THE BETA BULLETIN

BETA Machinery Analysis

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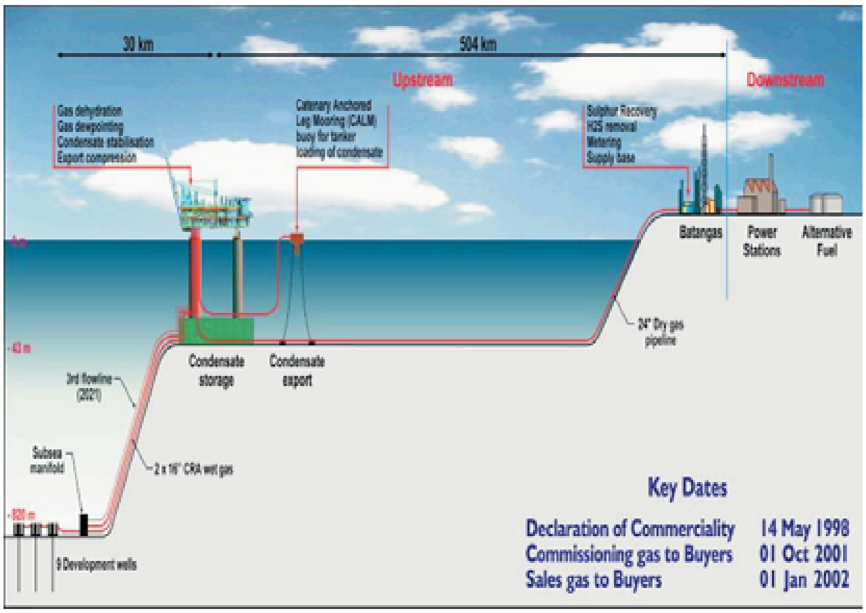
Beta Around the World: Malampaya, the Philippines



Malampaya Deep Water Gas to Power Project

Beta's engineers regularly visit the far corners of the world. Our expertise calls us to some very innovative and progressive project sites.

A recent assignment took us to the Malampaya project in the Philippines. Beta's engineers worked as part of a team tasked with ensuring the acceptability of machinery vibration levels. The focus during this field assessment was on the four reciprocating flash gas compressors. Two compressor trains, each train consisting of one 4-throw two-stage unit and one 6-throw two-stage unit are located on the platform. Vibration levels were evaluated on process piping, compressor frames and skids.

Along with the general unit assessment, field vibration checks were conducted on small bore piping attachments (2" diameter and under). Back in the Calgary office detailed finite element models were used to determine if the field measured vibration levels posed any increased risk to reliability.

The Malampaya Deep Water Gas to Power Project is a national flagship project of the Philippine Government, the owner of the resources. This project represents the largest and most significant industrial investment in the history of the Philippines. It will supply fuel slated to provide 2,700 megawatts of power to Luzon for a period of 20 years starting January 2002.

Developers of the upstream component are Shell Philippines Exploration B.V. together with joint venture partners Texaco Philippines and Philippine National Oil Company Exploration Corporation.

This project involved development and construction of the Malampaya Field facilities in 820 meter depth, a shallow water gas production platform some 30 kilometers from the wellheads, a 504 kilometer subsea pipeline, gas landing facilities in Batangas, and three power plants with a combined generating capacity of 2,700 megawatts.

Subsea wells and a manifold were installed at approximately 820 meters water depth and connected to the production platform via two 30 kilometer flowlines. On the platform, the dry gas is separated from the condensate which is temporarily stored in the caisson of the concrete gravity structure. The dry gas is then transported to a Catenary Anchored Leg Mooring (CALM) buoy and retrieved by tankers. The dry gas travels through a 504 kilometer, 24" carbon steel pipeline stretching from the platform to the gas treatment plant at Batangas where it undergoes final treatment prior to delivery to the power plants.

The task of extracting the natural gas at this water depth and transporting it to market a great distance away posed a great challenge in the field of deep water developments and required the use of the latest in gas technology and skills. The Malampaya field contains recoverable natural gas reserves of some 2.7 trillion cubic feet and 85 million barrels of condensate, enough fuel to supply 30% of the country's power for 20 years.

The principal technical challenge for Shell Philippines Exploration B.V. (SPEX) is to ensure continuous delivery of sales specification gas throughout the production chain while containing costs and maintaining stringent health, safety and environment standards.

INSIDE:

A GEARBOX VIBRATION - FACT OR FICTION

INTRODUCTION

This case history started with a phone call, requesting assistance to check the condition of a gearbox in a fertilizer plant. Overall vibrations collected with a periodic predictive maintenance program had shown a sudden and dramatic increase in vibration levels on the gearbox.

There was a heightened sense of awareness of gearboxes amongst the people in the plant because a similar gearbox had failed without warning a few months earlier. Furthermore, since it was late fall when we received the phone call, a plant shutdown to repair this gearbox would still be acceptable, though obviously undesirable. A full plant turnaround was planned for the spring. The concern was that an unplanned shutdown in the cold of winter would probably result in startup problems that might even prevent restarting the plant till spring.

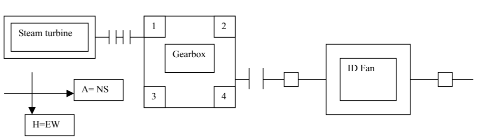
Despite the critical nature of the problem, there were limits placed on what we were able to do as part of our testing. We were not allowed to change the speed of the fan, let alone shut the fan down. There was no once-per-turn marker available on the system.

Our report (Reference 1) indicated that the gearbox would be acceptable till the next year, but that there were ways to improve the periodic vibration system being used in the plant.

SYSTEM DESCRIPTION

The layout of the system and a description of the gearbox is shown below. The steam turbine and ID fan are not described.

Layout of the machine



<u>Gearbox</u>

Horsepower 1113
Service Factor 2.44
SN 32671
Max. Input speed 5672 RPM
Max. Output speed 1167 RPM
Pinion Teeth 43
Bull gear Teeth 209
Max. gear-mesh frequency 4064Hz
Gear Ratio 4.860465

The following photographs show some views of the gearbox with testpoint locations indicated.

Picture 1: High speed side of the box, Test Points 3 and 4: (Turbine to the right)

Picture 2: View of Test Point 5 with accelerometer mounted with stud on nut glued to box in Horizontal orientation. The oil pump is on the turbine side of the low speed shaft.

Picture 3: Optical pickup on the low speed shaft. Once per turn mark was put on the shaft while it was running.

Picture 4: Gearbox base on the south side.

ANALYSIS OF THE GEARBOX VIBRATIONS

We collected vibration data from the gearbox. Time domain, frequency domain, and averaged data (time averaging and frequency averaging) were collected.

Figure 1 is representative of the data. It was collected in the horizontal direction, near the bearing on the output side of the low speed shaft. The accelerometer was mounted on a nut, which had been glued to the gearbox.

Based on the speed of the output shaft and the number of gear teeth given to us, the gearmesh frequency was 3494 Hz. Note in the spectrum plot in Figure 1, that there is a peak in the acceleration spectrum approximately at this frequency (3504 Hz). However, the major peak in the spectrum is at 4828 Hz, which is 1.38 times gearmesh frequency.

Close inspection did not disclose significant amplitudes at harmonics of gearmesh, or sidebands. The time domain data did not show any spikes. Generally, the gearbox looked to be in good condition based on normal dynamic indicators, except for the unusual peak at 4828 Hz.

Further analysis of the 4828 Hz peak could not be done because of restrictions on changing speed. However, it can be seen that there are many small peaks around the main 4828 Hz peak. This energy is consistent with the idea that there is a mechanical natural frequency (MNF) of the gearbox case that is at or near 4800 Hz. Changing the speed of the unit would have permitted us to test this theory. Experience has shown, however, that frequently there are MNFs which are excited in a gearbox. While it is not a particularly good sign, it is not necessarily an indication of a serious problem with the gearbox in the absence of harmonics and sidebands. It is more likely a sign that the gearbox manufacturing tolerances were looser than what can be achieved (but at what cost?).

Vibrations were collected in a similar manner at Test Points 3, 4 and 5 in the horizontal direction. Refer to Pictures 1 and 2 for details of the test point locations.

Refer to Picture 3 for details of the once-per-turn setup. As noted, the mark was put on the shaft without shutting down the unit.

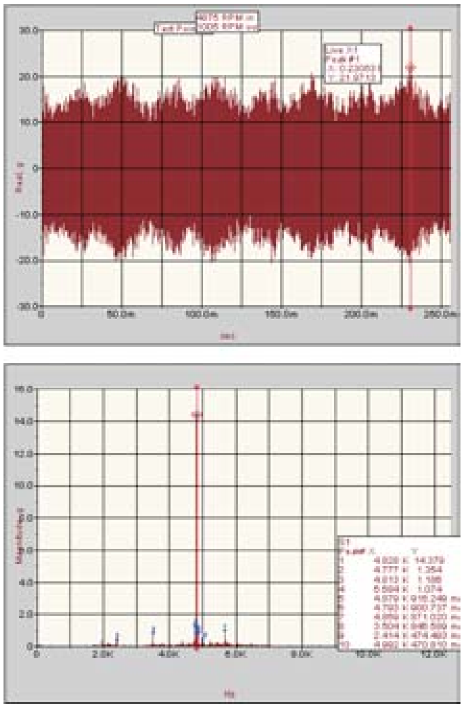


Figure 1: Test point 6H Acceleration Data

Picture 4 shows the feet of the gearbox on the south side. This is shown in connection with the soft foot check that was done. We checked for the effect of a soft foot on the vibrations by loosening and tightening one foot at a time. The change in vibrations was noted for each of the four feet. Foot 3 showed the largest change in vibrations. The overall vibrations (from the time domain data) decreased by 6%.

The amplitude of the vibration at 4828 Hz decreased by 22% when the anchor bolt was loosened.

CONCLUSIONS AND RECOMMENDATIONS FROM THE ANALYSIS

We concluded that the gearbox was in good enough condition to run till the planned turnaround next spring. We did, however, recommend doing a few additional checks to ensure that this conclusion was correct. We recommended checking the oil in the gearbox for wear metals. We also recommended checking the oil filters for any sign of metal.

At the time of writing this paper, the gearbox has run successfully over the winter. The plant is now in turnaround, as planned.

DISCUSSION OF THE PREDICTIVE MAINTENANCE DATA

The periodic collection of vibration (and other) data is generally a good concept. In order for the process to give maximum benefit, the planning and implementation of the system should be carefully considered, and reviewed from time to time.

In this case, we were not able to determine how the vibration data had been collected. Was the method the same before and after the increase in overall vibrations? It could have been collected with a stinger on the accelerometer (hand held), and then a change made to a magnet. Alternatively, a magnet may have been used, but the condition of the magnet may have changed. Similarly, we were unable to determine exactly where the test points were that had been used.

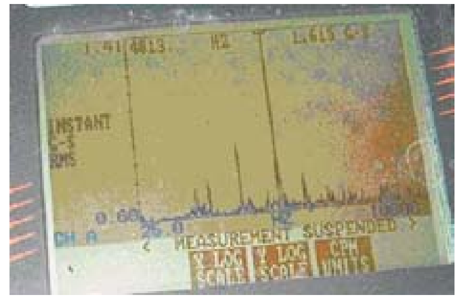
We did observe that all of the spectral information was taken with a maximum frequency (Fmax) of 2500 Hz. Unfortunately, this Fmax is less than gear mesh frequency, let alone the first or second harmonic of gear mesh.

(Note: There is a common tendency to incorrectly use the term "harmonic". A harmonic requires a fundamental frequency to be present. The first harmonic of the fundamental frequency is two times the fundamental frequency.)

Nonetheless, the overall vibrations did increase. Why? Although we were not sure of the frequency range included in the data used to calculate the overall amplitude, it could have been up to the Fmax setting for the spectra, or up to the maximum frequency of the instrument. As a minimum, the energy measured by the data collector had increased in the range of frequencies below 2500 Hz.

Based on experience, we were suspicious of the use of either handheld or magnetically mounted accelerometers for collection of data above 1000 Hz. We have tested the effect of a magnetic mount versus a stud mount. The results of the test are shown in two photographs of the screen of the instrument.

Picture 5: Vibrations at 6H with magnet mount



Picture 6: Vibrations at 6H stud mounted

Observe the dramatic change in vibration energy below the gearmesh frequency peak at 3500 Hz. Stud mounting increased the amplitudes of the peaks at gearmesh and 4820 Hz.

The good news is that all of the peaks are present, albeit at different amplitudes, in both spectra.
The biggest reason for not using a magnet to mount an accelerometer for high frequency measurements is the desire to collect data repeatably. The frequency response of the accelerometer is influenced by the connection of the magnet to the gearbox. Changes in the thickness of the paint will change the frequency response and produce an apparent change in the condition of a gearbox. Dirt under the magnet will do the same thing. Poorly mounting the magnet (not "wiggling" the magnet to ensure the magnet is solidly mounted) will do the same.

But one of the most significant sources of lack of repeatability is a loss of magnetic strength in the magnet. This comes with time and use (or misuse) of the magnet.

Instructions with magnets tell the user to slide the keeper (usually a steel washer) on and off the magnet. It is a natural tendency to let the keeper impact the magnet. This is bad. It tends to realign the magnetic particles in a random order, thus causing a loss of magnetic pull. In this case, the magnet that came with the collector was not particularly strong and it seemed to be even weaker than experience suggested it should have been. Changes in the magnet may have been the reason for the increase in overall vibrations.

Lest it be said that this case history suggests that periodic vibration collection is a bad thing, it does seem that it is better to have an occasional false alarm than to miss a real problem.

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CONCLUSIONS

Vibration measurements can tell us about the condition of a gearbox, if properly done. Other physical parameters besides vibrations can be helpful.

Trending of data should be helpful, but only if the data being trended is collected consistently.

Methods of mounting of transducers, such as accelerometers used for measuring high frequency data, are important. Repeatability of data collection is the most important consideration in choosing a mounting method.

Periodic data collection programs should be reviewed by an unbiased third party expert. Such a review could have benefit at the start of a program to ensure that the test points and frequency ranges are correct. After the program has been active for a few years, benefit might be derived from a review of the cost-effectiveness of the program.

News & Notes

This year at the Gas Machinery Conference in Nashville, Tennessee (October 8 - 9, 2002) the Gas Machinery Research Council celebrated 50 years of service. Beta was there celebrating our 35 years of service. Again this year the technical conference and trade show event lived up to its reputation as first class. Thanks to all who stopped by to see us at our booth. In addition to participating in the trade show, Beta personnel made two technical presentations summarized here:

API 618 Forced Response Studies
To meet the requirements of the API 618 4th Edition, Design Approach 3, the vibration and stress levels in the compressor manifolds and piping must be calculated. This paper not only demonstrates how these studies may be completed, but also show their value in the successful design of reciprocating packages. Case histories help make the connection between theory and reailty.

Pulsation/Vibration Guidelines
Basic theory of pulsation and vibration in reciprocating compressors is presented, course notes detail fundamental design guidelines that ensure a smooth, efficient reciprocating compressor package with minimal capital cost.

The CMVA Annual Meeting was held in Quebec City in October, 2002. The following papers were presented by Beta:

A Gearbox Vibration - Fact or Fiction (included in this issue)
Pulsation & Vibration Control for Small Reciprocating Compressors
A Case Study of Piping and Shaft Vibrations on a Centrifugal Compressor

The 4th International Pipeline Conference was held in Calgary from September 29 - October 3, 2002. Brian Howes and Shelley Greenfield presented “Guidelines in Pulsation Studies for Reciprocating Compressors”.

If you would like copies of any of these presentations, please contact us.

Ron Carpendale has joined Beta as our Operations Manager and will be located in the Calgary office. Ron comes with several years experience in the oilfield and gas processing industry.

Happy New Year, may 2003 hold peace, good health and prosperity for us all.

Beta Machinery Analysis engineers Bryan Fofonoff (left) and Luis De la Roche (right) explore Makati City in Manila.

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