THE COST OF

DYNAMIC PRESSURE DROP

BY:

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SEMINAR ON ACOUSTIC VIBRATIONS
AND
PRESSURE PULSATIONS IN ROTATING MACHINERY

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ABSTRACT

One of the least recognized reasons for performing an acoustical pulsation design analysis on reciprocating compressor piping systems is a phenomenon known as dynamic pressure drop. Oscillating flow causes dynamic pressure drop. Average flow causes static pressure drop. Dynamic pressure drop is most significant when there is a large oscillating flow at a discrete flow resistance, such as an entrance or exit nozzle on a vessel. The increase in pressure drop results in a decrease in compressor capacity or an increase in required power. Reducing the magnitude of the oscillating component of the flow is key to minimizing the pressure drop..

An acoustical pulsation design analysis can be justified for some installations solely by the increase in production from the compressor that results from lower system pressure drop. In practice, pressure pulsations and unbalanced forces due to t he pressure pulsations are usually high when flow fluctuations are high. Discrete pressure drops, introduced into the system as part of a pulsation control design, may produce a net saving in overall system pressure drop.

Introduction

Pressure pulsations in industrial equipment have been recognized as a problem for decades. Modelling of systems has been done on passive analog computers since the early 1950’s. The emphasis in the past has been on minimizing pressure pulsations and piping vibrations in reciprocating compressor and pump piping systems. Only in recent years, however, has the significance of flow pulsations been generally recognized.

Beta Machinery Analysis Ltd. introduced MAPAK, the first practical digital computer program for pulsation design, in the mid 1970’s. The development of this computer program was done by Dr. Bryan Long and Brian Howes. A large number of problems have been modelled over the intervening years by the company’s engineers. Some of the highlights are discussed below.

Several examples of actual problems are included in this paper to illustrate the significance of pulsating flow.

The Range of Problems Caused by Pulsations

Many different types of plane-wave pulsation problems can be experienced in industrial applications. The term “pulsation” refers to both pressure and flow. The range of possible problems is demonstrated by the following partial list of different types of machines, fluids, and systems that have been analyzed using MAPAK.

- reciprocating compressors
- screw compressors
- vane compressors
- reciprocating pumps
- centrifugal pumps
- complex systems
  - gas transmission compressor stations
  - multi-cylinder compressors
  - multi-plunger pumps
  - multi-unit installations
- flow-induced (Strouhal) problems
- high pressures (over 35MPa)

- low pressures (atmospheric)
- many gases - CO2, CO, H2S, natural gas, air, hydrogen, ammonia, etc.
- many liquids - water, amine, glycol, miscible fluids, ethane, hot oil, etc.
- metering problems caused by pressure pulsations and oscillating flow at orifice meters
- pressure drop problems caused by oscillating flow

Most of the problems alluded to in the above list involved concerns with piping vibrations. This major class of problem is not discussed in the present paper. Refer to References 1, 2 and 3 for more discussion of vibration problems.

Oscillating Flow and Its Effects

The main effect of oscillating flow is to increase pressure drop in a piping system. Metering error, and effects on centrifugal pumps and compressors are also areas of concern.

Understanding pressure drops caused by the flow of fluids in piping systems starts with an understanding of the Bernoulli Equation [4]. The dynamic plus static heads equal the stagnation head. The dynamic term, referred to as the velocity head ($\frac{1}{2}V^2$), is used in the classic pressure drop equation, Equation 1.

$$DP = \tfrac{1}{2}\rho V^2 K \qquad (1)$$

where ρ = fluid density
V = fluid velocity
K = a constant dependent on piping geometry, a summation of flow resistances for discrete and distributed properties of the piping
and dP = pressure drop

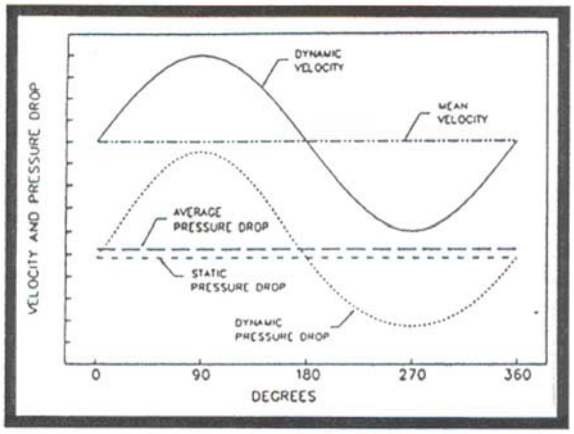
Pressure drop in a piping system is caused by many discrete resistances, such as changes in section, and distributed resistance (review, for example, the Darcy-Weisbach equation [4]).

Reciprocating compressors require more power to compress gas when system pressure drop increases. Alternatively, when the compressor driver is at full load, the addition of pressure drop results in a lower capacity for the system. In either event, there is a cost when pressure drop is increased.

In the case of plunger pumps, increased pressure drop on the suction side leads to less NPSH available. A recent discussion by Vandell et al. (1993) [5] deals with some of the pressure drop issues with pumps.

The pressure drop along a piping system increases when an oscillating flow term is added to the mean flow term. Refer to Figure 1 for a graphical representation of the effect of pulsating flow on pressure drop.

Figure 1: Oscillating Flow and Pressure Drop



The total flow at a point in the piping can be written in one of the forms shown below.

For a single frequency:

$$v = V_0 + V'\sin(2\pi wt) \qquad (2a)$$

For the general case:

$$V = V_0 + V(t) \qquad (2b)$$

where V_0 = mean flow,
V' = amplitude of the oscillating flow,
w = frequency
t = time
$V(t)$ = oscillating flow as a function of time.

Then, total pressure drop for a discrete flow resistance becomes:

$$dP = \tfrac{1}{2}\rho\,(V_0^2 + 2V_0V(t) + V(t)^2)\,K \qquad (3)$$

The dynamic pressure drop from Equation 3 can be written as

$$dP = \tfrac{1}{2}\rho\,(2V_0V(t) + V(t)^2)\,K \qquad (4)$$

Assuming that the oscillating term is a sinusoidal variation at one frequency (Equation 2a), then the dynamic pressure drop at a discrete resistance for one period of the sine wave can be written, after integration of Equation 4, as

$$dP = \tfrac{1}{2}\rho\,(\tfrac{1}{2}V'^2)\,K \qquad (5)$$

Equation 5 represents the average pressure drop across a discrete resistance caused by a single frequency oscillating flow. The instantaneous pressure drop for a discrete resistance will, of course, vary as the square of the instantaneous velocity. It can be shown that the pressure drop caused by more than one frequency of oscillating flow is a summation of terms like Equation 5.

Similarly, the pressure drop for a distributed resistance will vary both with time and distance. The standing wave nature of pulsations in piping systems causes the amplitude of the pulsating flow to vary down the length of the pipe, as well as with frequency. The average pressure drop, due to distributed resistance, summed along the pipe length depends on the number of pulsation wavelengths in the system at each frequency. Helpfully, in practice, the discrete resistances are usually more significant than the distributed resistances, in terms of causing pressure drop. A reasonable first approximation of the dynamic pressure drop term can be had from a summation of Equation 5 over a few harmonics of the speed, for all of the discrete resistances in the system.

Logically, a system with less oscillating flow amplitude will have less pressure drop, all other things being equal.

In summary, the oscillating flow caused by pulsations in a piping system plus the mean flow result in more pressure drop in a system than caused y the average flow. It can happen that adding pressure drop in the form of discrete pulsation suppression devices to reduce pulsations, rather than causing an increase in system pressure drop, can reduce system pressure drop.

Orifice meters are affected by oscillating flows. The measured capacity is assumed to be proportional to the pressure drop across the orifice plate. The average total pressure drop is greater than the mean flow pressure drop by the dynamic pressure drop as discussed above.

The effects of pulsations on centrifugal compressor or pump performance is beyond the scope of this paper. Suffice to say that there can be a strong interaction between reciprocating and centrifugal machines, or even between centrifugal machines and acoustical resonances in piping systems.

Case Histories

History 1

This case study is taken from a job at the Gardiner Oil and Gas compressor installation at Dimsdale, Alta. There were excessive pulsations and vibrations at this installation when the field engineer arrived on a trouble shooting job.

The recommendation that came out of the field visit was that acoustical modelling be done to design pulsation control devices. From the acoustical model it is possible to determine the total pressure drop that a system is experiencing. In addition, RMS meter error due to the oscillating flow can be predicted.

Figures 2 through 5 show the predicted system pressure drops for the “as found” and modified systems, first versus the system’s length and then versus the speed range of the engine. The system for this case begins just after the cylinder flange and ends just after the metering orifice, before the discharge control valve. Figure 2 and 3 show that the dynamic pressure drop has decreased to a negligible amount in the modified system. This decrease in dynamic pressure drop is due to pulsation control devices inserted in the system. The net result is that incorporating pulsation control devices resulted in a substantial decrease in total system pressure drop.
Figure 4 and 5 show the effect of the oscillating flow throughout the speed range of the compressor. This shows that the top of the speed range is the area where the dynamic flow creates the largest effect on the as found system. For the modified system, Figure 5 shows this problem has been controlled.

Figure 6 and 7 show the initial and adjusted flow comparisons between predicted and measured values. We predict flow based on GPSA standards which need to make assumptions on the general health of any given compressor. When the first comparison was completed the result was that the predicted flow in the area of 800 – 900 r/min was high. This was not expected as it was known from Figure 4 that the oscillating flow at 800 – 900 r/min was very low, thus there should be little error. The fact that the shape of the measured flow curve was curved upward also agreed with Figure 4. That is, very little error at the low speeds and gradually increasing to a large error near the top of run speed, with a “hiccup” at 950 r/min. For these reasons, we inferred that the compressor must not be running at an average condition, and is instead running in below average condition. The predicted flow was adjusted and the result is shown in Figure 7.

Figure 8 shows the predicted RMS metering error based on the computer model while Figure 9 shows the total metering error based on the measured flow. The measured error is larger than the predicted RMS error. The difference is due, in part, to an unpredictable error caused by instability of the vena contracta at RMS errors great then 0.5%.

Figure 2: **Static and Dynamic Pressure Drop for the As Found System at 1200 r/min**

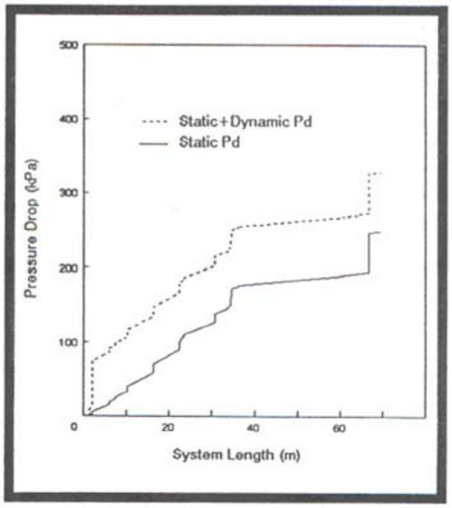


Figure 3: Static and Dynamic Pressure Drop for the Modified System at 1200 r/min

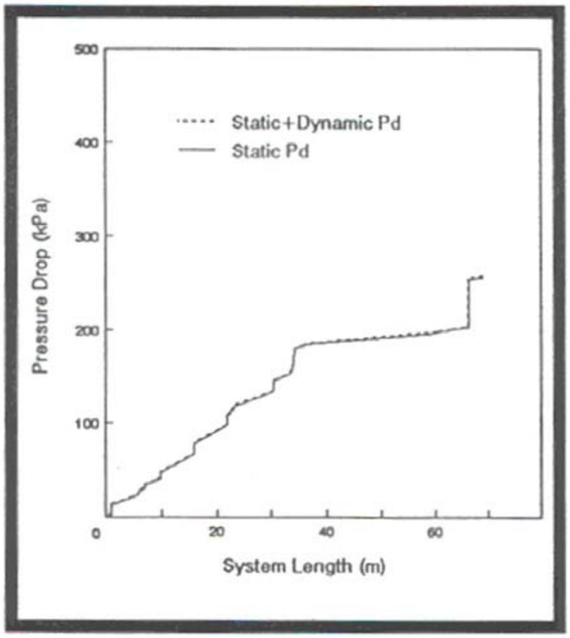


Figure 4: As Found System Pressure Drop

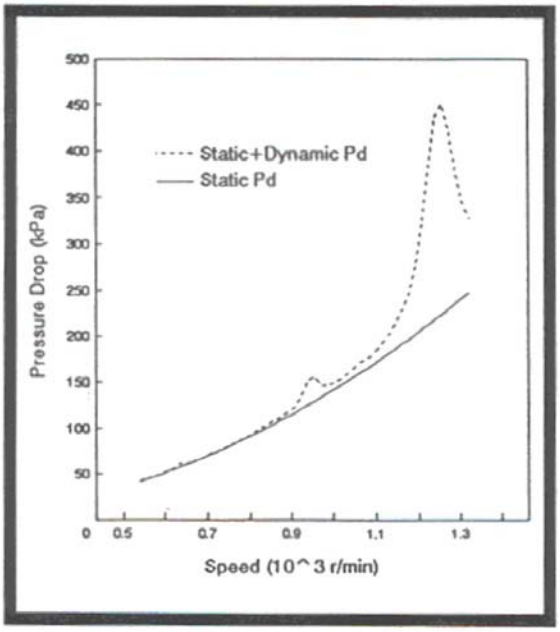


Figure 5: Modified System Pressure Drop

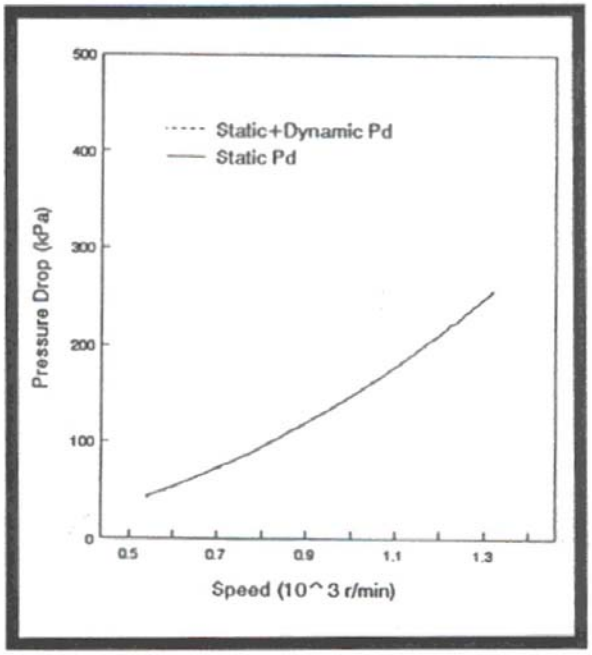


Figure 6: Flow Comparisons Using Field Data

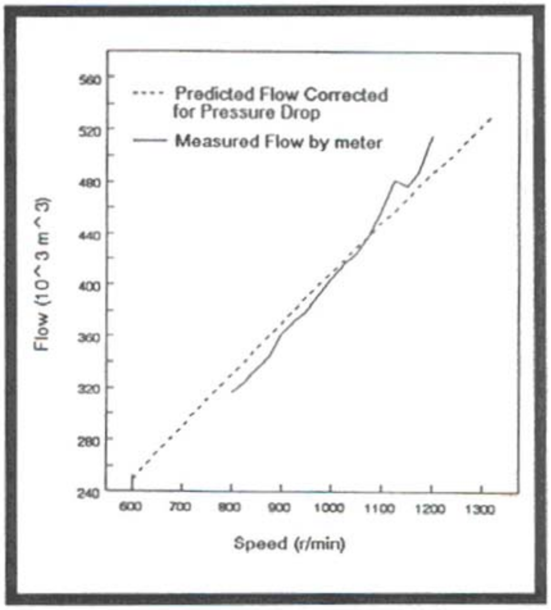


Figure 7: **As Found System Flow Comparisons, Adjusted Predicted Flow**

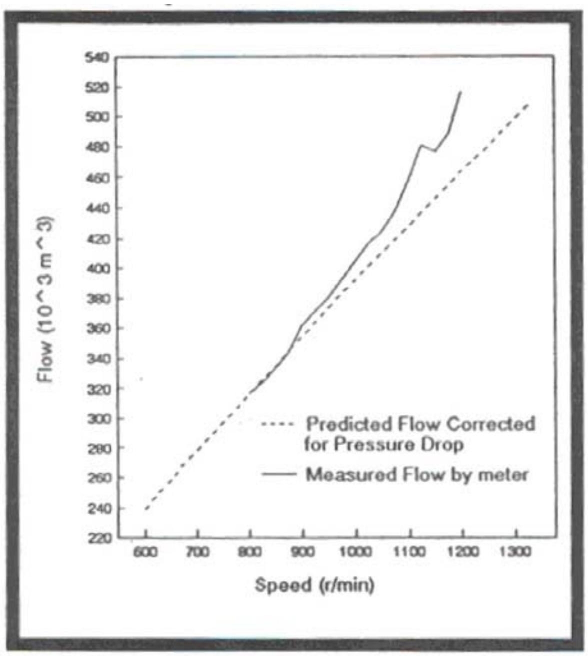


Figure 8: **As Found System Predicted Metering Error**

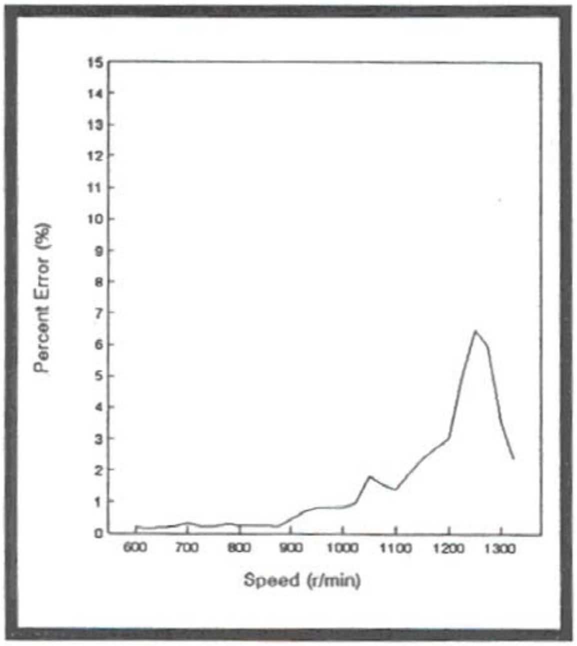
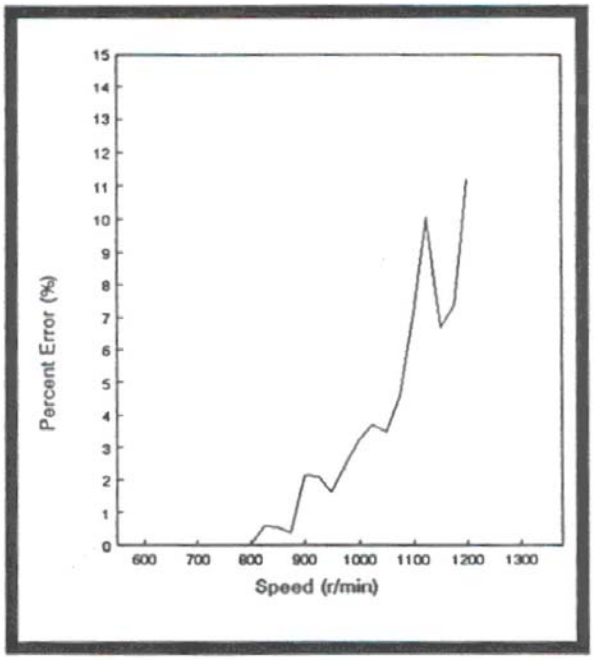


Figure 9: **As Found System Measured Metering Error**



History 2

The compressor station is a hydrogen service installation having two identical units. Each unit has six throws and two stages. They are fixed speed units running at 327 r/min and rated at 9.7 MW. After several months of operation it was discovered that high pressure drop in the interstate system was occurring.

High pressure drop was occurring for two reasons. First, the cylinder nozzle orifices were causing large amounts of pressure drop. This was due to the instantaneous flow through the orifice during the valve opening event. The second reason was large volume velocities occurring at the entrance and exit of the choke tube in the first stage discharge bottles. Figure 10 shows a sketch of part of the piping system. This large volume velocity at the choke tube was not due to a resonance in the system as it remained constant through the speed range. Refer to Figure 11. From Figure 12 it can be seen that the majority of the dynamic component of the pressure drop is located in the choke tube.

Pressure drops were reduced to acceptable levels by increasing the size of the cylinder nozzle orifice and moving the choke tube.

Figure 10: Case History Piping

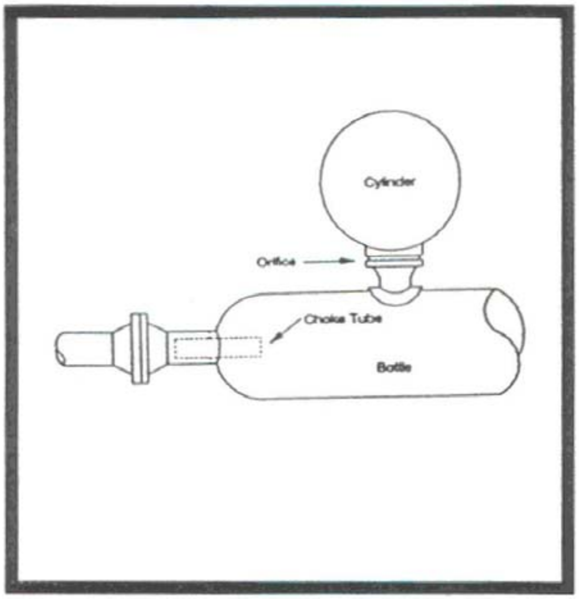


Figure 11: System Pressure Drop

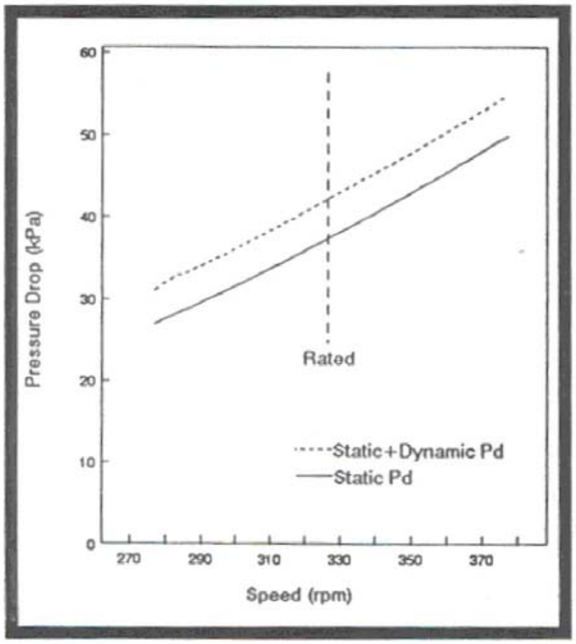
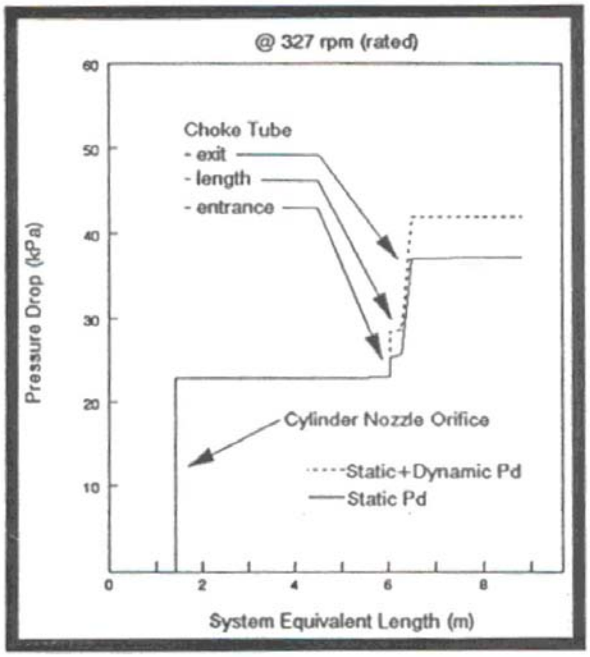


Figure 12: Static and Dynamic Pressure Drop



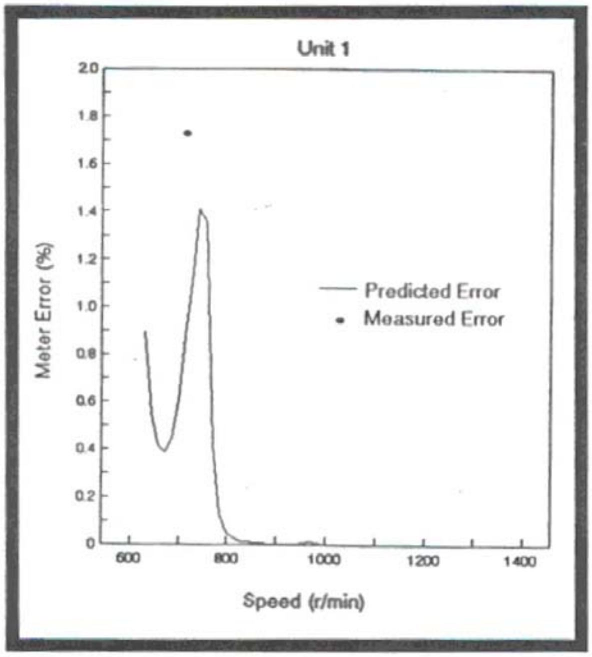
History 3

The compressor station was a new pipeline service installation having two 2 throw, 990 kW units operating in parallel over a speed range of 700 to 1200 r/min. The initial reason for our visit to site was that several vibration problems had been encountered at start-up. However, it was discovered that metering error was also present.

Accurate prediction and measurement of meter accuracy can only be performed when one unit is online. When both units are operating in parallel the phase relationship between the two machines is not constant. The data presented are for each unit online separately.

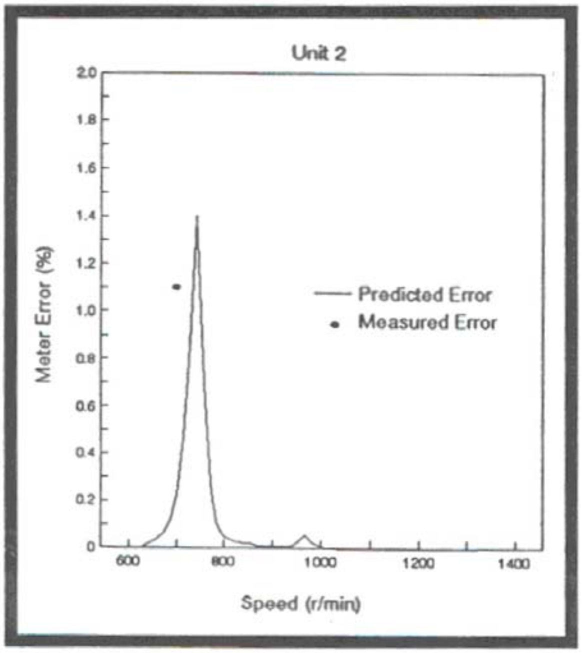
The first unit is predicted to cause meter error that is large at low speeds of the machine. Refer to Figure 13. A sharp resonance peak in meter error occurs at approximately 740 r/min. This prediction is consistent with what was observed and measured in the field.

Figure 13: Meter Error – Unit 1



The second unit is also predicted to cause a sharp increase in meter error at around 740 r/min. Refer to Figure 14. Again this prediction is consistent with observed and measured data.

Figure 14: Meter Error – Unit 2



To determine the relationship of pulsations and volume velocities of two machines, a vector addition involving phase is required. Due to the varying nature of the phase relationship between units, the meter error will also vary. The worst case assumption is that the total error with two units would be up to twice the sum of the individual errors. The best case assumption is that the error from each unit would counteract and the overall meter error would be minimal. If the worst case scenario occurs there would be significant meter error at 740 r/min.

Conclusion

Pulsations in piping systems include both pressure and flow effects. Pulsating flow can cause problems that are hard to detect in systems, unlike pulsating pressures.

Well designed piping systems will not have excessive pressure drop or metering errors due to oscillating flow. Unfortunately, systems designed with acceptable pressure pulsation control can still have unacceptable flow pulsations.

REFERENCES:

1. Howes, Brian; “Vibrations in Reciprocating Compressors”, May, 1992.

2. Howes, Brian; Eberle, Kelly; Zacharias, Val; “Cylinder Stretch as a Source of Vibration in Reciprocating Compressors”, May, 1991.

3. Howes, Brian; Greenfield, Shelley; Schuh, Christine; “Acoustic Pulsations in Reciprocating Machinery”, March, 1991.

4. Baumeister, T. (editor), “Standard Handbook for Mechanical Engineers”, seventh edition.

5. Vandell, Craig; Foerg, Wolfgang; Randall, James; “The Pluses of Positive Displacement”, Chemical Engineering, January, 1993.

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Introduction

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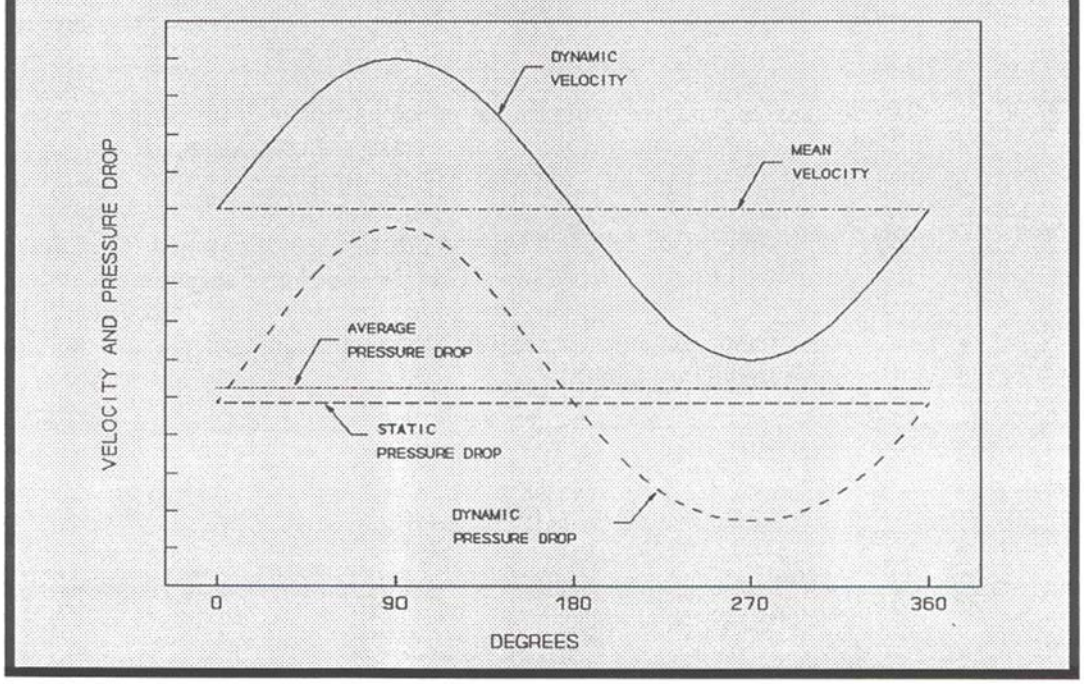
The pressure drop along a piping system increases when an oscillating flow term is added to the mean flow term. Refer to Figure 1 for a graphical representation of the effect of pulsating flow on pressure drop.

FIGURE 1: OSCILLATING FLOW AND PRESSURE DROP



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Logically, a system with less oscillating flow amplitude will have less pressure drop, all other things being equal.

In summary, the oscillating flow caused by pulsations in a piping system plus the mean flow result in more pressure drop in a system than caused by the average flow. It can happen that adding pressure drop in the form of discrete pulsation suppression devices to reduce pulsations, rather than causing an increase in system pressure drop, can reduce system pressure drop.

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HISTORY 1

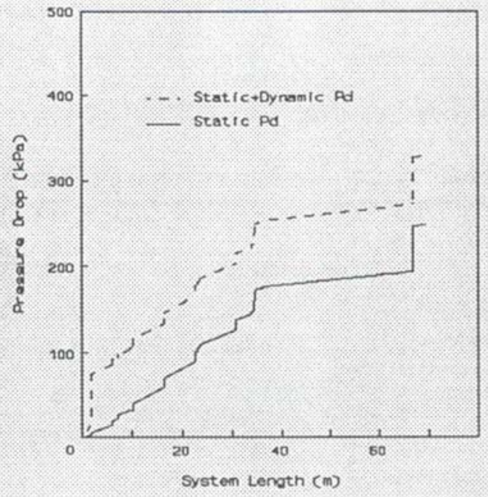


Fig. 2: Static & Dynamic Pressure Drop As Found System (1200 RPM)

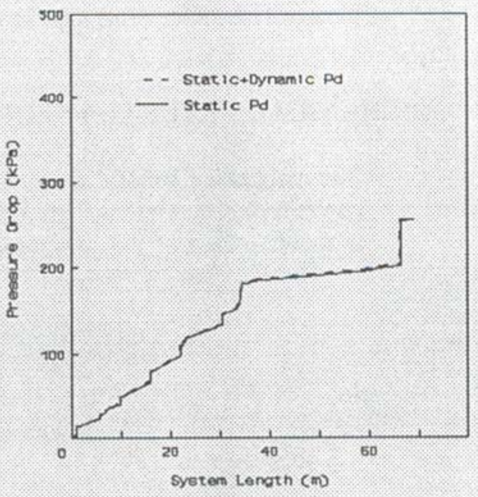


Fig. 3: Static & Dynamic Pressure Drop Modified System (1200 RPM)

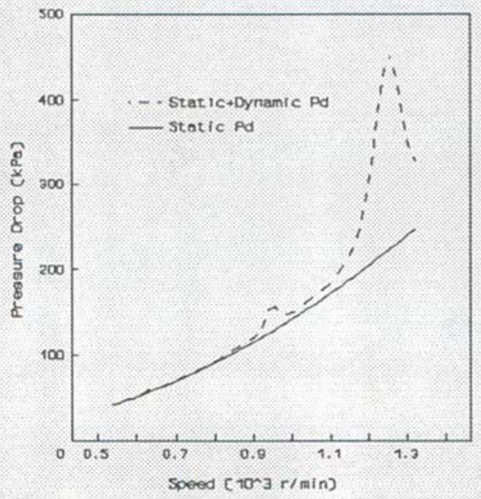


Fig. 4: As found System Pressure Drop

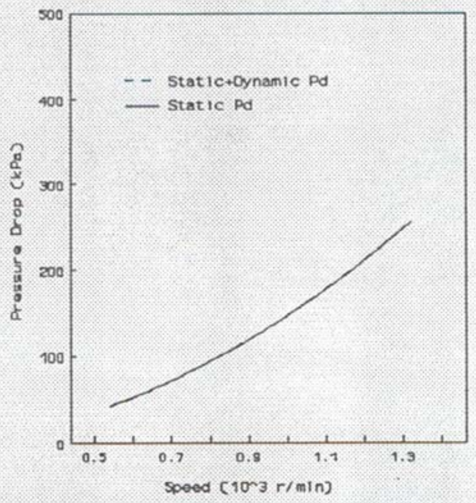


Fig. 5: Modified System Pressure Drop

HISTORY 1

Fig. 6: Flow Comparisons Using Field Data

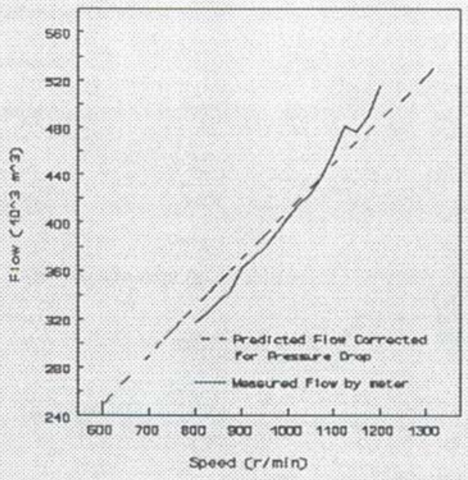


Fig. 7: As Found Flow Comparisons Adjusted Predicted Flow

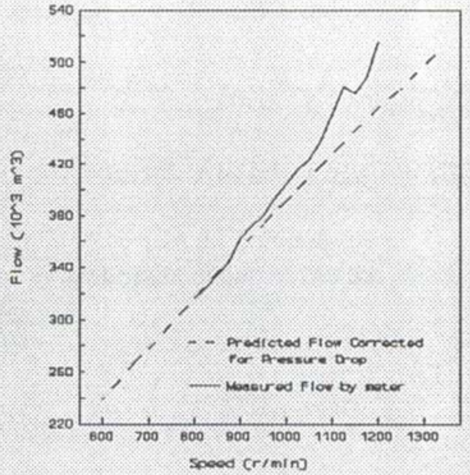


Fig. 8: As Found System Predicted Metering Error

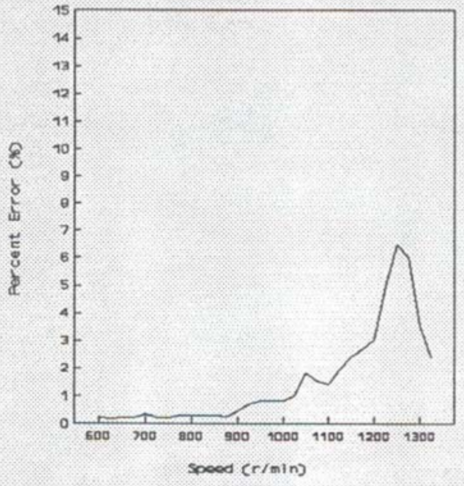
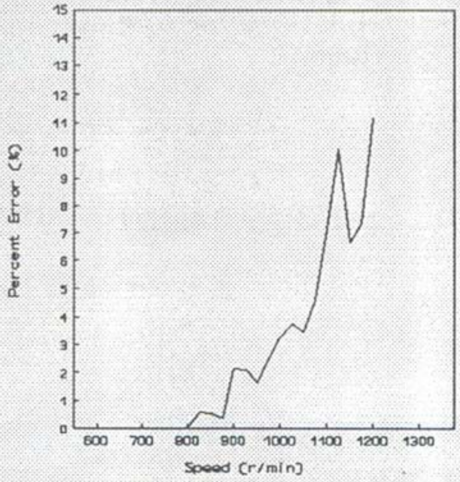


Fig. 9: As Found System Measured Metering Error



HISTORY 2

The compressor station was a new pipeline service installation having two 2 throw, 990 kW units operating in parallel over a speed range of 700 to 1200 r/min. The initial reason for our visit to site was that several vibration problems had been encountered at start-up. However, it was discovered that metering error was also present.

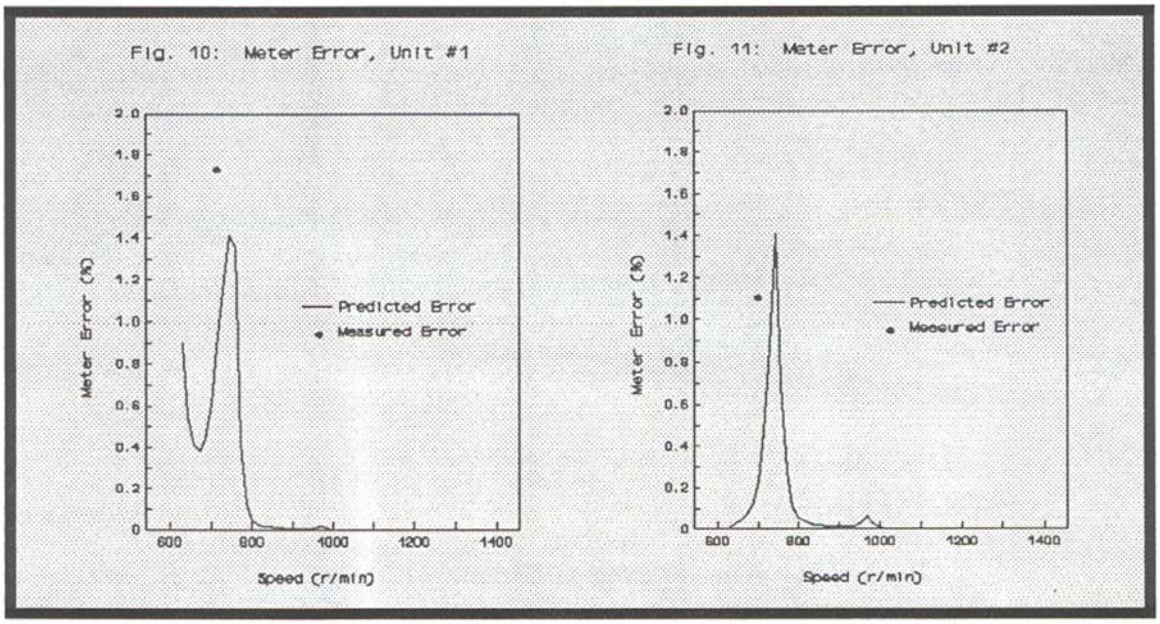
Accurate prediction and measurement of meter accuracy can only be performed when one unit is online. When both units are operating in parallel the phase relationship between the two machines is not constant. The data presented are for each unit online separately.

The first unit is predicted to cause meter error that is large at low speeds of the machine. Refer to Figure 10. A sharp resonance peak in meter error occurs at approximately 740 r/min. This prediction is consistent with what was observed and measured in the field.

The second unit is also predicted to cause a sharp increase in meter error at around 740 r/min. Refer to Figure 11. Again this prediction is consistent with observed and measured data.

To determine the relationship of pulsations and volume velocities of two machines, a vector addition involving phase is required. Due to the varying nature of the phase relationship between units, the meter error will also vary. The worst case assumption is that the total meter error with two units would be up to twice the sum of the individual errors. The best case assumption is that the error from each unit would counteract and the overall meter error would be minimal. If the worst case scenario occurs there would be significant meter error at 740 r/min.

HISTORY 2



History 3

This case study is based on the results of a field compressor analysis and a MAPAK analysis of a Cooper MW64 compressor installation. Suspected compressor performance problems were the reasons for the investigation.

At the beginning of the investigation performance-impairing pulsations near compressor valves, dynamic pressure drop, and fluctuating volume velocities at the orifice meters were suspected.

The result of both the field and MAPAK analysis was that compressor performance should be very near specification. Suspicions were then cast upon non-pulsation related metering errors. Subsequent to receiving reports of the results of the combined analysis, an investigation was carried out by plant operators which discovered that the plant fuel gas flows were being subtracted twice from the plant output. This metering error had resulted in the apparent shortfall in compressor performance.

The combined analysis did, however, identify static and dynamic pressure drops to be high in various locations throughout the system.

One area in which some improvement could be gained by reductions in static and dynamic pressure drops was the second interstage system. In the “as found” system static and dynamic pressure drops were high due, in large part, to a high loss coefficient, (“K”) at the entrance to a choke tube in the second stage discharge bottle. The acoustical analysis predicted a significant degree of oscillating flow at the entrance to the choke tube. See Figure 12 for static and combined static + dynamic pressure drop through the second interstage system over the compressor run-speed range.

The acoustical model was utilized to design a second interstage system that would result in less static and dynamic pressure drop while providing similar control of pulsations and unbalanced forces in the piping system and bottles. See Figure 14 for static and combined static + dynamic pressure drop versus system length for the as found system. See Figure 15 for static and combined static + dynamic pressure drop through the second interstage system over the compressor run-speed range.

The acoustical modifications were by the owner’s request to use existing, identical bottles off a decommissioned unit. Once removed from the “old” unit the bottles would then be modified internally and installed at the operating compressor installation when convenient. At the time of writing the bottles have yet to be modified, so field confirmation of these results remains in the future.

HISTORY 3

Fig. 12: Static & Dynamic Pressure Drop
As Found System (900 RPM)

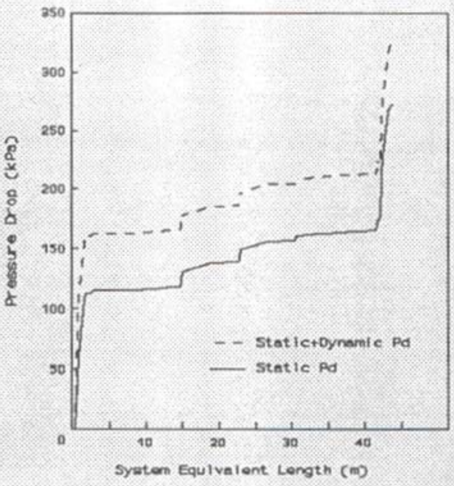


Fig. 13: As Found System Pressure Drop

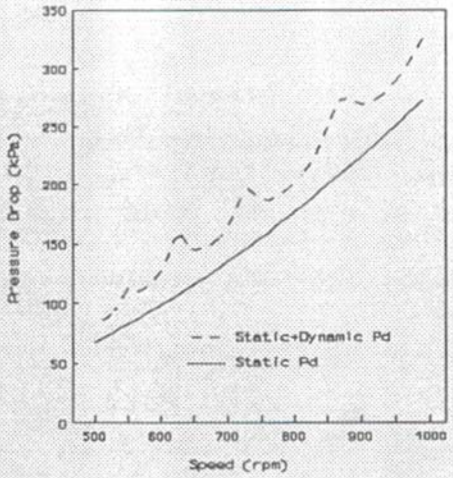


Fig. 14: Static & Dynamic Pressure Drop
Modified System (900 RPM)

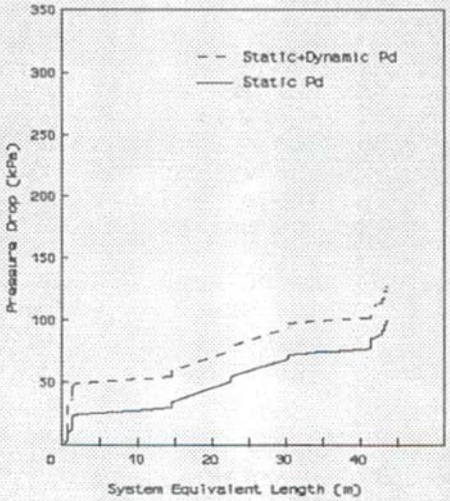
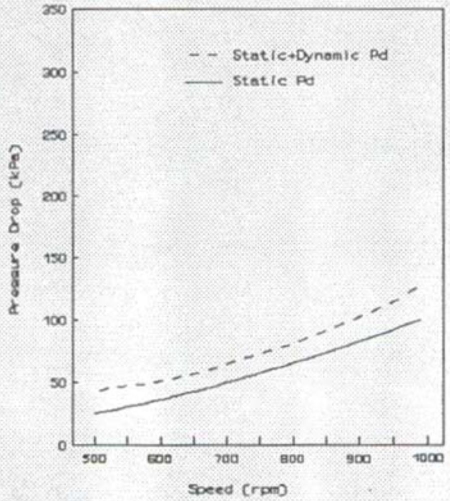


Fig. 15: Modified System Pressure Drop



Conclusions

Pulsations in piping systems include both pressure and flow effects. Pulsating flow can cause problems that are hard to detect in systems, unlike pulsating pressures.

Well designed piping systems will not have excessive pressure drop or metering errors due to oscillating flow. Unfortunately, systems designed with acceptable pressure pulsation control can still have unacceptable flow pulsations.

REFERENCES

1. Howes, Brian; “Vibrations in Reciprocating Compressors”, May, 1992

2. Howes, Brian; Eberle, Kelly; Zacharias, Val; “Cylinder Stretch as a Source of Vibration in Reciprocating Compressors”, May, 1991

3. Howes, Brian; Greenfield, Shelley, Schuh, Christine; “Acoustic Pulsations in Reciprocating Machinery”, March, 1991

4. Baumeister, T. (editor), “Standard Handbook for Mechanical Engineers”, seventh edition

5. Vandell, Craig; Foerg, Wolfgang; Randall, James; “The Pluses of Positive Displacement”, Chemical Engineering, January, 1993