International Compressor Engineering Conference | School of Mechanical Engineering

1974

Reliability and Performance Assurance in the Design of Reciprocating Compressor Installations - Part I: Design Criteria

Walter W. von Nimitz
Southwest Research Institute

RELIABILITY AND PERFORMANCE ASSURANCE IN THE DESIGN OF RECIPROCATING COMPRESSOR INSTALLATIONS

PART I - DESIGN CRITERIA

Walter W. von Nimitz
Director of Industrial Applications
Division of Applied Physics
Southwest Research Institute
San Antonio, Texas

ABSTRACT

While safe and efficient compressor installation is the fundamental design objective, most of the design criteria and techniques still in use today fall short of achieving the reliability and performance levels required and possible with the technology available today. On the basis of state-of-the-art analysis and worldwide experiences with the design and evaluation of more than 3,000 plants, this paper defines a new dynamic design criterion for reciprocating compressor and pump installations and presents the techniques required to assure safe and efficient plant operation. Comparison with actual field data further illustrates the accuracy and reliability of presented design methods and techniques.

INTRODUCTION

The intermittent flow of gas through compressor cylinder valves generates gas pulsations which can couple to the piping, producing vibrations and even fatigue failures. These gas pulsations also have detrimental effects on installation performance both in terms of efficiency and the required plant maintenance. While these byproducts of compressor operation are generally known, it is also essential to understand the effect of the piping systems attached to the compressor and the relation between pulsations and plant reliability.

The level of pulsations generated by a compressor is related to a number of parameters which include operating pressures, cylinder horsepower, and capacity being handled, compression ratio, cylinder clearance volumes, phasing between cylinders, thermodynamic gas properties and compressor cylinder and valve design. The generated pulsation levels are usually not excessive without further amplification by piping resonances, which, in low pressure drop systems, can range from a factor of 10 to as high as 300. Thus, it is the effect of the attached piping that is usually responsible for excessive pulsation buildup in the system.

Considering that compressor valves are pressure-operated devices, the suction (discharge) valve will open when cylinder pressure drops below (starts exceeding) the pressure just outside of valve. However, this pressure is not a dead weight line pressure, but a highly dynamic wave which must be known if the actual compressor cycle is to be determined. Similarly, the pulsation buildup in the piping cannot be determined without knowing the dynamic energy generated by the compressor, which, in turn, is dependent on the pulsations in the attached piping. Consequently, neither the compressor horsepower and capacity nor pulsations in the piping systems are definable on the basis of compressor geometry and operating plant parameters alone without a total system analysis including dynamic interaction between compressor and the attached piping systems.

The frequencies of discrete pulsation components generated by the compressor consist of the fundamental frequency corresponding to compressor RPM and multiples of this frequency. The still found practice of calling 2 X RPM the fundamental frequency of a double-acting compressor cylinder fails to recognize the fact that in real life the cylinders are not truly double-acting (except possibly those with a tail rod); even if they were, the opening of underloader pockets or lifting valves on one end of the cylinder would make the cylinders at least partially single-acting. Ultimately, it will be the response of the attached piping systems that will determine which pulsation components will be dominant.

Before vibrations can be produced, pulsations must couple to the mechanical system. Typical coupling points are closed ends of pipe or vessels, bends and flow restrictions such as orifices, reducers, etc. The produced acoustical shaking forces will excite vibrations the magnitude of which will depend on the location of mechanical natural frequencies relative to the frequencies of acoustical shaking forces and the amount of damping in mechanical system. The produced vibrations may or may not be a problem, depending on the cyclic stress levels they produce, since it is not the vibration but stress that fails pipe. Actually, it is the cumulative effect of cyclic stresses over the expected plant life that will determine failure probability of a proposed system.

REQUIREMENTS FOR A MEANINGFUL DYNAMIC PLANT DESIGN CRITERIA

The efforts to define dynamic design criteria are almost as old as the use of reciprocating compressors and pumps. Recognizing that piping vibrations were being excited primarily by pulsations, the first criteria were evolved limiting maximum residual overall pulsation levels to a fixed percentage of line pressure (usually 2%) at the line connection of pulsation filter. While this was progress, the experience has shown that it was inadequate to assure freedom from excessive vibrations and resulting fatigue failures.

Of the additional concepts developed, two became popular. One of the criteria adjusted the allowable overall residual pulsation levels to line pressure ($PUL\% = 15/\sqrt[3]{P}$ where P = average line pressure in psia) while the other related allowable residual pulsation levels to pulsation frequencies. (For normal service: 2% at fundamental frequency, 1% at second harmonic and 1/2% at higher frequencies). While these concepts are more realistic, they still neither assure freedom from fatigue failures nor efficient compressor operation.

The intent of such residual pulsation level criteria is to control piping vibrations, yet there is no direct relation between the overall pulsation levels and vibrations they will produce. This is illustrated in Figure 1 showing the amplitudes and frequencies of discrete pulsation and vibration components at the same point in the piping system. The highest

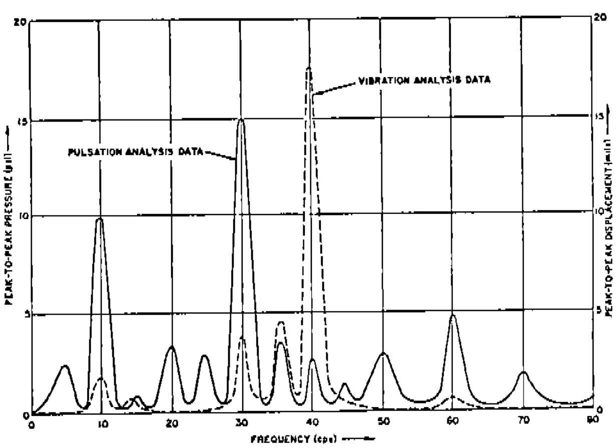


Figure 1

vibration levels do not correspond to the highest pulsation peaks; thus, eliminating these pulsation peaks and consequently reducing the peak-to-peak pulsation levels by a factor of 3:1 will not reduce vibrations unless the minor pulsation peak at 40 Hz is also reduced.

Typical complex pressure wave contains a number of significant components. However, their frequencies and, particularly, amplitudes usually cannot be determined without a harmonic analysis of the complex wave. This is clearly illustrated in Figure 2 showing a complex pressure recording and its harmonic content.

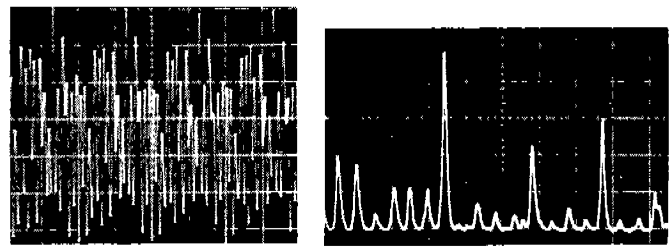


Figure 2

As was stated earlier, the level of pulsation buildup depends on the presence of acoustic resonances in the piping and the relation of acoustical resonant frequencies to frequencies of engine harmonics. Nevertheless, most residual pulsation level guarantees refer to unit operation at design speed and load. The problems this can lead to are shown in Figure 3. At rated speed the maximum pulsation level was 8 psi p-p at 51 Hz (20 psi p-p full scale) while dropping compressor speed only 15 RPM

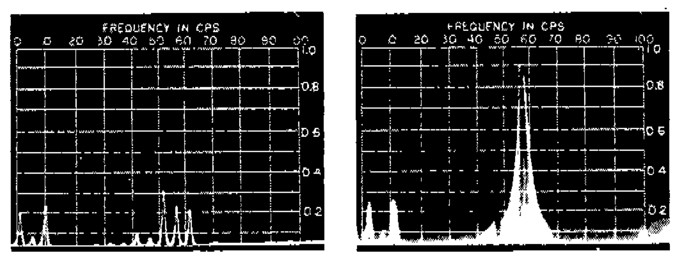


Figure 3

increased the maximum buildup to 18 psi p-p at 57 Hz. Consequently, design criteria must consider the anticipated operating range of the compressor units rather than the design condition.

Finally, the practice of specifying line flange at the snubber as a pulsation guarantee point is not realistic since pulsation levels further away from the compressor can be much higher than those at the snubber flange. An example of this happening is illustrated in Figure 4. Here the pulsation level at midpoint of the suction lateral (9.2 psi at 21 Hz) is 3.7 times higher than the maximum pulsation level at line flange of the snubber. Thus, to be meaningful, residual pulsation level criteria should apply to any point in the system.

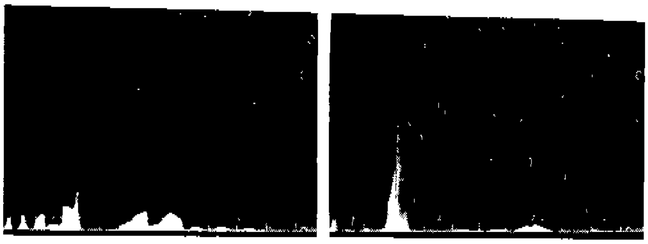
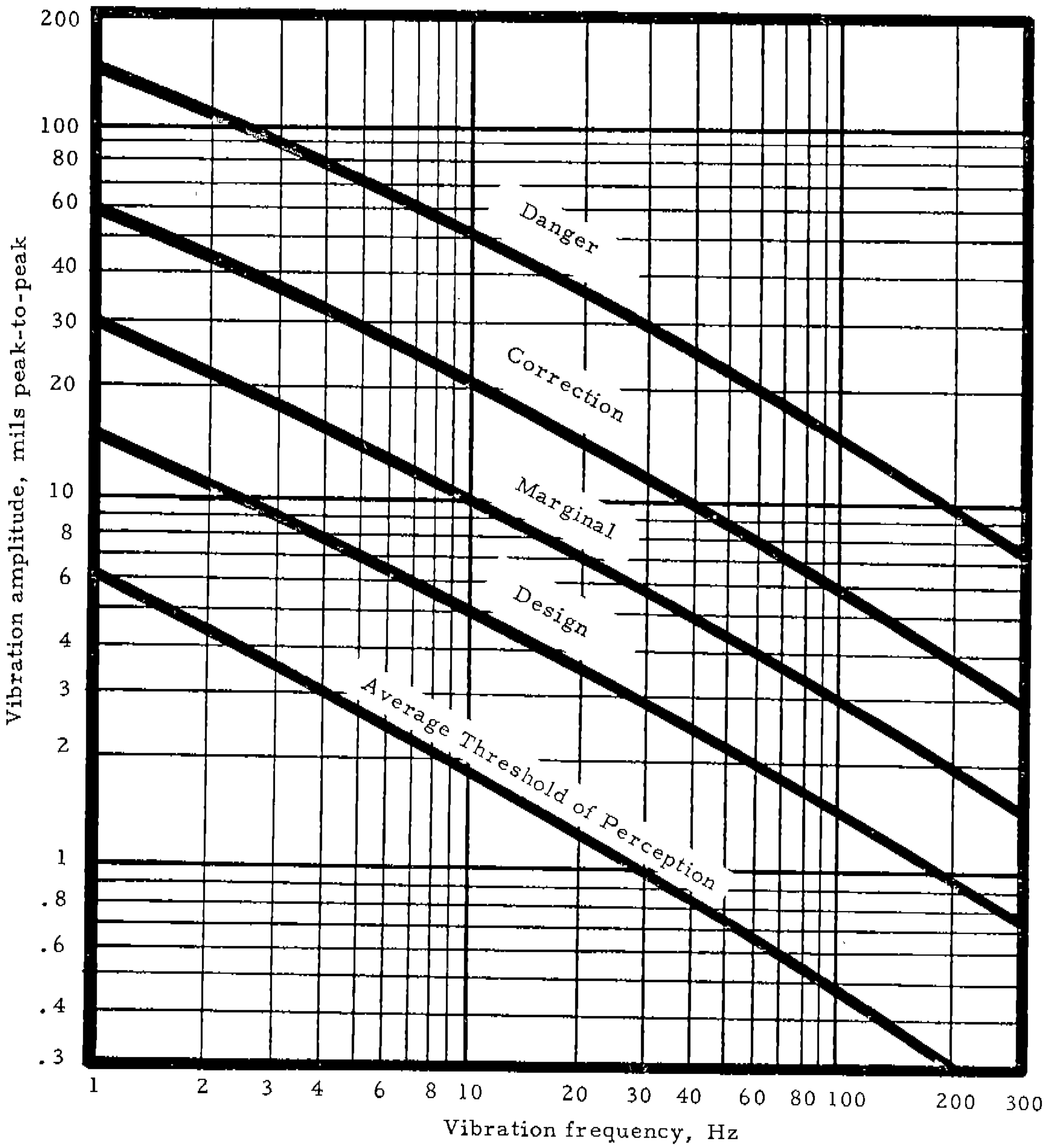


Figure 4

In more recent years increasing efforts have been directed toward introducing vibration levels in addition to or in place of maximum allowable pulsation level criteria. Since vibration control is one of the basic reasons for pulsation control, such criteria are much more meaningful. Starting with a fixed maximum allowable vibration level (typical: 8 mils p-p) such criteria progressed to "Frequency-Variable Allowable Vibration Level Criteria" as shown in Figure 5. In spite of statistical probability on



Note: Indicated vibration limits are for average piping system constructed in accordance with good engineering practices. Make additional allowances for critical applications, unreinforced branch connections, etc.

Figure 5

which such criteria are based, great caution should be used to achieve the desired vibration control.

The requirements for a truly meaningful dynamic plant reliability criteria relating pulsations to vibrations and to stress are shown in Table 1. As it can be seen, only cyclic stresses are directly related

While this PCRC Standard is still valid today, it is not specific as to the kind of physical evidence (pulsation, vibration, or stress levels) to result from the study. Consequently, this criterion is difficult to enforce other than to assure that the required level of effort was put into analysis of a compressor or pump installation design.

PULSATIONS ↓	Generated by compressor or pump action with the amplitudes controlled by the location of acoustic resonances and damping in pulsation filters and piping system.
ACOUSTICAL SHAKING FORCES ↓	Produced by coupling of energy at bends, closed valves or capped ends, restrictions, and similar points in the system with levels dependent on the degree of coupling.
VIBRATIONS ↓	Levels dependent on location of mechanical resonances relative to the frequency of shaking forces, point of coupling relative to the mode shape and amount of damping.
CYCLIC STRESSES ↓	Strains due to vibration of components relative to each other or to supports result in stresses the levels of which are also dependent on stress intensification factors present.
FAILURE PROBABILITY	Probability of fatigue failures determined not only on the basis of calculated cyclic stresses but also their cumulative effects during projected installation life.

Table 1

to failure probability and any use of pulsations and vibrations requires first a determination of cyclic stress levels they would be expected to produce.

About 20 years ago a group of 16 major gas companies formed a so-called "Pipeline and Compressor Research Council" (PCRC) under the auspices of the Southern Gas Association* and engaged Southwest Research Institute (SwRI) to develop new technology for compressor plant design and evaluation. The research efforts started with the development of better technology for control of pulsations at the design stage culminating in the development of the SGA-Analog Simulator, development of analytical computer programs for the analysis of mechanical system response and vibration control and the development of related technology required in the design and analysis of compressor installations. These efforts brought new understanding of the requirements in plant design and resulted in a formulation of minimum standards for a so-called "SGA-Compressor System Design" (see Table 2). The intent of these minimum standards was to assure a level of effort consistent with the technology that was available at that time, recognizing the need to combine acoustical and mechanical analysis and optimization of the design on system rather than component basis.

Minimum Standards for "SGA-Compressor System Design"

As a minimum requirement, an "SGA - Compressor System Design" shall consist of duplicating on an SGA Dynamic System Analog the compressor unit(s) and associated piping systems to a point where piping changes will have only insignificant effect on the parts of the system under study and in determining the acoustical characteristics of the design. Furthermore, the design shall consist of determining the mechanical and stress characteristics of the compressor and piping system with the help of digital computational techniques and other methods available to PCRC member companies and in evaluating these acoustical and mechanical characteristics to determine system parameters mutually compatible with all major design requirements such as pulsation and vibration control, compressor cylinder operation, economic considerations, operating and maintenance requirements, etc.

Table 2

*Current PCRC Member Companies are listed in the Appendix.

Performance aspects in the design of compressor and pump installations are generally a user's responsibility. There are three basic performance factors involved in controlling pulsations and vibrations in the design of a plant. Pulsation at compressor valves may cause deviations in cylinder loading and loss in compressor efficiency. They can also cause valve vibrations and thus excessive compressor maintenance. The pressure drop associated with the design of pulsation filters as well as the additional pressure loss in the piping systems due to the pulsations are direct performance losses. Finally, vibrations will increase maintenance requirements and may cause plant down time. It behooves one, therefore, to include design performance factors such as pressure drop and maximum allowable pulsation levels at compressor valves in plant reliability assurance criteria to provide a common denominator in evaluating various pulsation filter designs for a maximum tradeoff between performance and pulsation filter design and effectiveness.

To be complete, dynamic plant criteria should also define when and how it is to be used. For example, in case of a 12,000 BHP unit, no effort should be spared to base the design on as complete an analysis as technology permits, while in case of a 50 BHP field unit, a quick analytical check assuring control of overall pulsation levels and good installation practices might be quite adequate.

Meaningful dynamic plant design criteria should thus involve total system analysis and be based on allowable cyclic stress levels rather than vibration or pulsation since only the cyclic stresses are directly related to fatigue failure probability. Any use of pulsation or vibration as a measure of failure probability requires first relating them to the cyclic stress levels they would be expected to produce.

Finally, dynamic plant reliability criteria should not only assure freedom from fatigue failures, but also specify when and what kind of dynamic analysis is to be performed and to provide a common denominator for evaluation of effectiveness and economics of possible solutions.

DYNAMIC RELIABILITY AND PERFORMANCE ASSURANCE CRITERIA

The comments made earlier, based on 20 years of continuous applied research and the experiences with the design and evaluation studies of over 3,000 worldwide compressor and pump installations, provided the background for formulating meaningful dynamic reliability and performance design criteria. The recommendations listed below are thus consistent with the current practices and design technology available today and represent, in the author's opinion, the essential requirements in compressor plant design.

(1) All reciprocating compressor and pump applications require the use of pulsation suppression devices except instrument utility, starting air compressors which discharge directly into the air receiver, and similar auxiliary applications.

(2) Pulsation suppression devices may be:

- (a) Volume bottles defined as empty vessels with a diameter at least twice that of the line connection;
- (b) Pulsation filters and attenuators based on acoustical suppression techniques.

(3) The size of volume bottles and pulsation filters should be established in the design study; however, their per cylinder (plunger) volume shall not be less than 10 times the cylinder or plunger total swept volume.

(4) The design techniques which may normally be used for plant reliability and performance assurance analysis range from:

- A - The use of standard analytical techniques, to
- B - The design of pulsation filters with the aid of acoustical piping analog analysis, and to
- C - SGA-Analog study of complete system including acoustical and mechanical interaction of compressors, suppression devices, and piping systems.

Design methods (A) and (B) are basically limited to acoustic evaluation for the purpose of achieving the desired pulsation suppression. The use of piping analog analysis in design method (B) is thus also limited to pulsation analysis of the system under study.

The general objectives of design method (C) are to conduct a total system study simulating the entire compressor, pulsation suppression devices, piping, and equipment system considering dynamic interaction between all these elements, as well as to conduct a mechanical and dynamic stress analysis of the system to the extent required to achieve the design objectives. The steps required to accomplish the objectives of design method (C) should include the following:

(1) Determination of the acoustical response of the system including the amplitude and spectral frequency distribution of pulsations throughout the system. This analysis shall include the effect of dynamic interaction between cylinder - dampener - piping and dampener - piping effect on compressor cylinder performance.

(2) Determination of the mechanical response of the piping system including mechanical natural frequencies and mode shapes of the compressor cylinder - manifold system. This analysis should also establish allowable limits of pulsation-induced shaking forces based on the cyclic stress levels they can produce.

(3) Determination of the required pulsation suppression based upon the acoustical and mechanical responses and their interaction. To obtain the desired pulsation control, selective use should be made of both acoustical and mechanical control techniques. These include the elimination of coincidences between acoustical and mechanical resonances, the use of acoustical filtering techniques, and changes in mechanical configurations.

The selection of types of design and evaluation analyses should take into account equipment size and cost, the service it will perform with special consideration to critical or hazardous applications, performance requirements including significance of down time, present and future system and plant requirements, and the anticipated plant life. The recommendations listed below pertain to typical installations and should be used as a guide:

- For all units with driver horsepower in excess of 500 BHP, design method (C) is recommended. Design method (C) is also recommended for smaller units in critical service or in critical locations.

- For all other compressor units, design method (B) is recommended except for units with 2 or less cylinders per stage of compression,and for all units rated below 150 BHP with final discharge pressure of less than 500 psia design method (A) may be used.

Considering the guidelines presented in this section, the recommended design criteria for reliability and performance assurance of reciprocating compressor and pump installations are as follows:

(1) Pulsation-induced vibrations shall not cause cyclic stresses in excess of the allowable endurance limit of the material. For carbon steel below 700°F temperature, a stress value of 26,000 psi peak-to-peak should be used with all other stresses within applicable code limits.

Based on experience, this requirement is usually satisfied when the residual pulsation levels in the piping systems do not exceed values expressed by an empirically derived equation (1)

$$PUL\% = \frac{300}{\sqrt{P \times ID \times f}} \qquad (1)$$

Where:

PUL% = % allowable peak-to-peak pulsations at any point in the piping system.

P = average line pressure in psia

ID = internal pipe diameter in inches

f = pulsation frequency in Hz; i.e.,

$$f = \frac{RPM \times N}{60}$$

where RPM is the compressor speed and N = 1, 2, 3... corresponding to fundamental and harmonics of compressor speed.

A nomogram for convenient determination of allowable residual pulsation levels is presented in Figure 6.

(2) Unless specified by the user, pulsations of compressor cylinder valves shall not cause more than 5% compressor cylinder overload nor more than 3% loss in compressor cylinder efficiency.

This requirement is usually met when the pulsation levels at compressor valves do not exceed the levels defined by the empirically derived equation (2).

$$PUL\% = 8\sqrt{\frac{R-1}{R}} \qquad (2)$$

Where:

PUL% = % allowable peak-to-peak pulsations at compressor valves.

R = stage compression ratio.

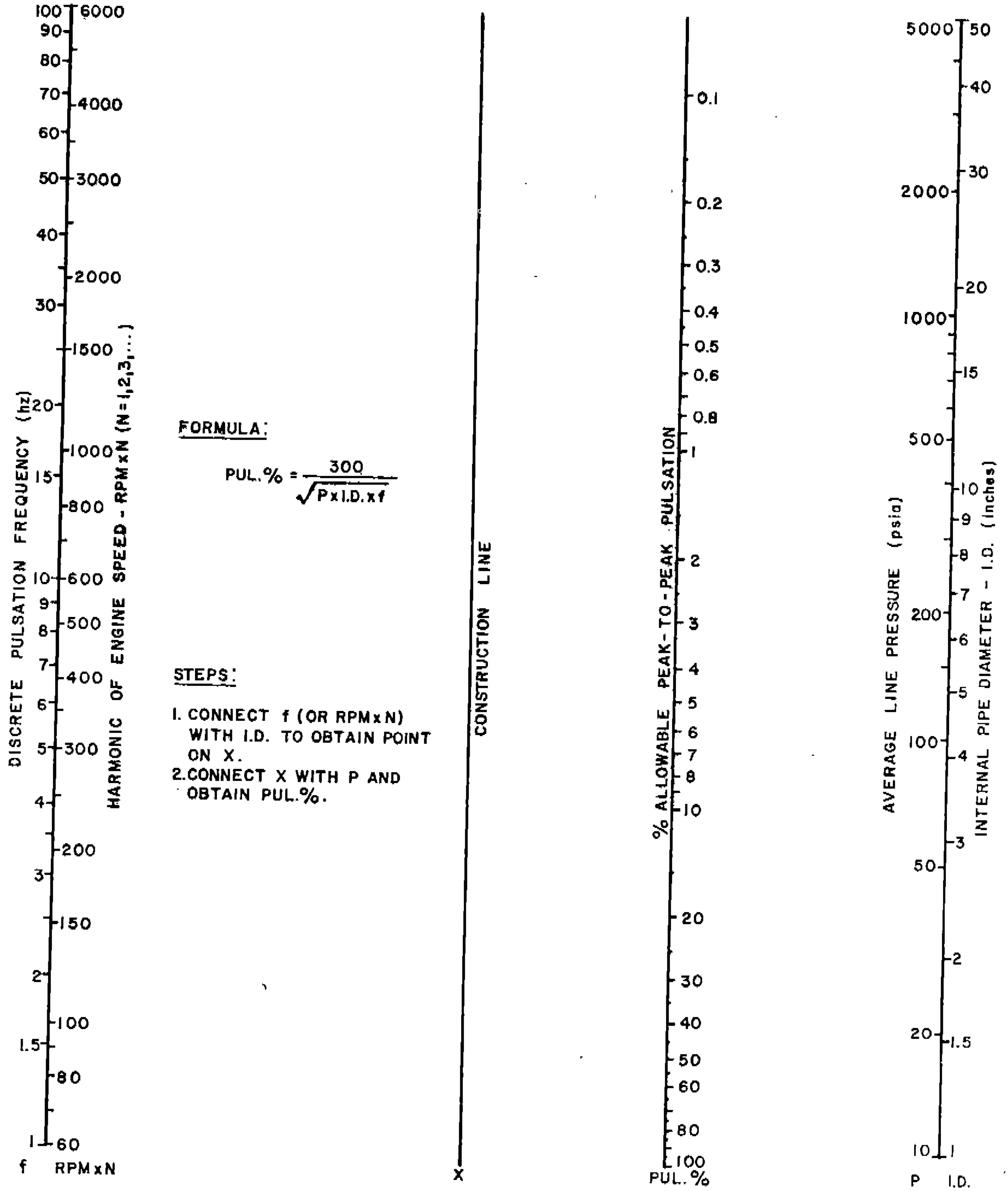


Figure 6

(3) Unless specified by the user, the dead weight pressure drop through the pulsation suppression devices shall not cause more than 3% loss in compressor efficiency.

This requirement is usually fulfilled if the dead weight pressure does not exceed the values expressed by the empirically derived equation (3).

$$\Delta P\% = \frac{5(R-1)}{3R} \qquad (3)$$

Where:

$\Delta P\%$ = % allowable dead weight pressure drop through pulsation suppression device.

R = Stage compression ratio.

CONCLUSIONS

Inadequacy of residual pulsation levels design criteria still in use today prompted the development of plant reliability and performance assurance criteria presented in this paper which is based on dynamic stress. In defining design techniques, recognition was given to established design methods available to the industry. In suggesting the level of design effort, the current practices of the major companies were taken into consideration. The performance considerations were left as user's decision, yet safeguards for assuring reasonable performance were incorporated.

The empirical equations correlating residual pulsations and pressure drop to resulting cyclic stresses and performance effects are quite reasonable for a large number of recent design studies; however, these equations are intended as a guide only and not as a substitute for basic requirements in the recommended design criteria. Further efforts could produce better correlation; however, a reliable and practical dynamic design concept using residual pulsations as the only criterion is not believed to be feasible.

Part 2 of this paper discusses available and required design technology for study of larger compressor units requiring the design method (C) type of study which calls for a total system analysis.

APPENDIX

PIPELINE AND COMPRESSOR RESEARCH COUNCIL
of the
SOUTHERN GAS ASSOCIATION

American Natural Gas Service Company - Detroit, Michigan
Amoco Production Company - Tulsa, Oklahoma
Arkansas Louisiana Gas Company - Shreveport, Louisiana
Brown and Root, Inc. - Houston, Texas
Burgess Industries - Dallas, Texas
Cities Service Gas Company - Oklahoma City, Oklahoma
Colorado Interstate Gas Company - Colorado Springs, Colorado
Columbia Gas System Service Corporation - Columbus, Ohio
Consolidated Natural Gas Service Gas Company, Inc. - Pittsburg, Pennsylvania
Cooper Industries, Inc. - Houston, Texas
De Laval Turbine, Inc. - Houston, Texas
Dresser Industries Machinery Group, Clark Bros. Co., Inc. - Olean, New York
El Paso Natural Gas Company - El Paso, Texas
Esso Research and Engineering Company - Florham Park, New Jersey
Florida Gas Transmission Company - Winter Park, Florida
General Electric Company, Schenectady, New York
Houston Pipe Line Company - Houston, Texas
Hudson Engineering Corporation - Houston, Texas
Ingersoll-Rand Company - New York, New York
Kansas-Nebraska Natural Gas Company, Inc. - Hastings, Nebraska
Lone Star Gas Company - Dallas, Texas
Mississippi River Transmission Corporation - St. Louis, Missouri
Monsanto Company - St. Louis, Missouri
Natural Gas Pipeline Company of America - Chicago, Illinois
Northern Illinois Gas Company - Aurora, Illinois
Northern Natural Gas Company - Omaha, Nebraska
Oklahoma Natural Gas Company - Tulsa, Oklahoma
Optron, Inc. - Dallas, Texas
Pacific Lighting System - Los Angeles, California
Peerless Manufacturing Company - Dallas, Texas
Phillips Petroleum Company - Bartlesville, Oklahoma
Pioneer Natural Gas Company - Amarillo, Texas
Shell Oil Company - Houston, Texas
Solar Division of International Harvester Company - San Diego, California
Southern Natural Gas Company - Birmingham, Alabama
Southern Union Gas Company - Dallas, Texas
Standard Oil Company of California - San Francisco, California
Tennessee Gas Transmission Company - Houston, Texas
Texas Eastern Transmission Corporation - Shreveport, Louisiana
Texas Eastman Company - Longview, Texas
Texas Gas Transmission Corporation - Owensboro, Kentucky
Transcontinental Gas Pipe Line Corporation - Houston, Texas
Trunkline Gas Company - Houston, Texas
United Gas Pipe Line Company - Shreveport, Louisiana
Vibration and Noise Engineering Corporation - Dallas, Texas
Worthington Corporation - Buffalo, New York