Dynamic Design Considerations When Modernizing a Pipeline Compressor Station

Presented by

Kenneth E. Atkins
&
Kile S. Watson
Engineering Dynamics Inc.

V. W. Vaughn, Jr.
Vaughn Engineering Consultants

At the

1998 Gas Machinery Conference

October 5th – October 7th
Adam's Mark Hotel
Denver, Colorado

Dynamic Design Considerations When Modernizing a Pipeline Compressor Station

By
Kenneth E. Atkins
and
Kile S. Watson
Engineering Dynamics Incorporated

V. W. Vaughn, Jr.
Vaughn Engineering Consultants

1. Introduction

In 1995, a major U.S. natural gas pipeline company initiated a modernization project for four of its critical main line compressor stations that were originally commissioned in the early 1970's. The modernization included improved emissions (pre-combustion chamber "PCC" technology) as well as other upgrades that increased the speed and therefore compressor capacity at certain locations. A total of five integral engine/compressors from four compressor stations were involved. At one station, two V12 engines were increased in horsepower from 4000 HP at 300 RPM to 5000 HP at 330 RPM. At two other stations, the V12/14 engines were upgraded from 4000 HP at 250 RPM to 6350 HP at 330 RPM, which included adding two power cylinders on each engine. All five engines were retrofitted with the "best available control technology" or BACT PCC upgrade packages from the respective OEMs. These upgrades were very successful from the standpoint of reduced emissions.

The effects of the speed increases are the major focus of this paper. Such changes have major ramifications regarding the dynamics of the system. Pulsation, vibration and performance characteristics, as well as the inertial forces generated by an engine/compressor are significantly affected by speed changes.

This paper discusses the substantial engineering effort undertaken by the pipeline company to ensure safe, reliable and efficient operation. Each of the following topics are discussed:

- The effects of the speed increase on the original pulsation control devices,
- The system modifications required to control pulsation levels,
- The conceptual differences in considering the rigid body shaking forces versus the individual main bearing loads when designing the tie-down system,
- Alternate counterweight configurations, as well as the interaction between counterweight configuration and the torsional response of the engine,
- The relevant issues of crankshaft phasing, cylinder configuration, and rotating and reciprocating balance weight design,
- Engine force-moment calculations as well as predicted and measured results.

2. Acoustical and Mechanical Effects

In this paper, only one compressor station is discussed; however, the problems and considerations involved were common to the upgrade of the entire system. The station under consideration has a single reciprocating compressor. It is a seven throw integral compressor, originally rated at 4000 HP, with 12 power cylinders and 4 compressor cylinders (20.75" bore). The unit was originally commissioned in the early 1970's, and operated satisfactorily at a maximum speed of 250 RPM. In 1995, the unit was upgraded to reduce emissions. This included increasing the maximum speed to 330 RPM, adding two power cylinders and increasing the horsepower to 6350 HP. The station has a complex yard piping configuration for bi-directional flow operation, and for flow into multiple main lines. Many of the problems encountered at this station were typical of those that occurred at the other upgraded stations.

After the speed increase, several acoustically and mechanically related problems were noted:

- Excessive vibration of piping in the downstream metering station began to occur.
- High vibration was noted on piping and valve operators in the yard, as well as on the suction scrubber.
- Noise was emanating from the piping several miles from the station, particularly on the upstream side.

To clearly identify the cause of the problems, several types of field measurements were taken:

- Piping vibration was measured using accelerometers placed at various locations in the yard area and on the compressor.
- Pulsation measurements were taken in the unit bottles, and in the yard piping. In order to locate any acoustical resonances, the measurements were made while the compressor running speed was swept from 250-330 RPM.
- Impact testing was performed on the compressor cylinders as well as on the suction scrubber.

These tests indicated several problems. High pulsation levels at 2x compressor running speed (11 Hz) were measured in the suction and discharge choke tubes (Figures 1 and 2) and in the suction yard piping (Figure 3). The dominant vibration measured in the yard piping was at this same frequency, and appeared to be acoustically excited. In particular, one of the valve operators was vibrating in excess of 70 mils p—p at 11 Hz. Additional pulsation measurements indicated that the dominant pulsation at 2x compressor speed extended to the main suction and discharge headers as well.

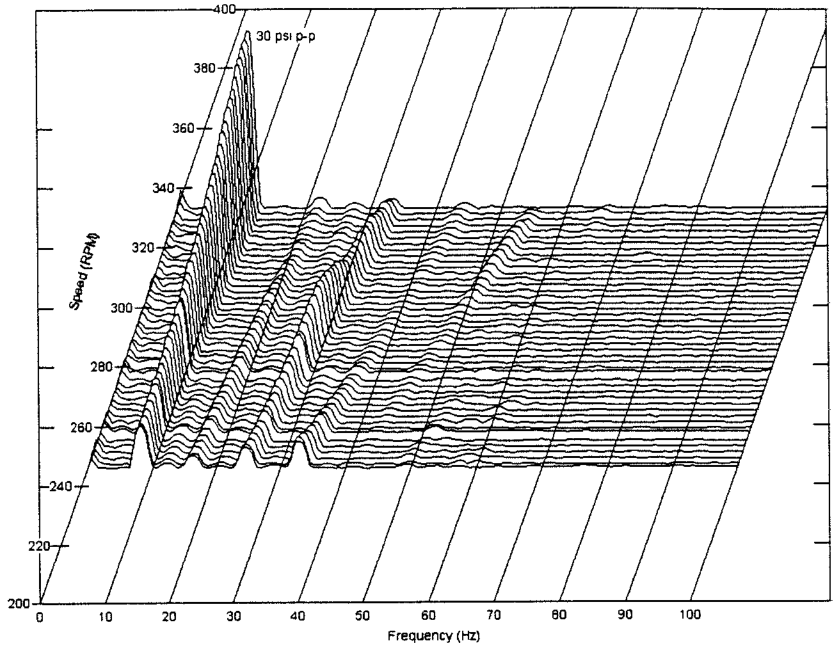


Figure 1 - Measured Pulsation in Suction Choke Tube

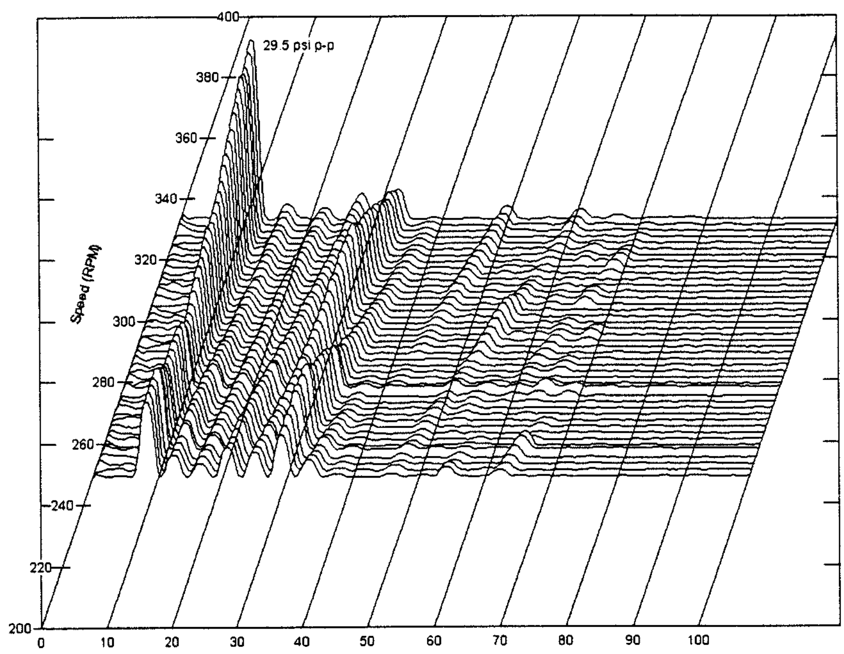


Figure 2 – Measured Pulsation in Discharge Choke Tube

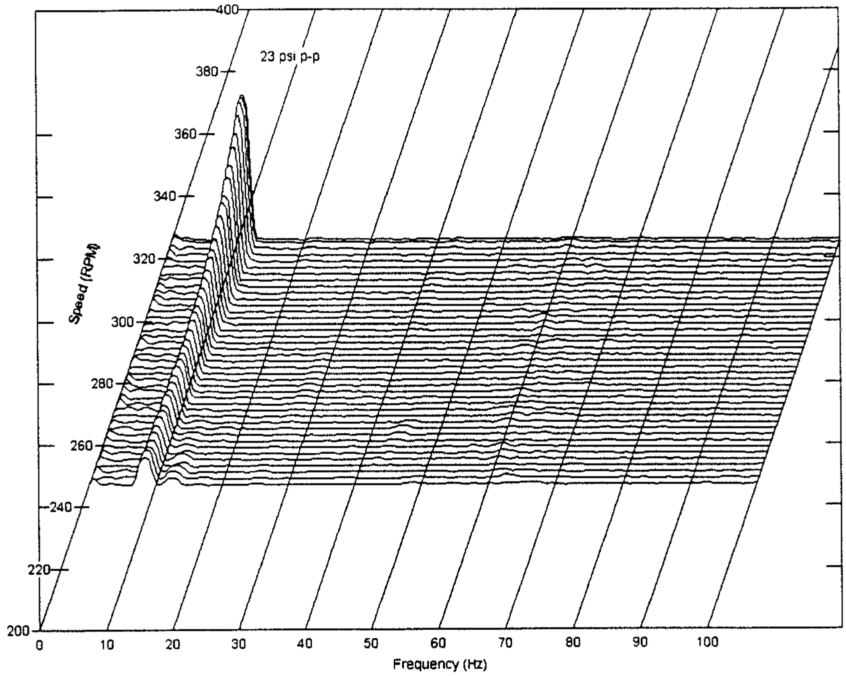


Figure 3 – Measured Pulsation in Suction Header

2.1 Acoustical Analysis

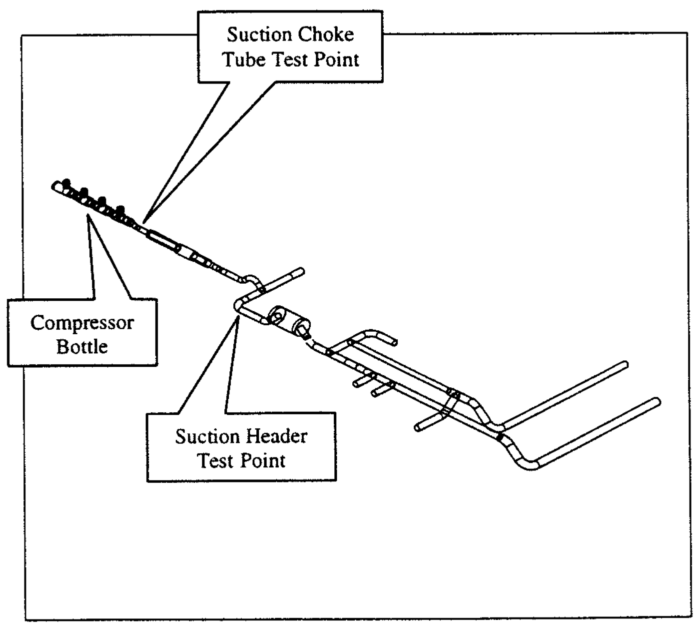


Figure 4 - Suction Acoustic Model

In order to further investigate the problem and develop a solution, a digital acoustical model of the system was created. Figures 4 and 5 are diagrams of the acoustical models.

The existing acoustical design for the station consisted of primary and secondary filter bottles on both the suction and discharge systems that formed an acoustical filter. Acoustical "filters" normally consist of two large volumes connected by a relatively small diameter pipe, or "choke tube". This arrangement is commonly called a volume-choke-volume filter. Such a filter has characteristics such that pulsation above a certain frequency (the so-called filter or Helmholtz frequency) is greatly attenuated. However, pulsation near the filter frequency is amplified. Therefore, it is generally desirable to design acoustic filters with filter frequencies below 1x compressor running speed so that all orders of pulsation are filtered.

The acoustical analysis indicated that the measured 2x pulsation levels were primarily due to the frequencies of the existing two-bottle acoustical filters on both the suction and discharge. The suction and discharge filter frequencies were just above 14 Hz. Since the system was originally designed to run at 250 RPM, these frequencies were originally between 3x and 4x. However, after the speed increase, the filter frequencies were only 30% above 2x, which resulted in an increase in 2x pulsation amplitude by a factor ranging from 2 to 5, particularly in the main suction headers.

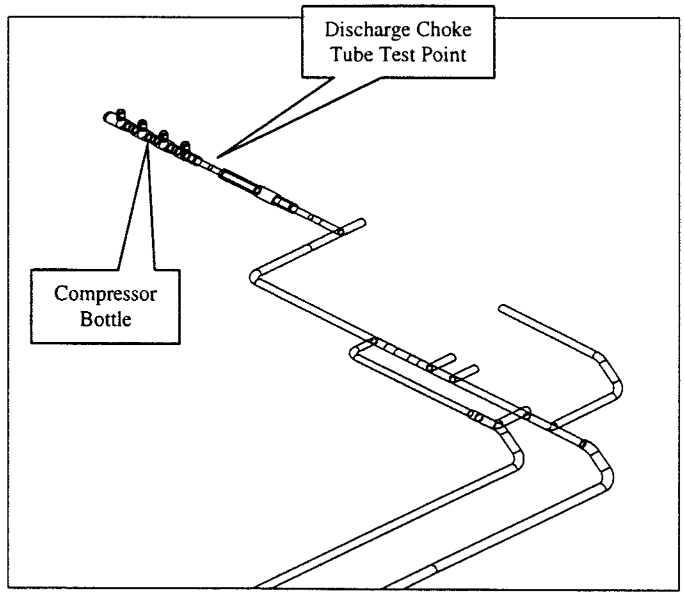


Figure 5 - Discharge Acoustic Model

2.2 Yard Piping Vibration

The measurements indicated that much of the yard piping vibration occurred at a frequency of 2x running speed. The higher acoustic excitation generated by the compressor resulted in increased shaking forces, which resulted in higher vibration levels. In addition, impact measurements of the suction scrubber and several of the valve operators indicated that mechanical resonances in the vicinity of 2x running speed were being excited.

2.3 Upstream Pipeline Noise

The increase in pulsation levels in the main suction headers was the cause of the noise upstream of the station. This noise was characterized as a rumble, or popping noise. In pipeline compressor stations, this type of noise is usually encountered from 1 to 5 miles away, and can come from the suction header, the discharge header, or both. The noise is actually caused by non-linear acoustic effects, which are similar to the effects that cause waves to break on the ocean. These non-linear effects cause the shape of the travelling acoustic wave to change, such that a sharp pressure discontinuity (shock wave) is formed. This shock wave is the cause of the popping sound that is heard coming from the pipe. The non-linear effects are proportional to the amplitude of the pulsation generated by the compressor, and they take some time to develop. That is why the sound is usually heard miles away from the station. The only remedy for this type of problem is to reduce the pulsation amplitude at its source. Damping effects eventually dissipate the pulsation. If the initial pulsation amplitude is kept low enough, the wave will be damped out before it has time to form the shock wave, and no significant noise will be heard.

2.4 Acoustical Design

The root cause of most of the noise and yard vibration at the station was increased pulsation, which was caused by the speed increase. Several system modifications were investigated to reduce the pulsation levels. Normally, it is best to design acoustical filters at the compressor with filter frequencies below 1x running speed. With such a design, the filter frequency does not have to be "tuned" between orders of running speed. The advantage of such a design is that it is generally very effective at controlling pulsation levels.

Unfortunately, installing filters with Helmholz frequencies below 1x running speed was not practical in this case. It would have required larger unit bottles, and there was not sufficient room. Therefore, alternate designs were investigated. The final modifications consisted of the following:

- On the suction side, the header downstream of the scrubber was extended by 40 feet to de-tune an acoustic resonance between the suction filter bottles and the scrubber.
- On the discharge side, a new bottle was added downstream of the existing secondary bottle to form a yard filter.

The acoustical analysis indicated that these modifications would work. However, modifications of this type are generally not optimal, for several reasons:

- Acoustically tuned piping can become a problem if changes are later made to the yard piping. The tuning is designed for one particular piping configuration. If this configuration is significantly changed, the tuner will probably have to be re-designed in order to be effective.
- The addition of yard filters can create additional acoustical resonances between the yard filter bottles and the compressor.

2.5 Results

The recommended acoustical modifications were implemented. The pipeline noise subsided, and the piping vibration was greatly reduced. Several important design considerations were illustrated in this process:

- Acoustical filters are generally designed for a specific operating speed range. Changing this speed range without considering the effects on the system acoustics can have adverse effects on both pulsation and vibration levels.
- When possible, the use of acoustic filters with filter frequencies below 1x is the preferred design. Such designs are less prone to have problems when the yard piping is changed, and generally result in lower pulsation levels.
- If proper acoustic treatment is chosen when the equipment is installed, many vibration problems can be avoided.

3. Engine Dynamics

3.1 Inertial Unbalance and Main Bearing Loads

As previously mentioned, the upgrade project involved a speed increase from 250 to 330 RPM, with a corresponding horsepower increase. The engine has a V14 power section with four 20.75" bore compressor cylinders on one side (Figure 6). The original configuration had no reciprocating balance weights ("dummy" crossheads) but did utilize four rotating counterweights to control the inertial unbalance.

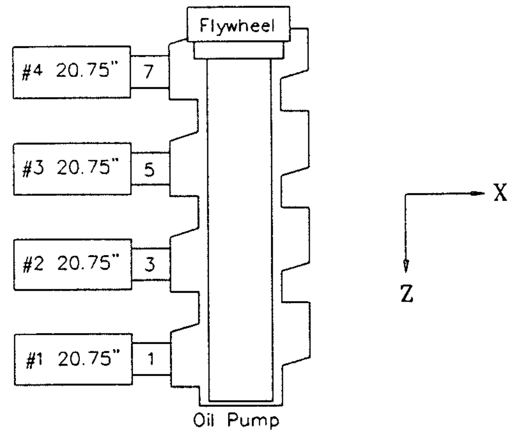


Figure 6 - Compressor Configuration

Table 1 lists the net shaking forces and moments reported by the OEM for the original configuration, based on nominal component weights.

Table 1. Inertial Unbalance Original System 250 RPM

	Primary	Secondary
Horizontal Force (lbs, 0-p)	0	0
Vertical Force (lbs, 0-p)	0	0
Horizontal Moment (lb-ft, 0-p)	248,000	272,000
Vertical Moment (lb-ft, 0-p)	288,000	119,000

The predicted forces were zero and the moments were within the manufacturer's experience limit of 350,000 ft-lbs. Increasing the speed causes these forces and moments to increase by the speed ratio squared. Table 2 lists the net shaking forces and moments at 330 RPM, again for nominal component weights.

Table 2. Inertial Unbalance Original System 330 RPM

	Primary	Secondary
Horizontal Force (lbs, 0-p)	0	0
Vertical Force (lbs, 0-p)	0	0
Horizontal Moment (lb-ft, 0-p)	432,000	474,000
Vertical Moment (lb-ft, 0-p)	502,000	207,000

The calculated forces were still zero; however, the moments now exceeded the 350,000 ft-lb limit. Because these moments were considered too high, the addition of three "dummy crossheads" on the side of the engine opposite the compressor cylinders (Figure 7) was recommended. Note that these reciprocating weights were mounted directly to the frame, which is a fairly common configuration, but includes no additional anchor bolts to the foundation.

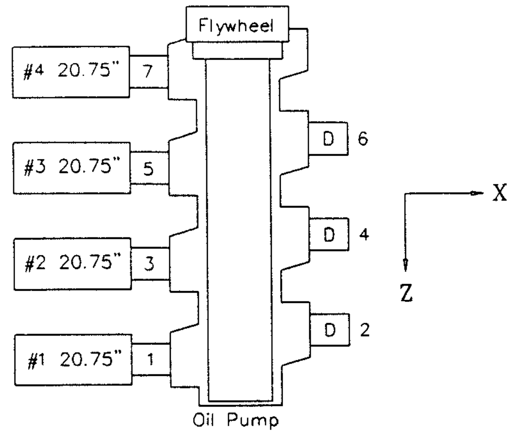


Figure 7 - Modified Compressor Configuration

The resultant inertial unbalance reported by the OEM with the dummy crossheads is listed in Table 3.

Table 3. Inertial Unbalance With Dummy Crossheads 330 RPM

	Primary	Secondary
Horizontal Force (lbs, 0-p)	0	0
Vertical Force (lbs, 0-p)	0	0
Horizontal Moment (lb-ft, 0-p)	80,000	276,000
Vertical Moment (lb-ft, 0-p)	75,000	187,000

The dummy crossheads are effectively reciprocating balance weights. As shown in Table 3, the unbalanced moments were reduced to below the recommended limit by the addition of the dummy crossheads. This appeared to be an acceptable solution.

One noteworthy modification that had been made to all engines many years prior to the upgrade project was to the original tie-down systems. Two-inch "Vibratherm" chock systems with high strength anchor bolts were installed on all of the engines, which performed successfully for many years at the original rated engine speeds. When the upgrade project was implemented, the use of precision "Super Nut" fasteners and stud tensioners was also implemented to carefully control the engine anchor bolt preload. The preload values were also increased.

The modifications were implemented and the unit was re-commissioned. After a short operating period at 330 RPM, several problems developed. The vibration levels of the cylinder and frame were of concern. Severe oil leaks were experienced. The composite chocks slipped at certain locations. Ultimately, both main bearing and crankshaft problems occurred.

The gas pipeline company carefully disassembled and inspected the engine. Weights of all reciprocating and rotating parts were obtained using a precision scale. The resultant inertial unbalance calculated based on the actual weights is listed in Table 4. Note that the forces are not zero due to the variation of component weights, however they were still very low. The moments were significant, but still not above the recommended values.

Table 4. Inertial Unbalance
With Dummy Crossheads
Actual Weights 330 RPM

	Primary	**Secondary**
Horizontal Force (lbs, 0-p)	962	13,065
Vertical Force (lbs, 0-p)	1,354	6,534
Horizontal Moment (lb-ft, 0-p)	195,972	332,591
Vertical Moment (lb-ft, 0-p)	44,832	345,993

The problems resulted from the frame/foundation system not being able to withstand the applied forces. The question is, "Are the forces too high, or is the system under designed?" For large integral compressors such as this, the frame alone is not usually capable of restraining the forces. That is, the frame is not infinitely rigid and the foundation must supply additional stiffness. The effectiveness of the foundation is also very dependent on the tie-down system (anchor bolts, grout, chocks, etc.).

The forces transmitted to the foundation are a function of the frame stiffness as shown in Figure 8. If the frame were infinitely rigid, the foundation would see the net inertial unbalance loads given in Table 4. However, at the other extreme, if the frame is very flexible, the foundation would see loads on the order of the individual main bearing loads. These forces can be significantly higher than the inertial unbalanced forces. It therefore becomes important to consider the main bearing loads when evaluating reciprocating machinery dynamics.

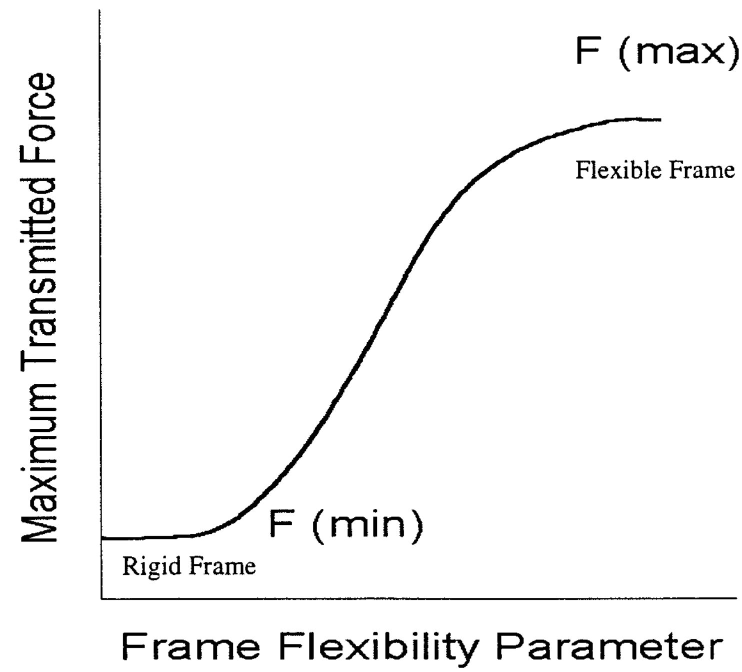


Figure 8 - Frame Flexibility Characteristics

Calculation of the main bearing loads is typically done as a function of crank angle as the engine progresses through a single revolution. All of the mechanical forces from the slider crank mechanisms and the pressure forces from the compressor and power cylinders are considered. Computer programs exist which utilize input from the engine/compressor design (geometry, mass properties, pressure data, gas properties, etc.) and generate all of the necessary information pertinent to engine dynamics such as main bearing loads, torque-effort diagrams, crankpin loads, gas forces, rigid body shaking forces and moments, etc. The main bearing loads are often presented in the form of a force hodograph as shown in Figure 9. This is a plot of the "tip" of the vertical versus horizontal force vector as one crankshaft revolution is traversed. The results are plotted both with and without gas forces, which shows that the inertial forces control the maximum loads for this type of engine.

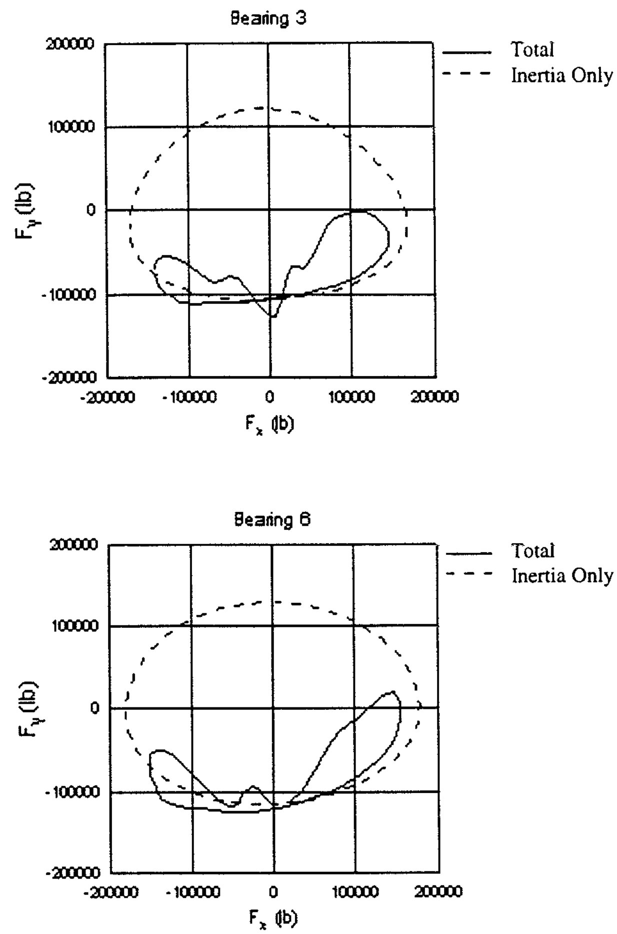


Figure 9 - Main Bearing Loads for Modified System

Maximum main bearing loads are listed for the modified system in Table 5.

Table 5. Maximum Main Bearing Loads Modified System		
Bearing	**250 RPM Force (lbs)**	**330 RPM Force (lbs)**
1	54,500	95,000
2	19,500	34,000
3	97,500	170,000
4	31,000	54,000
5	50,500	88,000
6	101,500	177,000
7	15,000	26,000
8	58,500	102,000

This shows that the main bearing loads nearly doubled as a result of the upgrade. It has been the authors' experience that large integral compressors behave closer to the flexible frame region of Figure 8 than the rigid frame region. Experience also indicates that it is difficult to adequately restrain horizontal dynamic forces larger than about 100,000 lbs (0-p) with typical engine tie-down systems. Therefore, the approach taken to solve this problem was to re-balance the engine to minimize the main bearing loads instead of the net forces and moments.

The main bearing loads are greatly influenced by the crankshaft phasing, the cylinder orientation, and the location of counterweights. The crank phasing and cylinder orientation are shown in Figure 10. This shows why main bearings 3 and 6 have the highest forces, since the crank throws on either side of these bearings are nearly in phase.

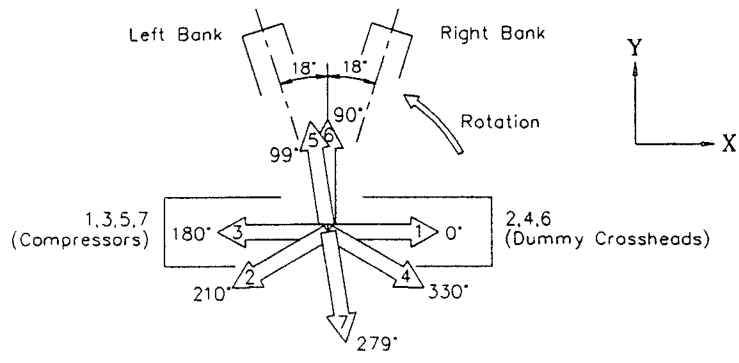


Figure 10 - Crankshaft Phasing

A schematic of the counterweight locations is shown in Figure 11. The original design utilized four rotating counterweights as shown. Minimizing the main bearing loads required a total of 13 rotating counterweights (also shown in Figure 11). The maximum resultant bearing loads with this "super balanced" configuration are compared with the original design in Table 6.

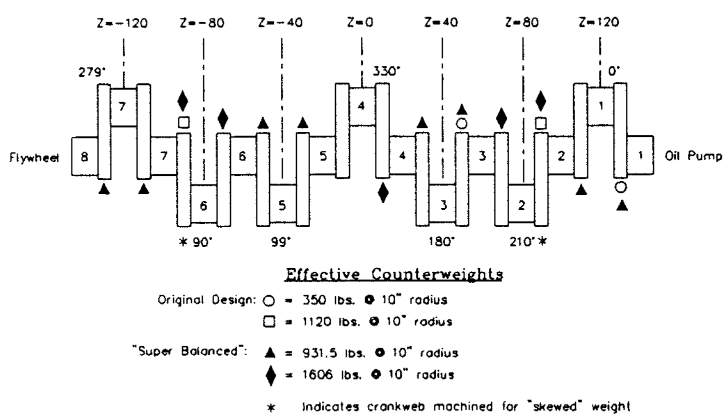


Figure 11 - Counterweight Schematic

Table 6. Comparison of Main Bearing Loads			
Bearing	Original Design		"Super Balanced"
	250 RPM Force (lbs)	330 RPM Force (lbs)	330 RPM Force (lbs)
1	54,500	95,000	74,000
2	19,500	34,000	14,000
3	97,500	170,000	115,000
4	31,000	54,000	39,000
5	50,500	88,000	77,000
6	101,500	177,000	110,000
7	15,000	26,000	19,000
8	58,500	102,000	74,000

As shown, the "super balanced" bearing loads are much lower at 330 RPM, and only about 10-20% higher than the original 250 RPM bearing loads. The net inertial unbalance is listed in Table 7.

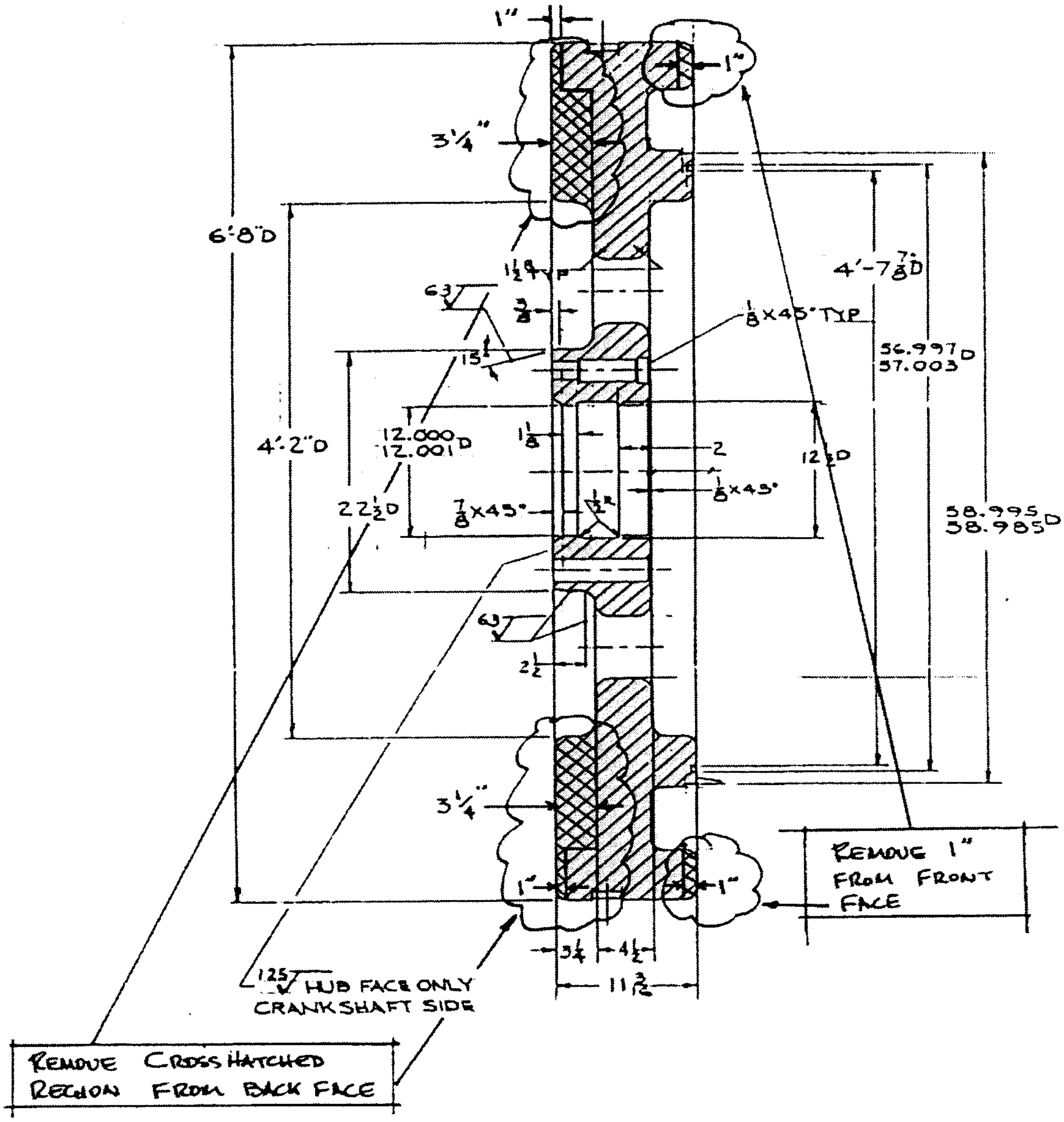
Table 7. Inertial Unbalance "Super Balanced" Condition Actual Weights 330 RPM		
	Primary	Secondary
Horizontal Force (lbs, 0-p)	962	13,065
Vertical Force (lbs, 0-p)	1,354	6,529
Horizontal Moment (lb-ft, 0-p)	104,220	332,591
Vertical Moment (lb-ft, 0-p)	295,817	345,993

3.2 Torsional Response of System

The addition of the rotating counterweights had a significant influence on the torsional natural frequencies of the system. The torsional mass properties of the original and super balanced systems are compared in Table 8.

Table 8. Torsional Mass Properties		
Throw	Original WR^2 (in-lb-sec^2)	Super Balanced WR^2 (in-lb-sec^2)
1	1,738	2,854
2	2,325	3,489
3	1,736	2,852
4	1,598	2,551
5	1,538	2,854
6	2,333	3,497
7	1,543	2,859
Flywheel	23,089	17,221
1st Torsional Natural Frequency	24.9 Hz	24.7 Hz

To maintain approximately the same first torsional natural frequency, the flywheel was trimmed to reduce the WR^2 to compensate for the added WR^2 from the counterweights. This was a fairly simple modification as shown in Figure 12.



	ORIG	MOD 3
CALC. WT:	9475 LBS	7233 LBS
WR²:	62,711 LBS-FT²	46,162 LBS-FT²
MAX SPEED:	358 RPM	

Figure 12 - Flywheel Modifications

3.3 Field Measurements

To verify the acceptability of the super balanced system, field data were compared before and after. The most significant improvement was in the frame and cylinder vibration levels. Figure 13 shows the before and after differential frame vibration at the crankshaft centerline elevation. Maximum differential vibration levels were reduced by more than a factor of two.

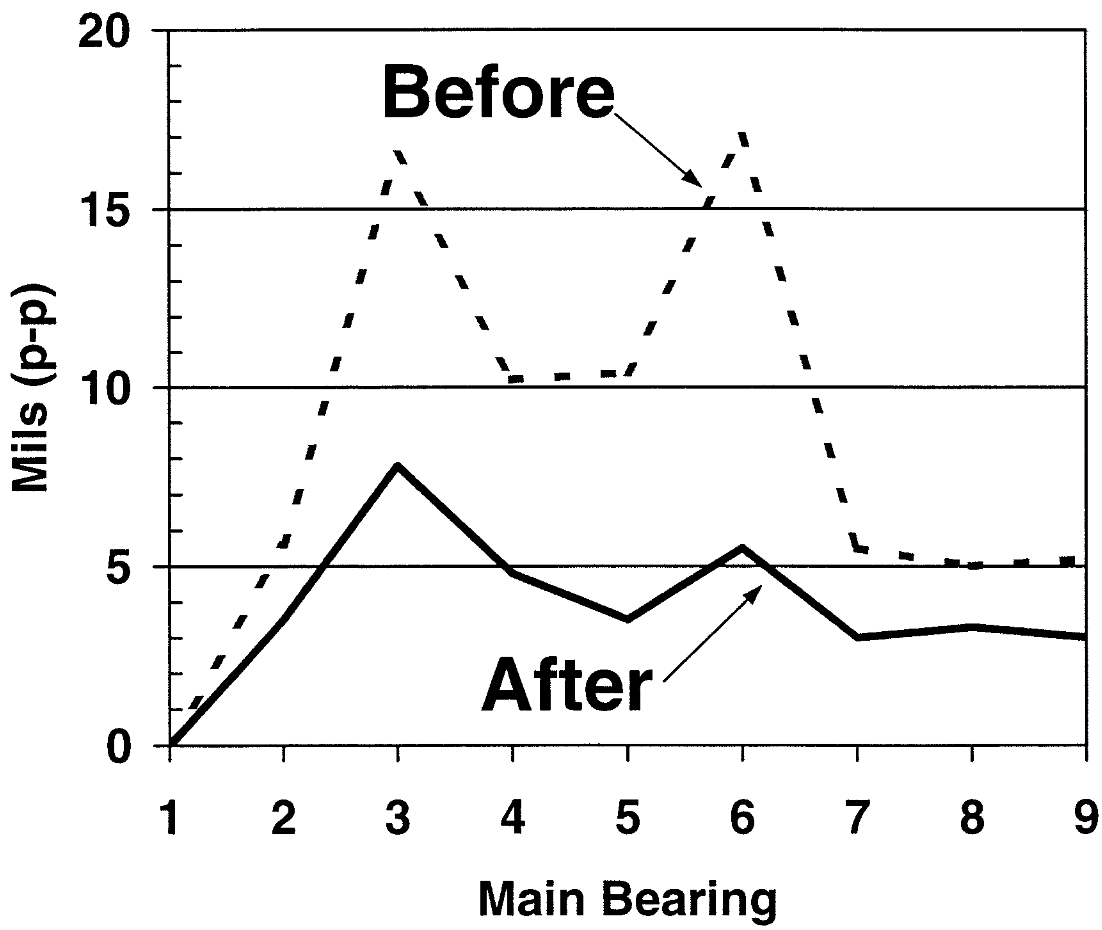


Figure 13 - Frame Differential Vibration

Operating Deflection Shape (ODS) measurements showed similar reductions in frame motion and base-to-grout differential motion (Figure 14). The oil leaks have been considerably reduced and no main bearing problems have recurred.

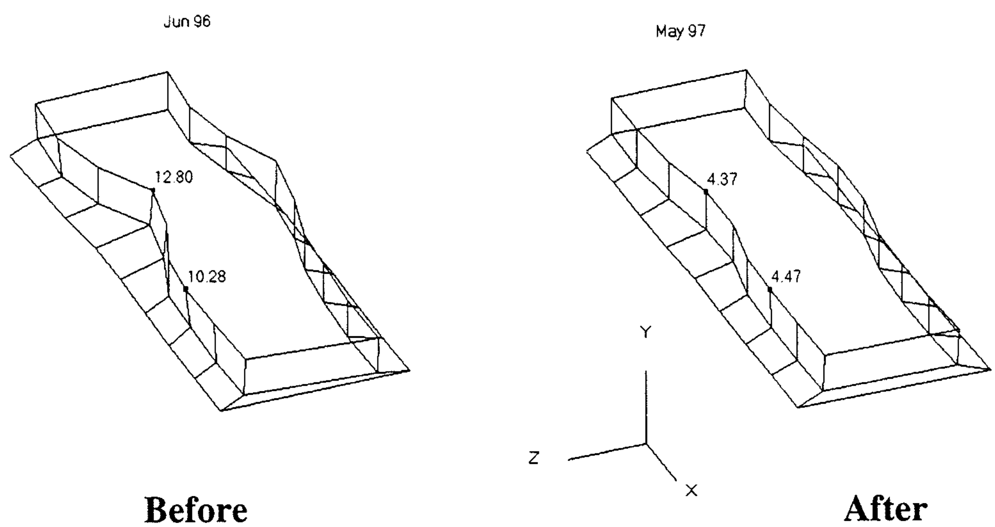


Figure 14 - Comparison of Frame Vibration Levels Before and After Super Balance

The measured torsional response showed that the first torsional natural frequency for the super balanced system is 24.8 Hz, Figure 15. This mode was excited by the 6th harmonic of compressor speed at 248 RPM as shown. This agrees well with the predicted first torsional natural frequency, which was 24.7 Hz. The torsional amplitudes also agreed well as shown in Figure 16. The first order (1X) component deviates the most due to effects from the torsiograph resonance near 3 Hz. However , the 2nd through 4th orders as well as the overall peak-to-peak torsional response showed excellent correlation. It should be noted that this degree of correlation required careful consideration of the exact operating conditions (load step, speed and pressures).

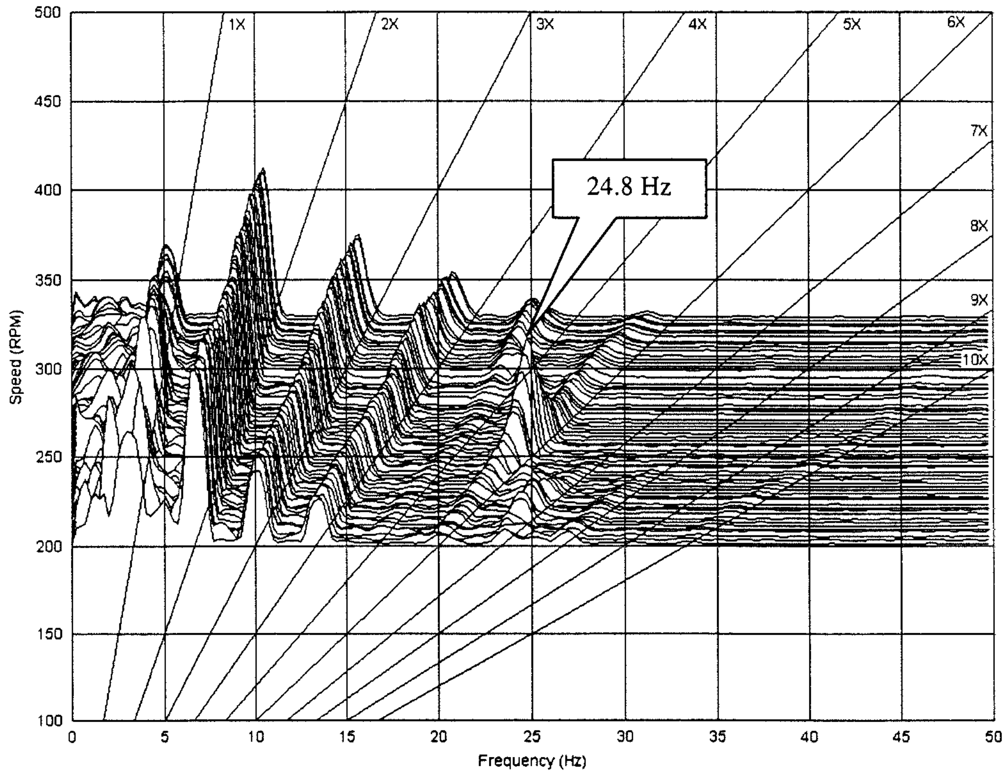


Figure 15 - Measured Torsional Frequency

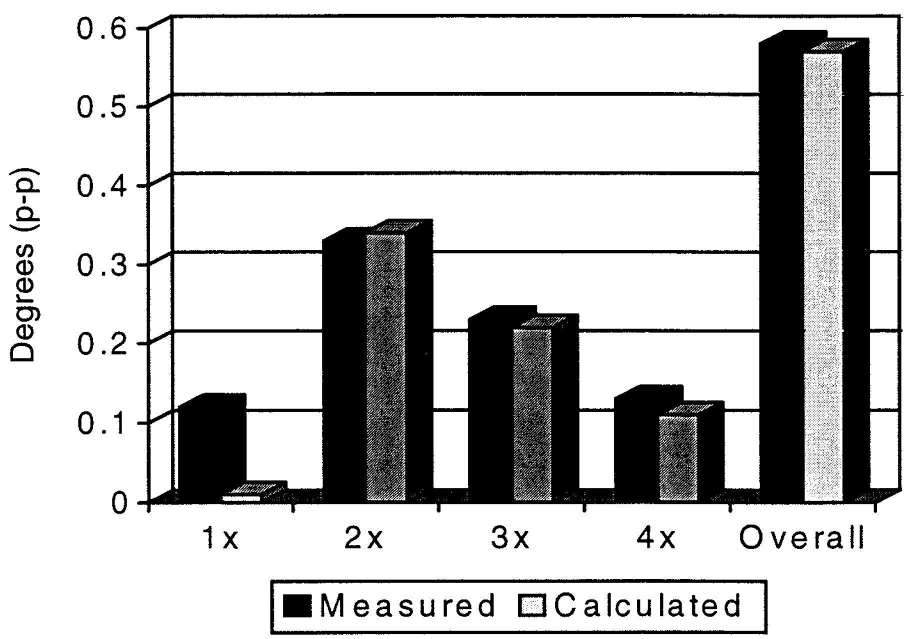


Figure 16 - Comparison of Measured and Calculated Torsional Amplitudes

4. Conclusions

This paper illustrates the need for detailed engineering whenever modifications to compressor stations are undertaken. The major dynamic influence in this case was the speed increase. However, seemingly small modifications, such as extending a header, can also have a significant impact on the pulsation levels at the station and even upstream and downstream in the pipeline.

The importance of evaluating main bearing loads, as well as inertial unbalance forces and moments whenever a speed increase is considered, was also explained. Controlling these forces often involves the use of both rotating and reciprocating counterweights. Re-configuring cylinders and counterweights can also have a significant influence on the torsional response of the system.

The following engineering analyses are recommended as part of any compressor station upgrade or re-rate:

- Acoustical Simulation (API 618 Design Approach 3)
- Evaluation of Main Bearing Loads and Inertial Unbalanced Forces and Moments
- Torsional Analysis (if cylinders or counterweights are re-configured)
- Field verification of tie-down system effectiveness and torsional characteristics

Bibliography

1. Atkins, K. E. and Price, S. M., "Evaluation of Reciprocating Compressor Foundations Using Vibration Measurements," *PCRC 7^{th} Annual Reciprocating Machinery Conference*, Denver, CO, September, 1992.
2. *Vibrations in Reciprocating Machinery and Piping Systems*, EDI Educational Seminar, San Antonio, TX, EDI Report 41450.
3. Smalley, Anthony J., "Dynamic Forces Transmitted by a Compressor to its Foundation," Energy-Sources Technology Conference and Exhibition, ASME Paper 88-ICE-13, New Orleans, LA, January 10-14, 1988.
4. Tison, J. D., "Computer Program RECIP," Proprietary Computer Program for Reciprocating Machinery Dynamics, Engineering Dynamics Inc., San Antonio, TX.