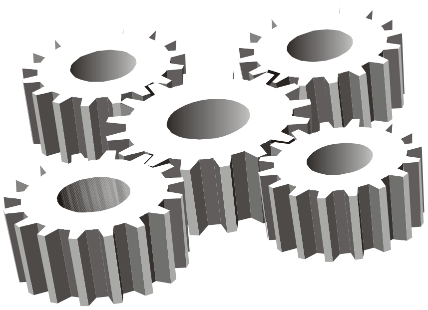
THE BETA BULLETIN



INSIDE VOLUME 9#3

Profiling of Reciprocating Compressors

Introduction

Beta Machinery Analysis Ltd. was retained to investigate high vibrations on a fuel gas package, consisting of three fixed speed, two-throw reciprocating compressors. Depending on the demand, up to two of the compressors would be run, with the third as a spare. This was a new installation, and the client was concerned because the vibration monitors were reading vibrations as high as 0.5 ips on the compressor frames of those machines that were running. As well, vibrations at levels of 0.2 ips were measured on the spare compressors. Vibration Profiling proved to be a good tool for analyzing this situation.

Profiling is a data reduction and presentation technique that succinctly outlines the problem areas to the analyst (see Beta Bulletin Vol 9 #1 for more detail on what Profiling is and how it works). Profiling looks at defined frequency windows in each spectrum and finds the highest amplitude component. These are displayed in a bar graph format for all the spectra taken on a machine, or across several machines. Historically, Profiling has been used with great success on paper machines, eliminating the need for the analyst to wade through hundreds of spectra to detect and diagnose machine problems. However, Profiling has not been used on reciprocating compressors. In this case, Profiling was used to greatly reduce the analysis time.

Profiling Results

In general, Unit 2 had the highest amplitude vibrations. Therefore, data presented here will be for that compressor. The first two profiles given in Figures 1 and 2 are those originally taken on Unit 2 in the as found condition (April 6th), when that unit was running alone. The profiles show overall vibration in inches per second peak. A list of locations is provided in Table I. The first point is MIB or motor inboard (drive end) with the three bars above representing the horizontal, vertical and axial vibrations. See Table II for a description of the directions.

These two profiles show that there were a number of points well above the guideline of 1 ips pk. In fact, the inboard end of the compressor frame had vibrations of over 1 ips in the horizontal direction. As well, the number 1 cylinder suction bottle was over 2 ips pk in the horizontal, vertical and axial directions at both ends. The number 2 cylinder suction bottle was also above guideline in the horizontal and axial direction. High vibrations were also found on Unit 2 piping, especially the relief lines.

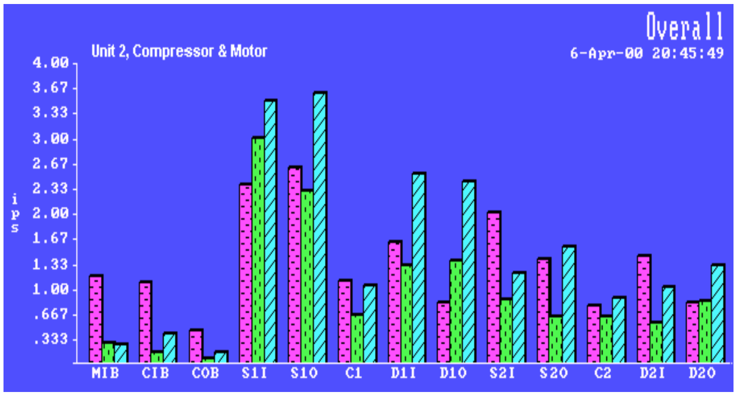


Figure 1. Overall vibration profile on Unit 2 as found, taken on compressor, motor, and bottles

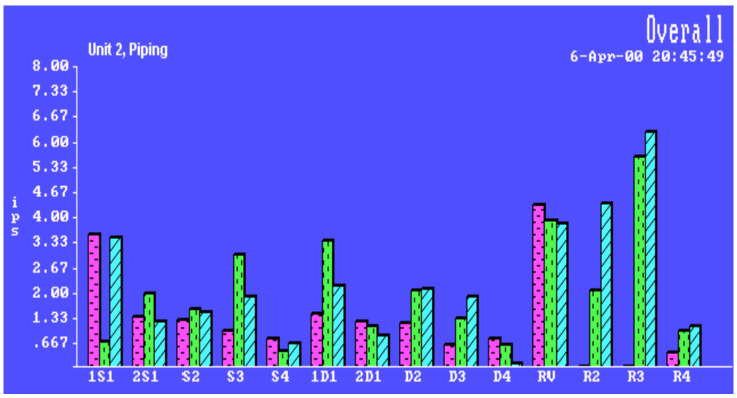


Figure 2. Overall vibration profile on Unit 2 as found, taken on piping.

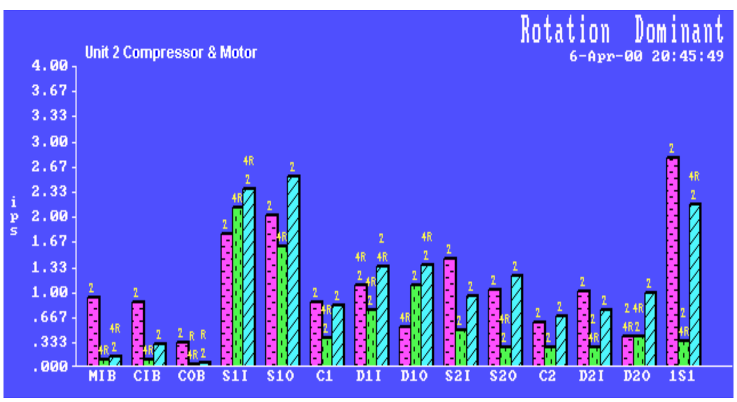


Figure 3. Rotation dominant profile of Unit 2 in the as found condition.

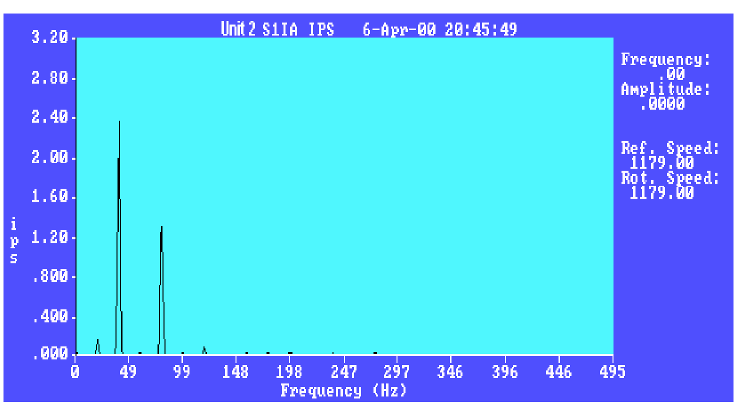


Figure 4. Spectra of the axial vibration measured at S1I on Unit 2 as found.

The two overall profiles immediately highlighted the areas of high vibration. But, what were the frequency components, and the corresponding amplitudes that were causing these high vibrations? Without Profiling, the analyst would have to look at all the corresponding spectra at the points with high overall vibration, and painstakingly build an understanding of the vibrations. Profiling can greatly reduce that time by using frequency windows to automatically produce those profiles. Although the actual mechanics of the Profiling process are simple, the information revealed by Profiling can be profound.

Figures 3 and 4 show the rotation dominant profiles. In this case, the rotation dominant profiles isolate the maximum amplitudes that occur at 1x, 2x, 3x, 4x, 6x and 8x compressor run speed. If we look at the number 1 suction bottle inboard end (S1I) in the axial direction (see Table II for a definition of directions), we see a bar higher than 2.33 ips. The “2” immediately above the bar indicates that the 2x compressor run speed (or approximately 40 Hz) was the dominant component. That is, the 2x component had an amplitude of 2.33 ips pk. The “4R” above the “2” indicates that the 4x component was more than 50% of the dominant component. This can be confirmed by looking at the spectrum for that point, as shown in Figure 4. As well, a quick scan across the profile reveals that at most of the points, the second order component (2x) was the highest amplitude.

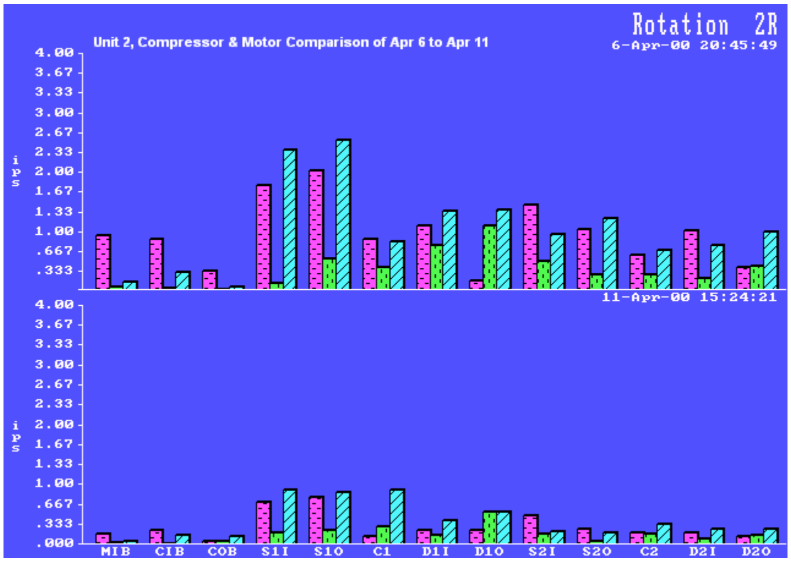


Figure 5. . Comparison of the second order vibration components on Unit 2 compressor, before and after modification.

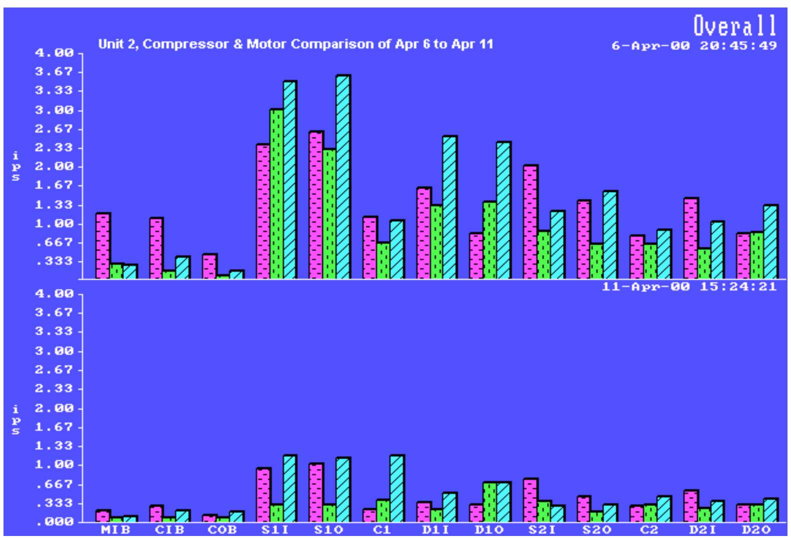


Figure 6. Comparison of the overall vibration profiles on Unit 2 compressor, before and after modification.

From this one graph we get a clear picture of the vibration at harmonics of rotational frequency. Without Profiling we would have to evaluate 42 individual spectra to try to develop the same understanding.

Further field analysis revealed that the compressors had a mechanical natural frequency near 2x or 40 Hz. The mode shape was a rotation of the compressor and the attached bottles and piping about a vertical axis near the center of the compressor frame. The relief header joining the three compressors also had a mechanical natural frequency at 2x. Acoustic analysis (computer modelling) revealed the presence of high pulsation forces at 2x. The acoustic forces, along with the inherent high unbalance couple at 2x in a two-throw compressor resulted in high amplitude vibrations of the compressors, cylinders, bottles and attached piping.

The solution was to cut the outboard cylinder supports. This lowered the natural frequency of the compressor below 2x run speed. The relief line and pressure relief valves were braced at the appropriate locations. As well, orifice plates were added to reduce the acoustic forces. The profiles of the second order vibrations measured on Unit 2 as found on April 6th and as modified on April 11th are compared in Figure 5. The difference in the amplitude of the second order component of the vibration is clearly seen. A comparison of the resulting overall vibrations on Unit 2 from the as found to the as modified is shown in Figure 6. The difference that the modifications made to the overall vibration levels is quite apparent. Although some of the levels exceeded the 1 ips guideline after modification, they were considered acceptable. Figure 6 was used in a wrap up meeting to quickly help the client understand the success of the modifications.

Table I. Point descriptions

Point	Description	Point	Description
MIB	Motor inboard (drive end)	2S1	2 side suction elbow riser tip
CIB	Compressor inboard	S2	Suction line at drain on compressor centerline
COB	Compressor outboard (non-drive end)	S3*	Main suction header at drain
S1I	Suction bottle 1 inboard end	S4*	Main suction line at inlet to separator (riser)
S1O	Suction bottle 1 outboard end	1D1	1 side first discharge tee or elbow
C1	Cylinder 1 head end	2D1	2 side first discharge tee or elbow
D1I	Discharge bottle 1 inboard end	D2	Discharge line at drain on compressor centerline
D1O	Discharge bottle 1 outboard end	D3*	Main discharge header at drain
S2I	Suction bottle 2 inboard end	D4*	Main discharge header at reheater
S2O	Suction bottle 2 outboard end	RV	Relief valve
C2	Cylinder 2 head end	R2	Relief vent line on compressor centerline
D2I	Discharge bottle 2 inboard end	R3*	Relief vent line half way between Units 2 and 3
D2O	Discharge bottle 2 outboard end	R4*	Relief vent line elbow at reheater
1S1	1 side suction elbow riser top		

* Indicates points that are common for all three units. Other points are taken on appropriate points on each unit.

Table II. Description of direction

Direction	Description
Axial	Parallel to the crankshaft
Horizontal	Parallel to the piston motion
Vertical	Vertical

Conclusion

For this reciprocating compressor troubleshooting job, Profiling was an excellent tool for defining the problem areas and then finding the largest amplitude frequency components. If the analyst were to do that manually, it would require looking over and understanding 80 spectra per unit. Profiling reduced the process to minutes, allowing the analyst to understand and compare the vibrations on each unit.

In addition, the before and after profiles proved to be an exceptionally understandable way of presenting the results to the client.

New Grande Prairie Location

Our Grande Prairie office is now at 10821 – 78th Avenue. Tim Klone, B.Sc., has left our Calgary office to join Roy Finlay at that location. The new office is bigger and brighter with lots of room for all of the equipment needed to service that area of Western Canada. We'll be hosting an Open House early in November to show off our new space, please call Roy at 780-831-7347 for a date and time.

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