PULSATION & VIBRATION CONTROL FOR SMALL RECIPROCATING COMPRESSORS

N. Sackney *B. Fofonoff*

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

ABSTRACT

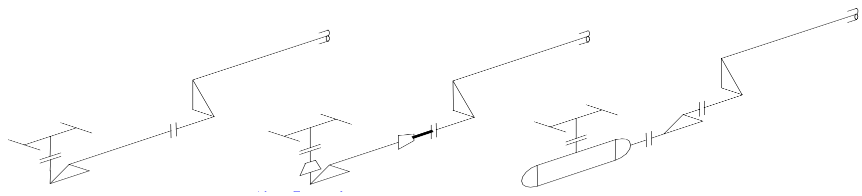
Low horsepower (below 400) reciprocating natural gas compressors normally don't get extensive design consideration when it comes to pulsation and vibration control. However, serious vibration, resulting from extreme pulsation levels, can still be a major concern on this size of unit. Typical industry design strategy on smaller units is to install a line expansion near the compressor cylinder. Field measurements, complemented by computer simulation, have shown that the line expansion can in fact make the situation worse rather than better. If a detailed acoustical design study cannot be conducted in the packaging stage, then the pipe should be sized for flow considerations and the line expansion omitted. The compressor skid and pipe spool design should be such that a small pulsation bottle could be easily added if pulsation is a problem after startup.

1. INTRODUCTION

Reciprocating compressors emit pulsations by virtue of their design. Pulsations travelling away from and to the compressor cylinders will set up standing wave patterns that result in unbalanced pressure forces in the piping system. These unbalanced forces can result in extreme levels of vibration on the compressor and associated piping. For larger units (above 400 hp) digital acoustical design studies are normally used to determine the optimum pulsation control for a given configuration and set of operating conditions. For smaller units, the acoustical design studies are not usually conducted. Instead, pulsation control volume sizes are determined using methods presented in API 618 and API Specification 11P. Typical industry practice is to incorporate the guideline volumes by expanding the piping directly attached to the compressor cylinders. Field data, complemented by computer simulations, indicate that in some cases adding the line expansion will result in increased vibration.

2. PULSATION CONTROL VOLUMES

Typical options for pulsation control in the discharge system of a reciprocating compressor are sketched in Figure 2.1. The piping can be left at the same diameter as the cylinder flange (straight pipe), the line size can be expanded for some length, or a pulsation bottle can be incorporated. The Formulae 2.1 and 2.2 are used in determining volume sizes.



Straight Pipe Line Expansion Small Bottle

Figure 2.1: Typical discharge system options.

Formula 2.1: API 618 4th edition Section 3.9.2.2.2

$$V_s = 7PD\left(\frac{KT_s}{M}\right)^{1/4} \qquad V_d = 1.6\left(\frac{V_s}{R^{1/k}}\right)^{1/4}$$

$$V_s \geq V_d \qquad V_d \leq V_s$$

$$V_s \geq 1 ft^3 \qquad V_d \geq 1 ft^3$$

Where:

V_s = minimum required suction surge volume
K = isentropic compression exponent
T_s = absolute suction temp (kelvin)
M = molecular weight
PD = total net displaced volume/revolution
R = stage pressure ratio at cylinder flanges (P_d/P_s)
V_d = minimum required discharge surge volume

Formula 2.2: API Specification 11P $V = A \times SV$

Where: V = minimum required surge volume
SV = swept volume
A = Multiplier as determined from Figure 2.2

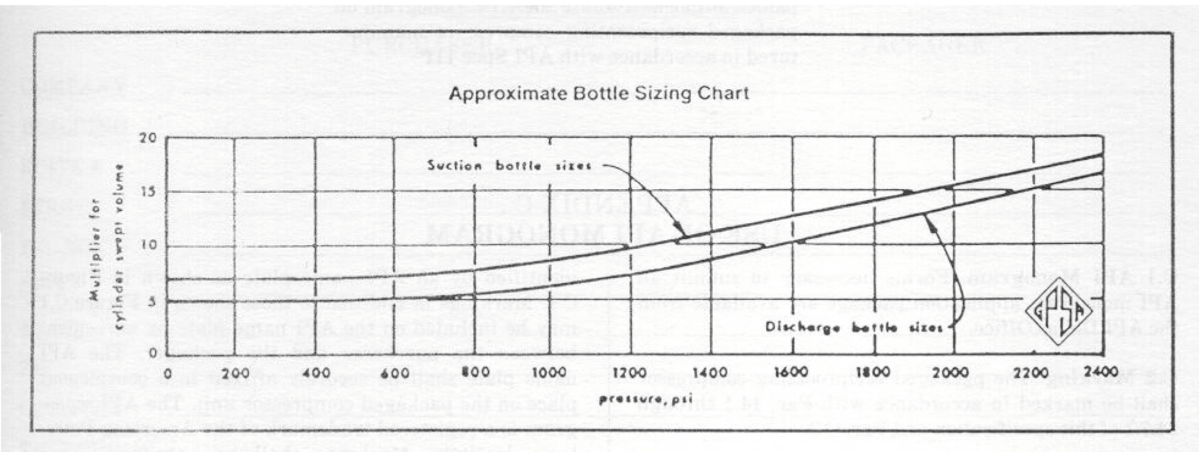


Figure 2.2: API Specification 11P Bottle Sizing Chart

3. FIELD CASE STUDY #1

3.1 Introduction

A 200 horsepower electric motor driver running at 1785 rpm is coupled to a two throw single stage reciprocating compressor. The system is used in a natural gas processing facility as a de-ethanizer compressor. No acoustical design study had been conducted on this unit. The unit was configured without pulsation bottles or line expansions. High vibration had plagued the unit since startup and the owner was considering the installation of line expansions in an attempt to lower pulsation and vibration. A field troubleshooting analysis was conducted to determine the cause of and possible solutions to the vibration problems. It was decided that a digital computer model should be employed to determine the effectiveness of a line expansion prior to making the major modification.

3.2 Field Troubleshooting Analysis

Vibration levels on much of the process piping were found to be above a 1.0 inch per second guideline. One area that exhibited high vibration was the compressor discharge piping between the cylinder and the cooler. Figure 3.1 and 3.2 are spectrum plots of the vibration levels on the discharge pressure safety valve and on the inlet flange to the cooler respectfully. The majority of the high vibration on the discharge system was at a frequency of 59 Hz corresponding to the second order of compressor run speed.

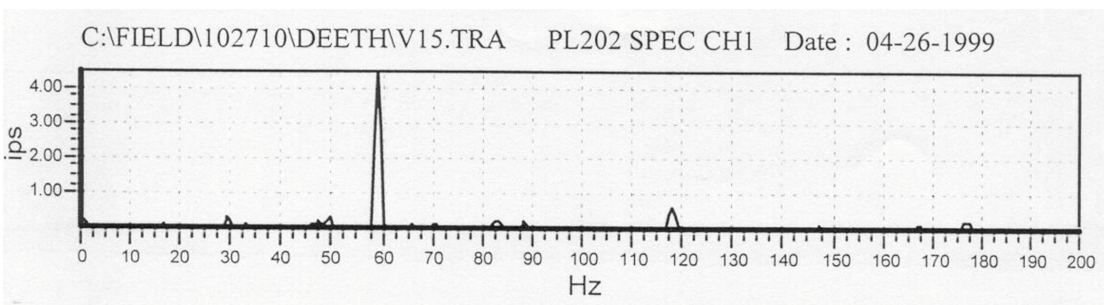


Figure 3.1: Vibration on discharge pressure safety valve.

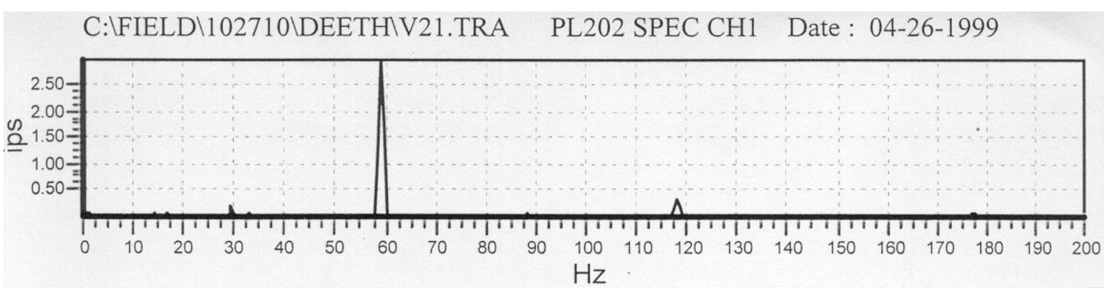


Figure 3.2: Vibration on cooler inlet flange.

3.3 Digital computer simulations

Digital simulations were undertaken and the proposed line expansion was modelled. Figure 3.3 shows the calculated unbalanced forces for the existing system with straight piping. Figure 3.4 shows the calculated forces for the system with a proposed line expansion. Figure 3.5 shows the calculated forces with a proposed small pulsation bottle added to the system. The simulations indicated that if the line expansion had been installed there would have been a significant increase in pulsation forces and vibration. The models showed that installing orifice plates at the cylinder nozzles could sufficiently reduce the system forces. Even further reductions could be achieved by installing a small bottle, however, for this unit that was not deemed necessary.

The continuous line running across each of the unbalanced force plots is used as a guideline to gauge the acceptability of the levels. The guideline is based on line diameter and for some of the plots the guideline level changes between modifications.

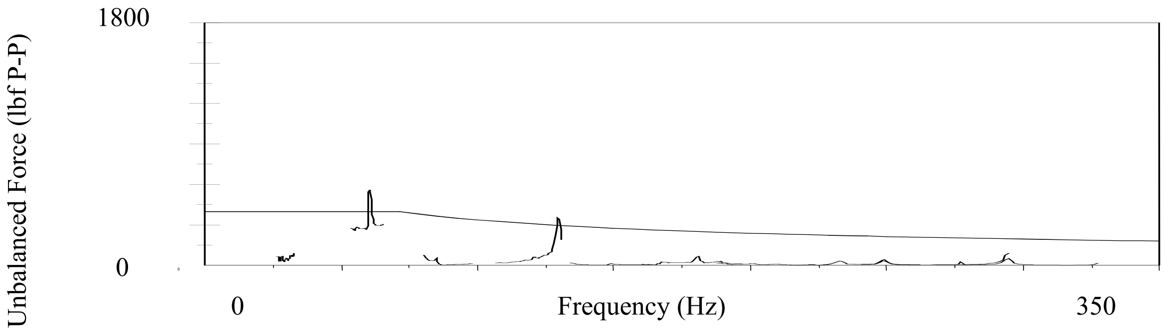


Figure 3.3: Predicted unbalanced forces with the as found discharge system.

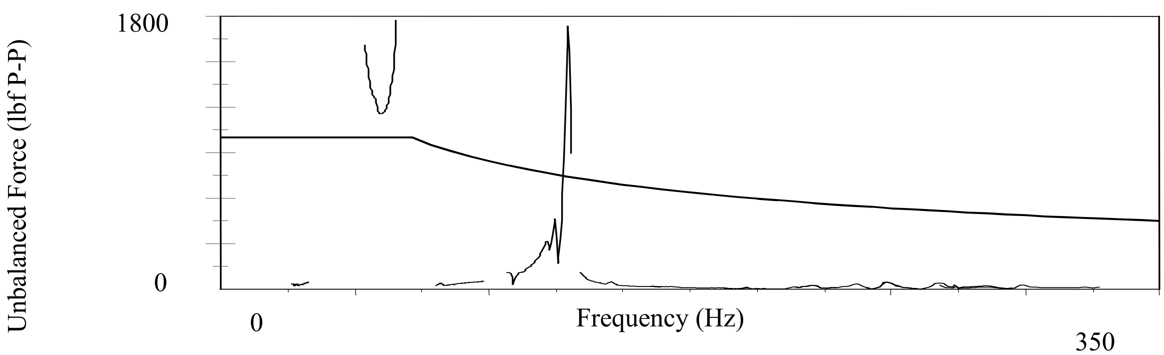


Figure 3.4: Predicted unbalanced forces with proposed line expansion.

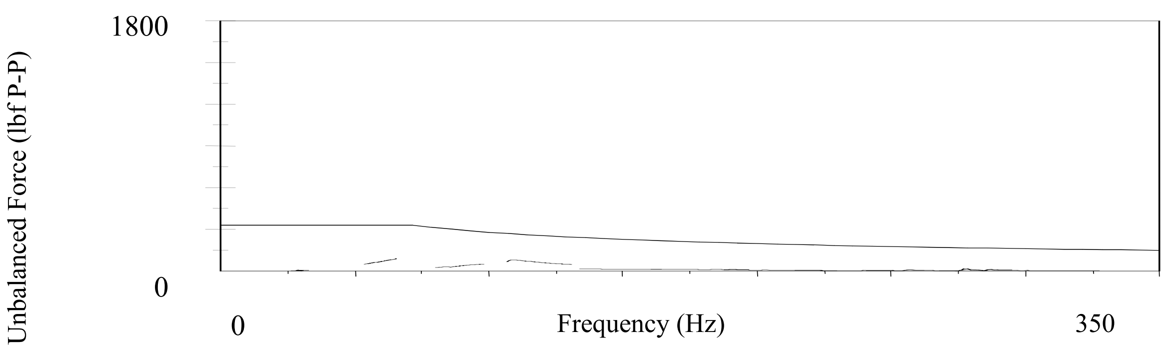


Figure 3.5: Predicted unbalanced forces with proposed pulsation bottle.

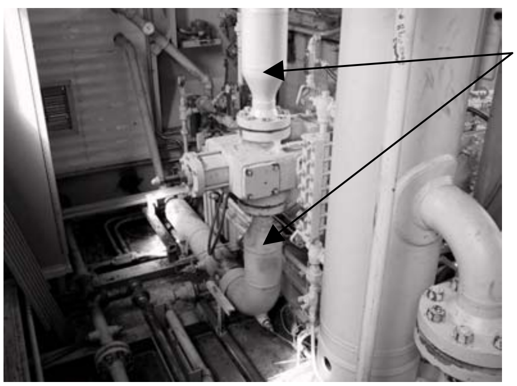
4. FIELD CASE STUDY #2.

4.1 Introduction

Field operators for a major gas producing company were reporting extreme vibration levels on a new reciprocating compressor installation. The subject unit was a single stage two throw compressor frame driven by a 140 horsepower 1800 rpm natural gas engine. No acoustical design study has been conducted on this unit and line expansions were incorporated in the process piping as an attempt at controlling pulsation. A troubleshooting field assessment was conducted to determine the cause of, and possible solutions to the high vibration.

4.2 Field analysis

Vibration levels throughout the package were excessive with levels in the extreme range on the piping and line expansion between one of the compressor cylinders and the gas cooler. Figure 4.1 shows how the line expansion was incorporated into the system.



Compressor cylinder with line expansions from 3" to 6" diameter.

The discharge system line diameter drops back down to 2" on the last leg to the cooler.

Vibration levels were extreme on the discharge lines in this corner of the package.

Figure 4.1: Compressor cylinder and line expansion.

The predominant frequency in the vibration spectra was approximately 120 Hz. This corresponded to the fourth order of compressor run speed. The frequency of the vibration indicated the high vibration was most likely caused by high pulsation-induced unbalanced forces in the piping system. Vibration midway up the discharge riser to the cooler was measured to be as high as 4.0 inches per second peak. An acceptable level would be about 1.0 inches per second at this location. On the final discharge line near this area, vibration levels as high as 8.0 inches per second peak were observed. The spectrum of the final discharge line vibration is shown in Figure 4.2.

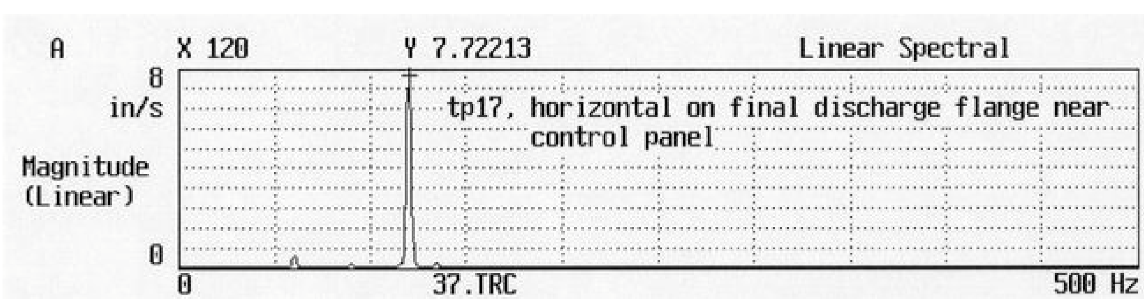


Figure 4.2: Vibration levels on the final discharge line.

4.3 **Digital acoustical models**
To determine an appropriate method of lowering the acoustical unbalanced forces a digital computer model was generated. Figure 4.3 is a plot of the predicted forces in the discharge line with the existing line expansion. Figure 4.4 shows what the forces in this line would have been if the line expansion had not been included in the piping system (i.e. if straight pipe had been used). Figure 4.5 shows how the forces in the system could be minimized if a pulsation dampening bottle design was incorporated.

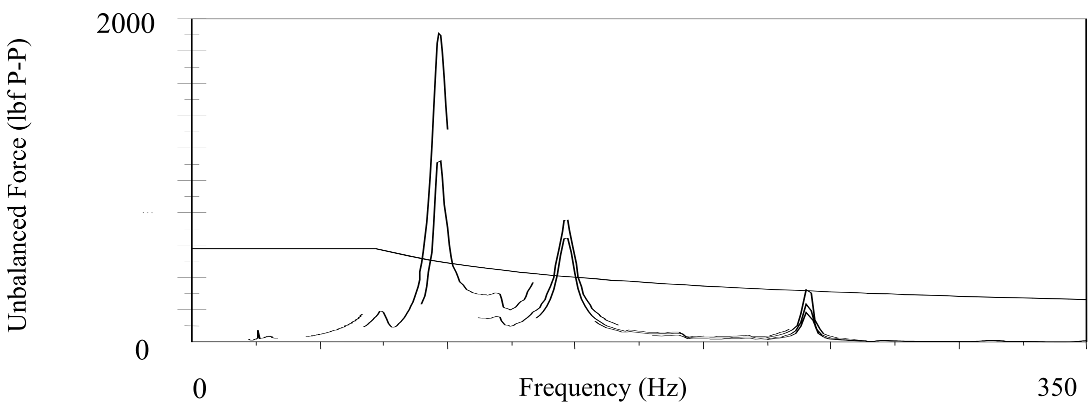


Figure 4.3: Predicted unbalanced forces with as built line expansion.

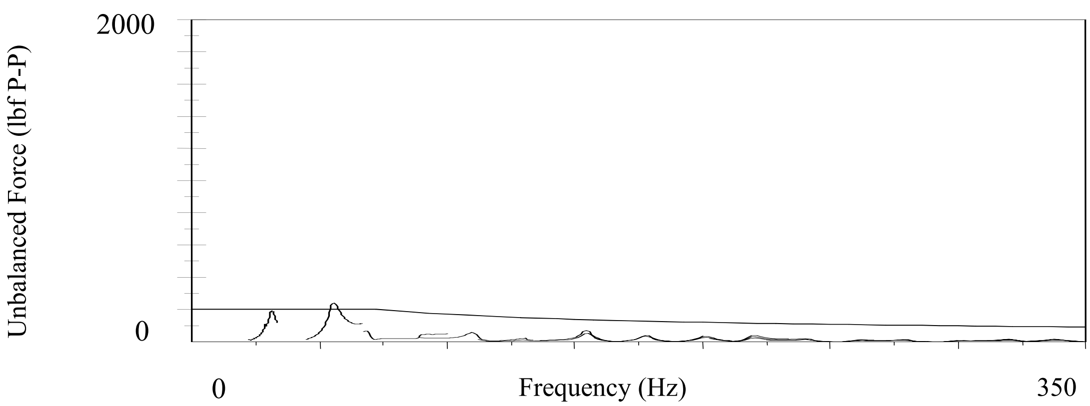


Figure 4.4: Predicted unbalanced forces with straight pipe in discharge system.

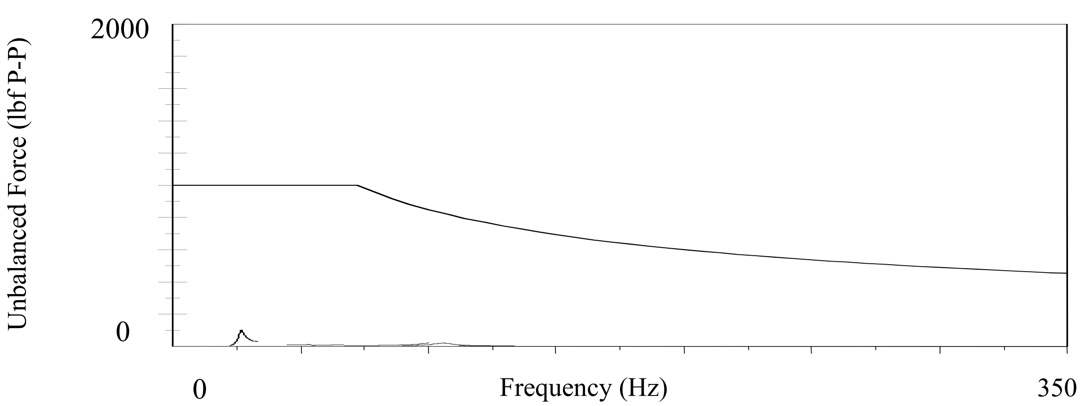


Figure 4.5: Predicted unbalanced forces with optimum size pulsation bottle.

5. DIGITAL SIMULATIONS

5.1 Introduction
To compliment the field case studies, digital models of two typical compressor configurations were generated and evaluated.

5.2 Simulation #1
A 300 horsepower electric motor driver running at approximately1800 RPM coupled to a two throw, two stage reciprocating compressor. Digital simulations were undertaken and several configurations were modelled. Figure 5.1 is a plot of the predicted forces in the first stage discharge line with a line expansion. Figure 5.2 is a plot of the predicted forces in the first stage discharge line with a straight pipe and no orifice plates. Figure 5.3 is a plot of the predicted forces in the first stage discharge line with a straight pipe and with orifice plates. Figure 5.4 is a plot of the predicted forces in the first stage discharge line with a small pulsation bottle.

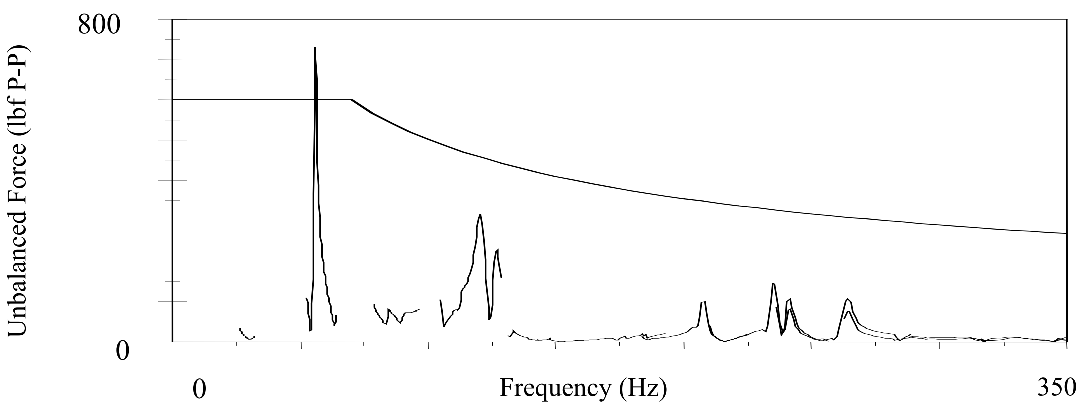


Figure 5.1: Predicted unbalanced forces with line expansion.

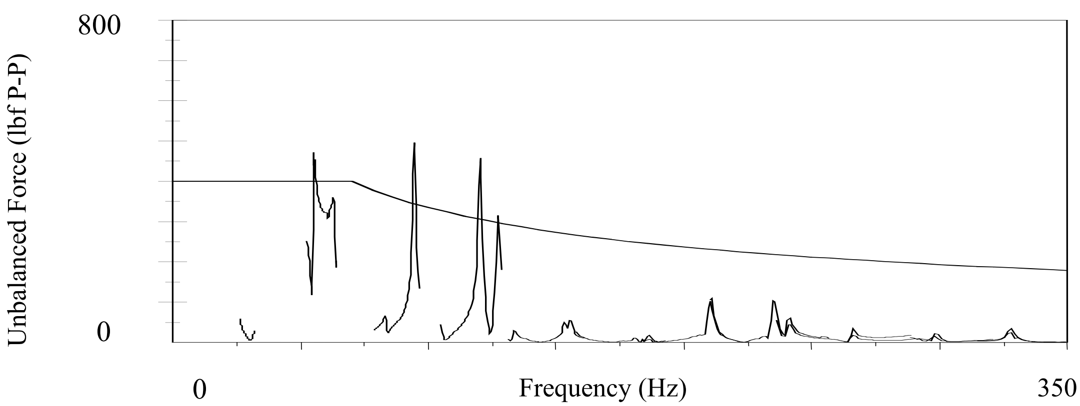


Figure 5.2: Predicted unbalanced forces with straight pipe (no orifice plates).

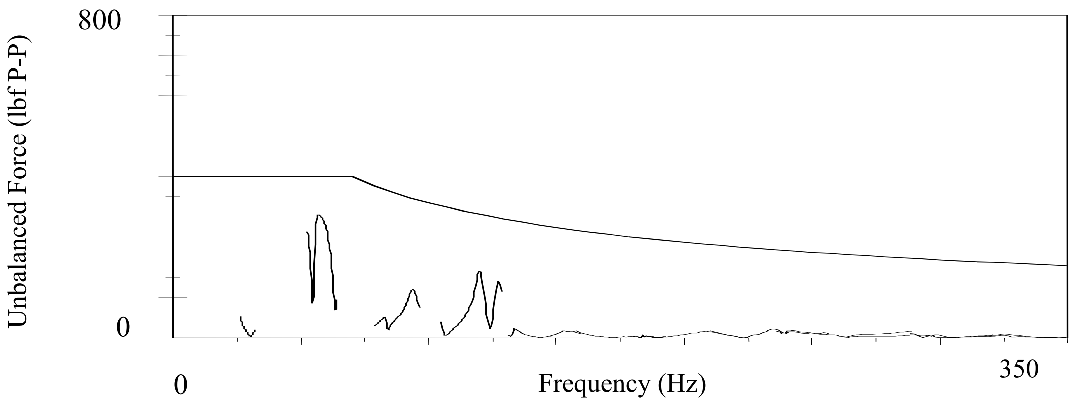


Figure 5.3: Predicted unbalanced forces with straight pipe (with orifice plates).

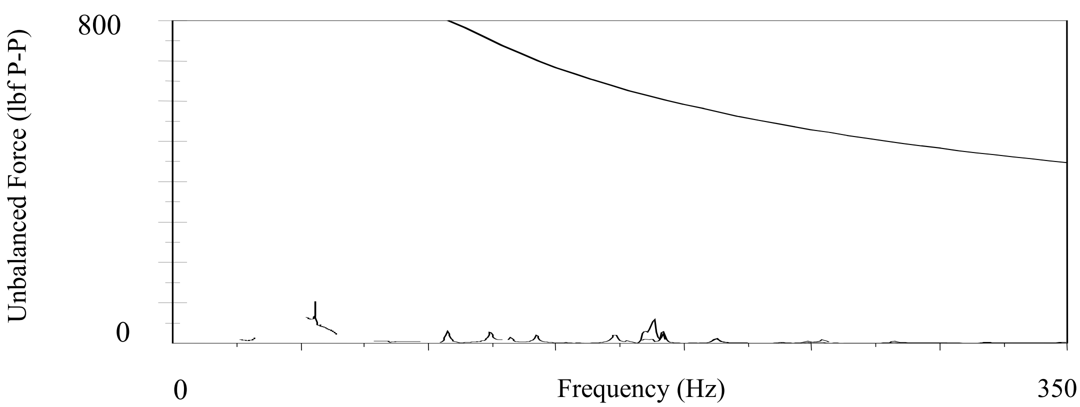


Figure 5.4: Predicted unbalanced forces with small bottle.

5.3 **Simulation #2**

A 360 horsepower electric motor driver running at approximately1800 RPM coupled to a two-throw two stage, reciprocating compressor. Digital simulations were undertaken and several configurations were modelled. Figure 5.5 is a plot of the predicted forces in the first stage discharge line with a line expansion. Figure 5.6 is a plot of the predicted forces in the first stage discharge line with a straight pipe and no orifice plates. Figure 5.7 is a plot of the predicted forces in the first stage discharge line with a straight pipe and with orifice plates. Figure 5.8 is a plot of the predicted forces in the first stage discharge line with a small pulsation bottle.

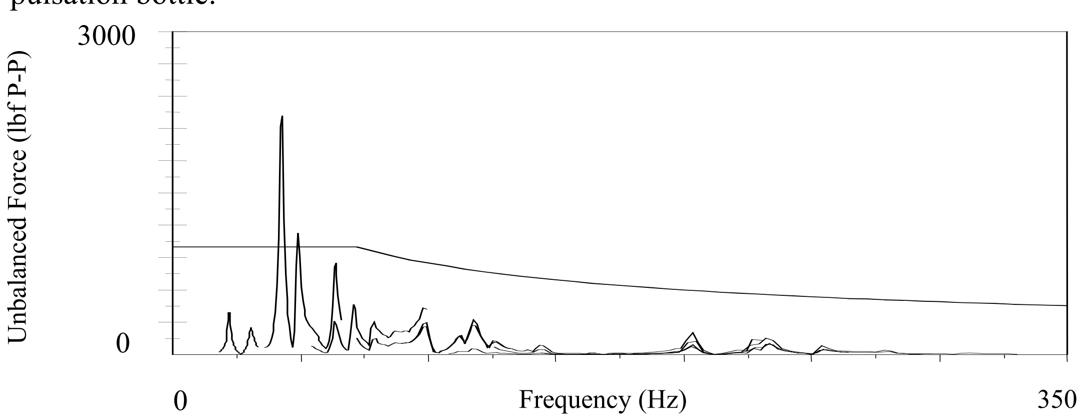


Figure 5.5: Predicted unbalanced forces with line expansion.

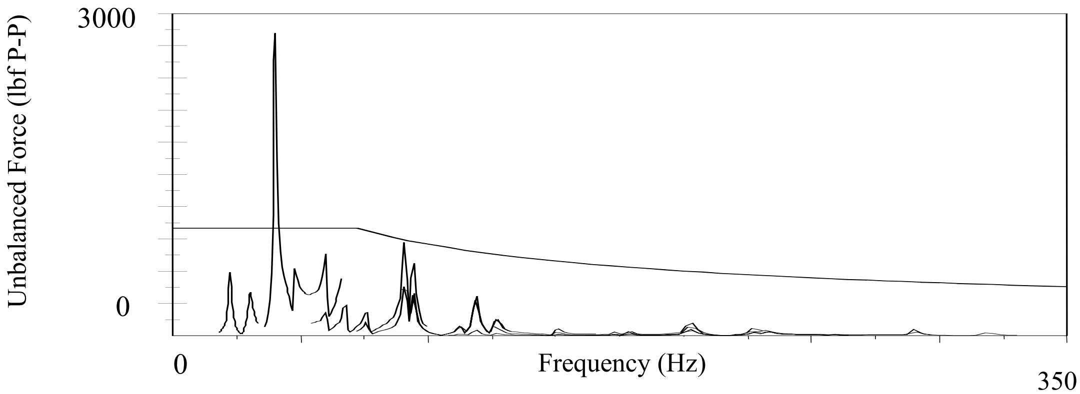


Figure 5.6: Predicted unbalanced forces with straight pipe (no orifice plates).

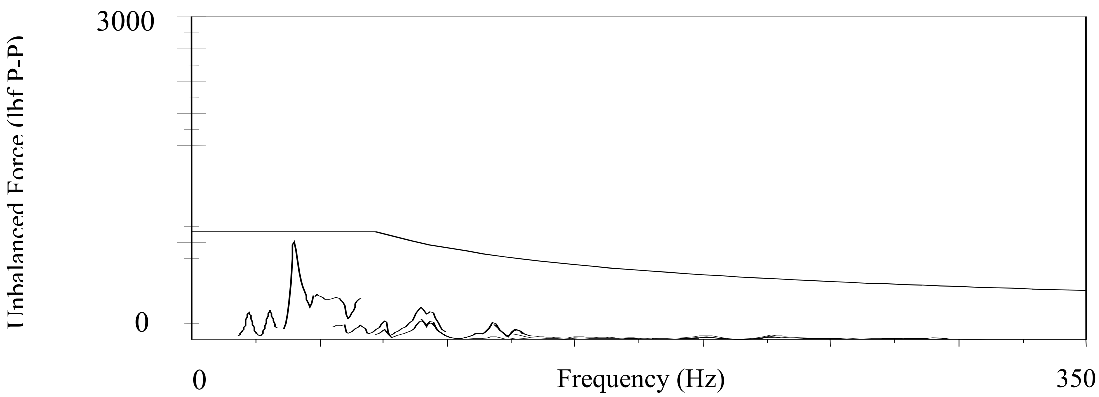


Figure 5.7: Predicted unbalanced forces with straight pipe (with orifice plates).

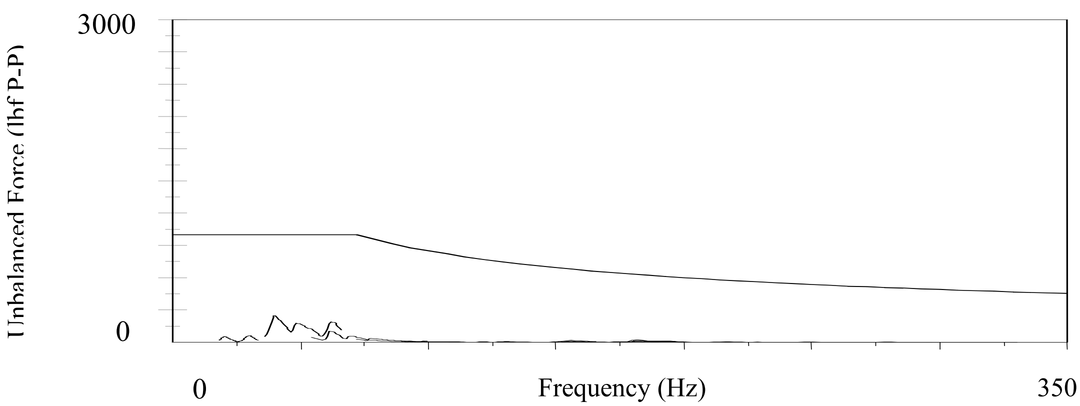


Figure 5.8: Predicted unbalanced forces with small bottle.

6. CONCLUSIONS

For the two field cases studied, the units saw, or would have seen, higher acoustical unbalanced forces with the installation of a line expansion as compared to straight pipe sized for flow considerations. For the digital simulations conducted, the line expansion did not provide adequate control over acoustical unbalanced forces. The best solution always included an appropriately sized pulsation bottle. Good end results were achieved by starting with straight pipe and including mild restricting orifice plates in the system.

In reciprocating compressor installations under 400 horsepower, when an acoustical design study is not being considered, the line expansions should be foregone. The skid and piping design should allow for the easy installation of a pulsation bottle should startup pulsation and vibration levels indicate better pulsation control is required.

Further design studies should be conducted to determine if the range of operating conditions and, or the size of the compressor to which these conclusions apply could be narrowed.

7. ACKNOWLEDGEMENTS

Digital acoustical simulations were conducted using proprietary software (MAPAK) developed and refined by Beta Machinery Analysis over the past 30 years.

Field data were collected with two channel spectrum analyzers. A Diagnostic Instrument model 2200 and a HP 3560A were employed using Wilcoxon velocity transducers.

8. REFERENCES

(1) Specification for Packaged Reciprocating Compressors for Oil and Gas Production Services. *API Specification 11P (Spec 11P), Second Edition, November 1, 1989.*

(2) Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services. *API Standard 618, Fourth Edition, June 1995.*

9. AUTHOR BIOGRAPHIES

The authors are both employed by Beta Machinery Analysis Ltd. Nolan Sackney obtained his Bachelor of Engineering degree from the University of Victoria in 2000. Nolan has spent the last 1½ years performing computer simulations for designing pulsation control on reciprocating compressor packages.

Bryan Fofonoff obtained his Bachelor of Science degree in mechanical engineering from the University of Saskatchewan in Saskatoon in 1991. Bryan's first five years with Beta were spent doing computer simulations designing pulsation control on reciprocating compressor packages. For the last six years Bryan has been full time troubleshooting pulsation and vibration issues on existing compressor installations.