3.1	2.7	
F	V	3.5	3.6	3.8	3.5	
	H	2.0	2.3	2.5	2.6	
	A	2.9	3.0	3.3	2.7	
G	V	2.5	2.5	2.4	2.6	
	H	3.5	3.5	3.6	3.8	
	A	1.5	1.6	1.6	1.7	
H	V	4.7	4.9	5.2	5.6	
	H	7.5	7.6	7.7	8.0	
	A	2.5	2.3	2.3	2.0	

Analysis of bearing A(H)

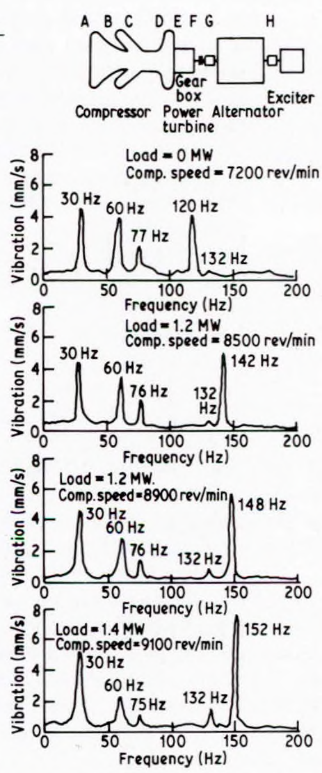


FIG. 10: Vibrations on gas turbine generator and real-time analysis of bearing A(H) before repair

CASE HISTORY 6: GAS TURBINE GENERATOR

The machine discussed in this investigation is one of four units on a North Sea oil platform. As a matter of course, this particular platform operator uses vibration monitoring techniques as an inherent part of the planned maintenance programme for all the machinery. During the course of monitoring it was noticed that the vibrations on one of the gas turbine generators had progressively increased throughout the set. A full and detailed study was therefore carried out to determine what had caused the vibrations to change.

Gas turbine units of this sort comprise a compressor unit which is completely disconnected from the main power turbine. In this particular instance, the main power turbine drove through a gearbox directly onto the generator rotor.

The project was commenced by taking a range of measurements at different loads in the vicinity of the bearings, as shown in Fig. 10. It was found that the vibrations were particularly large in the area surrounding bearings A and H. First, an analysis of the bearing H measurement showed that a large 30 Hz component was present, which indicated that the generator was out of balance.

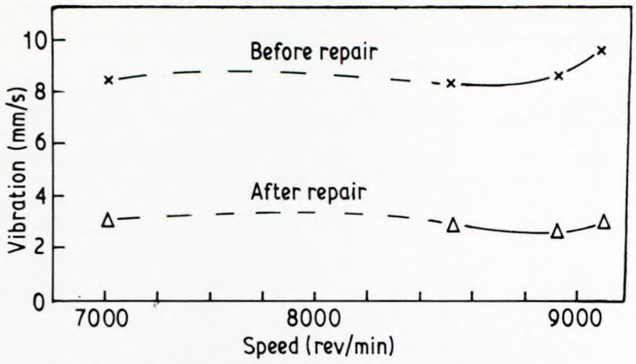


FIG. 11: Compressor non-drive-end vibrations

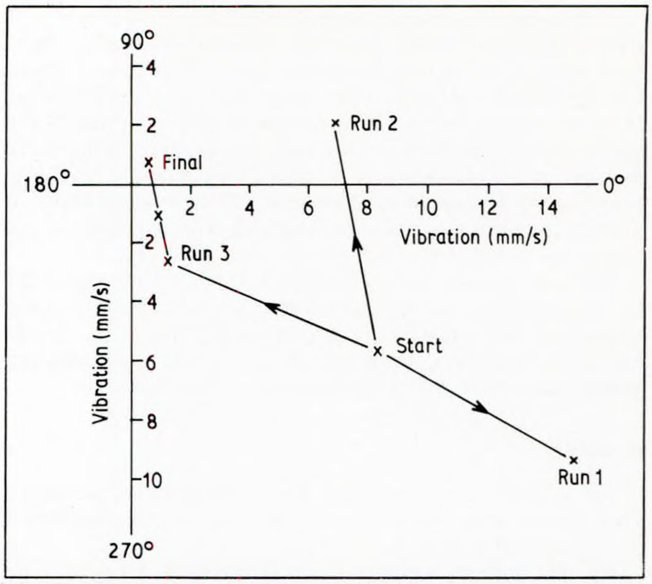


FIG. 12: Alternator non-drive-end vibrations showing progression of vector changes with improvement of balance

The pattern of vibrations on bearing A were much more complex so a real-time analysis trace was taken in the horizontal direction for each load condition, as can be seen from Fig. 10. This analysis reveals that the components of vibration and their most likely causes were as follows:

30 Hz This is very large and is due to the unbalance of the generator rotor.

60 Hz This is a twice per revolution vibration of the generator shaft and is caused by some misalignment between the generator and gearbox. It will be seen that as the load is increased and the unit heats up, the 60 Hz component reduces, most probably because as a result of the growth of the set, the alignment between the generator and gearbox tends to improve. Although this 60 Hz vibration is present, the overall level is relatively small.

76 Hz This is an unusual vibration which corresponds to the first critical speed of the power turbine rotor. In this case it is most likely caused by slight whirling of the power turbine rotor, but because it is so small it is not serious.

132 Hz The vibration at this frequency is almost negligible but it is due to the slight inherent unbalance of the power turbine rotor. As the levels are almost insignificant, one can deduce that this power turbine rotor is in good balance.

120 Hz
142 Hz
148 Hz
152 Hz The vibrations at these frequencies, which occurred at different loads, are from the compressor. It will be observed that both the frequency and vibration levels increased with load. This component of vibration is of concern because the magnitude was so large. It was deduced that the non-drive-end bearing of the compressor had worn.

After the analysis, the compressor non-drive-end bearing was physically examined and found to be worn, as had been predicted. Therefore, a new bearing was fitted and a high-speed in-position dynamic balance was carried out (see Figs 11 and 12).

The vector diagram (Fig. 12) shows the individual stages of balance and the plot of vibration against speed before and after repair are compared in Fig. 11. Once these improvements had been completed, another set of vibration measurements and real-time analysis traces were taken (see Fig. 13).

It can be seen that the overall vibration has reduced to a satisfactory level. The 30 Hz component from the generator was down to about 1 mm/s and the vibrations from the compressor were within International Standards.

One interesting feature of this investigation was that as the balance of the generator improved, the 60 Hz component reduced slightly. In theory, one would not expect this to happen, but it has been noticed on many occasions in the past that balancing does sometimes tend to improve second-order vibrations.

Unit Speed		*Power turbine 7900 rev/min*				*Generator 1800 rev/min*
Power (MW)		0	1.0	1.2	1.4	
Compressor Speed		72%	85%	89%	91%	
A	V	2.9	2.4	2.2	2.5	
	H	3.2	3.0	2.8	3.2	
	A	2.8	2.2	1.7	1.8	
B	V	1.5	1.4	1.5	1.5	
	H	1.7	1.6	1.5	1.4	
	A	2.3	2.1	2.0	2.0	
C	V	1.3	1.4	1.3	1.8	
	H	1.8	1.6	1.7	1.5	
	A	1.2	1.2	1.3	1.5	
D	V	1.5	2.1	2.4	2.5	
	H	2.0	2.2	2.4	2.5	
	A	1.0	1.2	1.4	1.6	
E	V	1.3	1.4	1.5	1.9	
	H	2.3	2.4	2.5	2.5	
	A	2.2	2.0	2.2	2.5	
F	V	3.5	3.5	3.7	4.2	
	H	1.5	2.0	2.3	2.3	
	A	2.5	2.7	2.8	3.0	
G	V	0.5	0.4	0.4	0.5	
	H	0.5	0.5	0.5	0.5	
	A	0.5	0.6	0.5	0.6	
H	V	0.4	0.4	0.4	0.4	
	H	0.8	0.7	0.7	1.0	
	A	1.8	1.6	1.6	1.6	

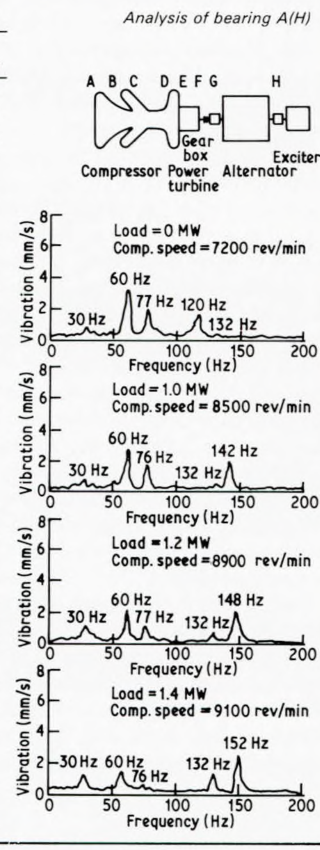


FIG. 13: Vibrations on gas turbine generator after balance and real-time analysis of new bearing A(H)

CASE HISTORY 7: WATER INJECTION PUMP

Water injection pumps, which are used on oil platforms, often comprise a prime mover operating through a gearbox to a high-pressure pump. In this investigation the prime mover was an electric induction motor which ran at 3550 rev/min via a step-up gearbox to the water injecton pump. The complete assembly was mounted on a large fabricated frame which, in turn, was supported from the floor by isolation mounts. Initially, vibration measurements were taken in the vicinity of the bearings at the lettered positions shown in Fig. 14.

It was found that the vibration levels were high, as can be seen from Fig. 14. Therefore, selected positions were tape recorded and these were later processed through a real-time analyser and a hard copy of the vibration trace taken using an *X–Y* plotter. Copies of the real-time analysis trace are shown in Fig. 14.

From the traces, it can be seen that the largest level of vibration at all positions is at a frequency of 59.2 Hz. This corresponds with the once per revolution component of the low-speed shaft, ie motor and gearbox, and indicates that there is unbalance. It will also be seen from the trace of the motor bearing B in the horizontal direction that there is a component of vibration at 118.4 Hz, ie twice the speed of the motor. This indicated some misalignment between motor and gearbox.

The real-time analysis traces for the gearbox were taken over a much wider frequency range to include the effects of gear teeth vibration. Two relatively small components existed at about 1.5 kHz, but they were insignificant, thus indicating that the gear teeth were satisfactory.

The vibration measured on the compressor was mainly at 59.2 Hz and was being transmitted from the low-speed drive of the motor.

Next a range of vertical measurements were taken on the machinery and base frame, as shown in Fig. 15. Because of the flexibility of the isolation mounts, the base was being allowed to twist, which affected the alignment.

From the compressor there was a main discharge pipe supported by means of a clamp and anti-vibration pad. Because this pad was much stiffer than the flexible supports under the machine base, it was tending to restrict the movement at the right-hand lower corner of the frame.

A further test was carried out by taking measurements in a vertical direction down the side of the machine and base frame (Fig. 15). On a normally supported machine one would expect the vibration levels to decrease going down towards the base, but it was noticed in this case that the vibrations tended to increase in the vicinity of the base. Also, the level of vibration

Position		Vibration velocity (mm/s) Total
A	V	3.75
	H	4.0
	A	3.75
B	V	2.0
	H	5.5
	A	3.5
C	V	2.0
	H	11.0
	A	7.5
D	V	6.0
	H	10.5
	A	5.5
E	V	14.5
	H	11.25
	A	3.75
F	V	5.0
	H	12.5
	A	9.5
G	V	5.5
	H	4.5
	A	3.5
H	V	4.0
	H	5.5
	A	3.0

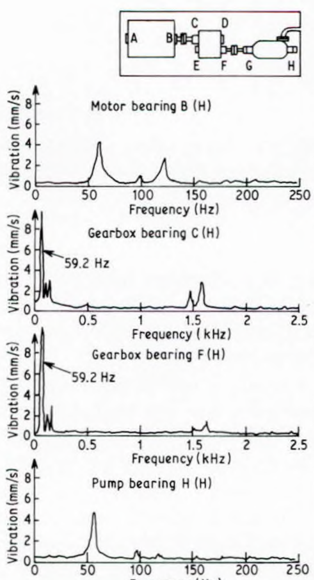


FIG. 14: Vibration levels and real-time analysis for water injection pump

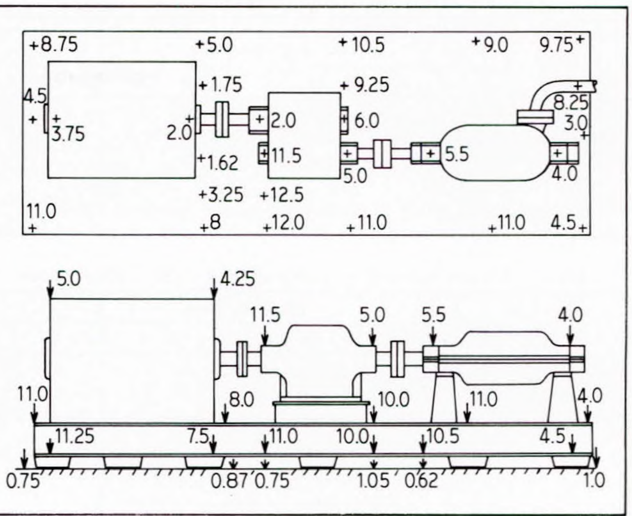


FIG. 15: Vertical vibrations (in mm/s) on water injection pump. Upper: plan view. Lower: side view

on the floor structure under the isolation mounts were acceptable in that the levels varied between about 0.5 and 1.0 mm/s.

Clearly there was a very large difference in vibration between the floor and base frame, which indicated that the isolation mounts were far too flexible, with the result that they most probably amplified the overall vibration, rather than attenuating it! This is a common problem with isolation mounts which either have been incorrectly selected in the first place, or have deteriorated through wear.

To prove that the isolation mounts were too flexible, temporary wedges were placed between the base frame and floor to short-circuit the isolation mounts. This resulted in the vibrations being reduced to about a third of their original value. Therefore, the solution in this instance was to balance the low-speed drive and to install much stiffer isolation pads between the frame and floor.

CASE HISTORY 8: PIPEWORK PROBLEMS

Many of the problems investigated involve structures, pipes, boilers, tanks etc. In the case of problems with structures, these are usually excited by propeller, main engine or auxiliary machinery, whereas large vibrations in boilers and certain pipes are often caused as a result of the flow conditions. However, this is not necessarily the case as external forces can also produce severe vibrations in these areas.

Normally the main problem with structures, pipework etc. occur because they are too weak or flexibly supported, with the result that they become prone to resonance. Where resonance is a problem it is indicated by a small force producing very large levels of vibration, which sometimes result in cracks or failures.

To illustrate this point, a problem which was encountered on a section of ten year old high-pressure steam pipe which ran from twin boilers has been selected. The arrangement of the pipework is shown in Fig. 16 and measurements of vibration were taken at each of the positions denoted by letters. The test was repeated for the following conditions:

1. HP1 valve open and HP2 valve closed.
2. HP1 valve closed and HP2 valve open.
3. HP1 valves open and HP2 valve open.

It was noted that the vibration pattern was similar for all three conditions, but with both valves open the levels were slightly larger, as shown in Table II.

Table II: Pipework vibrations (in mm/s) excited by HP steam

	Pipe position											
	A	B	C	D	E	F	G	H	I	J	K	L
Vertical	3.8	3.0	5.0	3.7	7.5	3.8	3.3	13.7	20.0	15.0	10.0	7.5
Horizontal	3.5	2.5	6.2	3.5	2.3	2.5	2.2	7.5	10.0	7.5	5.0	4.5
								High vibrations in this region				

Note: Main component of vibration is at a frequency of 29 Hz

Table III: Steam excited vibrations (in mm/s) – after repair

	Pipe position											
	A	B	C	D	E	F	G	H	I	J	K	L
Vertical	1.4	1.0	1.8	1.4	2.2	1.4	1.1	4.5	5.4	5.1	3.7	2.5
Horizontal	1.3	1.0	2.3	0.9	0.7	0.9	0.7	2.5	3.2	2.8	1.3	1.7

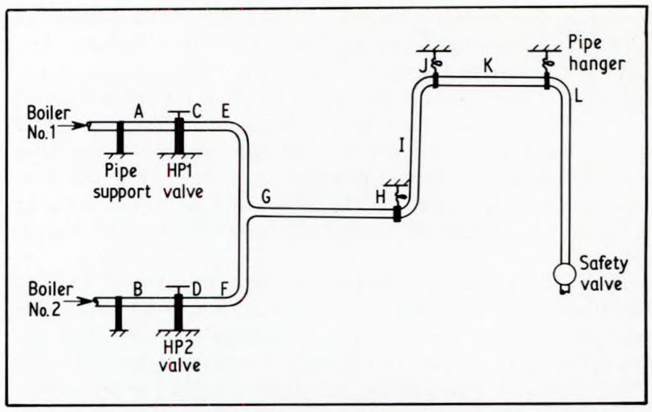


FIG. 16: High-pressure steam pipes from twin boilers

It can be seen from Table II that the vibrations in the vertical direction are almost twice those measured horizontally and that the main component of vibration was at 29 Hz. Clearly, the region between positions H and K was experiencing excessive levels, so much so that the pipe joints were continually leaking and cracks were appearing at the flanges.

Although steam pulsations were producing the vibration in the first instance, it was felt that the natural frequency of the pipework was being excited (ie a small force was causing a large vibration). Consequently, natural frequency tests were carried out with the plant shut down and an external variable frequency exciter attached to the pipe midway between positions H and I.

A vibration transducer was placed at position I in the vertical direction and vibration measurements taken at 5 Hz increments from 0 to 50 Hz. The results were plotted in the form of a 'resonance curve', as shown at the top of Fig. 17. From this curve it can be seen that the natural frequency of the pipework was at 30 Hz, ie very close to the 29 Hz component, measured on line.

This test, therefore, proved that the 'on-line' forces were exciting the pipework natural frequency and thus causing the various failures previously encountered.

After the resonance test, the exciter was run at a steady speed corresponding to 29 Hz (ie the steam exciting force) and vibration measurements taken in the vertical and horizontal directions at each of the lettered positions. As the vertical levels were the largest these were plotted in the lower graph of Fig. 17. It is evident that the area of concern was between positions H and K.

After the test, the pipe hangers at positions H and J were examined. These were found to be badly worn and the springs had wasted away in certain places. Therefore, they were not giving adequate restraint to the pipework. It was decided to take two courses of action:

1. To refurbish all the pipe hangers with stiffer springs.
2. To add an extra hanger in the vicinity of position I, with the object of raising the natural frequency of the pipework beyond the steam exciting frequency range.

After the work had been completed, another set of vibration measurements was taken with the pipework carrying high pressure steam and the two valves open. The measurements are shown in Table III from which it will be noted that the vibration levels with the new and refurbished hangers had been reduced considerably.

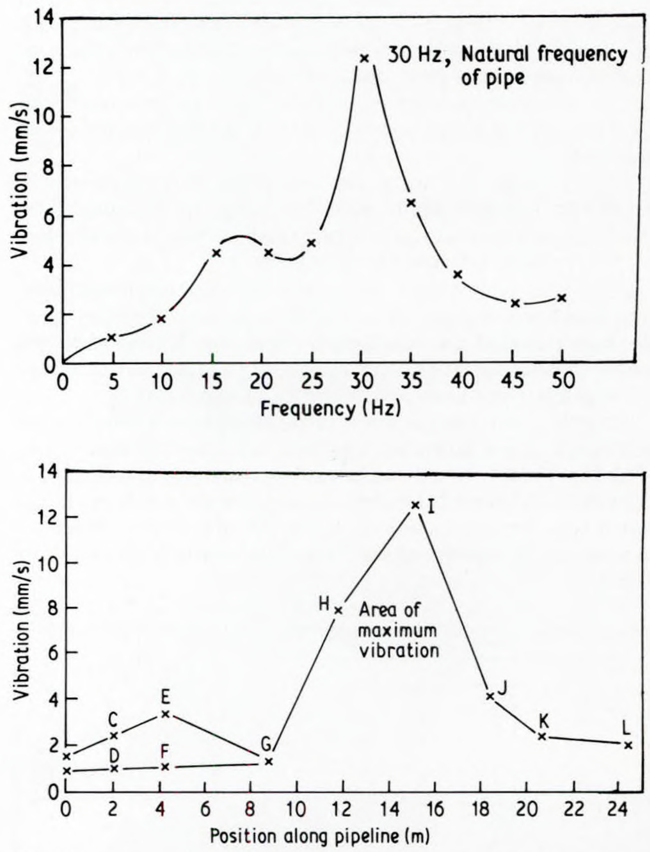


FIG. 17: Tests using vibration exciter. Upper: Resonance test, position I(V). Lower: Vibration deflection along pipe(V)

CASE HISTORY 9: GYRO GENERATOR

One fleet of seven gas carriers which operates from Japan has a vibration monitoring programme onboard each vessel. This programme is used in conjunction with the planned maintenance programme and on one of the vessels it was found that the vibration levels on the forward gyro generator were high. This can be clearly seen in Table IV, where the original measurement for both the forward and aft generators are compared.

Clearly, it can be seen that the vibration levels of the forward generator are too high and it was analysed that the machine was unbalanced. Therefore, a high-speed in-position balance was carried out which reduced the levels to a satisfactory International Standard, as shown.

Normally, after a machine of this type has been balanced, the reduced levels of vibration should remain more or less constant, but it was noticed that after the ship had been drydocked the levels on the forward gyro had suddenly increased. A sudden change of this nature usually indicates that something has moved or broken off the rotating element.

Table IV: Vibration levels (in mm/s) in gyro generators

	Position						
	Top bearing (A)			*Bottom bearing* (B)			
	Pt/ Stb	*Fwd/ Aft*	*Axial*	*Pt/ Stb*	*Fwd/ Aft*	*Axial*	
Aft gyro	1.0	1.5	0.5	0.75	0.62	1.0	
	Satisfactory						
Fwd gyro	15.5*	6.75	2.0	6.5	2.75	2.0	*Analysis* *Total = 15.5
	Vibration excessive due to unbalance						57.5 Hz = 15.5
Fwd gyro After in-position balance	3.5	2.0	1.25	1.75	2.25	1.25	
	Satisfactory						

Note: Warning level = 4.5, high warning = 6.75. Speed 3450 rev/min.

Table V: Vibration levels (in mm/s) in gyro generator before and after stripping

	Position						
	Top bearing A			*Bottom bearing B*			
	Pt/ Stb	*Fwd/ Aft*	*Axial*	*Pt/ Stb*	*Fwd/ Aft*	*Axial*	*Analysis*
Fwd gyro	11.75*	2.25	2.0	3.5	3.5	2.0	*Total = 11.75
	Rough, after drydocking						57.5 Hz = 11.75
Fwd gyro, nut on poles removed	3.5	2.0	1.1	2.5	2.5	1.5	
	Satisfactory						

The analysis of the vibration in Table V indicates that the main harmonic was at the fundamental frequency, indicating the machine was out of balance again.

On being informed by the Chief Engineer that this particular generator had been serviced and overhauled, it was decided to strip the machine and examine it carefully. This was carried out and it was found that a loose nut was being held semi-captive by one of the magnetic poles of the generator. However, the nut was slowly being thrown out by centrifugal forces towards the windings. In time, this nut would have damaged the windings and could have caused a complete failure of the machine. The nut must have been picked up by the magnets when the machine was stripped for overhaul in drydock.

The offending nut was removed and the machine re-assembled. The vibration levels were again re-monitored and it was found that the levels returned to those measured after the machine had been balanced.

The use of vibration techniques clearly saved this particular machine from electrical and mechanical destruction and has shown how a fault can be detected prior to dismantling a machine.

ACKNOWLEDGEMENT

We would like to thank those clients who gave permission to review the problems presented in the paper and their subsequent solution. To protect the interests of these clients, the exact source of each investigation was not specifically indicated so as to preserve confidentiality.

BIBLIOGRAPHY

1. B. H. Thomas, 'Vibration and balancing parameters for ships.' Dissertation for Shell International Marine Ltd.
2. 'Criteria for assessing mechanical vibrations of machines'. VDI 2056 (October 1964). Translation published by Peter Peregrinus Ltd. (1971).
3. 'Criteria for assessing the state of balance of rigid rotating bodies.' VDI 2060 (October 1966). Translation published by Peter Peregrinus Ltd. (1971).
4. 'Measurement and evaluation of mechanical vibrations of reciprocating machinery.' VDI 2063, Handbach Schwingungstechnik (August 1982).
5. 'Mechanical vibration and shock: Guidelines for the overall evaluation of vibration in merchant ships.' ISO 6954 (1984).
6. 'Mechanical vibrations in rotating and reciprocating machinery.' BS 4675 Part I (1976).
7. 'Mechanical balancing of rotating bodies.' BS 5265 (1979).
8. 'Code for the measurement and reporting of shipboard data.' ISO/DIS 4867 (1984).
9. 'Code for measurement and reporting of local vibration data for ships' structures and equipment.' ISO/DIS 4868 (1984).
10. 'Mechanical vibration of large rotating machines with speed range from 10 to 200 rev/s–measurement and evaluation of vibration severity in situ.' ISO 3945 (1977).
11. 'Military standards mechanical vibrations of shipboard equipment.' MIR Std. 167B (Ships), American Bureau of Shipping (August 1969).
12. 'Survey based on preventative maintenance techniques.' (1984). American Bureau of Shipping.
13. 'Recommendations designed to limit the effects of vibrations onboard ships.' Bureau Veritas (1979).
14. M. Baudin and P. Morel. 'How to adapt vibration studies to shipbuilding sequence requirements.' Bureau Veritas technical papers (May 1984).
15. 'Technical and research code C4: Local shipboard structures and machinery vibration measurements.' Society of Naval Architects and Marine Engineers (Dec. 1976).
16. 'Technical and research code C1: code for shipboard vibration measurement.' Society of Naval Architects and Marine Engineers (January 1975).