THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
One World Trade Center, Suite 1369, New York, N.Y. 10048
Paper to be presented at the Ship Vibration Symposium
Arlington, Va., October 16–17, 1978

Vibration Signature Analysis as a Preventive Maintenance Tool Aboard Ship

J. B. Catlin, Jr., Visitor, IRD Mechanalysis, Inc.,
Columbus, Ohio

ABSTRACT

This paper describes vibration measurements of shipboard machines which are now being utilized in a program of preventive maintenance as a supplement to standard shipboard maintenance procedures. While similar in many respects to vibration measurements used ashore, the shipboard program must take into account environmental vibrations that occasionally mask the vibration of shipboard machines. Illustrations are shown of these environmental vibrations that are caused primarily by propeller blade passing frequencies and random turbulence from the propeller and hull.

Implementation of a shipboard program using two portable instruments, a vibration meter and an analyzer/XY recorder, is briefly described. The value of such a program results from its ability to detect machine deterioration in its earliest stages while there is sufficient time to correct problems before they reach the critical stage. Examples of vibration signatures indicating machinery defects are given.

Measurement techniques and procedures are also discussed which have been developed to pinpoint defects and separate environmental vibrations from those associated with the machines.

INTRODUCTION

The use of vibration analysis as a *routine preventive maintenance* tool aboard ship is only about ten years old, although there are a number of isolated cases of its use prior to that time. Even today its widespread use is just starting because, as with any new technique, it has required time to gain the confidence of both management and operating personnel before being routinely accepted as a standard tool.

The marine industry's efforts in the area of vibration analysis are not completely without precedent, since industry ashore has been utilizing this preventive maintenance technique for almost 40 years. It is probably safe to say that most large industries throughout the world, and countless small, now rely on vibration analysis as the backbone of their preventive maintenance programs. The reason for this is, of course, vibration analysis' unique capability to diagnose machinery mechanical condition while a machine is operating, and pinpoint even minor mechanical problems (unbalance, defective bearings) while they are still in the incipient stage.

There are, however, significant differences between the application of vibration analysis aboard ship and that ashore, so it is not surprising that the transfer of technology has not been more rapid.

Perhaps, the biggest difference between vibration analysis ashore and vibration analysis aboard ship is environmental vibration. Although environmental vibration is occasionally encountered in fixed machinery installations ashore, it is certainly not as pervasive as aboard ship.

When it does occur aboard ship it tends to mask a machine's self-generated vibrations and thus obscure the true mechanical condition of the machine. It, therefore, requires a different selection of measurement instruments, different measurement procedures, and different interpretation techniques from those used ashore.

For purposes of this paper environmental vibration can be defined, at least as it relates to vibration analysis of shipboard machines, as follows:

Environmental vibration is that vibration, measured on a machine, whose source is other than that directly resulting from the operation of that machine.

For example: vibration of the main H.P. turbine aboard ship which is caused by vibration from the propeller and main reduction gears would be considered machine vibration associated with the operation of the H.P. turbine. However, vibration of the ship service turbogenerator which is caused by the propeller would constitute environmental vibration, since the propeller is not directly associated with the operation of the turbo-generator.

Aboard ship the predominant sources of environmental vibration come from the blade passing frequencies and flow turbulence of the propeller plus the wave action and flow turbulence impinging on the hull. To a lesser extent there can also be environmental vibration caused by the transmission of one machine's vibration to another.

In the sections which follow, instrumentation and measurement techniques are discussed which have been successfully used to implement vibration signature analysis as a preventive tool aboard ship. Examples are given of vibration signatures which uncovered defective machines. Further examples show cases where environmental vibrations of different types were encountered, and methods used for analyses under these conditions.

INSTRUMENTATION AND PROCEDURES FOR A SHIPBOARD VIBRATION MEASUREMENT PREVENTIVE MAINTENANCE PROGRAM

A complete shipboard vibration preventive maintenance program involves the use of a handheld vibration meter and a portable narrow band vibration analyzer which can be connected to an XY recorder to produce hard copy vibration signatures. Figures 1 and 2 show examples of such instruments.

Fig. 1 Example of Marine Vibration Meter

Fig. 2 Example of Vibration Analyzer with XY Recorder

In this paper the use of the handheld vibration meter will not be specifically discussed; however, it should be mentioned that the reason for utilizing two types of instruments is to minimize the measurement time required while still maintaining the capability to detect and pinpoint machinery problems in their incipient stages. The vibration meter is used for quick periodic checks of the machinery to determine whether it is in good mechanical condition, or in need of maintenance and/or repair. On the other hand, the analyzer is used to pinpoint any defects detected by the vibration meter.

To startup a program aboard ship, it is necessary to take a complete set of vibration meter readings and a set of vibration signatures with the analyzer. These may be taken in port or at sea depending upon the ship's schedule and the effects of environmental vibration. This is a vital step which serves two purposes:

First, it provides baseline data against which future measurements can be compared. While all machines have certain common predictable vibration amplitudes and frequencies, they also have individual characteristics which are the result of manufacturing, assembly tolerances, and their foundationing. By taking baseline data at the start of a program, these individual characteristics can be accounted for, and greatly simplify the task of pinpointing defects, should any occur at a later date.

Second, it provides a means of evaluating the machinery mechanical condition at the start of the program. Although it is not necessary to run machines at maximum speed or load for these baseline measurements, it is important that any subsequent measurements be taken under the same operating conditions. While this initial baseline data will not provide the details of machinery condition because no previous vibration measurements will be available for comparison; nevertheless, most of the significant and even some minor problems can be caught with this data.

Once this set of measurements is complete, a schedule of periodic, routine vibration meter checks is set up to coincide with the ship's operating schedule. These checks are designed to catch any machinery mechanical problems before they become serious, and provide the lead time necessary to enable the ship to correct the problem at a convenient time.

No additional vibration signatures are required after the initial baseline signatures are taken until such time that the vibration meter indicates a mechanical problem in the machine. In such case the analyzer and XY recorder are brought back aboard ship, and a new set of signatures obtained for the machine. These are then compared with the baseline measurements for that machine to indicate the specific frequencies which have changed, thereby, pinpointing the source of the trouble.

As was implied above, it is only necessary to maintain the vibration meter aboard ship. The analyzer, because of its relatively infrequent use, can be left ashore at some strategic location; and brought to the ship only when necessary to analyze a problem.

Limiting the shipboard equipment to a simple handheld meter is desirable for several reasons. First, the crews aboard ship have been reduced to a point where there are very few manhours available for special tasks. By using the vibration meter, measurements require only a few seconds; hardly more than the time it takes for a man coming on watch to place his hand on machine bearings to check their temperature. Recording these measurements also takes only a matter of minutes. Second, the simplicity of meter operation and data interpretation requires a very minimum of training, so it is not necessary for crew members to attend any lengthy training sessions.

The analyzer, on the other hand, which requires somewhat more skill in its operation and data interpretation, can be handled by shoreside personnel, such as a port engineer, who would be more readily available for training.

At this point it might be worth noting that while the primary benefit of vibration analysis aboard ship is derived from its use in preventive maintenance, it can also prove valuable in a number of other areas, namely:

1. As an inspection tool for new construction to verify mechanical condition of main and auxiliary machinery prior to ship delivery.
2. As a troubleshooting technique to pinpoint the cause of suspected machine faults.
3. As a means of verifying that machinery repairs have been done correctly.
4. As a pre-overhaul technique to aid in determining what repair work is required.
5. As a post-overhaul technique for verification that repair work has been properly accomplished.

The specific signatures shown in the figures of this paper which follow were for the most part generated with a manually tuned swept frequency narrow band (5 percent bandwidth) analyzer which was connected to an XY recorder to provide the hard copy. All measurements were made on the bearing housings of the machines with a velocity type transducer. These signatures were the result of preventive maintenance program startups, troubleshooting, as well as pre and post overhaul measurement.

SOME EXAMPLES OF MACHINERY PROBLEMS PINPOINTED BY SIGNATURE ANALYSIS

Some marine engineers are able to detect defects in shipboard machines by changes in sound or touch, although this capability requires considerable knowledge and skill plus extended time aboard the ship.

As an aid to the shipboard engineer, vibration signature analysis offers two primary advantages for machinery defect detection. It provides a positive indication of specific faults such as unbalance, misalignment, or defective bearings which would be difficult to distinguish by sound or touch, and which would not be indicated by temperature or pressure. It also gives a quantitative number (i.e., vibration level) which can be used to evaluate the severity of the defect, and aid in the decision as to whether to shut down or to run; and if the decision is to run, how long it will last. Guidelines indicating acceptable machinery vibration levels are available from: machinery manufacturers, industrial associations, U.S. Navy (1), SNAME (2), IRD Mechanalysis, Inc. (3), and others.

By using vibration signature analysis along with the traditional inputs of temperature, pressure, sound and touch, the engineer is in a much better position to minimize operational problems. This also gives the shipboard engineer the opportunity to reduce open-and-inspect work, as well as minimizing repairs through knowledge of the specific repairs which are needed.

Given below are some case histories which illustrate the use of vibration signature analysis as a preventive maintenance tool.

Forced draft blowers are one of the types of shipboard machines which are more prone than others to mechanical problems. Figure 3 compares the vibration signatures for two of these units which were made during a pre-overhaul vibration check as an aid to machinery overhaul planning. The one forced draft

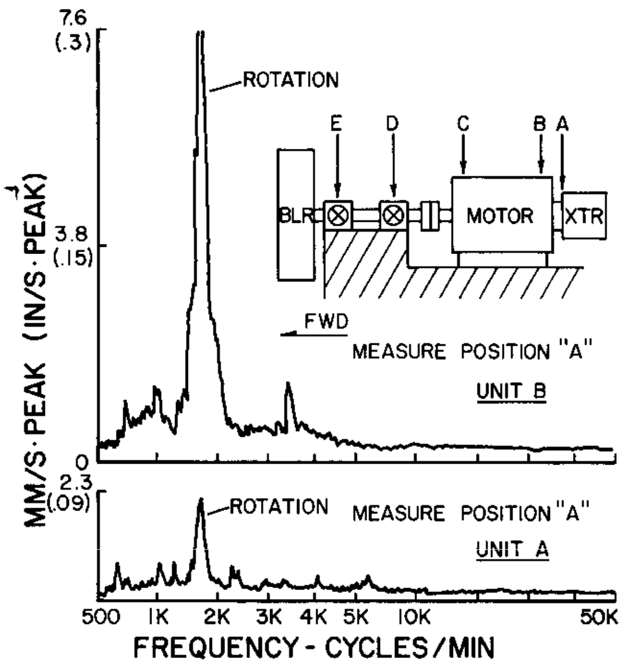


Fig. 3 Comparison of Vibration Signatures of Two Forced Draft Blowers

blower, unit A, was found to be in very good mechanical condition with low amplitude of about 1.8 mm/s peak (.07 in/s peak) or 20.3 µm peak-peak (.8 mils peak-peak), at the rotational frequency and no other vibrations of significance. The other blower, however, shows a major vibration peak of 12.4 mm/s peak (.49 in/s peak) or 152.4 µm peak-peak (6 mils peak-peak), at rotational frequency caused by rotor unbalance. No other important vibration peaks were found indicating that, except for the unbalance which was recommended for correction to avoid rapid wear of the bearings, the blower was in good mechanical condition.

On board ship during the same pre-overhaul vibration survey defective ball bearings were found on the turbine driven tank washing pump. To show the vibration signature characteristics of these bad bearings, Figure 4 compares the vibration on this pump bearing, Unit A, with that of an identical pump on a sister ship which had bearings in good mechanical condition. As is often

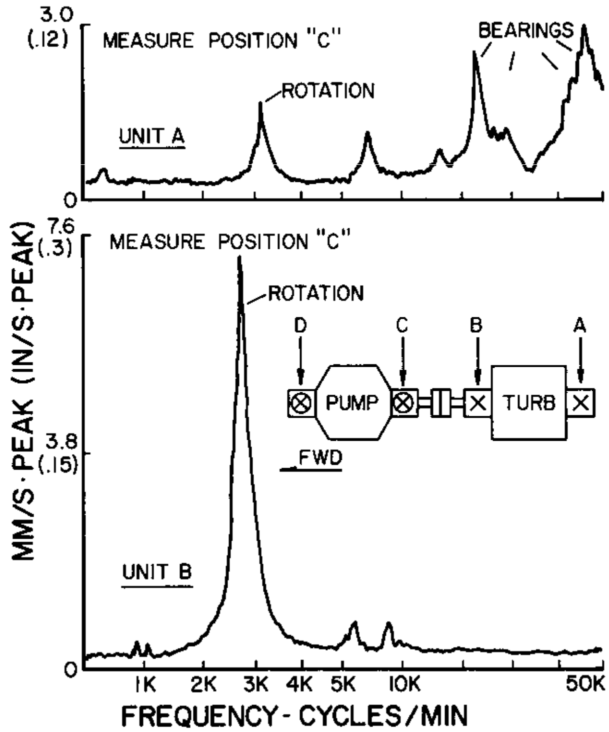


Fig. 4 Vibration Signatures Showing Difference Between Good and Bad Ball Bearings on Turbine Driven Tank Washing Pump

typical of defective bearings, a high frequency broadly peaked region of vibration occurs. In this case it is seen from 40,000 to 60,000 cycles/min. The lower peak from 20,000 to 30,000 cycles/min is also attributed to the bearing. In contrast, the same pump bearing on Unit B shows no vibrations of significance in the frequency regions 10,000 to 60,000 cycles/min. This Unit B, however, does exhibit moderate amount of unbalance as indicated by the 6.1 mm/s peak (.24 in/s peak) or 38.1 µm peak-peak (1.5 mils peak-peak), vibration at pump rotational frequency.

As a further comparison, Figure 5 shows the bearing housing of both the turbine and the pump. Of particular note is the fact that the bearing vibrations are not transmitted to the turbine which makes identification of the defective bearings quite straightforward.

On still another ship a main circulating pump was observed to be vibrating excessively. A vibration signature analysis revealed that it was not unbalance as was suspected, but misalignment, since the large vibration peak was occur-

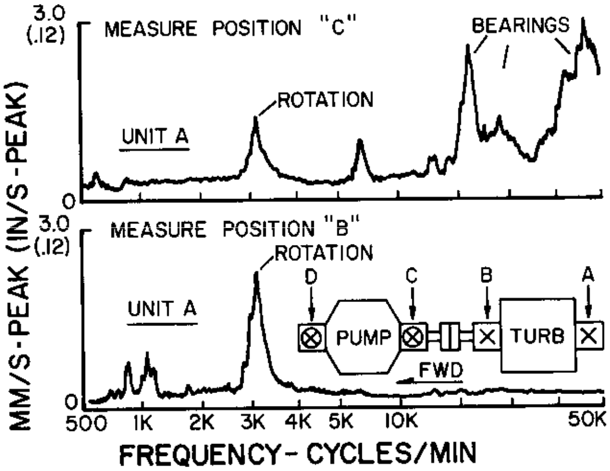


Fig. 5 Comparison of Vibration Signature on Turbine with that on Pump

ring at twice rotational frequency rather than at rotational. This is shown in Figure 6. The effect of this misalignment was aggravated by the fact that the

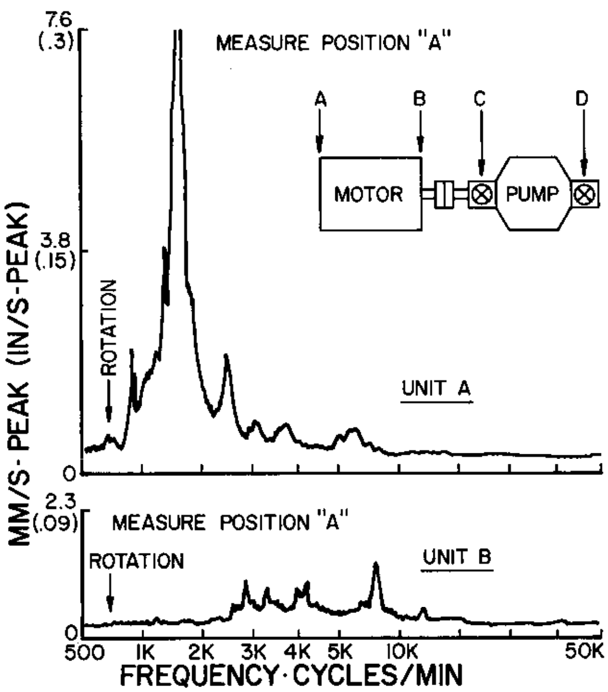


Fig. 6 Example of Main Circulating Pump with Shaft Misalignment Problem which was Aggravated by a Resonant Foundation

machine foundation had a natural frequency which coincided with twice rotational frequency. This frequency was checked by "bump" test in which the foundation was bumped and resulting natural frequency automatically registered on the analyzer.

The very high level of about 20.3 mm/s peak (.80 in/s peak), or 254 µm peak-peak (10 mils peak-peak), called for priority action to avoid excessive wear on the coupling and bearings. Proper alignment was recommended to reduce the vibrations to an acceptable level, although it was also suggested that the foundation be stiffened in the athwartship direction to eliminate the resonant condition. Again, for comparison purposes, an identical pump and motor unit, with the exception of the foundation which had considerably less height, is shown as Unit "B" in Figure 6. In this unit, no important vibrations are present, indicating that it is in good operating condition.

Another forced draft blower which was checked is shown here in Figure 7 as

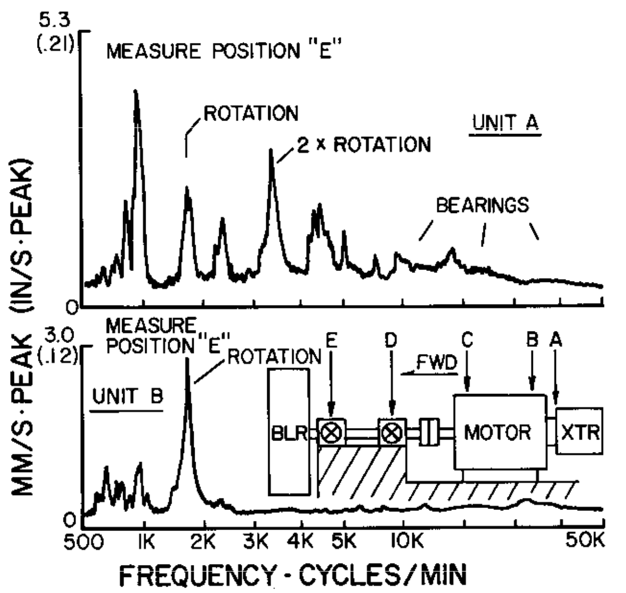


Fig. 7 Forced Draft Blower with Multiple Problems: Misalignment, Looseness, Bearing Deterioration

an illustration of a unit having multiple problems. This measurement was on a pillow block bearing supporting the forced draft blower shaft. The twice rotational frequency peak which is greater than the rotational is indicative of a small amount of misalignment, while the multiple harmonics of rotational shows some looseness in the bearing. In addition, the broad-band vibration extending from below 5,000 cycles/min to above 20,000 cycles/min shows some minor bearing deterioration. While this analysis indicated that the bearing could probably operate for some time, it was recommended that vibration checks be made at frequent intervals to watch for further deteriorations, and plans made to change bearings at a convenient time in port.

In the sections which follow, the effect of environmental vibration from propeller, hull and adjacent machines which occasionally interfere with machinery measurement is discussed.

HULL VIBRATION GENERATED BY SHAFT ROTATION AND PROPELLER BLADE FREQUENCIES

To obtain a true indication of a machine's mechanical condition it is necessary to eliminate the effects of any significant environmental vibration from the vibration measurement. It is worthwhile, therefore, to identify the characteristics of those environmental vibrations which may affect vibration readings.

As a result of vibration measurements on a number of ships, some tentative observations can be made concerning hull environmental vibration. These observations are limited primarily to vibration measurements on auxiliary machinery, and not the hull girder itself.

The most significant of the hull environmental vibrations is generally caused by the propeller blade frequency (i.e. number of propeller blades times shaft RPM) and its harmonics. On present merchant ships fundamental blade frequencies typically run from a low of about 320 cycles/min to a high of 600 cycles/min. The higher harmonics, as measured at the thrust bearing, generally drop off in amplitude, with the second harmonic, perhaps, one-half, or less, the amplitude of the fundamental and the higher harmonics still lower. An example of this is shown in Figure 8 for a commercial ship at full speed in calm seas. Such is not always the case for propeller blade frequencies measured on the auxiliary machines. The second, third, or even fourth propeller blade harmonics can have amplitudes almost as high or higher than the fundamental. The measurement data suggests that this is caused by the natural frequencies of the auxiliary machines and their foundations, which apparently are often in the range of the higher harmonics rather than that of the fundamental.

While the amplitudes of these frequencies vary from point to point because of different structural transmission paths and local natural frequencies, for engine rooms located at the stern amplitudes appear to be typically highest on machines farthest aft, and decrease with distance forward and upward in the ship.

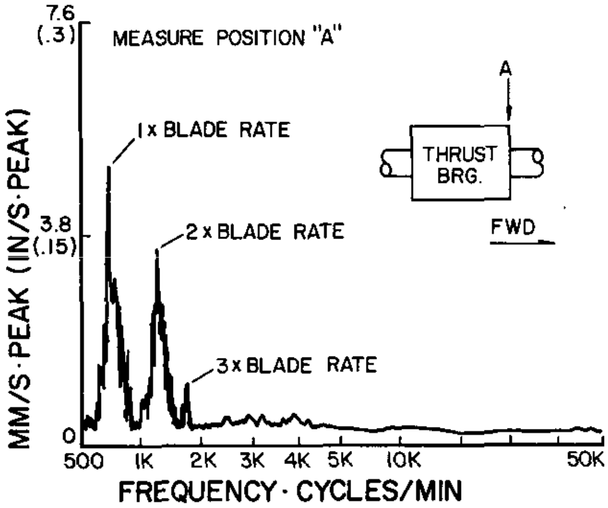


Fig. 8 Hull Girder Vibration As Measured at Thrust Bearing

This observation is based upon limited data, however, and it is recognized that there are cases where low vibrations of the hull can be magnified at the bridge or other areas of a ship.

From the standpoint of reduced environmental vibration, the engine room located at some distance from the stern offers considerable advantage; however, most present designs place the engine room as far aft as possible to increase cargo carrying capability.

Many other factors also affect the machinery vibration amplitudes of the propeller blade frequency vibrations on the auxiliary and propulsion machines. Ship displacement, horsepower, hull form, stern design, hull vibration modes, wake velocity profile in way of the propeller and the draft and trim are some of the more important of these.

In proceeding with a machinery vibration measurement program, there are many ways in which the effects of propeller blade frequency vibrations can be eliminated.

1. It should first be stated that in many instances the propeller blade vibrations are of sufficiently low amplitude as to be of no consequence in the machinery measurement. Figure 9 compares the vibration signature of an SSTG set in port with that obtained underway with the ship at full speed. The in-port and at-sea loads were 160 kW and 190 kW, respectively. For this particular case the vibration signatures are almost identical. Propeller vibration frequencies

are completely absent.

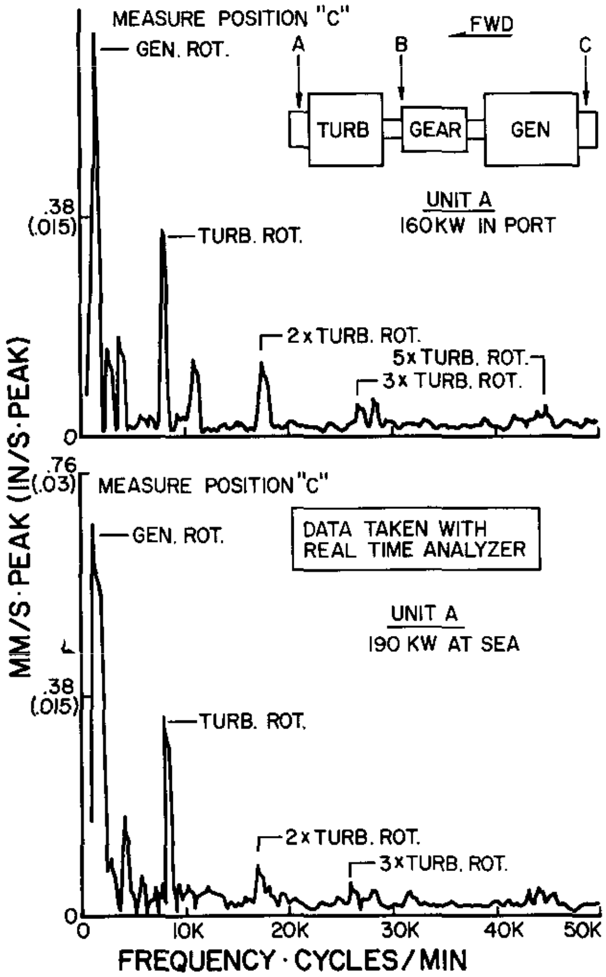


Fig. 9 Comparison of In-Port and At-Sea Vibrations of a Ship Service Turbogenerator Set

2. It is also often the case that the propeller blade frequencies of significant amplitude are below the frequency range of interest for the machine being measured. The majority of shipboard machines have rotational speeds above 1400 r/min, while propeller blade frequencies of significance are usually of 1200 cycles/min and below. Under these conditions an instrument measuring overall vibration which includes a suitable high pass filter can be used to eliminate the propeller vibrations and measure only those vibrations associated with the machine itself. The vibration signature will also be unaffected by such propeller vibrations, as shown in Figure 10.

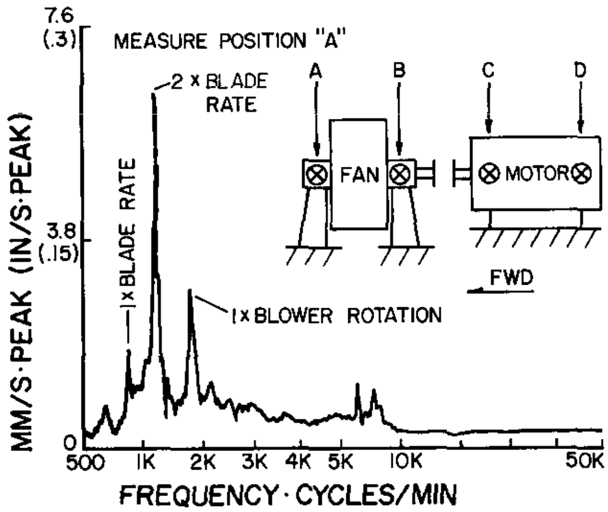


Fig. 10 Vibration Signatures Showing Case Where Propeller Blade Frequencies are Lower Than Frequencies Associated with the Machine

3. Where significant propeller blade frequencies extend up into the frequency region of machine vibrations, it is often still possible to separate the two when the propeller vibrations occur at different frequencies from those of the machine. In addition, propeller blade frequencies typically have a distinctively different characteristic appearance than those generated by machinery problems such as unbalance and misalignment when analyzed by a narrow band slowly sweeping frequency analyzer. It should be mentioned that a fast sweeping analyzer will not show these differences. This provides a further means of identifying and separating the two sources. Even when propeller blade frequencies and machine frequencies coincide, the difference in the characteristics still enables the operator to obtain an indication as to which of the two vibrations is dominant. Figure 11 illustrates the different appearance of these two types of vibrations. This difference is caused by the fact that the propeller blade frequency vibrations are generated by a random (flow) forcing function which produces a "hashy" vibration peak, whereas machine unbalance and misalignment are caused by mechanical forces which are typically almost sinusoidal and produce a smooth

vibration peak.

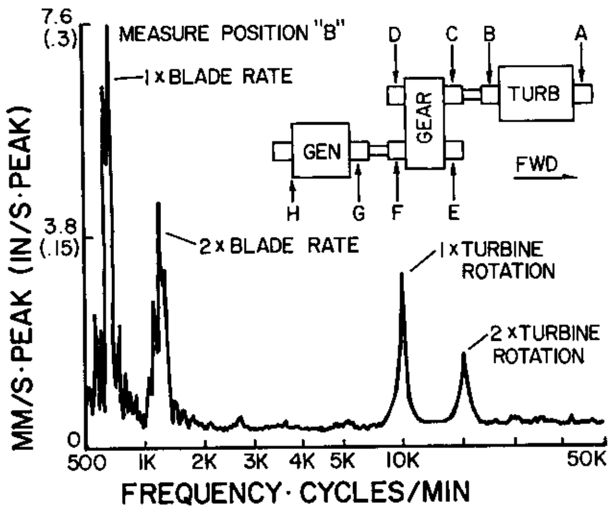


Fig. 11 Illustration Showing the Difference in Appearance Between Propeller Blade Vibrations and Vibrations Caused by Machine Unbalance and Misalignment

4. When, as it infrequently occurs, the propeller blade vibrations are coincident and/or dominate the machine vibration spectrum, the machinery vibration measurements should be made only at reduced ship speed (with consequently lower propeller blade vibrations), or in port. Several examples which follow compare machine vibration signatures taken at full ship speed with those taken in port.

Figure 12 compares the at-sea with in-port signatures of a large vertical shaft ballast pump. The at-sea measurement was with the ship at full speed, calm sea and normal loaded draft. The at-sea signature is dominated by the first, second, third and fourth harmonics of the propeller blade frequencies. In the in-port signature it is interesting to note that the frequency at the second harmonic is still present, and is, in fact, not primarily caused by the propeller, but by the pump's own turbulent flow which is exciting a natural frequency in the pump and its foundation.

Figure 13 shows the in-port and at-sea signatures for a main circulating water pump. Again, the measurement at sea was taken with the ship at full speed in calm seas with normal load draft. The first and fourth propeller blade frequencies control the at-sea signature, making it necessary to measure this unit in port in order to determine the machine's true vibration signature.

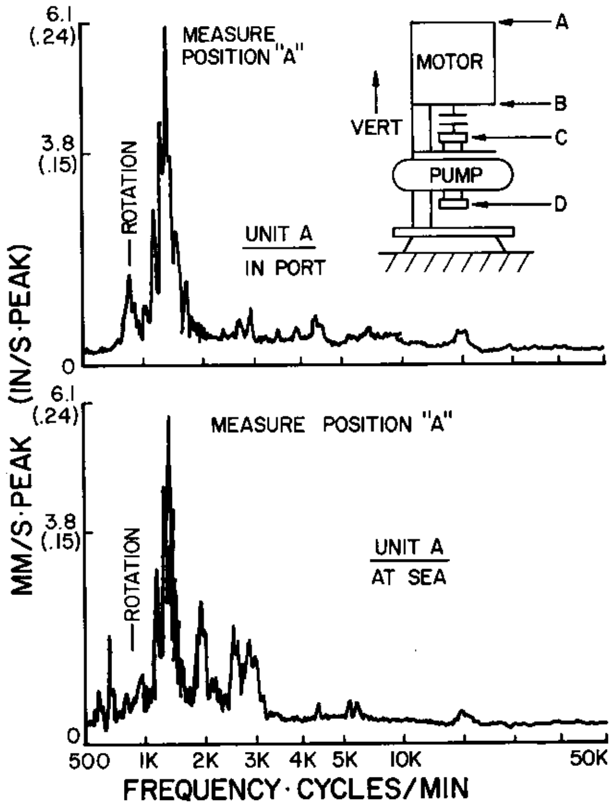


Fig. 12 Comparison of In-Port and At-Sea Vibrations of Large Vertical Shaft Ballast Pump

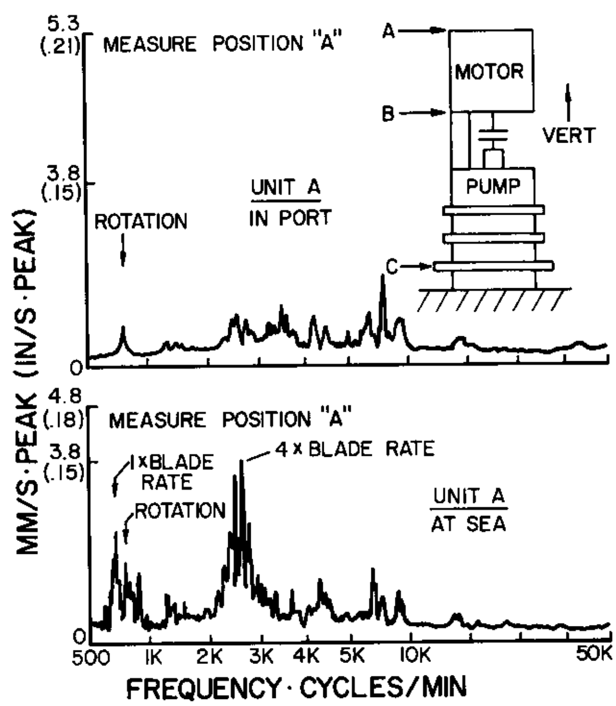


Fig. 13 At-Sea and In-Port Vibration Signatures of Main Circulating Water Pump Showing Presence of Propeller Blade Frequencies

HULL VIBRATION GENERATED BY HULL AND PROPELLER RANDOM FLOW TURBULENCE

Machine vibration caused by hull and propeller generated random flow turbulence is not often a problem in machinery vibration measurement but can occassionally result in masking a machine's vibration signature. Impact type vibrations can also occur which are caused primarily by wave action, but these are generally intermittent and do not have a significant effect on the vibration signatures. It is the random turbulence generated vibration which is not easy to measure and identify because it lacks the distinctive discrete frequency characteristics which are associated with machine vibrations and those of the propeller blade frequencies. Identification is primarily by deduction through the process of elimination of identifiable frequencies. This process is aided somewhat by the fact that it's random broad band nature produces a distributed "hashy" type signature when plotted by a slow swept narrow band frequency analyzer. This distributed energy often forms into broadly peaked frequency regions, and occasionally is exhibited as fairly sharply pronounced frequency peaks. This peaking appears to be the result of weak, or strong excitation of local hull and/or foundation structural natural frequencies. The extent of the peaking is felt to be related to the degree of damping in the structure, although some periodic flow vortex generation could be a contributing factor.

As might be expected, flow turbulence appears to be most pronounced toward the stern; and, therefore, when it is seen it generally affects machinery installed close to the stern and particularly those mounted low in the ship.

Since the various pumping systems aboard ship can develop flow excited vibrations in themselves as a result of the fluids passing through the pumps and piping, it is often difficult to separate these vibrations from those generated by the hull and propeller flow. In such cases, where it is suspected that hull and/or propeller may be contributing to the vibrations measured on a pumping system, it is necessary to take measurements in port or at reduced ship speed where hull and propeller flow were felt to be contributing, as shown in the following examples:

This first example, shown in Figure 14, involves vibration measurements on a general service pump located on the lower level of a ship with a stern configured engine room. The two signatures show the same measurement point for the pump operating at slow speed under calm at-sea conditions with the ship at full speed, and in-port also with the pump at slow speed. While there are some indications of propeller blade frequencies, the broad band nature of the at-sea signature is attributed to the hull and propeller random turbulence flow excited vibrations.

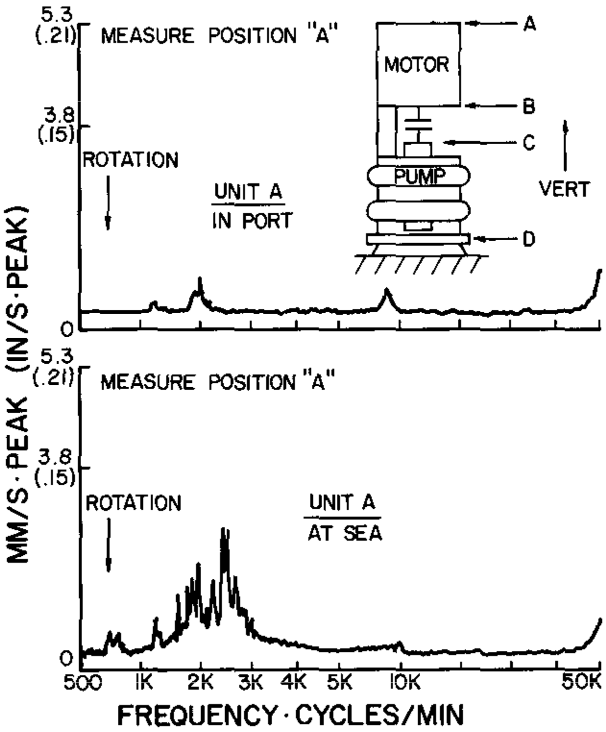


Fig. 14 Comparison of At-Sea and In-Port Signatures of General Service Pump Which Shows the Characteristics of Hull and Propeller Random Turbulence Flow Excited Vibrations

The second example, shown in Figure 15, illustrates the effect of hull and propeller generated flow vibrations on a main circulating pump. Again, this pump was installed low in the ship in an aft located engine room. The two conditions under which the vibration measurements were taken were: under calm sea state with ship at full speed, and in-port. It can be seen that the vibration signature of the at-sea measurement from 600 cycles/min to about 4,000 cycles/min was dominated by propeller and hull vibrations which include both the propeller blade frequencies and the random turbulence.

MACHINE VIBRATION CAUSED BY TRANSMISSION OF VIBRATION FROM ADJACENT MACHINES

A ship's hull and associated machine foundationing would appear to be an ideal transmission path for vibratory energy, and that vibrations occur-

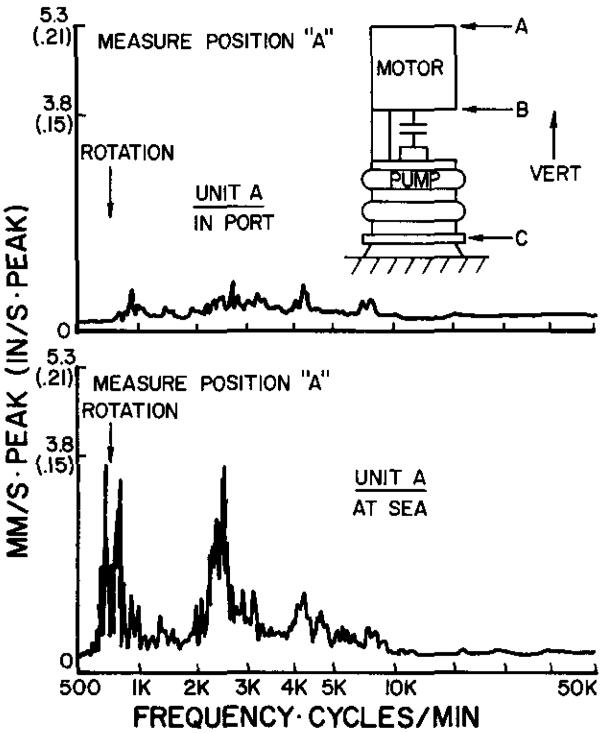


Fig. 15 At-Sea and In-Port Signatures of Main Circulating Pump Whose At-Sea Signature was Dominated by Propeller Blade Frequencies as Well as Random Turbulence

ring in one machine would be readily transmitted to adjacent machines with resulting difficulty in vibration measurement. Surprisingly, however, for the conditions encountered in machine vibration measurements, this does not appear to be a problem in the majority of cases. Several reasons can be suggested for this:

1. Adjacent machines often involve spared units, which means that only one unit is running at one time.

2. The inherent design of the foundations makes the structural path a relatively poor transmitter because of changes in structural impedance; auxiliary machine foundations generally have a relatively low mechanical impedance in comparison to the ship frames, longitudinals and columns.

3. Machine vibratory energy is dissipated by transmission in many directions, rather than by being transmitted solely to an adjacent machine.

Of course there are occasional cases where there is noticeable coupling; but this usually occurs where a machine is vibrating excessively because of a mechanical problem, which is readily recognized and pinpointed.

One specific example of vibration transmission involved adjacent Forced Draft Fans. These Fans were mounted on structural foundations as shown in Figure 16. Both were required to be

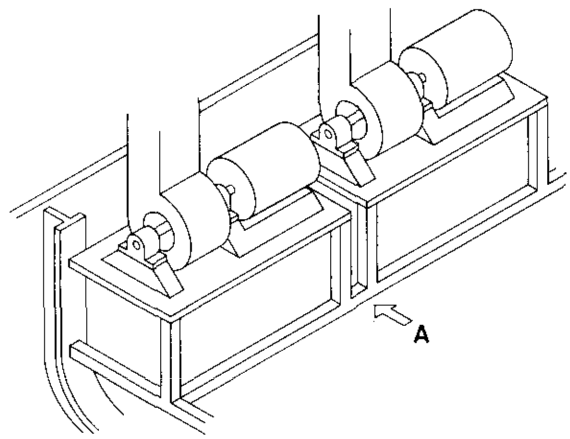


Fig. 16 Diagram Showing the Structural Foundations of Two Force Draft Fans

operating at the same time under normal steaming conditions. The two adjacent support columns at position (A) were close together and supported by the same structural member. The increased stiffness at this point was not sufficient to prevent the transmission of the fan vibration between the two F.D. fans. Measurements on both fans indicated high vibration levels at rotational speed indicating major rotor unbalance. This is shown in Figures 17 and 18.

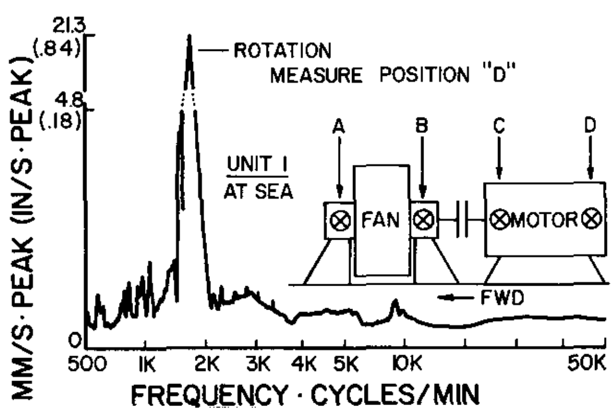
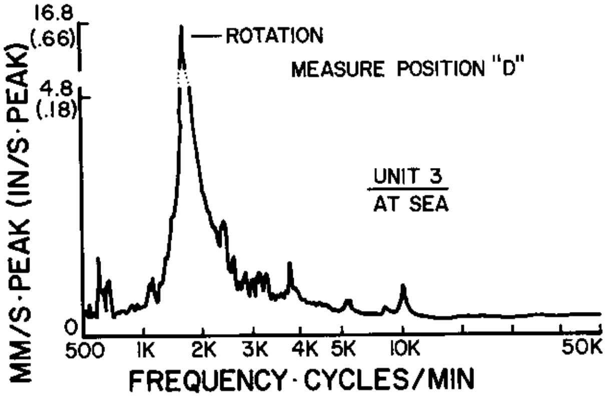


Fig. 17 Signature of Forced Draft Fan #1 Whose Unbalance Vibration Caused a Beat with Fan #3



Fib. 18 Signature of Forced Draft Fan #3

What indicated the vibration transmission between the two units, however, was the wide, slowly varying amplitude at rotational speed on each machine indicating a "beat" between the vibrations of both machines. When each machine was run individually, (i.e. one at a time), the beat completely disappeared, and the mechanical condition of each machine could then be evaluated. As seen in Figures 17 and 18, machine #1 with a level of 21.3 mm/s peak (.84 in/s peak) at rotational speed had significantly more unbalance than machine #3 with an unbalance level of 16.8 mm/s peak (.66 in/s peak).

Interestingly, measurements of two identical forced draft fans, with the same foundationing, located on the opposite side of the ship, had somewhat lower vibration levels and had no indications of a beat frequency. The signatures of these machines are shown in Figures 19 and 20. This showed that by balancing

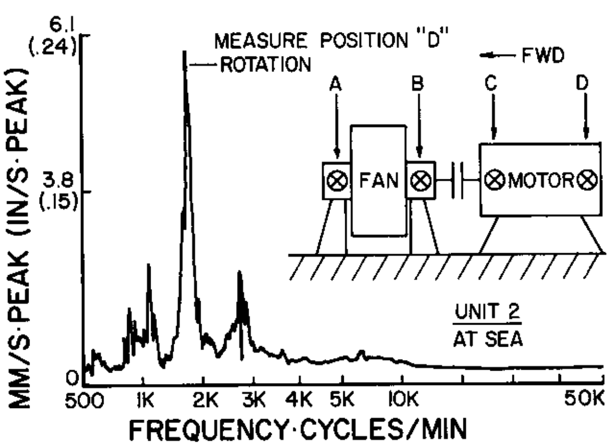


Fig. 19 Signature of Forced Draft Fan (Identical to #1 in Figure 17) Which Exhibited No Beating

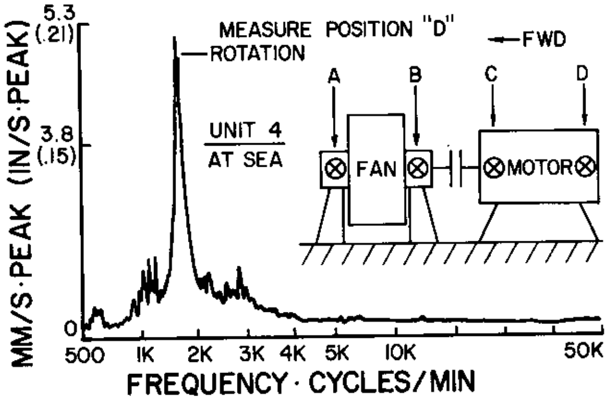


Fig. 20 Signature of Forced Draft Fan (Identical to #3 in Figure 18) Which Exhibited No Beating

the machines with the high vibration levels that the problem could be solved; and that it would not be necessary to stiffen the foundationing at point A, although this might be desirable to reduce the vibration coupling between the machines. It also showed that for low vibration levels the structural path was not sufficient to interfere with preventive maintenance vibration checks of each machine, even when both were running.

SUMMARY

The use of vibration measurement aboard ship for preventive maintenance is only about ten years old, but has proved to be a valuable supplement to other standard shipboard maintenance procedures. Vibration Measurement aboard ship differs from that ashore because of environmental vibration which arises primarily from propeller blade passing frequencies, random turbulence generated by the propeller and hull, wave action and, to a limited extent, transmission of vibration from one machine to another.

A shipboard program can be implemented with two portable instruments: a handheld vibration meter which is used for quick periodic checks of machine condition, and a vibration analyzer/XY recorder which provides graphic vibration signatures for each machine to pinpoint defects detected by the vibration meter.

Many different mechanical problems can be caught in their incipient stage using vibration measurement which then permits planned corrective maintenance, rather than emergency shutdowns. Some of the more important of these include: unbalance, misalignment, defective bearings, looseness, and gear problems.

Under some conditions environmental vibrations can interfere with the true measurement of machine vibrations. A number of procedures are available for dealing with this. Special electronic filtering in the vibration meter will reduce the effect of low frequency propeller vibrations, and careful signature analysis can separate environmental vibrations from those of the machine. For some machines in areas of high environmental vibration it may be necessary to carry out the measurements in port or at low ship speeds. Where environmental vibrations are caused by vibration transmission from adjacent machines, foundation stiffening may be required, but more often the problem can be solved by reducing the vibrations of the offending machine by balancing, or as otherwise required. This improves the machine's mechanical condition and eliminates the environmental vibration problem.

Machinery preventive maintenance and a knowledge of each machine's condition is of vital concern aboard ship for several reasons: 1) ships are often at locations where it is difficult to obtain spare parts and repair work, 2) shipboard machinery is forced to function in a hostile environment of corrosion and vibration, and 3) a machinery failure at sea is always a serious and sometimes dangerous occurance. Against this, vibration measurement provides one more preventive maintenance tool for the shipboard engineer.

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