

## A CASE STUDY OF A FLOW-INDUCED TORSIONAL RESONANCE

*William F. Eckert, P.Eng., Ph.D.*  
*Field Services Manager*

*Brian C. Howes, M.Sc., P.Eng.*  
*Chief Engineer*

Beta Machinery Analysis Ltd., Calgary, AB, Canada, T3C 0J7

### ABSTRACT

A centrifugal compressor driven by electric motor had been modified from its original 450 HP motor to a 600 HP variable frequency drive. Upon startup, the coupling failed (broken bolts) several times. Each of the failures occurred after less than two weeks of run time. The coupling was a flexible disk type with a spacer. The compressor manufacturer had done a torsional analysis and had noted that the predicted torsional natural frequency was approximately 38 Hz. They also noted a possible excitation of that natural frequency by a 12<sup>th</sup> order harmonic from the VFD. In our tests a digital strain gauge telemetry system was used to measure the actual torque carried by the coupling spacer.

The fundamental torsional natural frequency did in fact show up in the speed sweep of the unit. What was not expected, was that the torsional natural frequency was excited throughout most of the run speed range. Flow induced pulsation caused by vortex shedding of the mean flow past a manway in the suction piping was the culprit. This is the first case, in our knowledge, of excitation of a torsional natural frequency by flow induced phenomena.

Typically a resonant sidebranch is required to amplify the pulsation to a sufficient level to cause vibration issues. In this case, the pulsation seemed to be amplified by an acoustical resonance of the suction piping itself.

The compressor was also surged at startup. This was apparently not atypical of this unit during startup and operation. The torque measurements during the surge event also showed an extreme excitation of the torsional natural frequency. The torsional response was such that the coupling was definitely overloaded.

### 1. INTRODUCTION

An evaporator compressor at a processing facility had run successfully for years with a 450 HP variable frequency drive (VFD) electric motor. For production reasons, the unit was modified to a 600 HP VFD. With the larger motor, the coupling unit failed several times (broken bolts with less than two weeks of run time for each failure). The coupling is a Thomas Series 52 flexible disk with spacer. See Figures 1 and 2.

The compressor itself is a single stage unit, with a large overhung centrifugal wheel. The suction piping enters the compressor axially, and the discharge piping exits the housing tangentially in the vertical direction.



**Figure 1.** Broken coupling bolts



**Figure 2.** Coupling on 600 HP motor

During engineering of the drive modification, the compressor manufacturer had done a torsional design analysis. A torsional natural frequency of the modified unit was predicted at 38 Hz (2280 rpm), with a possible excitation of the natural frequency by a 12<sup>th</sup> order harmonic from the VFD. The operating speed range of the unit is from 2600 to 3600 rpm.

Beta Machinery Analysis Ltd., was retained to determine the reasons for the coupling failures. A field measurement of the torsional vibration was conducted on February 25, 2001 with both the 600 and 450 HP drives in place. The following measurements were taken: drive torque by strain gauge on the coupling spool piece; current on one phase of the motor input using an inductive pickup and current transformer; lateral vibration in the horizontal and vertical directions on the drive ends of the motor and compressor; and vibration at several points on the inlet piping to the compressor. Startup, loading of the compressor, and loaded steady state operating conditions were examined.

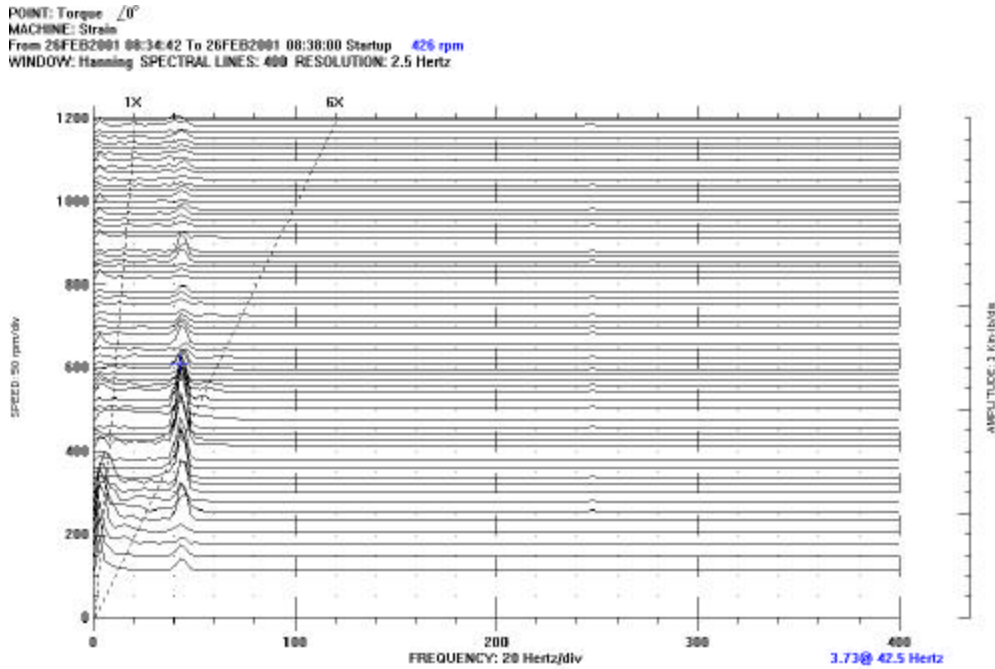
The measurements were taken using a Bently Nevada Adre digital acquisition system, a Binsfeld Torque Track 9000 digital telemetry strain gauge torque meter, a Fluke 80I-1010 current transformer, and several Wilcoxon 799LF low frequency accelerometers.

## **2. MEASUREMENTS**

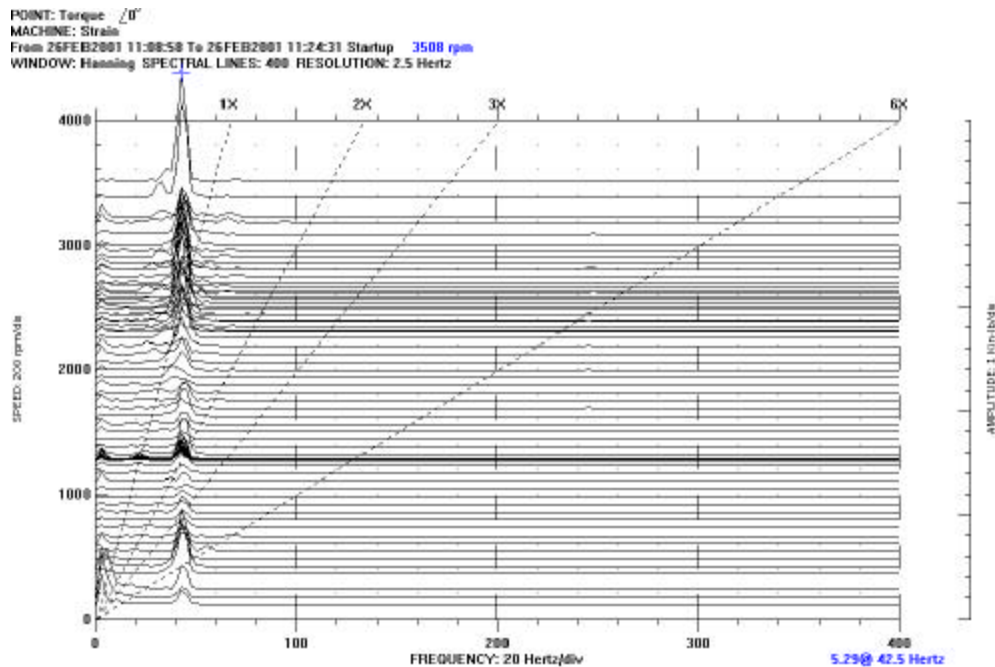
The first test was a start-up with the new 600 HP motor. The torque meter was set to a full-scale reading of 13,000 in- lb which was believed would adequately cover the start-up torque. The initial start-up was aborted at 1200 rpm. A cascade plot of the torque as the unit was run-up is shown in Figure 3. As the unit was started, a torque fluctuation at shaft speed was seen, but that died out above 500 rpm. As well, the unit's torsional natural frequency (TNF) was found at 42.5 Hz versus the predicted frequency of 38 Hz. The TNF was excited by a 6x component from the VFD at a speed of 426 rpm. The maximum torque fluctuation was 3730 in- lb pk during that event.

On the second start-up, the full speed of 3600 rpm was achieved. A cascade plot of the torque during that start-up is shown in Figure 4. Interestingly, the torsional natural frequency locked in and was present throughout a speed range of 200 to 3600 rpm. The torsional natural frequency was excited by VFD components at 1x, 2x and 6x shaft speeds. Most importantly during this

run, as the speed approached 3000 rpm, the compressor surged. While surging, the TNF was excited but the amplitude was highly variable. For much of the time, the instrumentation was overloaded and could not capture readings. During overload, the peak torque had to exceed the full scale range of 13,000 in-lb. The highest reading, that we were actually able to measure, was a peak torque of 5,300 in-lb at 42.5 Hz.



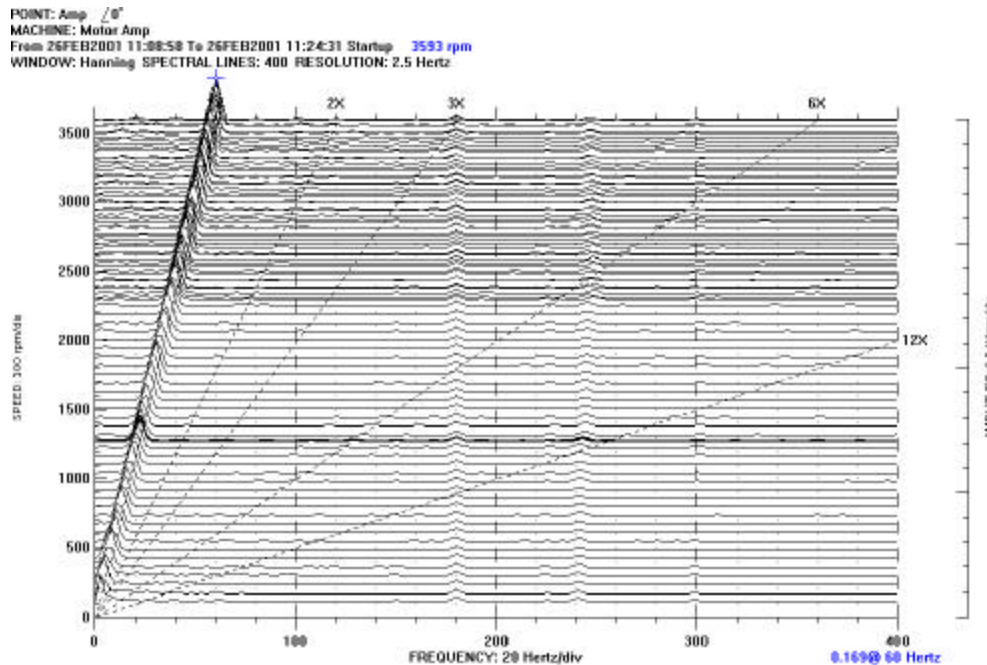
**Figure 3.** Cascade plot of torque with 600 HP VFD during initial start-up to 1200 rpm



**Figure 4.** Cascade plot of torque with 600 HP VFD during start-up to 3600 rpm, with surge

As well during surge, the instrumentation was bouncing on the table, and the horizontal vibrations measured at the driven end bearing of both the motor and compressor peaked at 2 times operating speed. However, there was no component at the torsional natural frequency in the horizontal vibration spectra. As well, the current measured on one of the phases showed only one major component at 1x shaft frequency (see Figure 5).

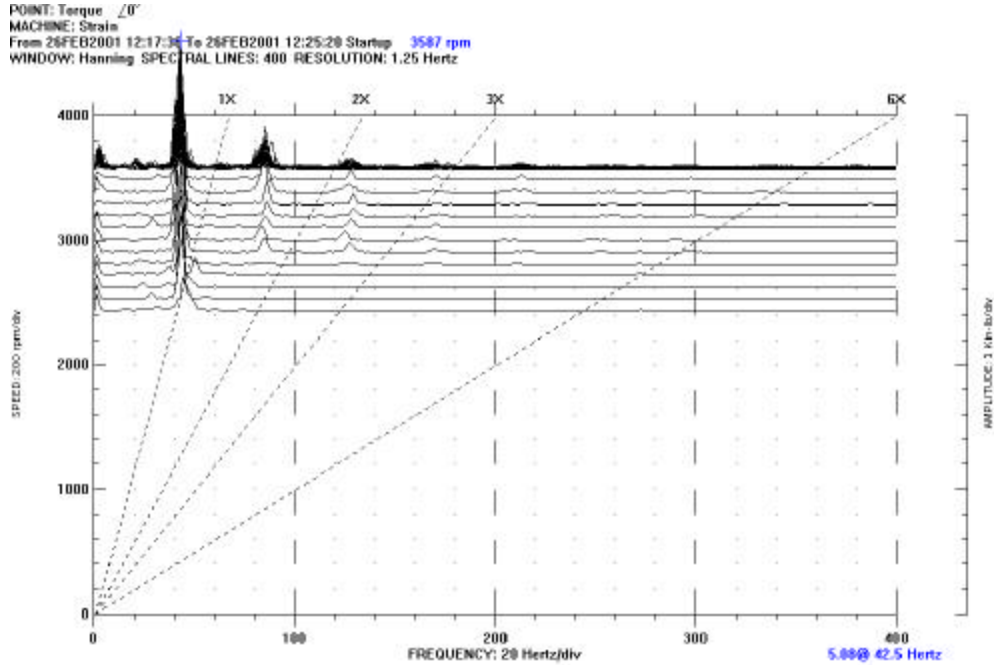
Plant personnel reported that the surge event observed was normal during start-up and loading of this unit.



**Figure 5.** Cascade plot of current to motor during run-up to 3600 rpm and surge

During the third test, the compressor was started up and carefully loaded so that there was no surge. The cascade plot of the torque during this event is shown in Figure 6 (from 2500 to 3600 rpm). Again, the torsional natural frequency of 42.5 Hz was present throughout the test. Under load at 3600 rpm, the torque fluctuation at 42.5 Hz was a maximum of 5,080 in-lb pk, and there was moderate variation in amplitude.

Table I gives a comparison of vibration, loaded current and coupling design factor at 3600 rpm when surged and when run under load without surge. Interestingly, the horizontal vibrations of both the motor and compressor drive end bearings were not significantly higher during surge. As well, during surge both the current drawn and the mean torque transmitted were low. However, the torque fluctuations at the torsional natural frequency were high and the resulting design factor (safety factor with respect to fatigue) on the coupling was much less than one. This indicated that the coupling was definitely sustaining fatigue damage during surge events. We had found the most likely culprit in the coupling failures. But the question remained what was exciting the torsional natural frequency during normal loaded operation?

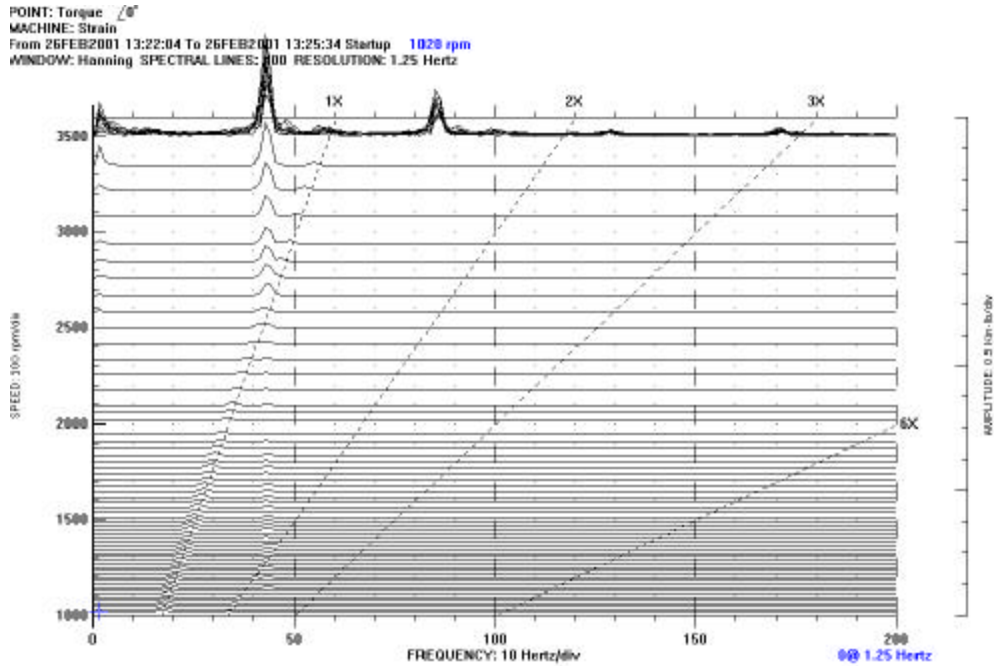


**Figure 6.** Cascade of the torque with the 600 HP motor and no surge

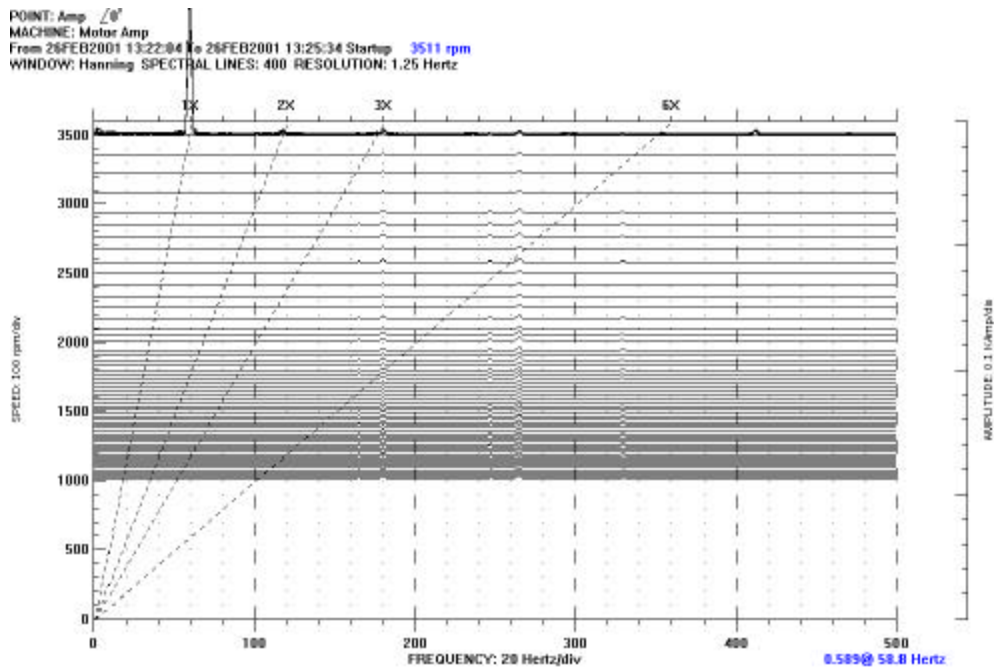
**Table I.** Comparison of 600 HP Tests with Surge and without Surge at 3600 rpm

	Surge	No Surge, Loaded
Max Vibratory Torque at 42.5 Hz	>13,000 in-lb pk	5,080 in-lb pk
Mean Torque	3,000 in-lb	7,700 in-lb
Coupling Design Factor	<0.6	1.0
Current at one phase	170 amps	590 amps
Motor DE Bearing Vibe Hor.	0.20 in/s pk at 120 Hz	0.14 in/s pk at 120 Hz
Comp DE Bearing Vibe Hor.	0.06 in/s pk at 120 Hz	0.04 in/s pk at 120 Hz

In the both of the previous tests, the torsional natural frequency was excited throughout the run speed ranges, and was obviously lightly damped. To check if the excitation was coming from the VFD, the unit was tripped from 3510 rpm. As expected the current dropped immediately, but the torsional vibration at 42.5 Hz existed until the speed was well below 2500 rpm (see Figures 7 and 8). The original concern raised by the torsional design check was that a 12<sup>th</sup> order frequency in the current from the VFD might excite the torsional natural frequency. From the start-up tests and the trip test it did not appear current fluctuations from the VFD were driving the torsional natural frequency. The only other possible cause seemed to be pressure fluctuations due to some unknown aerodynamic phenomena.



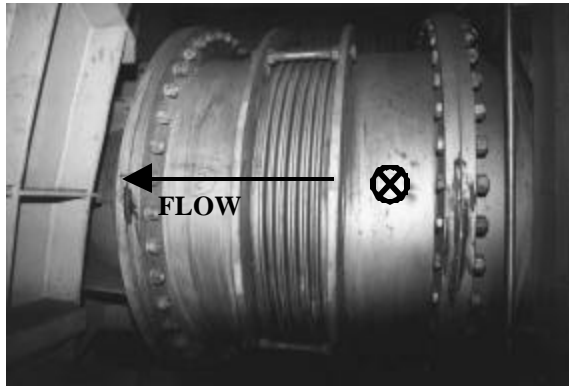
**Figure 7.** Cascade of the torque produced by the 600 HP motor after the unit was tripped



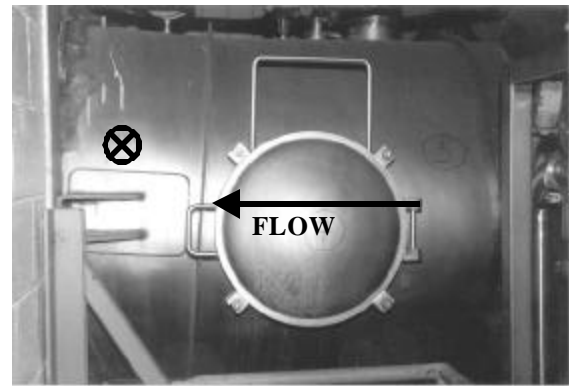
**Figure 8.** Cascade of the current (one phase) drawn by the 600 HP motor after trip

So additional testing was performed with the 600 HP VFD motor. The unit was again brought up to full speed and loaded. Vibration measurements were then taken on the suction piping. The suction piping showed small responses at the torsional natural frequency of 42.5 Hz at two

locations. These were both downstream of an inspection manway in the suction piping, and upstream of an expansion bellows adjacent to the compressor itself (see Figures 9 and 10). Since the piping had low response at the TNF frequency it was not mechanically resonant.



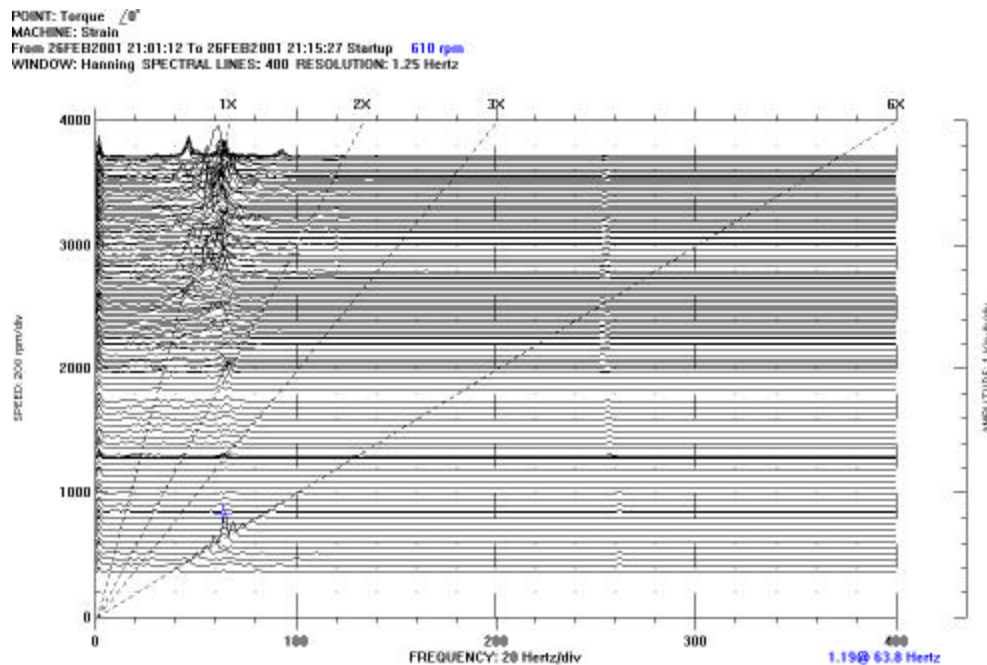
**Figure 9.**



**Figure 10.**

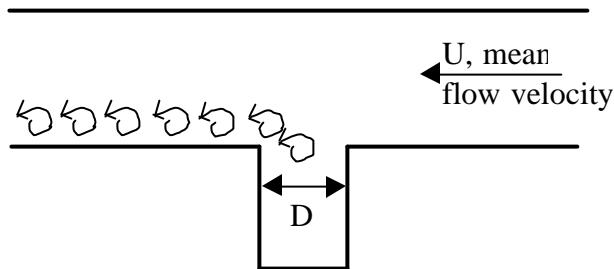
Suction piping with the compressor at left end of Fig. 9. The suction piping showed vibration at the torsional natural frequency between the manway shown in Fig. 10 and the bellows shown in Fig. 9 (at the spots marked with X). That vibration component was not found upstream of the manway or downstream of the bellows expansion joint.

A final test was run with the 450 VFD motor in place. The measured torque was substantially different (see Figure 11). With the smaller driver, the torsional natural frequency was 64 Hz. It was excited by a 6<sup>th</sup> order component from the VFD during the ramp up in speed. The torsional natural frequency was also excited at higher speeds by flow. However, the frequency was not locked in over a wide speed range, and the magnitude was not excessive.



**Figure 11.** Cascade of torque with 450 HP VFD, no surge

Was the flow past the manway responsible for exciting the torsional natural frequency under steady state conditions? Normally, for a significant vortex shedding phenomena to exist, a resonant sidebranch would be required (Figure 12). However, the compressor manufacturer representative cited an instance where flow past a manway, such as exists at this location, caused strong enough vortex shedding to excite a blade natural frequency resulting in failure. The piping vibration at 42.5 Hz only appeared downstream of the inspection manway – this seems consistent with the manway being the cause of the flow disturbance.



**Figure 12.** Flow past a dead leg will shed vortices

Vortex shedding phenomena produces pulsation with a broad range of frequencies. However, the center frequency has the highest amplitude and is given by:

$$F = S_n U / D$$

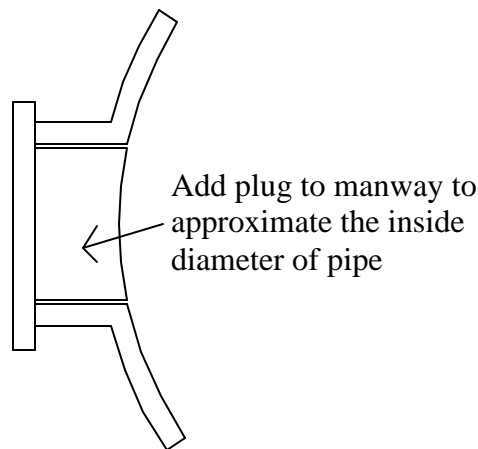
where,  $F$  = frequency (Hz),  $S_n$  = the Strouhal number,  $U$  = average flow velocity (ft/s), and  $D$  = a characteristic dimension (ft). For flow past a dead leg, like the inspection manway, the Strouhal number varies between 0.3 and 0.6. The manway was approximately 1 ft in diameter and the average flow velocity approximately 100 ft/s. The possible range of Strouhal or vortex shedding center frequencies would then be:

$$F = 0.3 \times 100\text{ft/s} / 1\text{ft} = 30 \text{ Hz} \quad \text{to} \quad F = 0.6 \times 100\text{ft/s} / 1\text{ft} = 60 \text{ Hz}$$

So the resulting vortex shedding frequencies are in the approximate range that would excite the torsional natural frequency of the unit. However, the vortex shedding frequency would normally have to match some acoustic natural frequency in the inlet piping to create pulsation of sufficient strength to excite the mechanical or torsional natural frequencies of the compressor. Assuming an approximate speed of sound of 1200 ft/s, at 42.5 Hz the wavelength would be approximately 28 ft. The length of suction piping between the evaporator and the compressor suction could support a  $\frac{1}{2}$  or  $\frac{1}{4}$  wavelength resonance depending on the end conditions. During surge events, flow reversals are likely to strongly excite that organ pipe resonance, producing pressure pulsation at the impeller. These in turn would excite the lightly damped torsional natural frequency.

With the 450 HP motor, the unit's torsional natural frequency was on the flank of the vortex shedding frequencies rather than near the center frequency. The magnitude of the pulsation would be lower than near the center frequency, resulting in a lower magnitude torsional response. With the 450 HP motor, the torsional natural frequency was being excited, but the vibration levels were low enough to allow reliable operation.

In this situation, reducing the excitation forces as well as reducing the response of the system was desirable. Reduction of the excitation could be achieved by avoiding surge and smoothing the flow. A recommendation was made that the control system should be modified to allow startup with out surge. In our opinion, avoiding surge would have the largest positive impact on unit reliability. However, to also reduce any vortex shedding excitation of the torsional natural frequency during normal operation, we recommended that the manway should be modified to accept an internal plug that fills the "dead leg", and creates a smooth inside diameter (see Figure 13). Finally, a recommendation was made to consider installation of a soft coupling to introduce extra torsional damping to the system and to move the torsional natural frequency somewhat.



**Figure 13.** Recommended for inspection manway

### 3. CONCLUSIONS

Surge was definitely the largest contributor to the failures of the coupling bolts. However, with the new 600 HP motor and under steady state conditions at full speed, the torsional natural frequency was still being excited. So what was the forcing function? We are confident that current fluctuations from the VFD were not the source. There was only circumstantial evidence that vortex shedding caused by flow past the manway was the forcing function. The calculated Strouhal frequency for vortex shedding and the length of the suction piping were consistent with an acoustic resonance at 42.5 Hz. Will pressure pulsation in suction piping due to an excited acoustic resonance cause a coincident system torsional natural frequency to respond? It certainly appears so.

#### **4. REFERENCES**

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Jungbauer, D.E., and Eckhardt, L.L., 1997, “Flow-induced Turbocompressor And Piping Noise And Vibration Problems – Identification, Diagnosis and Solutions”, *Proceedings of the 26<sup>th</sup> Turbomachinery Symposium*, Houston, Texas, pp. 79–85.