

# **OBSERVATIONS ABOUT ROTATING AND RECIPROCATING EQUIPMENT**

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## **ABSTRACT**

This paper discusses several small issues that have occurred in the last year that may be of interest to machinery users. Included is a follow-up from one of the case histories from a paper presented last year regarding contributions to vibration from Variable Frequency Drives.

### **1. INTRODUCTION**

The purpose of this paper is to discuss several small machinery issues and to update a paper from last year.

### **2. FOLLOW-UP FOR A VFD CASE HISTORY FROM REFERENCE 1:**

Last year, we presented a paper at the CMVA annual conference that discussed some experiences with vibration problems that had occurred in systems that had variable frequency drives (VFDs).

Subsequent to the presentation of that paper, more information was obtained during a follow-up visit to one of the sites. More detail than will be presented here is given in Reference 2, a paper presented at the 2005 Gas Machinery Conference.

Tuning of the VFD resulted in lower torsional vibrations, and a successful machine installation.

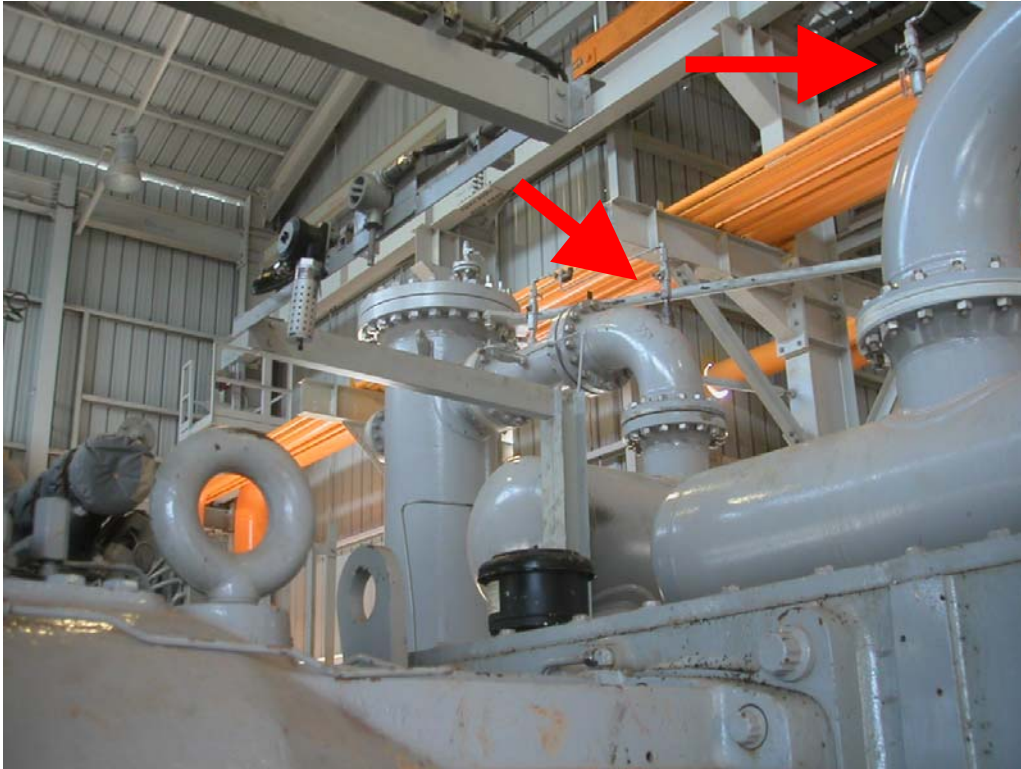
Prior to the VFD tuning, shaft failures had occurred in the motors.

### **3. SMALL BORE PIPING ISSUES**

These comments apply to a range of rotating and reciprocating equipment. There have been many failures of small piping attachments on process piping, vessels, and machines in the past and more continue to happen in the present. Failures have been associated with screw compressors, centrifugal compressors, plunger pumps and reciprocating compressors.

The common theme has been resonant vibration of attachments. The excitation of the attachments is via base motion. That is, the big pipe usually vibrates in the axial or tangential directions (with respect to the big pipe) and the attachment amplifies the motion due to resonance.

The following pictures show examples of small bore attachments on a reciprocating compressor installation. Failures occurred on some of these valves at the nipple-to-weldolet location.



Picture 1 Some small bore attachments.



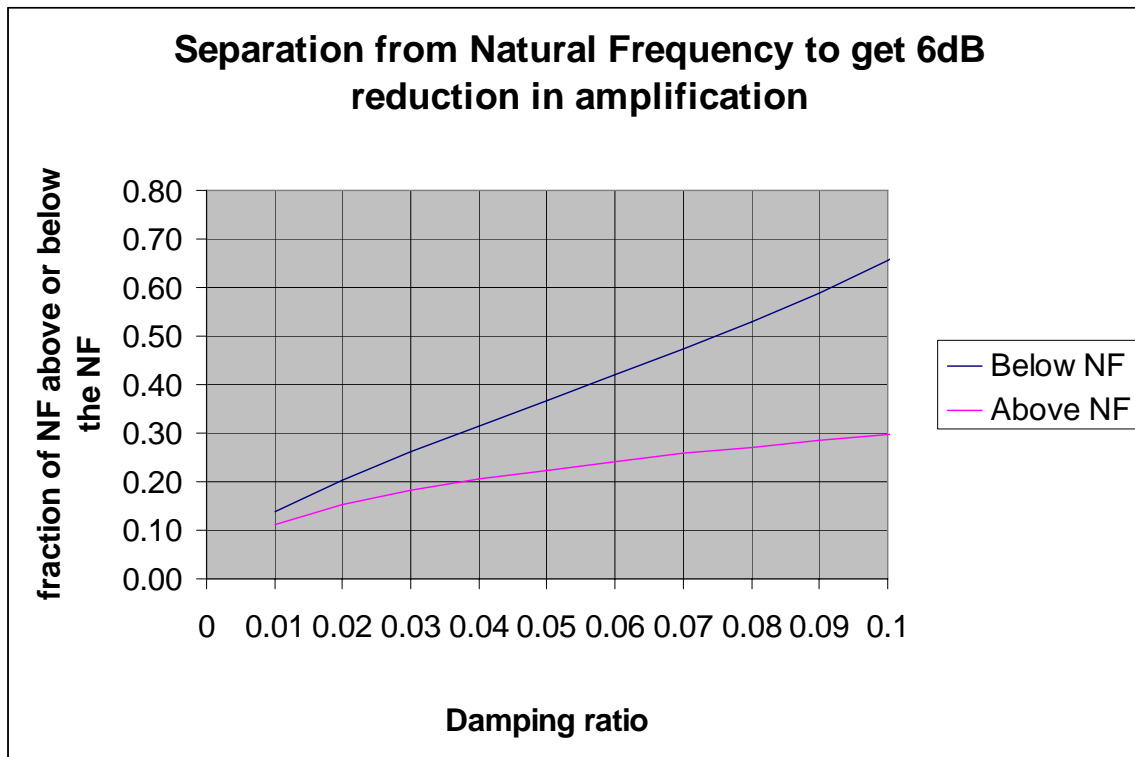
Picture 2 More small bore attachments.

#### 4. RESONANCE

The definition of resonance is worth reviewing. Resonance is usually explained as occurring when a forcing frequency coincides with a natural frequency (NF). It is necessary to define the degree of coincidence for this definition to be practical. In API 541 Standard, Reference 3, resonance is defined as occurring when the forcing frequency is in a frequency range that extends on either side of the response peak down 6 dB from the maximum amplitude. We can quibble over the details, but at least this is a usable definition of resonance.

Refer to the graph below. (The data for the graph were calculated for a damped single degree of freedom spring-mass system.) The graph shows the separation from the natural frequency that is required to achieve 6dB decrease in amplification either below or above the natural frequency. Traditionally, 10 to 15% separation has been used as a rule of thumb for minimum frequency separation from a critical speed or natural frequency. The graph suggests that this would be sufficient for lightly damped systems, which is how most mechanical or shaft systems would be defined. The graph seems to describe the separation required better as a function of damping.

Another way of interpreting the data on the graph, would be to say that it is not necessary to have 6 dB reduction from the peak response when the system is more heavily damped. Using that reasoning, less frequency separation may be acceptable at higher damping ratios.



In most cases, resonance of an attachment should be avoided with the major sources of dynamic forces in a machine. For example, with a fixed speed screw compressor with 4 pockets, small bore attachments should not have mechanical natural frequencies (MNFs) “near” 1X, 2X, 4X or 8X run speed.

A demonstration of two frequency ranges to avoid for MNFs of small bore attachments came from a centrifugal compressor problem. This problem occurred when a piece of metal got caught in the housing of the compressor and resulted in high pressure pulsations in the discharge piping at shaft rotational frequency and at vane passing frequency. These pressure pulsations caused shaking forces in the piping, which caused vibrations of the piping. The drain lines on the discharge line were then subjected to higher “base motion” vibrations than were normal. We were contracted to determine if the stresses due to the higher than normal vibrations would be acceptable until the unit could be repaired in a planned shutdown. The vibrations were worse than they needed to be if the drain attachments had been designed to avoid having MNFs near the frequency range of compressor shaft speed or vane passing frequency.

Plunger pumps can experience the same problem. However, we did see a problem with an attachment (Reference 4) on a pump that was a serious problem (failures occurred) even though individual frequencies of vibration were acceptable according to normal industry guidelines. Measurements of overall vibrations in displacement terms showed that the vibrations of the attachments were in fact too high. This is an unusual example which suggests that overall true peak differential displacements should be measured on small bore attachments, in general, to determine the safety of systems.

On reciprocating compressors, we have seen situations where small bore attachments have had natural frequencies that were close to the horizontal cylinder natural frequencies (horizontal=in the direction of piston motion). The horizontal cylinder stretch motion tends to be higher at the orders of run speed around this natural frequency of the cylinder. This high motion in turn leads to more energy on the bottles and piping to which small bore attachments append. In other words, there is more energy in the base motion at or near the natural frequencies of the attachments as a result. Failures have been seen in many instances. The solution in these cases is to change the natural frequencies of the attachments away from the natural frequencies of the compressor cylinders (if the small bore attachments cannot be removed).

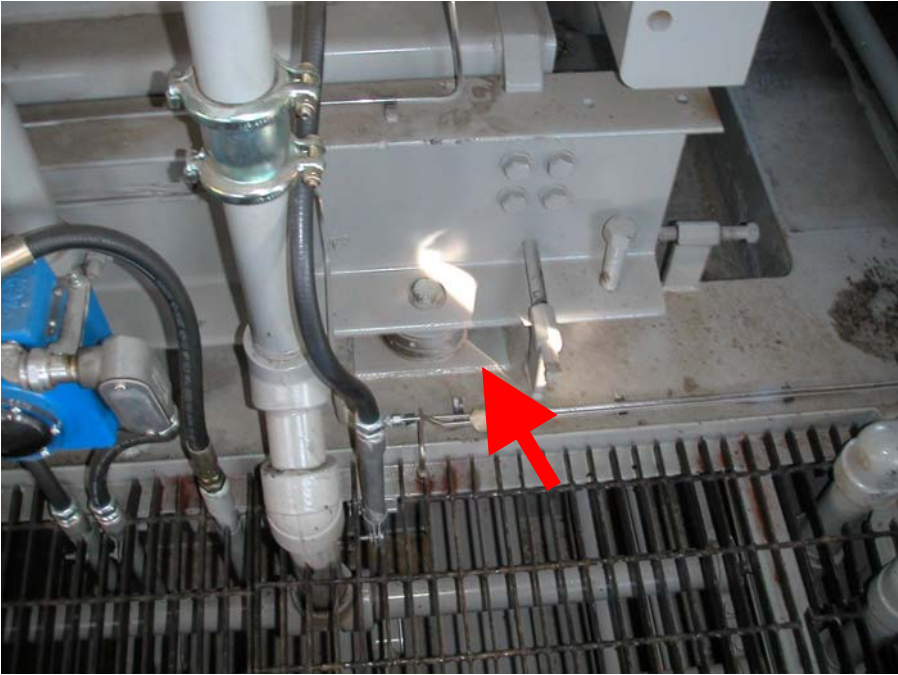
This course of action (changing NFs) is counter-intuitive for a machine that generates pressure pulsations which are usually the first suspected cause of all vibration problems.

## **5. ADJUSTABLE CHOCKS**

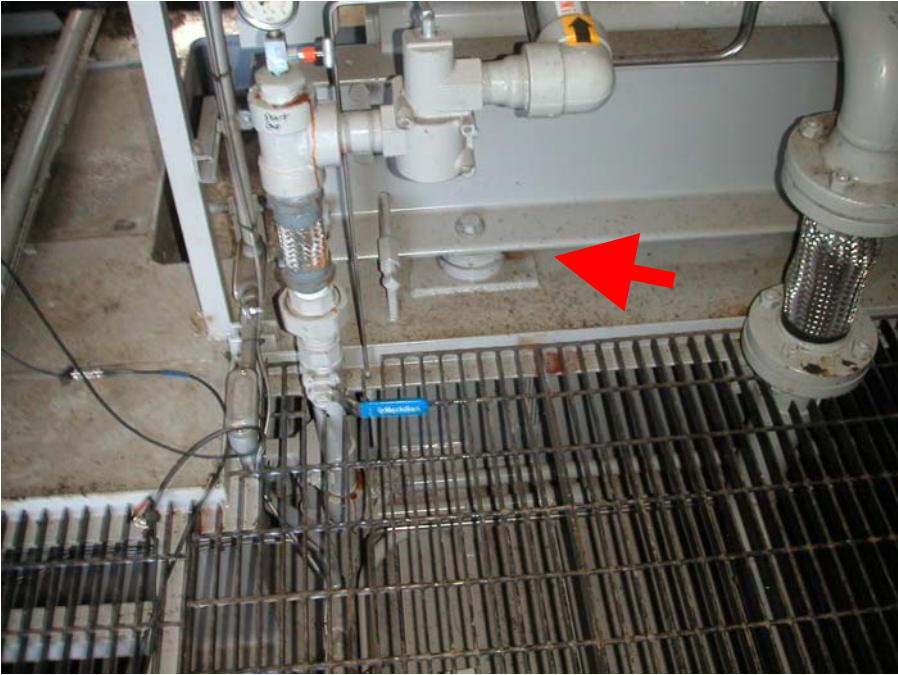
Installers of machinery look for easier ways to install, level and align equipment (motors, engines compressors, etc.). Many techniques have been invented. Some methods are more time consuming and therefore expensive.

One system is called Vibracon SM, “The Universal Adjustable Chock”. The goal of these devices is to eliminate the need for shims, allow rapid and accurate height adjustment and eliminate soft feet problems.

Pictures 3 and 4 show adjustable chocks used successfully under an engine.



Picture 3 Chock under front of engine.



Picture 4 Chock under back of engine.

We have seen two situations in which the adjustable washers do not constitute an acceptable connection between the equipment and the skid. The lateral vibration above the washers, at the equipment feet, was approximately 50% higher than the vibration just below the washers, at the skid beam where the washers were mounted. If the vibration at the top of the skid is marginal, or if the installation operates close to resonance, the resulting vibration levels at the centerline of the machine (compressor or engine) or at the compressor cylinders is not within our guidelines due, in part, to the amplification across the adjustable washers (chocks).

The vibration amplification has been observed at low and at high frequencies. Further investigation is recommended to understand the limitations of these devices in terms of equipment type, size, skid design and lateral forces. Many equipment packagers mention successful installation of these devices under small and medium size motors and under engines. Small reciprocating compressors have also been mounted on adjustable washers, and they, reportedly, show acceptable vibrations.

## **6. TORSIONAL VIBRATION NOTES**

We were recently asked to travel to Indonesia to assess some engine-compressor installations for both linear and torsional vibrations. Failures had been occurring.

The three units were nominally identical from the point of view of torsional vibrations, but not from the point of view of linear vibrations.

Torsional vibration (TV) measurements showed that the torsional natural frequencies (TNFs) were indeed the same on all three units.

However, the damping associated with the compressor mode of TV was different from unit to unit. This was exhibited by the fact that the amplitudes of TV at the TNF were significantly different on the 3 units, even though they were operating at essentially the same conditions and load step.

As the result of computer models, we recommended a change to the rotational inertia of one of the compressors, which had the highest measured TV and had experienced an oil pump failure.

Reports from the field indicated that linear vibrations on the frame of the compressor decreased after the addition of the inertia on the compressor crankshaft. The frequency of the linear vibrations that have apparently decreased was not the same as the TNF.

We expect to collect more vibration data from this unit in the future, which may be more definitive.

## **7. STRESSES IN MOTOR ROTORS**

We were asked to determine if vibration-induced stresses in a large motor rotor could be high enough to cause concern. We constructed finite element models of the motor rotor with the original design and with the modified design.

We were reminded of an interesting observation related to alternating stresses in a rotor that may be useful when analyzing vibrations on machines with rotors.

If the whirl of the shaft is circular, and at the same frequency as rotation, then the bending stresses in the rotor are constant with respect to angle of rotation. This means that there are no alternating stresses in the rotor to cause fatigue damage.

On the other hand, elliptical whirl shapes and frequency components in the whirl that are not synchronous with the shaft rotation will cause alternating stresses in a rotor.

## **8. CONCLUSIONS**

The world of machinery vibrations is an interesting place.

## **9. REFERENCES**

1. Howes, B.C., "Perplexing Variable Frequency Drive Vibration Problems," presented at the CMVA, Edmonton, AB, 2004.
2. De la Roche, L, "Lateral & Torsional Vibration Problems in Systems Equipped with Variable Frequency Drives," presented at the GMC, Covington, Ky, 2005.
3. API 541, Standard of the American Petroleum Institute.
4. Howes, Brian C., Berg, Lyle, and Zacharias, Valerie, 1996, "Incorrect Valve Selection on Plunger Pumps Results in Undetected High Frequency Vibration and Costly Failures, A Case Study," presented at the CMVA, Edmonton, AB.

## **10. AUTHOR BIOGRAPHY**

Brian graduated from the University of Calgary with a Master of Science in Solid Mechanics. His thesis was entitled *Acoustical Pulsations in Reciprocating Compressor Systems*.

Brian has worked with Beta Machinery Analysis since 1972. In his present position as Chief Engineer for Beta, he has performed troubleshooting services all over the world.

Brian has many technical papers to his credit. The range of machinery problems they cover includes all manner of reciprocating and rotating machinery and piping systems, balancing and alignment of machines, finite element analysis, modelling of pressure pulsation, torsional vibration testing and modelling, flow induced pulsation troubleshooting and design, pulp and paper equipment such as pulp refiners, etc. He has also worked on hundreds of reciprocating compressor installations.