REALTIME
Update

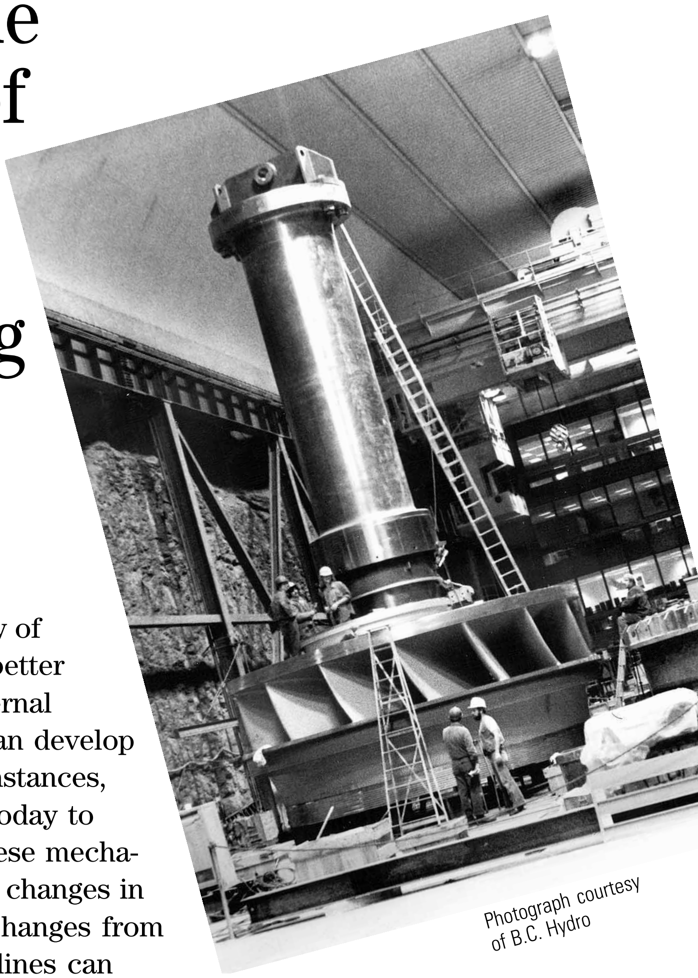
Hewlett-Packard Summer 1994

A Newsletter for Noise, Vibration and Electromechanical Test Professionals

Also in this issue:

Proving the Viability of Machine Condition Monitoring

by Doug Franklin
Powertech Labs Inc.



Photograph courtesy of B.C. Hydro

Improving the reliability, efficiency and availability of machinery starts with a better understanding of the internal fault mechanisms that can develop in machinery. In many instances, technology is available today to monitor and measure these mechanisms on-line and detect changes in machine performance. Changes from normal, acceptable baselines can provide the information needed to detect and diagnose progressing fault conditions. Used effectively, this new information can lead to significant savings of both time and money. The benefits include better predictions of failure timelines, more effective maintenance strategies, and reduced risks of costly immediate and secondary damage which could result if problems progress undetected.

Machine condition monitoring (MCM) is not as common in hydroelectric generating units as it should be. The reasons for this include the low speed at which hydro electric generators rotate (much lower than most other machinery monitoring applications); the large variety of components and subsystems that must function together; the variety of configurations, sizes and manufacturers; a tendency to stick with tried-and-true traditional maintenance procedures, and limited awareness in the industry about what is possible.

However, MCM promises significant benefits in both maintenance and operations. A development program underway at B.C. Hydro (British Columbia, Canada) is designed to realize four major goals:

1. Increase capacity of existing equipment through planned overloading. With a better understanding of the behavior and limits of each generator, we'll be able to add to peak capacity by operating safely above rated levels without significantly affecting equipment life or maintenance costs.

2. Reduce maintenance costs. The fundamental benefit here is shifting from time-based (calendar-driven) maintenance to condition-based maintenance. By scheduling only the maintenance that is really needed — and only when it is really needed — we avoid unnecessary maintenance and plant shutdowns and avoid creating new problems in the teardown and rebuild processes. Further, the potential for early detection decreases the chance of catastrophic failures and minimizes the costs of all the failure-related damage.

3. Extend the life of our equipment. An effective MCM system would enable the informed deferral of equipment rebuilds, as compared to the present system of calendar-driven equipment rebuilds.

4. Capture energy lost in present spills. A "spill" is water that is lost when there is excess water behind a dam or when generating units are out of service for maintenance or repairs. MCM has the potential to capture some of the energy potential from spills because both planned overloading and reduced maintenance and repair downtime will let us run more water through the generating units.

In addition to these key benefits, MCM promises a number of other positive changes, from improved safety to reduced spare parts inventories..

The MCM Initiative

As a first step in designing an MCM solution for hydro plants, we teamed up with technical experts from B.C. Hydro and other utilities to analyze the frequency and severity of failures in a wide range of hydroelectric subsystems. We ranked the failures by overall impact based upon reported hours of forced outage and on the judgment of operating personnel.

This solid historical database led to a proposal for a comprehensive, modular MCM system. Figure 1 shows the eight modules envisioned for a complete configuration. We opted for the modular concept to give individual plants and utilities the freedom to implement as many pieces as unique plant requirements dictate and budgets allow. In addition, a modular approach is easier to expand; it allows individual generating stations to match their processing capabilities with their monitoring needs, and it allows a more efficient approach to project development.

Due to the scope of the initiative, which has been defined as a five-year, $13 million dollar program, B.C. Hydro decided a consortium composed of interested utilities and manufacturers would be the ideal way to share the cost and risk of development.

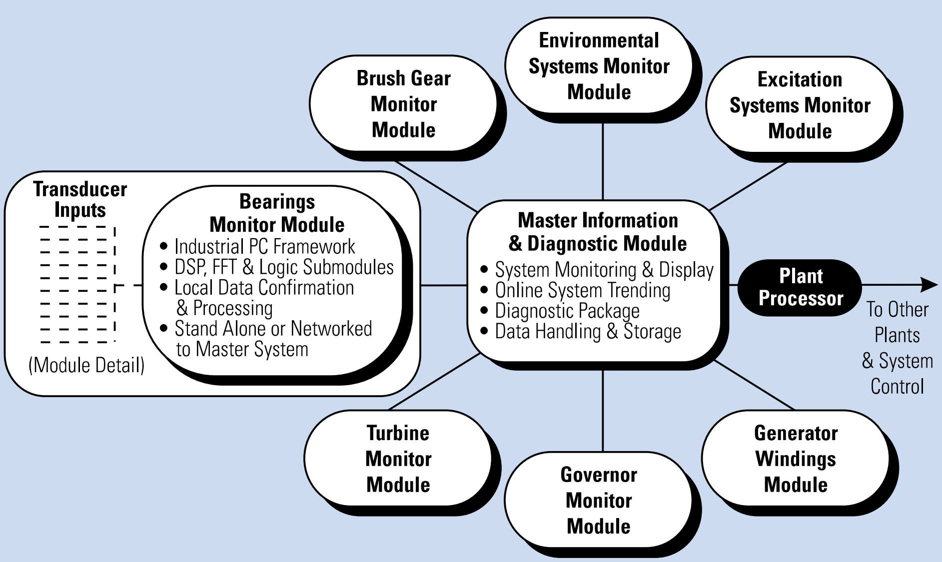


Figure 1
Modular MCM System
The MCM system will be designed in a series of modules in order to remain cost effective and flexible.

Figure 2
Making the Test with Existing Technology
Hewlett-Packard demonstrated how existing products can be used to do on-line analysis of these machine faults

Proving the Concept

In order to attract other utilities to join the consortium, B.C. Hydro designed a series of tests to prove the viability of hydro MCM. We focused on the bearing condition monitor module and performed the tests on a 150 MW hydro turbine scheduled to be replaced. The tests involved instrumenting the turbine and the generator to obtain key dynamic information under both normal and simulated fault operating conditions. The goal was to demonstrate the feasibility of detecting particular fault conditions on-line; table 1 summarizes the five tests. Hewlett-Packard, one of two equipment manufacturers who participated, used the HP 3567A PC spectrum/network analyzer to measure and display the test data on-line (see figure 2). The tests took place over three days in March of 1993 at the Kootenay Canal Generating Station in British Columbia.

For all the tests except bearing clearance, we sequentially introduced faults over the three-day test period, while conducting both on-line measurements and data collection for post-test analysis. The following sections give you a closer look at two of these tests, bearing clearance and mechanical imbalance.

The Bearing Clearance Test

The purpose of this test was to investigate methods of measuring clearance on both generator and turbine bearings. Using two proximity probes, we measured the shaft position during the first 10 revolutions during start-up. Figure 3 shows the results of using the HP 3567A to simulate strip chart recorder output. You can see how the generator shakes significantly as it runs up, then settles down once it approaches normal operating speed.

Figure 4 shows how the two probes are positioned to produce an orbit diagram of the shaft, and figure 5 shows a B.C. Hydro staffer installing them on the turbine.

On the resulting orbit plot (see figure 6), the distance from the center indicates magnitude of movement, and the radial direction indicates phase relative to a known reference

Table 1 - Bearing Module Concept Test

Test Name	Description	Test Scenario & Expected Results
Bearing Clearance	To show the methods of measuring bearing clearance in the turbine and generator bearings, and to study associated phenomena.	The random orbit of the shaft in the first 10 cycles of start-up shows the maximum radial excursion of the shaft within each bearing. The Unit will be warmed under load, while on-line bearing clearance is monitored using different eddy probes. The start-up method will be repeated after warm-up to confirm a change in warm clearance.
Mechanical & Magnetic Imbalance	To show the methods of measuring the effects of mechanical and magnetic imbalance of a hydro turbine generator, and to study associated phenomena.	A known weight will be added to the generator rotor to provide an imbalance force. The Unit will be brought up to speed and the electric field turned on (flashed), showing the effects of magnetic imbalance. The unit is then tripped and allowed to run down while its mechanical balance is monitored. This is then repeated with warm bearings.
Shaft Alignment	Seal rubs are common cause of thermally induced shaft alignment problems on hydro units. We plan to simulate the problem and show methods of monitoring for it.	An artificial seal rub is installed on the turbine shaft above the turbine bearing. Thermal expansion on the side of the rub should result in bowing of the turbine shaft. Once a sizeable change has occurred, the artificial rub will be removed and the unit stopped. The shaft will be allowed to cool and the unit will then be restarted to show that the shaft was returned to normal.
Shear Pin Failure	A shear pin failure typically results from a combination of static and dynamic radial loads. In the past, turbine bearings have been wiped because of an inability to detect the problem in time.	A shear pin is pulled from one of the wicket gate linkages. The unit is started and loaded up. As the unit comes up through the rough load zone, two characteristics should become evident. A high Nx vibration resulting from the blade passing frequency of the runner should begin to show up, and the turbine shaft should exhibit an offset toward the disabled wicket gate.
Hydraulic Imbalance	Hydraulic imbalance can result from a number of potential failures in the turbine. This imbalance can occur where part of the rotating hydraulic passage is interrupted.	A flow restriction will be installed between two blades of the turbine runner. The Unit will be loaded to show the effects of the resulting hydraulic imbalance. The imbalance occurs because of the increased impedance to flow and a resulting reduced power conversion in that section of the runner. The imbalance should be characterized by a sharp increase in 1x amplitude as load approaches 100%.

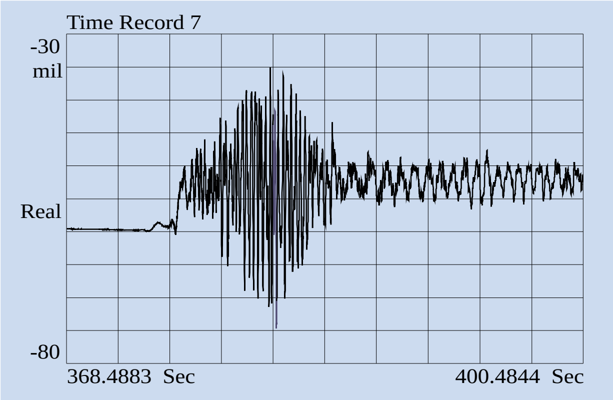


Figure 3
Vibration During Start-Up
This time view from the HP 3567A shows the generator shaft bumping into the bearings during the first revolutions of a start-up.

point. The orbits have a distinctly elliptical shape, and an elliptical fit of the cold-start data verified this. Information such as this is extremely helpful when we try to establish a normalizing curve for each generator, which will allow us to track clearances over time. Demonstrating that we could measure bearing clearance using vibration analysis was a key step in proving our MCM concept.

We also compared bearing clearance after the generator had warmed up to normal operating temperatures. The comparison showed substantially larger vibrations at lower temperatures (such as at a cold start). This information could be used to establish a lookup table for normalizing the bearing clearance to a standard temperature. Having long-term data can assist in identifying when to do maintenance procedures such as clearance adjustments.

The Mechanical Imbalance Test

The effects of mechanical imbalance on shaft runout can be isolated from other sources of imbalance by observing the shaft runout or vibration during run down, after the field breaker has been opened and the unit is coasting to a stop. For this analysis, the data are valid from the time the field breaker is opened until the brakes are applied to the rotor (usually at about 20% normal operating speed).

Mechanical imbalance can be caused by an unequal distribution of weight around the rotating portions of the system or by the rotor or turbine being radially misaligned with the shaft at the coupling above or below the last bearing. An unequal distribution of the mass about the center of the shaft will cause a displacement or vibration of the shaft at a frequency of 1X the rotating speed. The amplitude of the runout caused by mechanical imbalance is proportional to the square of the rotating speed.

A simple waterfall diagram in the frequency domain shows the amplitude of the runout or vibration at the 1X rotating speed as the speed of rotation decreases. A diminishing amplitude at this frequency indicates a runout or vibration caused by mechanical imbalance. The dramatic drop-off in amplitude shown in figure 7 is a clear indication of a mechanical imbalance.

(The plot in figure 7 shows the vibration in the order domain. In other words, the frequency axis was adjusted for each trace so that the 1X rotating speed stays in the same location along the X-axis. If we had viewed this result with a regular

Figure 5
Installing the Probes
A B.C. Hydro employee installs the displacement probes on the turbine guide bearing.

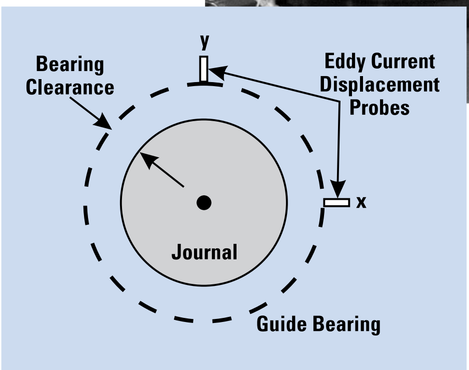


Figure 4
Instrumenting for the Orbit Diagram
Probes were positioned 90 degrees apart to create an orbit for the guide bearing clearance measurement.

frequency domain plot, the 1X rotating speed frequency would've shifted to the left on each trace as the rotating speed decreased.)

Figure 8 provides further confirmation of mechanical imbalance, plus it shows some evidence of misalignment as well. This polar plot uses amplitude and phase to chart the shaft vibration [the 1X rotating speed component] during coast down. If the runout is caused solely by imbalance, a line drawn through these points will point to the center of the plot. If there is some misalignment in the system, however, the amplitude and phase will trend towards the runout caused by the misalignment alone as the speed decreases.

Polar plots of shaft vibration measurements provide another key diagnostic benefit because they also help us identify the radial location of the heavy spot and evaluate changes after we've added balance weights.

Next Steps

The overall results from all the tests were very encouraging. We used existing products and technologies to demonstrate that MCM is in fact a valid and valuable approach for hydro plants to increase uptime and decrease costs. We've moved ahead in forming a consortium of major utilities and important industry associations to fully design and implement a comprehensive solution that should benefit hydro plants worldwide. ■

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For more information on the HP 3567A used in these tests, check 1 on the Reply Card.

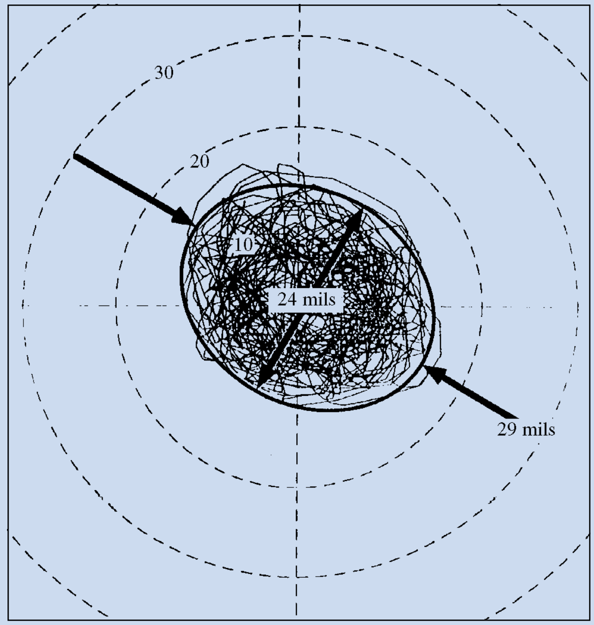

Figure 6
Discovering an Out-of-Round Condition
An elliptical fit of the cold bearing clearance measurement taken by the HP 3567A reveals an out-of-round condition in the setting of the generator guide bearing.

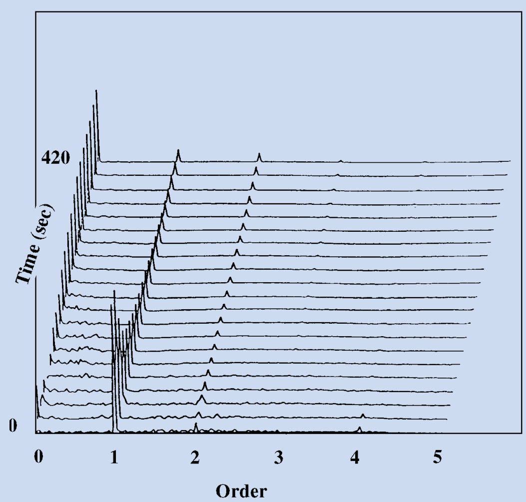

Figure 7
Frequency Spectra During Coast-Down
The rapid decrease of the 1st order frequency component during the early stages of the coast-down indicates an existing mechanical imbalance.

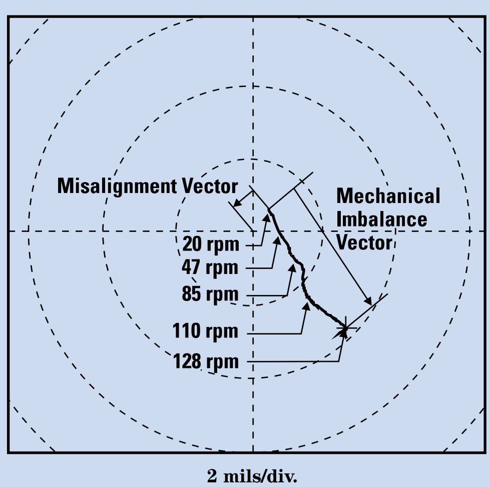

Figure 8
Polar Plot During Coast-Down
A polar plot during the coast-down shows the effects of the imbalance and of some misalignment as well.

Realtime Basics:

Minimizing Noise in Low Frequency Measurements

By Richard M. Barrett, Jr.
Wilcoxon Research, Inc.

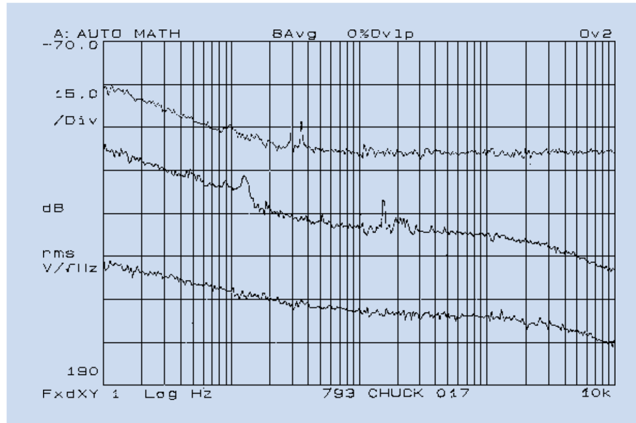

Figure 2
Accelerometer Noise Levels
This plot compares the noise level of accelerometers with sensivities of 100 mV/g, 500 mV/g, and 10 V/g.

Low frequency vibration measurements are important for monitoring structural movement, measuring mechanical isolation of precision equipment and conducting condition monitoring of slow speed machinery. These applications, which range from civil engineering to precision optics to industrial process monitoring, generally require measurements from 0.1 to 10 Hz, and at these low frequencies, noise becomes a significant factor in the measurement process. This article identifies the common sources of noise and suggests steps you can take to boost signal-to-noise ratios in your measurements.

Low frequency applications push the performance limits of the measurement system because motion below 10 Hz produces such low levels of acceleration (see figure 1). Accurate measurement of these low acceleration amplitudes requires special sensor designs and low noise electronics to maximize the signal-to-noise ratio. (Note that even though displacement increases at low frequencies, you can't always use displacement transducers; LVDTs are not as sensitive as accelerometers, and proximity probes can't be used in seismic applications where you don't have a stable reference surface.)

Signal noise is the primary consideration when performing low frequency measurements. This noise can obscure spectral data, alter amplitude information and render measurements useless. When integrated (to convert acceleration to velocity, for instance), the noise is amplified to produce the familiar "ski slope" response that often interferes with low frequency measurements.

Advanced sensor designs and low-noise electronics are an important first step, but to get the full benefit, users need to understand the causes and cures of noise. Three sources contribute to signal noise: electrical noise generated by the sensor, electrical noise generated by the analyzer, and various electrical and mechanical noises present in the measurement environment.

Sensor Noise

Electronic noise from accelerometers is directly related to the charge output of the piezoelectric sensing element and to the design of the amplifier. All amplifiers contain a variety of electronic noise sources, including resistors, diodes, and transistors. At low frequencies, amplifier noise is inversely proportional to frequency and resembles Schottky (1/f) noise. This noise determines the low frequency measurement limit (see figure 2). The low frequency noise of an accelerometer is proportional to the gain (amplification) of the circuit and inversely proportional to the charge sensitivity of the piezoelectric sensing element.

Increasing the charge output of the sensing element (output before the amplification) reduces the need for gain and increases signal-to-noise. The charge sensitivity can be increased only by adding more seismic mass or by using a more active sensing material. In low frequency applications, piezoceramic lead zirconate-titanate (PZT) should be used to maximize the charge output of the sensing assembly.

Analyzer Noise

The analyzer's contribution to the noise problem is a function of electronic design (including component noise as described earlier), the voltage input from the sensor, dynamic range performance, and instrument setup.

Piezovelocity transducers (PVT) and higher sensitivity (>500 mV/g) accelerometers significantly improve low frequency response by presenting

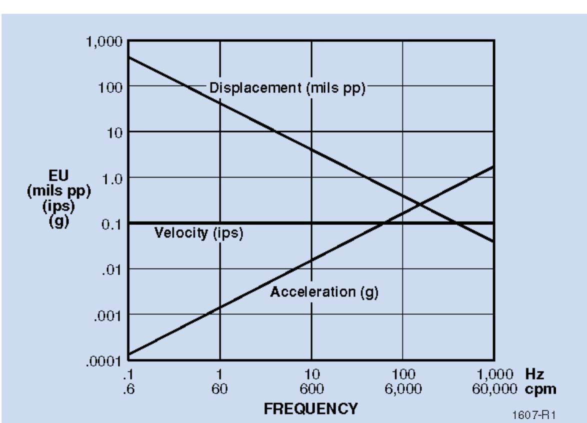

Figure 1
Relationship Between Acceleration and Displacement
This graph shows the relationship between acceleration and displacement at constant velocity as frequency approaches 0 Hz.

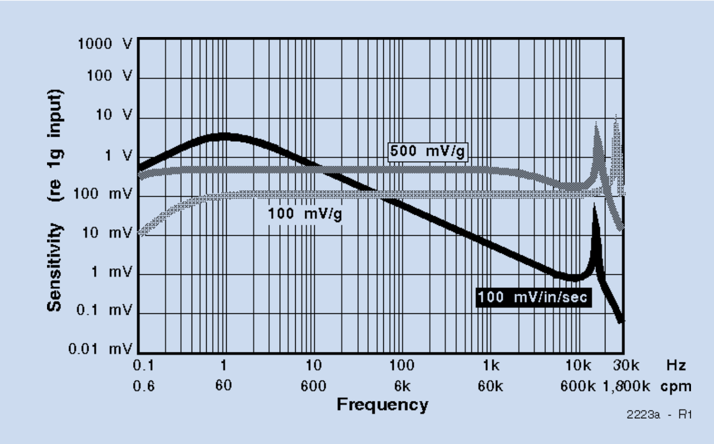

Figure 3
Comparison of Sensor Outputs
This graph compares the sensor output levels from an accelerometer with 100 mV/g sensitivity, an accelerometer with 500 mV/g sensitivity and a piezovelocity transducer with 100 mV/ips sensitivity.

a higher voltage to the analyzer's input. This higher input voltage improves signal-to-noise by reducing the effects of the analyzer's noise contribution. PVTs provide additional improvement in dynamic range by attenuating high frequency signals before the instrument input. Figure 3 provides a graphical comparison of equivalent voltage outputs for three different sensors excited by a constant 0.3 ips vibration.

Dynamic range considerations require matching the sensor output with instrument processing capabilities. You need to be sure that the analyzer has sufficient dynamic range and that the input range selection is optimized for the signal level you're measuring.

The setup factors to consider include integration, resolution and averaging. Analog integration within the analyzer usually increases low frequency noise and lowers signal-to-noise. The integration circuit converts acceleration to velocity by amplifying low frequency signals and attenuating high frequencies. Low frequency gain also amplifies and accentuates low frequency noise of both the accelerometer and instrument. Double integration from acceleration to displacement requires even more amplification and so introduces even more noise.

Finer instrument resolution improves signal fidelity by reducing spectral amplifier noise, which is defined in terms of volts (or equivalent units) per square root of a 1 Hz frequency band. If resolution is increased so that the linewidth is less than 1 Hz, noise will decrease.

For example, consider a sensor with a specified spectral noise of 1.0 μg/√Hz at 2 Hz, with an analyzer set up for 1600 lines of resolution over a 0 to 10 Hz bandwidth. The linewidth of the measurement (neglecting windowing) is:

(10 Hz - 0 Hz)/1600 lines = 0.00625 Hz (0.375 cpm)

The spectral noise improvement of the sensor is:

$(1.0\ \mu g/\sqrt{Hz})(\sqrt{0.00625\ Hz}) = 0.079\ \mu g$.

The trade-off with finer resolution is increased data collection time, as you can see from the example in table 1.

Increased averaging lowers noise by smoothing out random noise signals. Over time, the random noise contribution is reduced and periodic signals strengthened. As with finer resolution, however, the downside of increased averaging is increased data collection time.

Combined System Noise

You can determine the minimum electronic signal-to-noise ratio in your system by evaluating the spectral noise of both the sensor and the analyzer at the lowest frequency point of interest.

Given that the power spectral noise is measured over a 1 Hz band of frequency resolution and evaluated at the minimum frequency of interest (f_{min}), the signal-to-noise ratio (S/N) is:

$$S/N\ (min) = \frac{S}{\sqrt{[(N_S)^2 + (N_A/S_V)^2]\Delta f}}$$

Where:

S =	Lowest signal amplitude required in EU (Engineering Units — g, m/s², ips, mm, etc.)
N_S =	Spectral noise of the sensor in EU/√Hz at f_{min}
N_A =	Spectral noise of the analyzer in V/√Hz at f_{min}
SV =	Sensitivity of the sensor in V/EU

Environmental Noise

Environmental noise interference can be caused by a variety of external sources, both electrical and mechanical in nature. Potential sources include the machine under test, nearby machinery, and the plant structure and ambient environment. Some noises interfere with the measurement directly; others interfere indirectly.

Indirect Sources: High Frequency Vibration Noise

Indirect noise originates at high frequency and interacts with the measurement system to produce low frequency interference. Several common examples of indirect mechanical noise include pump cavitation, steam leaks and compressed air leaks. These sources produce high amplitude, high frequency vibration noise that can overload the sensor amplifier to produce low frequency distortion. This type of interference is a form of intermodulation distortion commonly referred to as "washover" distortion; it usually appears as an exaggerated ski slope.

Low frequency accelerometers are generally more susceptible to high frequency vibration noise and washover distortion than general purpose accelerometers. This is due to their lower resonance frequency and higher sensitivity. Piezovelocity transducers, where applicable, eliminate washover distortion by attenuating high frequency vibration noise.

Table 1
Resolution Effects Using a 0-10 Hz Bandwidth
Increased resolution decreases spectral noise; the trade-off is longer measurement times.

Lines of resolution	400	800	1600	3200
Electronic spectral noise of a low-frequency sensor (1 mg/√Hz)	0.16	0.11	0.079	0.056
Measurement time	56.6 sec	80.0 sec	113.1 sec	160.0 sec

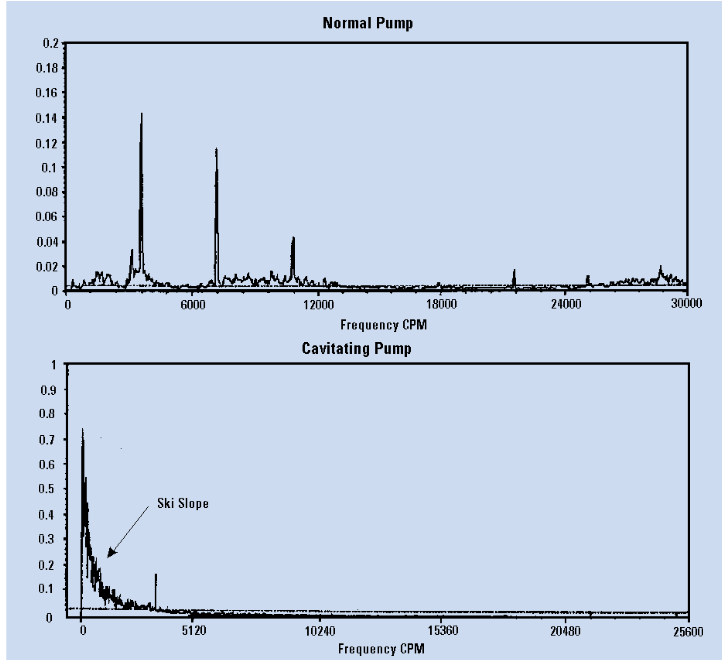

Figure 4
Sensor Overload from Pump Cavitation
High frequency noise from pump cavitation resulted in overloading the sensor at low frequencies.

Pump cavitation is a good example of this type of noise. The collapse of the cavitation bubbles creates high frequency noise that can cause problems at low frequencies. The spectra in figure 4 show measurements from identical pumps using a 500 mV/g low frequency accelerometer. The first plot displays expected readings from the normal pump; the second shows the ski slope due to pump cavitation and the resulting washover distortion.

Indirect Sources: Electrical Noise

Indirect electrical events from electromagnetic radiation and electrostatic discharge can induce noise directly into your measurement system. When mounting or cabling sensors near radio equipment, ignition wires or machinery with high voltage corona discharge, low frequency interference becomes a concern. Unless properly protected, the sensor amplifier can rectify very high frequency signals to produce low frequency distortion products. Make sure the sensors you're using have overload reduction circuitry to prevent the sensor amplifier from operating as a radio detector.

Direct Noise Sources

Direct environmental noise is caused by mechanical events within the measurement frequency range. Primary sources include thermal transient pick up, mechanical strains and interference from unwanted low frequency vibration sources.

Thermal transients and mechanical strain at the mounting surface can cause low frequency expansion and contraction of the sensor housing. This mechanical strain signal is then transmitted to the piezoelectric sensing element. Low frequency sensors must be designed for low strain sensitivity in order to prevent these thermal transient disturbances.

Direct vibration noise from the rumble of nearby machinery and equipment can limit low frequency measurements in many plant environments. Low frequency energy propagates easily through most structures; at very low frequencies, even passing vehicular noise can produce measurement interference.

Figure 5 shows the influence of environmental noise on low frequency measurements. Vibration measurements were made on an agitator gear reducer at a soap factory. The reducer vibration was then simulated in a laboratory on a low frequency shaker. Comparison of the laboratory and plant spectrums clearly show increased noise due to the plant environment.

Keeping Your Measurements Clean

You can see that low frequency measurements require careful attention to the selection and use of vibration measurement equipment. The low acceleration amplitudes are beyond the limits of general instrumentation and measurement techniques, so concerted efforts to improve signal-to-noise ratios are key to getting reliable answers. ■

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For a copy of the full paper from which this article was abstracted, check 4 on the Reply Card.

This article also appeared in full in *P/PM Technology*, a bimonthly publication covering all aspects of predictive maintenance. You may receive a complimentary 6-month subscription by calling 702-267-3970 and mentioning this newsletter offer.

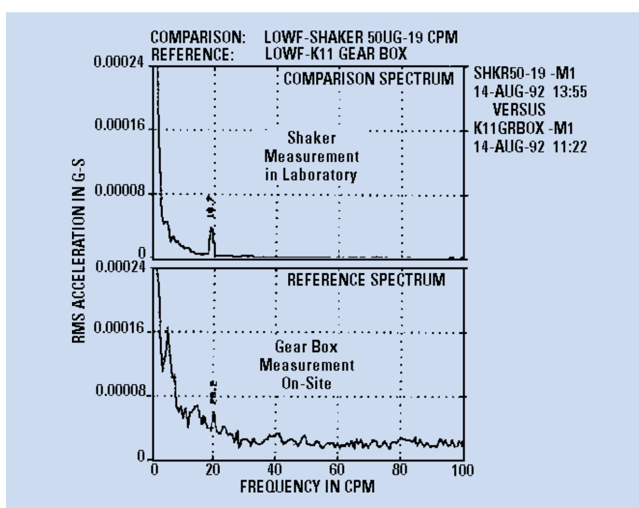

Figure 5
Environmental Noise
A comparison test simulated in the lab shows the effects of environmental noise on this gear-box measurement.

Realtime Answers

Question:
What's the best way to transfer data from an HP 3560A to a PC?

Answer:
The HP 3560A can store the equivalent of 1 floppy disk in its nonvolatile memory. Here are the steps to take for a trouble-free transfer.

Cabling: The HP 3560A uses a 9-pin male RS-232 serial connection. Connecting this to a 9 pin male serial interface requires an HP 24542U cable. Connecting the analyzer to a 25-pin female serial interface requires an HP 24542G cable. Be careful when using cables labeled as equivalent, not all cables contain the full number of lines required by the HP 3560A. Here are the cable pinouts in case you need to refer to individual lines:

HP 24542U
(9 pin female - 9 pin female)

HP 3560A	PC
DCD 1	7 RTS
RXD 2	3 TXD
TXD 3	2 RXD
DTR 4	8 CTS, 6 DSR
GND 5	5 GND
DSR 6, CTS 8	4 DTR
RTS 7	1 DCD
RI 9	9 RI

HP 24542G
(9 pin female - 25 pin male)

HP 3560A	PC
DCD 1	4 RTS
RXD 2	2 TXD
TXD 3	3 RXD
DTR 4	5 CTS, 6 DSR
GND 5	7 GND
DSR 6, CTS 8	20 DTR
RTS 7	8 DCD
RI 9	22 RI

Setting up the PC: You need to know which serial port is being used for data transfer. We recommend connecting the HP 3560A to either Port 1 or Port 2, since Ports 3 and 4 are not supported by some versions of the SDF Utilities. (Note that when connecting cables it is always good practice to first touch the metal outer shells of the cable and the port together. This will dissipate any static charge that may have accumulated on the analyzer and prevent static damage to the HP 3560A or the PC and its peripherals.

The transfer requires the HP SDF Utilities, which are included with your HP 3560A documentation. You need to run the utilities from the DOS prompt. (If you're using a DOS prompt in Windows, the DOS screen should be maximized by making that window active and pressing Alt-Enter simultaneously.)

The SDF Utilities will need to be available for batch files to access them. If you have copied the utilities to your hard disk, make sure they are available in the directory path. (See your DOS documentation for more information on paths.) If the current directory contains the SDF Utilities, you'll end up with data files and the utilities mixed in one directory. To keep them separate, put the SDF Utilities in a separate directory and add that directory to your DOS path definition.

Setting up the HP 3560A: The data transfer conditions for the analyzer are displayed when you press the [Utility] key on the front panel. Move the cursor to the RS-232 menu, which controls the settings for data transfer. Select Parity: NONE, Bits: EIGHT, and Handshake: CTS, and any Baud rate you want (the most common is 19200). During the data transfer procedure you will need to specify the Baud rate entered in this field.

The next step is to determine which files you want to transfer. The selection is made by pressing the [Save/Recall] key on the front panel. The HP 3560A offers two choices, transferring the current catalog entry or the entire contents of memory.

To transfer one file: Move the cursor to the OPERATION menu. Select the CATALOG command and press [Enter]. The HP 3560A will display all files in memory. Move the cursor to the desired file and press [Enter]. The HP 3560A will return to the Save/Recall menu. Move the cursor to highlight the XFER ONE operation. Now proceed to the step "Transferring the Data"

REALTIME Update

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To transfer all the files: Move the cursor to the OPERATION menu. Select the XFER ALL command. Now proceed to the step “Transferring the Data”

Transferring the Data: At this point you should have the PC in DOS and the HP 3560A ready to transfer a file. For the process to work correctly it must be initiated from the PC, and the easiest way to make the transfer is with the 60_SDF.BAT batch file.

To execute this file, you must specify both the port number and baud rate as follows:

60_SDF <port> <baud>

For example: To specify Port 2 with 19200 Baud: 60_SDF 2 19200 [Enter]

Once the command is initiated the PC will echo back the following message:

Using COM1...
Using 19200 baud...
Waiting ...

While the PC is waiting, execute the transfer operation on the HP 3560A. Press [ENTER] with the XFER ONE or XFER ALL operation highlighted.

The PC will display a small rotating ‘\’ marker while the transfer is in progress, and the HP 3560A will display “Transfer in progress” at the bottom of its screen.

Once the transfer is complete, the 60_SDF batch file will process all transferred data into SDF files. The transferred files will be named with the catalog number and a .DAT suffix, such as 111.DAT. If you transferred all of the files, the DOS DIR *.DAT command will list all the names for you. Be careful if you are adding files to an existing set of data as the 60_SDF utility will overwrite previous files of the same name.

Troubleshooting hints:

Symptom: PC does not transfer data, displays “Waiting ...” message.
Likely Cause: Wrong cable or serial port number.

Symptom: PC transfers data but then displays “corrupt file” or “overflow” message.
Likely Cause: Baud rate on PC and HP 3560A do not match. Extended memory programs interfering with transfer.

Symptom: Transfer All command does not transfer all files.
Likely Cause: Firmware and SDF incompatible. Contact HP for new HP 3560A firmware.

Symptom: “Can not find ...” or “Can not open” messages appear on PC.
Likely Cause: SDF Utilities are not in the current path definition.