1982

Pulsation and Vibration Control Requirements in the Design of Reciprocating Compressorand Pump Installations

W.W. von Nimitz

PULSATION AND VIBRATION CONTROL REQUIREMENTS IN THE DESIGN OF RECIPROCATING COMPRESSOR AND PUMP INSTALLATIONS

by

Walter W. von Nimitz, Ph.D.
Director of Industrial Applications
Southwest Research Institute
San Antonio, Texas

ABSTRACT

A number of recommendations presented by the author in an earlier paper(1) have been adopted in the Second Edition of API Standard 618 for Reciprocating Compressors in General Refinery Services. While these recommendations found wide acceptance in the industry, the advancement of the state of the art over the past eight years and the author's experiences with the application of API-618 to several hundred plant design studies, provided the basis for evolving improved methods for assuring the reliability of reciprocating compressor and pump installations at the design stage.

Specifically, this paper presents new methods for sizing surge volumes, maximum allowable pulsation levels at compressor valves and in the piping systems, and improved pressure drop criteria based on performance. Selection of the design methods and a discussion of the extent of the study are also presented together with a summary of recommended dynamic design criteria.

INTRODUCTION

The first requirement in the design, construction, and operation of every centrifugal or reciprocating compressor or pump system is the need to operate in a safe manner. Even so, there are differences both from plant to plant and within each plant in the sizes and types of equipment and the way in which they are used. An additional requirement for any compressor and pump installation is to operate in an efficient manner with a minimum of downtime.

Consequently, efforts to develop meaningful pulsation and vibration control criteria have been on going ever since the industry began using such equipment. For the past 30 years, the author's institution has been carrying out an extensive research effort for the 66 member companies of the Pipeline and Compressor Research Council (PCRC) directed specifically to the development of improved plant design and evaluation technology. This effort and practical experience with the design and evaluation studies of more than 5000 compressor and pump installations worldwide provided the basis for developing improved pulsation and vibration control requirements.

In an earlier paper(1) the author presented recommendations for reliability and performance assurance in the design of reciprocating compressor and pump installations, most of which have been accepted by the American Petroleum Institute's code committee for inclusion in the Second Edition of API Standard 618. World-wide use of this standard for the past 8 years confirmed its usefulness in ensuring safe and efficient compressor installations; however, it has also identified areas needing improvement. This paper responds to these needs on the basis of recent advances in the state of the art.

Of course, basic pulsation, vibration, and cyclic stress requirements are the same regardless of the type of compressor or pump unit used. Consequently, while this presentation will discuss reciprocating compressor installations, most of the conclusions and recommendations will apply to any compressor or pump installation.

Specifically, this presentation will discuss (1) minimum surge volume requirements, (2) maximum allowable pulsation levels at compressor valves, (3) maximum allowable pulsation levels in the piping systems, and (4) maximum allowable pressure drop through the pulsation suppression device(s). Furthermore, it will discuss basic design approaches and their selection to assure achieving safe and efficient compressor and pump installations as cost effectively as possible and a summary of recommended criteria.

MINIMUM SURGE VOLUME REQUIREMENTS

Surge volume has the inherent acoustical property of opposing a change in the applied pressure. Placed at compressor valves, surge volumes will reduce pulsations and thereby minimize the deviation of compressor cylinder performance from ideal.

Over the years it became accepted practice to use a surge volume equal to at least ten times the total double-acting displacement volume of all compressor cylinders to be manifolded in the surge volume.

Later examinations of this empirical rule indicated that while it was a good choice for typical natural gas service, for other gases such as propane or hydrogen, and for high pressure services, this was not a realistic guideline.

A review of basic thermodynamic and acoustical impedance relations between cylinder volume and the attached surge volume, assuming that the impedance of the two systems will remain the same, yields the basic relationship between the two volumes:

$$V_2 = V_1 \frac{\rho_2 \, C_2^2}{\rho_1 \, C_1^2} \tag{1}$$

Where:

V_1 = Volume of System 1.

V_2 = Volume of System 2.

ρ_1 = Gas density at operating conditions of System 1.

ρ_2 = Gas density at operating conditions of System 2.

C_1 = Velocity of sound in System 1.

C_2 = Velocity of sound in System 2.

For an isentropic process and assuming that the pressures in both systems will remain the same, the above equation can be reduced to:

$$V_2 = V_1 \frac{k_2}{k_1} \tag{2}$$

Where:

k_1 = Isentropic compression exponent at operating pressure and temperature in System 1.

k_2 = Isentropic compression exponent at operating pressure and temperature in System 2.

Finally, using typical natural gas service as the reference (System 1) and substituting 10 x PD for V_1 and 1.25 for the isentropic compression exponent k_1, we obtain a simple yet thermodynamically correct relationship for selecting minimum equivalent surge volume for other gases over a wide range of operating pressures and temperatures:

$$V = 8 \cdot k \cdot PD \tag{3}$$

Where:

V = Minimum required surge volume in ft^3.

k = Isentropic compression exponent at the operating pressure and temperature.

PD = Total double-acting displacement volume of all compressor cylinders to be manifolded in the surge volume in ft^3.

While equation (3) represents a substantial improvement over the earlier practice, it nevertheless does not do a good job matching the final surge volume selected in the actual design study. The basic cause of this apparent "discrepancy" is the fact that the surge volume also serves as a part of a pulsation suppressor design where the velocity of sound is the primary influencing factor. More specifically, it was determined from extensive analysis of actual design cases that modifying the relationship in equation (2) to the square root of the ratio of the corresponding velocities of sound provided a very realistic method for estimating the ultimately required surge volume size. This modified relationship is shown below:

$$V_2 = V_1 \left(\frac{C_2}{C_1}\right)^{1/2} \tag{4}$$

Substituting 10 x PD for V_1 and a typical velocity of sound in methane of 1400 fps for C_1 and using a simplified expression for velocity of sound:

$$C = 223 \left(\frac{k \cdot T \cdot Z}{M}\right)^{1/2} \tag{5}$$

we can arrive at a simple expression for minimum required suction surge volume:

$$V_s = 4 \cdot PD \cdot \left(\frac{k \cdot T_s}{M}\right)^{1/4} \tag{6}$$

Where:

V_s = Minimum required suction surge volume in ft^3.

k = Isentropic compression exponent at suction pressure and temperature.

T_s = Suction gas temperature in degrees Rankine (460 +°F).

M = Molecular weight of gas.

In the above equation (6), the supercompressibility factor of gas (Z) was set equal to one for the sake of simplicity. This is a reasonable assumption for typical suction conditions.

With the above equation establishing the minimum required suction surge volume, the corresponding discharge surge volume (V_d) can be calculated as follows:

$$V_d = \frac{V_s}{R^{1/k}} \tag{7}$$

A graphic illustration of various suction surge volume requirements as a function of the velocity of sound is shown in Figure 1.

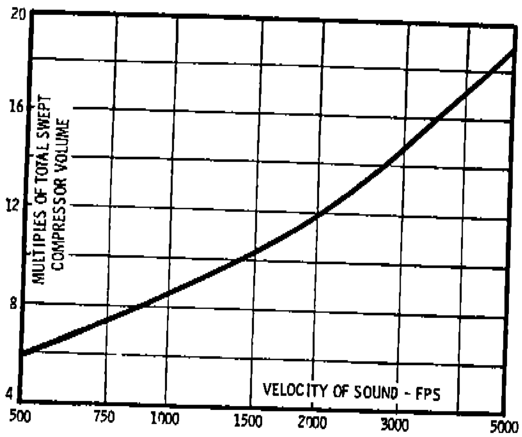

Fig. 1 Minimum Suction Surge Volume Requirements for Various Sound Velocities.

In addition, the discharge surge volume requirements as a function of compression ratio and isentropic compression exponent is shown in Figure 2.

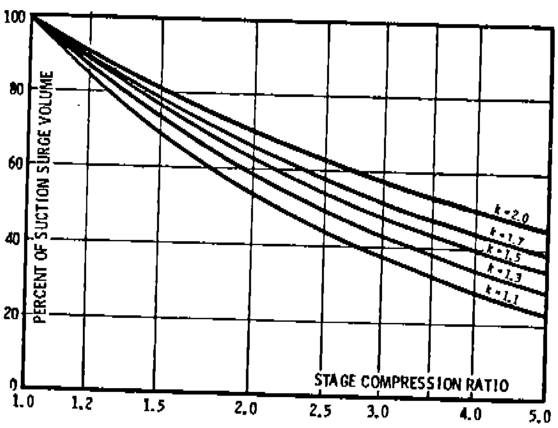

Fig. 2 Minimum Discharge Surge Volume Requirements as a Function of Compression Ratio and Isentropic Compression Exponent.

CONTROL OF PULSATION AT COMPRESSOR VALVES

Effective control of pulsations at compressor cylinder valves is needed to minimize their adverse effects on compressor cylinder operation and compressor valve life.

Analysis of numerous compressor cards suggests that in order to keep the deviation in compressor cylinder performance to a maximum of 5 percent of the indicated cylinder horsepower, the maximum peak-to-peak pulsation levels at the compressor cylinder valves should be in the range of 10 percent of the stage differential pressure ($P_d - P_s$). The effect of pulsations is also dependent on the frequency and phase of discrete pulsation components, cylinder volumetric efficiency and compression exponent; however, the primary factor is the overall level of pulsations. Using a maximum of 10 percent as the first approximation, the maximum allowable pulsation levels would be:

$$P_V\% = 5\left(\frac{R^2 - 1}{R}\right) \tag{8}$$

Where:

$P_V\%$ = Maximum allowable peak-to-peak pulsation levels at compressor valves as a percentage of the average absolute line pressure.

R = Stage compression ratio.

The effect of pulsations on compressor valve life is a function of the overall pulsation levels, the relative location of discrete pulsation frequencies and the mechanical natural frequencies of the valves, valve type and construction, and the amount of damping present. The first approximation is again the overall level of pulsations with the generally accepted industrial practice of limiting pulsation levels to some 6-8 percent of line pressure. For the purpose of establishing a criterion, an average value of 7 percent will be used which is also supported by major U.S. compressor manufacturers. Thus, the second requirement is to limit the peak-to-peak pulsation levels at the compressor valves to a maximum of 7 percent of the average absolute line pressure.

The third and final consideration is the practical low limit for control of pulsation levels at the compressor valves. The problem is not just that of economics but more fundamentally, that of the physical space available for the required size of surge volumes and their mechanical stability (such as supporting a large heavy volume bottle on small compressor cylinder nozzles). In addition, the extent of pulsation reduction is limited by the effect of internal cylinder gas passage volumes. Recognizing these limitations, the maximum required pulsation level at the compressor valves should, typically, not be less than 3 percent of the average absolute line pressure. The penalty for this requirement is the possibility of greater deviation in compressor cylinder loading and efficiency at very low compression ratios.

The expression for the maximum allowable pulsation levels at compressor valves in the Second Edition of API-618 is as follows:

$$P_V\% = 8\left(\frac{R - 1}{R}\right)^{1/2} \tag{9}$$

This equation (9) is in general agreement with the above listed requirements with the exception of maximum allowable pulsation levels at low compression ratios.

To correct this deficiency, a simpler and actually better approximation satisfying the three requirements outlined above would be to limit the maximum allowable peak-to-peak pulsation levels at compressor cylinder valves ($P_v\%$) to 7 percent of the average absolute line pressure or the value expressed by equation (10) below, whichever is lower:

$$P_v\% = 3 \cdot R \tag{10}$$

Where:

R = Stage compression ratio.

The relation between the current API-618 equation and the recommended procedure is shown in Figure 3 below.

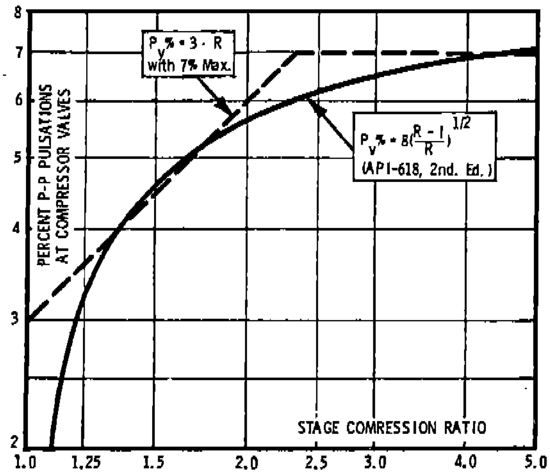

Fig. 3 Allowable Pulsation Levels at Compressor Valves Based on API-618, 2nd. Ed. and The New Recommended Procedure.

ALLOWABLE PIPING PULSATION LEVELS

Pulsations as such do not cause piping failures; however, they can couple to the piping at closed ends of a line, bends, restrictions, etc., to produce acoustical shaking forces. These forces, in turn, excite vibrations depending on the relative location of discrete acoustical force frequencies and the mechanical natural frequencies, the point of excitation, and the amount of damping present. Whether the vibrations can cause fatigue failure depends on the cyclic stress levels they produce and the cumulative effect of stress considering the cyclic endurance limit of the piping material. This process is illustrated in the Piping System Reliability Chart shown in Figure 4.

The ultimate measure of piping system reliability is thus the cyclic stress level produced by pulsation-induced vibrations. The maximum safe cyclic design stress level specified in the Second Edition of API-618 is 26,000 psi peak-to-peak which corresponds to the cyclic endurance limit run-out value at 10^6 cycles for carbon and alloyed steels under 700°F operating temperature. Subsequent studies by the ASME Code Committee and others extending the endurance curve to 10^7 and even 10^8 cycles as well as the analysis of numerous failure cases suggest that the cyclic endurance limit run-out at 10^7 cycles (which is 20,000 psi peak-to-peak) is more appropriate to ensure that the design will be free from the pulsation induced cyclic fatigue failures.

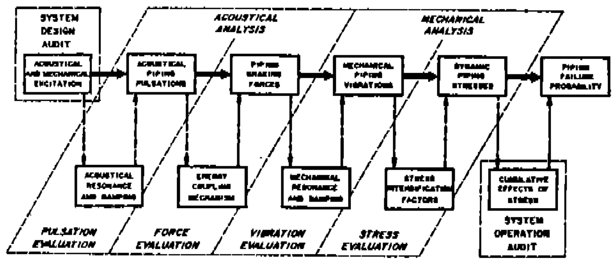

Fig. 4 Piping System Reliability Chart

Calculation of the expected cyclic stress levels at the design stage requires, first, determination of the amplitudes and frequencies of discrete pulsation components and corresponding acoustical shaking forces. Next, it is necessary to perform a mechanical analysis of compressor manifold and piping systems to determine the mechanical natural frequencies and mode shapes and the resulting vibration levels produced by the acoustical shaking forces. Finally, cyclic stress analysis must be performed (considering the stress concentration factors present) to determine resulting cyclic stress levels. While analytical tools are available to perform the above evaluation steps, practical considerations demand that such complete analysis be performed only when simpler criteria indicate a need for them.

The simplest but least reliable of such criteria is the relation used in API Standard 618, Second Edition, for installations requiring Design Approach 1 or Design Approach 2 without an acoustical simulation study:

$$P\% = \frac{10}{P_L^{1/3}} \tag{11}$$

Where:

P% = Maximum allowable overall peak-to-peak pulsation levels at the line connection of a pulsation suppressor expressed as a percentage of the average absolute line pressure.

P_L = Average absolute line pressure in psia.

One of the basic shortcomings of equation (11) is not considering pulsation frequencies. On the other hand, another method frequently used in the industry for analytically designed systems considers engine harmonics (frequency) but not the operating pressure level. A more meaningful approach is to combine both line pressure and frequency requirements. This improved method for analytically designed systems is presented in equation (12) below:

$$P_1\% = \frac{15}{P_L^{1/3}}; \quad P_2\% = \frac{P_1\%}{2}; \quad P_3\% = \frac{P_1\%}{4} \qquad (12)$$

Where:

$P_1\%$ = Maximum allowable peak-to-peak pulsation levels at the line flange of the pulsation suppressor at the fundamental frequency (compressor RPM) and second harmonic of compressor speed (2 x RPM) expressed as a percentage of P_L.

$P_2\%$ = Same definition as for $P_1\%$ except for the third (3 x RPM) and fourth (4 x RPM) harmonics of compressor speed.

$P_3\%$ = Same definition as for $P_1\%$ except for frequencies above the fourth harmonic of compressor speed.

P_L = Average absolute line pressure in psia.
Use values for 125 psia if line pressure is less than 125 psia.

Maximum allowable pulsation levels specified in the above equation (12) are presented graphically in Figure 5.

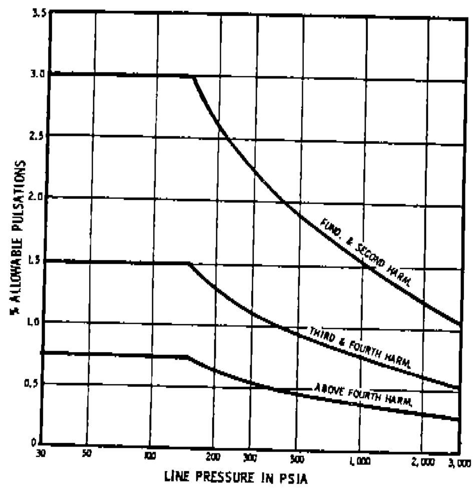

Fig. 5 Maximum Allowable Pulsation Levels at the Line Flange of Pulsation Suppressor as a Function of Line Pressure and Compressor Harmonics.

While this modified method is an improvement over the original equation (11) used in the Second Edition of API Standard 618, its use is not a dependable indicator that the resulting cyclic stresses will be at a safe level.

A much more meaningful criterion was developed earlier[1] from the analysis of a large number of plant designs and adopted by API-618, Second Edition, for systems requiring Design Approach 3 or Design Approach 2 when an acoustical simulation study is specified. This effort established an empirical relationship defining the amplitudes and frequencies of discrete pulsation components throughout the piping system which normally would not be expected to produce cyclic stress levels in excess of the endurance limit of the material used, assuming that good engineering practices are followed in the construction and support of compressor manifold and piping systems. This "rule of thumb" equation is presented below:

$$P\% = \frac{300}{(P_L \cdot ID \cdot f)^{1/2}} \qquad (13)$$

Where:

$P\%$ = Allowable percent of discrete peak-to-peak pulsation amplitudes at any point in the piping beyond the pulsation suppressor(s).

P_L = Line pressure in psia.

ID = Internal diameter of the pipe in inches.

f = Frequency of the compressor harmonics in Hz.
($f = \frac{N \cdot RPM}{60}$; where N = 1, 2, 3, ...)

The data base used for arriving at the above empirical relation covered compressor installations with operating pressures from about 50 psia to some 3000 psia. Consequently, the use of the above equation should be limited to that pressure range. Specifically, it is recommended that for pressures below 50 psia the pulsation levels corresponding to 50 psia line pressure should be used. The use of the above equation for line pressures above 3000 psia is not recommended and the allowable pulsation levels should be determined on the basis of a cyclic stress analysis. Even within the 50-3000 psia operating pressure range, the equation should be used only as a guide. Allowable pulsation levels may exceed those determined by equation (13) if calculated cyclic stress levels indicate the system will be safe from cyclic fatigue failures.

PRESSURE DROP CONSIDERATIONS

Pressure drop is not a system reliability consideration but, rather, an economic factor which is likely to be different in each design case. Nevertheless, a general pressure drop criterion is needed to assure that the design will meet certain minimum efficiency requirements and will permit comparison of various pulsation suppressor designs on an equal compressor efficiency basis.

The effect of pressure drop is to increase the compression ratio which the compressor will see thereby increasing horsepower requirements to deliver the same volume of gas. Some earlier criteria, including API Standard 618, used a fixed percentage of line pressure such as 1 percent or 2 percent as the allowable pressure drop through pulsation suppressor device(s). Such "fixed percentage" criteria are totally inadequate economically since the corresponding loss in compressor efficiency will vary widely depending on the compression ratio. The effect of compression ratio on compressor efficiency is shown in Figure 6.

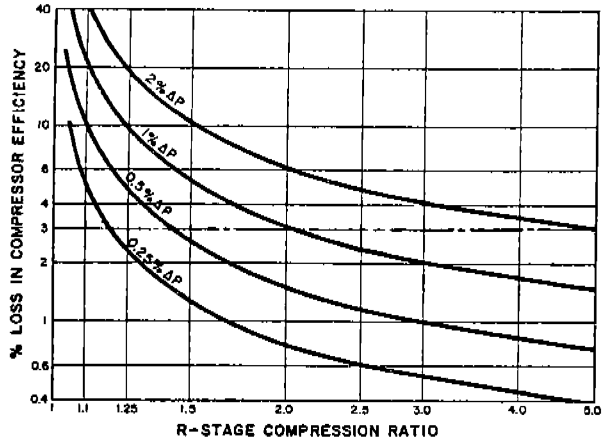

Fig. 6 The Effect of Fixed Percentage Pressure Drop on Compressor Efficiency at Various Compression Ratios.

With this understanding, we can proceed to examine the factors affecting compressor efficiency and develop meaningful criteria for allowable pressure drop assuming that the non-recoverable steady-state pressure drop is the total pressure drop in the system. Specifically, for a reciprocating compressor, efficiency (E) can be defined as:

$$E = 43.6 \left(\frac{k}{k-1}\right)\left(R^{\frac{k-1}{k}} - 1\right) \tag{14}$$

Where:

- E = Compressor efficiency in BHP/MMSCFD.
- k = Isentropic compression exponent.
- R = Stage compression ratio.

If we assume equal percent pressure drop on the suction and on the discharge ($\Delta P\%$), we can define the compression ratio the compressor will see with pressure drop (R_Δ) as:

$$R_\Delta = R\,\frac{100 + \Delta P\%}{100 - \Delta P\%} \tag{15}$$

and the corresponding fractional loss in compressor efficiency (L) as:

$$L = \frac{R_\Delta^{\frac{k-1}{k}} - R^{\frac{k-1}{k}}}{R^{\frac{k-1}{k}} - 1} \tag{16}$$

Solving equation (16) for R_Δ and substituting this value into equation (15), we can obtain the expression for percent pressure drop ($\Delta P\%$) as:

$$\Delta P\% = 100\,\frac{\left[L\left(R^{\frac{k-1}{k}} - 1\right) + R^{\frac{k-1}{k}}\right]^{\frac{k}{k-1}} - R}{\left[L\left(R^{\frac{k-1}{k}} - 1\right) + R^{\frac{k-1}{k}}\right]^{\frac{k}{k-1}} + R} \tag{17}$$

The above equation (17) is a general expression relating allowable percent pressure drop to the specified fractional loss in compressor efficiency as a function of stage compression ratio (R) and the isentropic compression exponent (k).

A simplified empirical equation for pressure drop corresponding to approximately 3 percent loss in compressor efficiency is presented below:

$$P\% = 1.67 \left(\frac{R-1}{R}\right) \tag{18}$$

This equation was introduced earlier[1] and was subsequently adopted by the API for the Second Edition of API Standard 618. At compression ratios of 1.30 and lower, it calculates slightly high values. Its basic shortcoming, however, is that it calculates lower than necessary pressure drop values at higher compression ratios. For example, at R = 5.0 the specified pressure drop is 1.34 percent and the corresponding loss in compressor efficiency is only 2.05 percent.

To correct these shortcomings, a modified equation was developed for the simplified calculation of the maximum allowable pressure drop through pulsation suppression device(s):

$$\Delta P\% = 1.5 \left(\frac{R-1}{R} + \frac{R-1}{10}\right) \tag{19}$$

This modified equation ensures that the loss in compressor efficiency will not exceed 3 percent ± 0.25 percent up to a compression ratio of 5.0 and beyond.

A comparison of resulting loss in compressor efficiency corresponding to pressure drop calculated with the original equation (18) and modified equation (19) is shown in Figure 7.

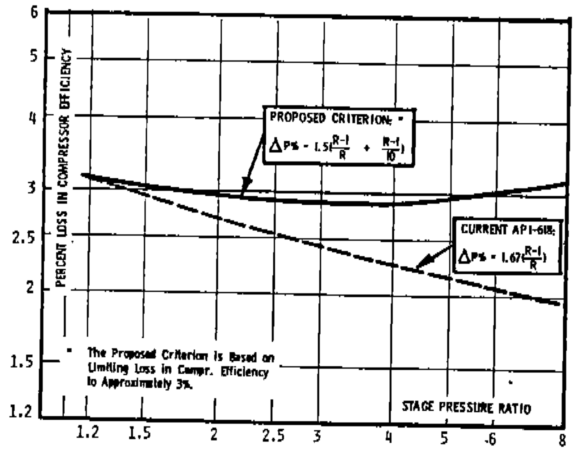

Fig. 7 The Effect of Allowable Pressure Drop on Compressor Cylinder Efficiency using Equations 18 and 19.

A review of actual compressor manifold configurations indicates that the lowest practical pressure drop limit is about 0.25 percent of the average absolute line pressure. Consequently, we will set the maximum allowable pressure drop through the pulsation suppression device(s) at 0.25 percent or the value calculated from the equation (19) above, whichever is higher. The penalty for setting the lowest pressure drop at 0.25 percent will be a greater than 3 percent loss in compressor efficiency at compression ratios below 1.175.

DESIGN APPROACHES AND THEIR SELECTION

It stands to reason that a small 50 BHP air compressor operating at 100 psi will not require nor can economically justify the extent of dynamic analysis required to optimize the design of a 5,000 BHP hydrogen unit with over 2000 psi discharge pressure. Indeed, considering a wide range of compressor power ratings and operating pressures, the level of effort required to arrive at a satisfactory design can range from simple surge volume calculations and avoidance of undesirable acoustic lengths in the piping system to a complete dynamic acoustic, mechanical, and stress analysis of the compressor units and the associated piping systems.

Over the years, three basic design approaches evolved as described below:

Design Approach 1 - Analytical evaluation using standard acoustical techniques to size volume bottles, orifices, and to select the preferred lengths of piping elements. (The PCRC defines this approach as an "Analytical Analysis.")

Design Approach 2 - Pulsation analysis consisting of a simulation of the compressor and associated piping systems including dynamic interaction between them using proven acoustical simulation techniques to arrive at effective control of pulsations at compressor cylinders and throughout the piping systems. (The PCRC defines this approach as an "SGA Pulsation Analysis" or an SGA Analog Study.)

Design Approach 3 - Acoustical and mechanical analysis including acoustical simulation of compressor and associated piping systems and their interaction using proven acoustical simulation techniques and a mechanical computer analysis of the compressor manifold and associated piping systems including interaction between acoustical and mechanical system responses. Both acoustical and mechanical methods are used to arrive at the most efficient and cost effective plant design. (The PCRC defines this approach as an "SGA Compressor System Design.")

The SGA Dynamic Compressor and Pump System Simulator represents a proven acoustical simulation technique whose validity and accuracy have been verified over the years by numerous user companies, engineering companies, and equipment vendors worldwide. It is based on a "physical modeling" technique which overcomes some of the basic limitations inherent in the use of digital or analog computer models. A sample of field recorded pulsation data and corresponding predictions made by the SGA Dynamic Simulator is shown in Figure 8. The case studied is the final discharge system of a gas processing plant.

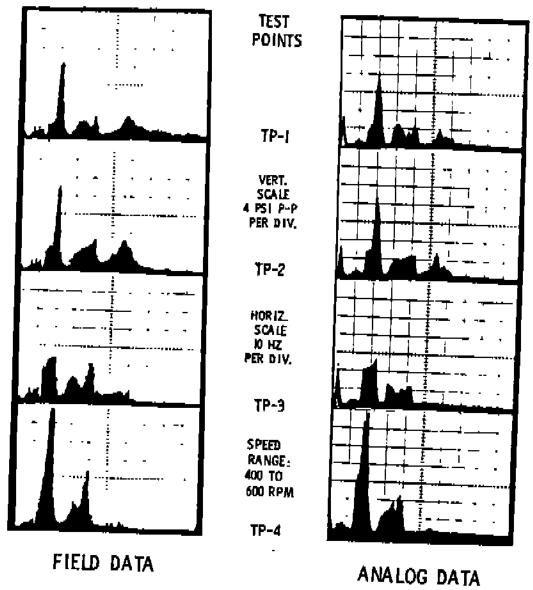

Fig. 8 Comparison of Field Pulsation Data and SGA Dynamic Simulator Predictions.

The author is not aware of any other design technique available today which is capable of determining acoustical system response to the degree of reliability and accuracy indicated in Figure 8.

In addition to using proven acoustical simulation techniques, the acoustical simulation of compressor piping systems in Design Approaches 2 and 3 must extend from a major plant vessel or volume on the intake side of the unit to a major plant vessel or volume on the final discharge side of the unit. In general, the simulation termination points are valid when piping changes beyond these points have only insignificant effect on pulsations in the compressor and piping systems being evaluated.

The selection of the Design Approach should be based primarily on equipment size and operating pressure range. However, consideration should also be given to the service the unit will perform with special consideration to critical or hazardous applications, plant locations, significance of downtime, etc. The method presented in Table 1 below applies to typical installations and should be used only as a guide.

Table 1
Design Approach Selection

Line Pressure (psia)	0–150 BHP	150–500 BHP	above 500 BHP
above 1000	2	3	3
500–1000	1	2	3
0–500	1	2	3

Rated Compressor Horsepower

Note: The numbers in blocks correspond to Design Approaches 1, 2, and 3 described above.

SUMMARY OF DESIGN CRITERIA

With the improved design considerations defined and the basic design approach requirements established, we can now summarize the recommended design practices for ensuring safe and efficient operation of compressor and pump installations. The overriding requirement can be formulated as follows:

> In the design of all compressor and pump installations, cyclic stress levels produced by the pulsation-induced vibrations shall not exceed the cyclic endurance limit of the material used. For carbon and alloyed steels up to 700°F operating temperature, an endurance limit of 20,000 psi peak-to-peak is recommended with all other stresses within allowable code limits.

The specific design requirements for various sized units and operating pressures are summarized below.

A. Design Approach 1 Systems

1. Minimum suction and discharge surge volume sizes should be determined in accordance with the equations listed below:

$$V_s = 4 \cdot PD \cdot \left(\frac{k \cdot T_s}{M}\right)^{1/4} \qquad (6)$$

$$V_d = \frac{V_s}{R^{1/k}} \qquad (7)$$

2. Maximum percentage of the peak-to-peak pulsations at various compressor harmonics should be limited at the line connection of the pulsation suppressors to the levels determined from:

$$P_1\% = \frac{15}{P_L^{1/3}}; \quad P_2\% = \frac{P_1\%}{2}; \quad P_3\% = \frac{P_1\%}{4} \qquad (12)$$

3. Unless another criterion is specified, the maximum pressure drop through the pulsation suppressor should be limited to 0.25 percent of line pressure or the values calculated from the equation below, whichever is higher:

$$P\% = 1.5 \left(\frac{R-1}{R} + \frac{R-1}{10}\right) \qquad (19)$$

With the three above requirements satisfied, any additional vibration control, if required, will normally be achieved by providing adequate mechanical piping supports.

B. Design Approach 2 and 3 Systems

1. Initial sizing of suction and discharge surge volumes will be determined as for Design Approach 1 Systems. However, final required surge volume sizes will be determined in an acoustical simulation study of compressor and piping systems.

2. Maximum allowable overall peak-to-peak pulsation levels at compressor cylinder valves shall be limited to 7 percent of the average absolute line pressure or the values determined from the relation below, whichever is lower:

$$P_v\% = 3 \cdot R \qquad (10)$$

higher levels may be acceptable if it can be shown in an acoustical simulation study that they will not cause excessive loss in performance or adverse effects on valve life.

3. For systems operating betweeen 50 and 3,000 psia line pressure, the maximum allowable peak-to-peak amplitudes of discrete pulsation components at any point in the piping beyond pulsation suppressors shall be limited to the levels calculated from:

$$P\% = \frac{300}{(P_L \cdot ID \cdot f)^{1/2}} \qquad (13)$$

For systems operating below 50 psia line pressure, use pulsation amplitude calculated for 50 psia operating pressure. For systems operating above 3,000 psia, the above equation should not be used but rather the resulting cyclic stresses should be calculated. In general, the calculated maximum pulsation levels may be exceeded if it can be shown that the resulting cyclic stresses will be within the allowable cyclic endurance limit of the material used.

4. Unless another criterion is specified, the maximum allowable pressure drop through the pulsation suppression devices should be determined as for Design Approach 1 Systems (equation 19).

5. Additionally, for Design Approach 3 Systems, proven mechanical computer modeling techniques shall be used to determine vibration responses and the resulting cyclic stress levels in the compressor manifold system to ensure compliance with the allowable cyclic endurance limits. Such computational techniques should be capable of three-dimensional modeling of the system including three-dimensional coupling between vertical, axial, and transverse responses. Similarly, mechanical response and cyclic stress evaluation of critical piping configurations (coolers, scrubbers, by-pass systems, etc.) should be performed to ensure that the allowable cyclic endurance limits are not exceeded.

While Design Approach 2 relies primarily on acoustical techniques for control of vibrations and resulting cyclic stresses, Design Approach 3 utilizes both acoustical and mechanical techniques to achieve the same objective, which is usually not only more efficient but also more cost effective. For example, a simple pipe clamp may be just as effective in controlling vibrations by separating the mechanical natural frequency from the frequency of acoustical shaking forces as the much more expansive modification to the pulsation suppressor design.

The dynamic criteria presented in this paper are expected to provide greater assurance that a given compressor or pump installation will operate in a safe and efficient manner. Of course, criteria alone are not sufficient without the latest acoustical and mechanical design technology at one's disposal and a highly experienced staff to develop practical solutions for any given design requirement.

REFERENCES

1. Walter W. Von Nimitz, "Reliability and Performance Assurance in The Design of Reciprocating Compressor and Pump Installations," 1974 Proceedings of the Purdue Compressor Technology Conference, July 1974.

2. American Petroleum Institute, API Standard 618, Second Edition, for Reciprocating Compressors in General Refining Services, New York, NY, 1974.