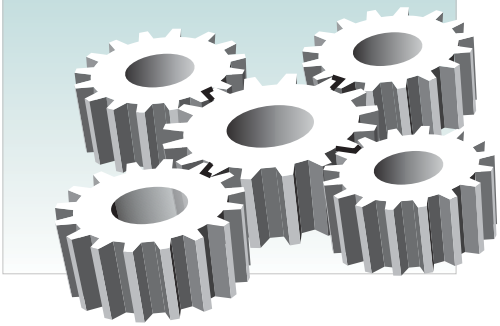
THE BETA BULLETIN

BETA

Machinery Analysis



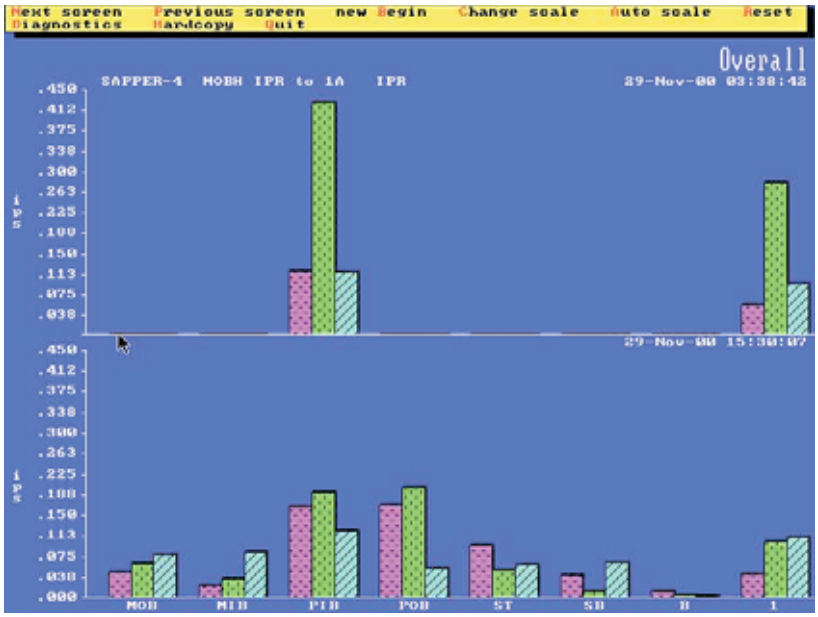
INSIDE VOLUME 9 #4

Centrifugal Pump Vibration

In a recent assignment, Steve Lawn, senior engineer with BMA, encountered some very complex, unusual and just generally interesting vibration phenomena on some vertical pumps.

Excessive vibration when not running

There are a number of pumps of differing sizes interconnected through header and valve arrangements. Variations in plant flow rates are accomplished by operating different pump combinations. Steve found that in some cases, pumps had very high vibration *when not running!* The figure below illustrates one of these cases.



Here vibration data is presented in profile format (See Vol 9, No 1 for explanation of vibration profiles).The upper profile shows the overall vibration at various test points in pump 4 when shut down, but with pumps 1 and 2 running. Note that the amplitudes shown are in units of ips *rms.*

The lower profile is the comparable results when no. 4 pump is running with all others down.

The highest overall vibration on no. 4 when running is considerably less than half the level when not running!! (Possible recommendation: never shut it down!!)

The explanation for this seemingly bizarre behavior lies in the pulsation environment set up in the interconnecting piping systems. Although we were not able to measure the pressures in the fluid, we believe that there is an acoustical natural frequency at about 29 Hz and that the associated mode is strongly excited by some operating configurations.

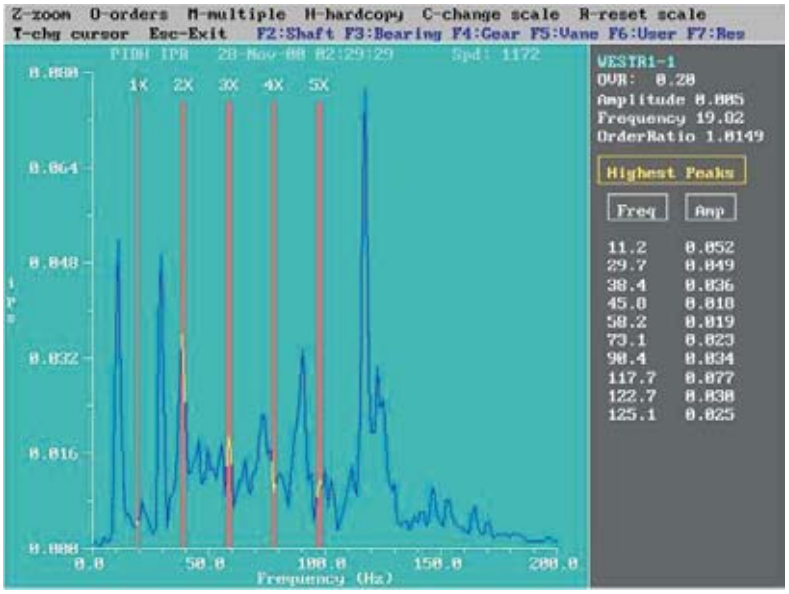
Strongest excitation sources in systems like this tend to be vane passing frequency and its harmonics. When these frequencies are close to acoustical natural frequencies, the energy can be amplified to very large pulsation levels.

Should the owner care about these high “not running” vibrations? For sure. This situation will shorten bearing life, open up bushings and seals, tend to loosen clamps, brackets and fixtures.

Non-harmonic excitation of MNF

Pumps 1 and 2 at one pump station had overall vibration levels somewhat exceeding acceptance levels. These pumps run at 1200 RPM nominally, and have three vane impellers. A sample spectrum from no. 1 pump inboard is shown below. The largest component is at twice vane passing frequency (i.e. 6X, not marked), but with substantial contributions at other frequencies. There is also a significant component at 2X. Much of the content is non-harmonic.

An unusual aspect of the pattern is that the energy content at rotational frequency, 1X, is very low compared to non-harmonic energy.



Part of the assignment was to conduct bump tests to determine mechanical natural frequencies of the main modes of vibration. For no. 1 it was determined that one of the so-called “reed frequencies” was 12 Hz.

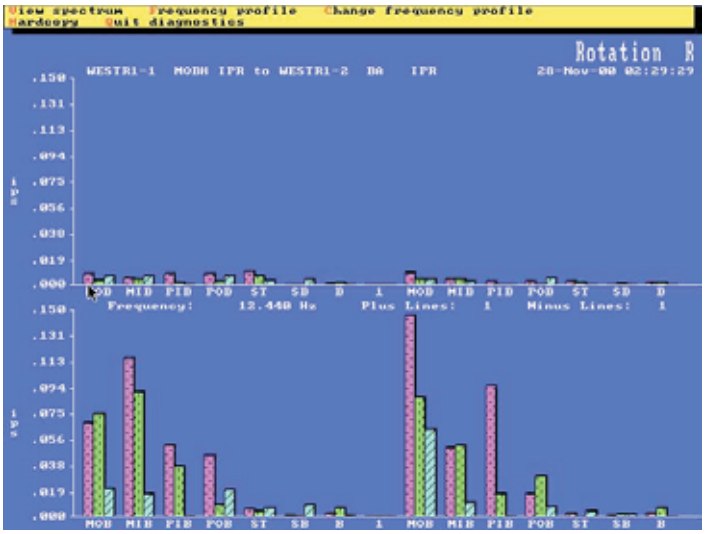
The reed frequency modes are the basic cantilever beam modes, in two perpendicular directions, that one would expect on vertical pumps.

It is no doubt excitation of this mode that causes the peak reported at 11.2 Hz.

This behavior is repeated across all the test points of No 1 and No 2 pumps, as illustrated by the profile below. The upper profile is the 1X pattern while the lower one is a band around 12.4 Hz.

An obvious question is “Why is there so much non-harmonic excitation?”. The reason is almost certainly the pump hydraulics. The pump is believed to be running well off its best efficiency point and probably at inadequate suction head.

One curiosity: the peak at 29.7 Hz is almost exactly half of vane passing frequency. And when we profile the results, the pattern of this 29.7 Hz vibration is the same as the vane passing pattern. Can anyone explain that?



Additional thoughts

The owner of these pumps has been planning to implement a vibration-based predictive maintenance (PdM) program. As is usual with PdM programs, the objective is to reduce maintenance costs and improve equipment availability.

The examples cited above, along with numerous other findings in Steve's work, illustrate the importance of conducting a diagnostic baseline analysis prior to starting the program. Design problems that impact maintenance cost and availability should be corrected first. If correction is not possible, the study provides the background information needed to interpret PdM program findings.

New Website

Three years ago we set up our first website, www.betamachinery.com, and it has served us well. Now, in an attempt to keep pace with the fast-changing areas of communications and information we feel we need our website to be bigger and better. BETA is working on a new site and plans to have it up and running by January 1, 2001. Please visit this page, same address. Keep in mind it is still a work in progress. We would love to have your comments and suggestions on what you see there.

Technical Tips

Dynamic Range

For an instrument dynamic range of 80 db, what is the lowest signal that I can measure for a given input range?

80 db = 20 log (K) where K = sensitivity resolution
log (K) = 80/20 = 4
K = 10,000

For an input range of 2 volts, the lowest signal increment theoretically measurable. would be 2/10000 or .0002 volts or 0.2 mv.

For an instrument with a 16 bit A/D Converter, what is the theoretical Dynamic Range?

K = 2 to 16th power = 65536 where K = sensitivity resolution
Dynamic Range = 20 log (K) = 20 log 65536 = 96.33 db

Actual Dynamic Range is less than the theoretical range. It generally decreases at higher frequencies and also at lower input voltages where noise becomes a factor.

Falling Transducer Sensitivity

Transducer sensitivity or output often falls off with time. A drop of 15% depending on usage, wear and tear is not uncommon.
Transducers should be checked regularly by employing a calibrated source or in a side by side comparison with a reference transducer of known sensitivity.
The inversion (2g's) test often recommended for accelerometers is tricky to measure reliably.

Seminars

Pulsation/Vibration and Turbine Seminars are scheduled for our Calgary office in February and March of the New Year. Please contact our office if you are interested in these training opportunities.

Best wishes to all our valued friends and customers for a Happy Holiday Season and a Prosperous New Year.

Rather than send greeting cards at this time of year, our practice at Beta is to make a contribution to the food bank as a symbolic thank you.
We trust you approve.

News & Notes

The Gas Machinery Conference, held every year in the first week of October, is regarded by many at Beta as the premiere technical conference and trade show event of its type. This year it was held in Colorado Springs from October 2-4. In addition to participating in the trade show, Beta personnel made three technical presentations. If you would like copies of any of these presentations, summarized below, please contact us at info@betamachinery.com or phone or fax one of our offices.

Analyzing Engines and Compressors in Economic Terms - Bryan Long
This course shows the technical specialists how to develop and communicate results in economic or near-economic terms. Topics include: management ready information-need and description; calculation of economic consequences of performance degradation, communication of mechanical degradation in risk terms, and consideration of return on investment when making maintenance decisions. Numerous examples and illustrations are incorporated.

Balancing Risk and Design Efforts in New Compressor Installations - Frank Fifer
It is difficult to justify a significant investment in technical modeling for compressor installations that are very close duplicates of other previously successfully installed, are not in production critical service, and/or are small. In contrast, new unique designs, units in critical, unspared service and/or large and very costly installations generally deserve a great deal of extra design effort. How should the project engineer decide what level of technical analysis, if any, should be employed for a given set of circumstances? This presentation attempts to clarify this issue.

Case Studies Illustrating Improvements in Reciprocating Compressor Design - John Harvey
This paper uses two real cases to illustrate improvements in compressor system design. One case involved a compressor with three nozzle suction and discharge bottles. Several nozzle failures were experienced. The second case involved an existing compressor with high vibrations measured on suction and discharge vessels and piping.

One development evident at the Gas Machinery Conference and other events attended by Beta personnel this year is the acceptance of remote monitoring and diagnostics (RM&D) and remote control. Ten years ago or more Beta offered a RM&D service that offered substantial cost savings compared to conventional on-site services. There was limited interest. So why is the level of acceptance of this sort of thing greater in 2000 than in 1990? Undoubtedly the Internet. It is now commonplace to send information of all kinds from the source to another site for evaluation and/or action.

There have been bold predictions that the Internet will change the way business is done. That seems unrealistic, at least in the near term. But it can help us improve our business processes. We have executed a number of RM&D assignments, with excellent results.

For those who have been around the oil and gas industry for a few years will remember the talk about the “gas bubble”; that is, the “temporary” excess of supply over demand. Apparently “temporary” meant about 10 years. With spot prices hitting astronomical heights, and average prices that make our home heating bills resemble mortgage payments, we can probably conclude that the gas bubble is gone. It is reported that some industrial facilities that use natural gas as a feedstock are shutting down production in order to resell natural gas that they bought at last year's prices, because they can make more money!

Now that the new millennium is actually about to start, is anyone worried about Y2.001K compliance?

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