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Reliability and Performance Assurance in the Design of Reciprocating Compressor installations - Part II: Design Technology

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RELIABILITY AND PERFORMANCE ASSURANCE
IN THE DESIGN OF
RECIPROCATING COMPRESSOR INSTALLATIONS

PART II - DESIGN TECHNOLOGY

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INTRODUCTION

The major effort to develop reliable plant design technology goes back some 20 years when a group of some 16 major gas companies formed a so-called Pipeline and Compressor Research Council (see Part I of this paper) and engaged Southwest Research Institute to undertake a research effort to develop the needed technology. The initial emphasis was on the development of better pulsation control techniques using mathematical computer modeling. Soon it was discovered, however, that the simplifications and assumptions required to make computer programs manageable and economically feasible would make the analysis inadequate. The effort was then put on the development of dynamic physical modeling technique which culminated in the design of the SGA-Analog Simulator. The "SGA-Analog", as it is commonly known, has been modified and refined over the years and today represents the most accurate, reliable, and economical method for analysis and control of pulsations in compressor plant design.

Mathematical computer analysis has been the major tool in mechanical system response studies and vibration control. Starting with a simple mathematical model of compressor-manifold system calculating the lowest resonant frequencies, the programs in use today compute not only system resonant frequencies and mode shapes but also the vibration amplitudes and corresponding dynamic stresses. Similarly, the 3-dimensional dynamic piping system analysis programs include forced vibration response analysis and thus are capable to determine the vibration and stress levels in piping system elements.

These acoustical and mechanical design techniques, together with the developed supporting technology such as computer programs for more reliable calculation of thermophysical properties, improved pressure drop calculation techniques, etc., permit compressor plant reliability and performance assurance to the degree impossible only a few years ago.

PULSATION CONTROL TECHNIQUES

Pulsations in reciprocating compressor piping systems propagate as plane waves obeying acoustical laws. Such well known relations as the "organ pipe" and "Helmholtz" equations describe the conditions of acoustic resonance of piping elements or combination of elements such as chokes and volumes. The real systems, however, are not ideal text book examples; consequently, even such an elementary system as an empty surge volume can have resonances quite different from what can be predicted considering the length of the volume bottle. This is illustrated in Figure 1 where the resonances of the actual system were at 60 Hz while the calculated half-wave length resonant frequency was 73 Hz. The reason for the difference is the

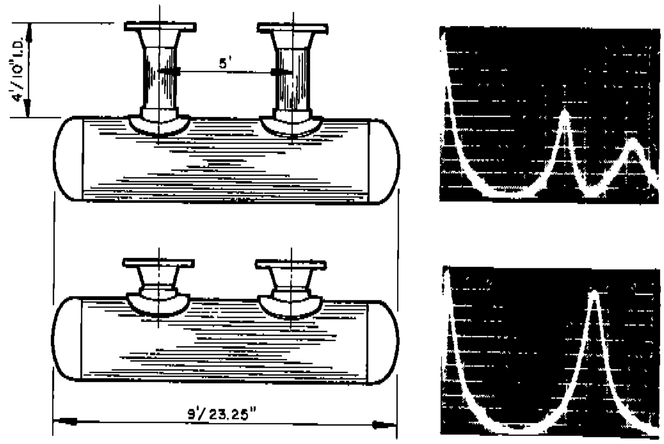

Figure 1

effect of nozzles on surge volume response and, indeed, cutting the nozzles off moves the surge volume resonance to the calculated value of 73 Hz. In typical piping systems the differences between the ideal and actual responses are so great that the aid of a computer or a simulator becomes essential for prediction of piping responses.

The solution to all these problems was found (by SwRI) through the development of an electro-acoustical model of compressor and piping systems. Olsen and others have shown that it is possible to reduce the study of acoustical systems to the analysis of the equivalent electrical networks. Murphy and Chilton, among others, have suggested electrical analog simulation methods for pulsation studies, but it was not until 1955 that the first dynamic system analog of reciprocating compressor installations was developed by SwRI into a practical design facility.

Selecting a modified classical analogy between acoustical and electrical systems (Table 1) and choosing a set of conversion factors, the electrical circuits analogous to the basic system components were designed and constructed. As built,

Acoustical Quantity	Electrical Quantity
Pressure	Voltage
Mass Flow	Current
Acoustical Frequency	Electrical Frequency
Acoustical Wave Velocity	Electrical Wave Velocity
Acoustical Impedance	Electrical Impedance

Table 1

this system analog is actually an electrical model of compressor units and associated piping systems with circuits simulating the generation of pulsations, or sound waves, by the compressor and modification of these waves by the piping. In addition, it duplicates the static pressure and flow conditions from the suction piping through the compressor cylinder and into the discharge piping.

The basic elements of the "SGA-Analog", as it is commonly known are: (1) Compressor cylinder analog duplicating either single- or double-acting cylinders of any dimension; (2) Compressor crankshaft analog permitting phasing of individual cylinders in accordance with the throws on the crankshaft; (3) Crankshaft driver analog duplicating any desired compressor speed or speed range; (4) Line termination analog providing the analog of static line pressures and characteristic impedance terminations, (5) Compressor piping analog duplicating the original piping usually in increments of one foot; and (6) Calibration, control and measurement instrumentation.

Continuous improvements and refinements of the SGA-Analog Simulator greatly extended its capabilities and accuracy. One of the models presently in use at SwRI is shown in Figure 2. It consists of compressor console, suction and discharge

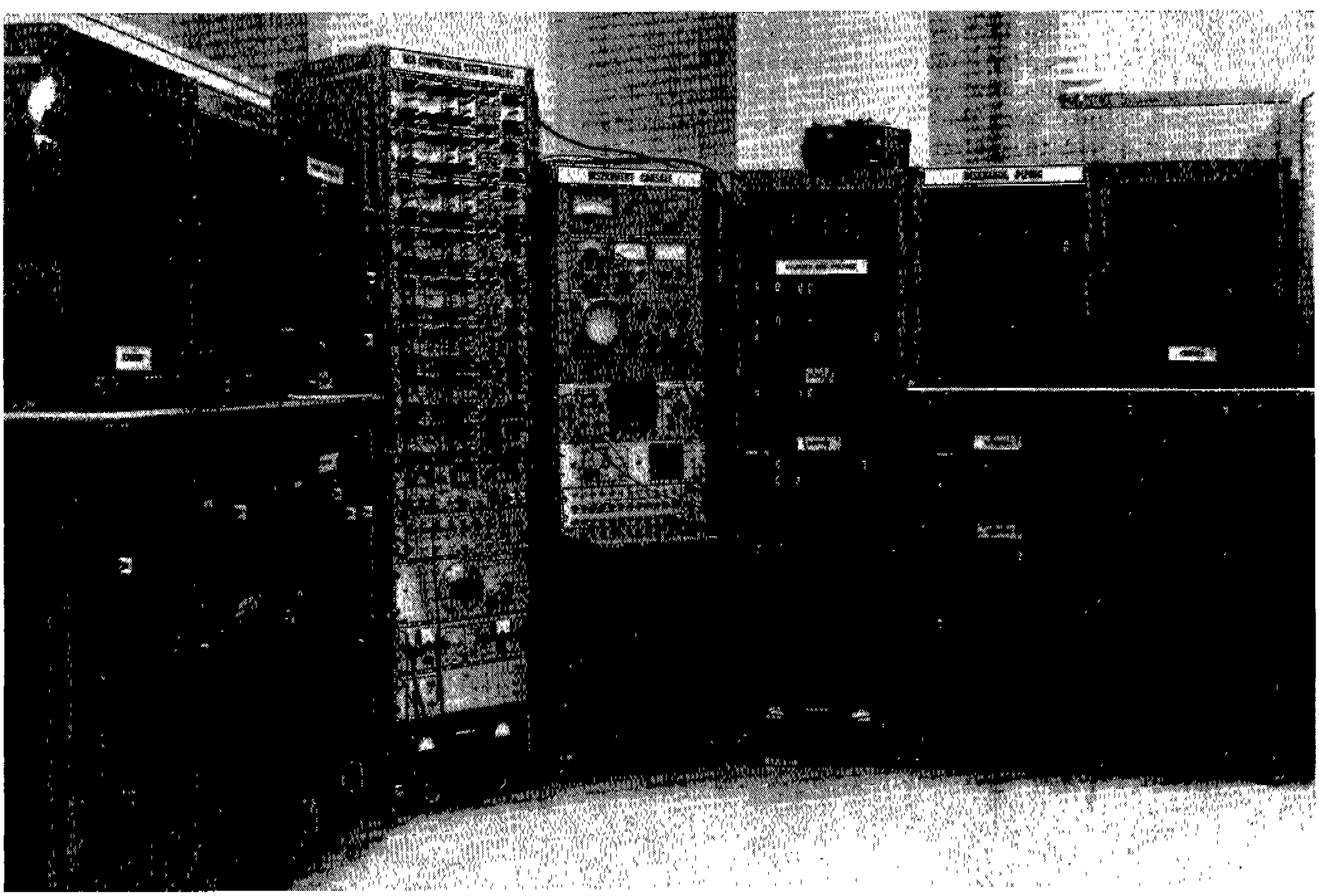

Figure 2

piping consoles, and instrument console. The significant improvements and additions of recent years include a compressor valve analog permitting duplication of such valve parameters as pressure required to open valve, equivalent valve flow area, and valve pressure drop. Improved compressor cylinder phasing techniques permit accurate phasing in critical cylinder arrangements. Nonlinear acoustical resistance analogs duplicate orifices and restrictions in critical applications. Finally, the change to "high Q" piping analog components brought the pressure drop per foot of pipe on the analog close to the required value. Consequently, predicted pulsation amplitudes, even at higher engine harmonics, correlate directly without further re-interpretation and compensation of recorded levels.

The current model of the compressor cylinder analog is shown in Figure 3. In addition to the ability to duplicate any size cylinder from vacuum

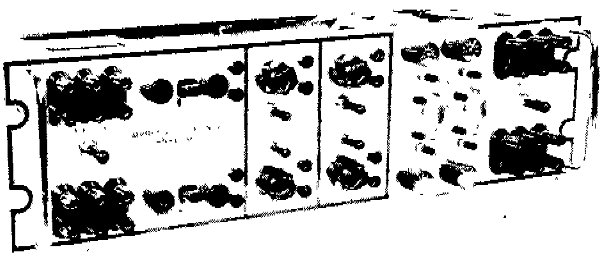

Figure 3

to very high pressures and setting compressor valve parameters, it has provision for convenient opening and closing of unloader volumes, lifting of head-end and crank-end valves and introduction of infinite (in practice: very large) volumes at compressor valves to isolate the effect of pulsations on compressor cylinder operation.

In using compressor piping analog (Figure 4) sections as short as one inch can be duplicated if

Figure 4

required in the study. This means elements with changing diameter or shape such as reducers, spheres, etc., can be easily and accurately represented. The piping analog sections also have provisions for adding additional resistance when this is required for closer correlation of acoustical and electrical Q's.

The information obtainable with the SGA-Analog starts with the determination of acoustical piping response. This data represents pulsation transmission characteristics of a given system with the peaks corresponding to acoustic resonances and valleys, indicating attenuation bands. Typical frequency transmission recording is illustrated in Figure 5. Since the compressor is not operating in such a passive response analysis, a relative amplitude scale is used while the recorded frequency scale is directly in Hz.

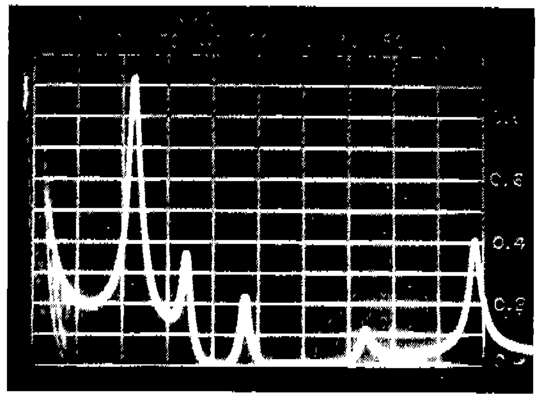

Figure 5

However, the basic information obtainable with the SGA Analog is the amplitudes and frequencies of pulsation components (as well as the overall pulsation levels which can be observed at any point in the system. A sample of such pulsation analysis data is shown in Figure 6. The frequency scale is calibrated directly in Hz and the amplitude scale

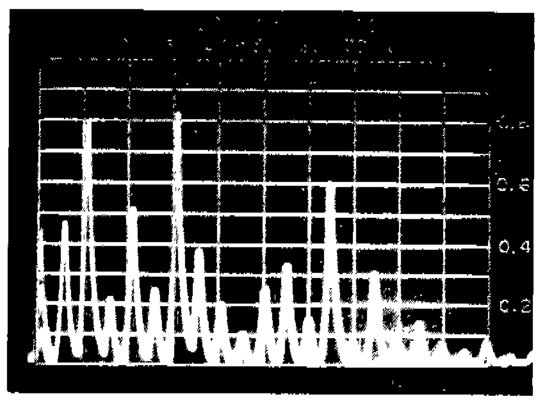

Figure 6

directly in psi. In addition, unit speed can be automatically varied on the analog over the specified unit speed range while pulsation analysis data is being recorded. The result is pulsation signature data showing the envelope of pulsation buildup at a given point in the system, which is more meaningful in the design analysis than the fixed speed pulsation data.

The extension of the pulsation analysis information is the recording of acoustical shaking forces data in pulsation filters, surge volumes, capped headers and similar elements. Special multi-channel vector adders permit the determination of unbalanced acoustical forces directly in lbs. force. Again, such information can be recorded at a fixed compressor speed or as acoustic shaking force signature data which can be obtained for a specified operating speed range of the compressor unit.

The compressor performance information obtainable on the SGA Analog Simulator includes compressor pressure-time and pressure-volume cards and dynamic rod loading data. The most basic of these are the compressor pressure-volume (PV) cards since they display cylinder loading and capacity data. In recording the PV cards, a practice has been adopted to superimpose an ideal PV-card (obtained by placing "infinite" volumes at compressor valves eliminating pulsations) on the actual PV card with pulsation effects. A sample of such ideal and actual PV card data is shown in Figure 7. In the illustrated case, the cylinder overload due to pulsations is about 26%; however, overloads on the order of 50 to 100% and more have been observed on some occasions.

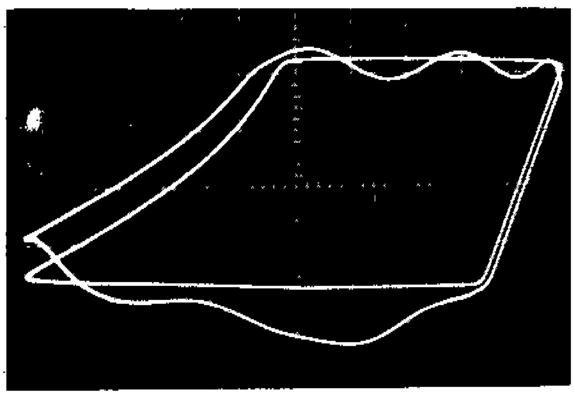

Figure 7

The application of the SGA-Analog Simulator to compressor installation design studies involves calculation of setup and calibration parameters on a computer and assembly of the plant piping systems on the piping analog consoles. Typically, the installation is duplicated and the compressor is "pumping" by the end of the first day. A complete set of pulsation, unbalanced force and compressor performance data is then recorded and analyzed to identify potential problems and possible solutions. The study proceeds with the simultaneous mechanical and performance analysis to assure that the final design represents the most economical solution meeting plant reliability and performance assurance criteria. Typically, the design studies of compressor installations with single stage units, and many of the two stage units, are completed within a period of one week. Since the SGA Analog Simulator is a direct model, it has no computational time. The total effect of any piping change or change in operating conditions is immediately available for evaluation. The engineering time is thus spent on the evaluation and interpretation of the results, determination of feasible alternatives, and physical layout of possible modifications including both the acoustical design changes and the mechanical means of vibration and stress control.

In addition to its proven accuracy and reliability, the SGA Analog Simulator is unique in its ability to fully account for the dynamic interaction between the compressor and attached piping systems. The efficiency and economics of its operation are unmatched and are likely to remain so in the foreseeable future.

One frequently overlooked feature of the Simulator is that it does not have a machine language as is the case with computers. Each piping component on the board is marked as to the length and diameter of pipe it represents. Compressor speed, pulsation amplitudes and frequencies, acoustical shaking force levels, etc., are all marked or measured in the original engineering units; thus, a new design engineer can work with the Simulator within a matter of days.

VIBRATION CONTROL TECHNIQUES

In the design of compressor installations, the dynamics of mechanical systems are an equally important aspect to the analysis and control of pulsations. Excessive vibrations in piping systems are usually associated with coincidence of mechanical natural frequency with the frequency of major pulsation components in the system. Effective pulsation suppression with acoustic filtering could be used to control vibration levels; however, often it is also possible to add a simple clamp detuning mechanical resonance and thus providing equally effective and much more economical solution.

The application of classical beam equations to the evaluation of mechanical piping response requires proper interpretation of end conditions. For example, in real life one finds such terminations such as welded, supported, bolted, anchored, change-of-plane, etc., and most of these are something other than the fixed end beam conditions. Consequently, empirical correction factors have been developed based on actual field experiences, and if properly applied, they give reasonable results. Nomograms for quick calculation of even such configurations as unsymmetrical "Z" bends and "U" bends with concentrated mass have been developed and are in use today. Graphical solutions have also been developed for 3-dimensional bends with or without concentrated mass for in-plane and out-of-plane vibrations.

More complex piping configurations require the help of computer programs. Typical gas scrubber, heat exchanger and bypass piping systems such as the scrubber arrangement shown in Figure 8 can be

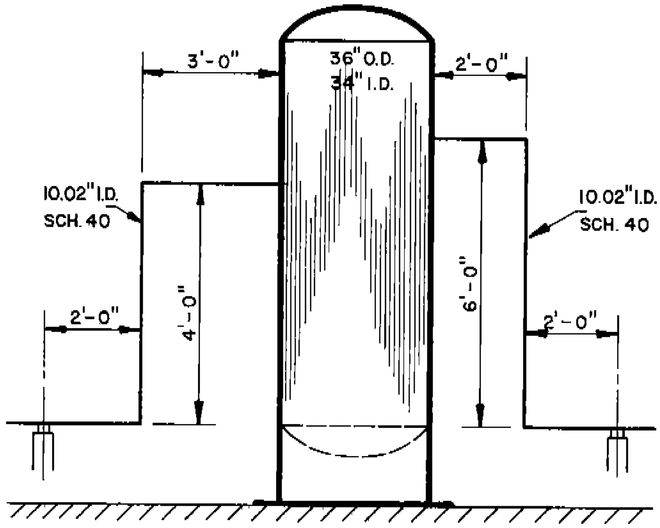

Figure 8

evaluated with a 10-mass Eigenmatrix, Eigenvalue structural computer program calculating the mechanical natural frequencies and corresponding mode shapes as well as vibration amplitudes for a given force input. Sample of such computer printout for the snubber system shown above is illustrated in Table 2.

```
  [illegible]
1 DIAGONAL MASS MATRIX
   1.0400F-01  3.1200E-01  3.6700E-01  4.6030E+00  3.3750E+00
   4.1300F-01  4.1300E-01  1.0300E-01  3.1200E-01  3.6700E-01
   4.6030E+00  3.3750E+00  4.1300F-01  4.1300E-01  3.1200E-01
   3.6700E-01  4.6030F+00  3.3750E+00  4.1300E-01  4.1300E-01
0     1 = SEQ          1.3340E+00 = FREQ (CPS)  OUT OF PLANE
0 AMPLITUDE =     5.3702E-01
   8.0113E-30  5.2639E-30  4.2804E-30 -1.3852E-30  6.3192E-31
  -1.0607E-29 -2.2240E-30  6.8118E-30 -3.1924E-30 -3.0681E-30
  -2.3500E-31  9.5727E-32 -4.1244E-31  2.0182E-30 -3.2818E-02
  -5.6179E-01 -7.3449E-01 -3.6717E-01 -1.0000E+00 -1.5600E-02
0     2 = SEQ          1.2845E+01 = FREQ (CPS)  IN PLANE
0 AMPLITUDE =     5.6505E-01
  -7.4291E-01 -7.4298E-01 -5.7554E-01 -6.0788E-01 -3.0544E-01
  -7.1141E-01 -9.9995E-01 -1.0000E+00 -1.4371E-01 -1.4565E-01
   5.8905E-05  2.9456E-05  1.7845E-01  1.7567E-01  6.6975E-29
   7.6864E-29  4.4053E-30 -1.9023E-29 -1.7914E-29 -7.2975E-29
0     3 = SEQ          4.0458E+01 = FREQ (CPS) * - INDIVIDUAL
0 AMPLITUDE =     1.0918E+00                      PIPING SPANS
   4.6551E-28  2.5219E-28  1.3315E-28 -4.4996E-29  1.8095E-29
  -3.2955E-28  1.2925E-28  4.0509E-28 -1.1848E-28 -1.1618E-28
  -8.2743E-30  2.2858E-30 -1.0957E-29  5.9225E-29 -1.5609E-01
   8.3491E-01  5.9591E+02  3.0017E-02 -1.0000E+00  5.8857E-01
0     4 = SEQ          5.3359E+01 = FREQ (CPS) *
0 AMPLITUDE =     1.2502E+00
  -2.0178E-01 -2.0162E-01 -1.2084E-01 -1.2643E-01 -6.5323E-02
  -6.7613E-02  9.9922E-01  1.0000E+00 -3.9603E-02 -3.8754E-02
  -2.3274E-03 -1.1661E-03 -1.0545E-01 -1.2720E-01  7.9661E-29
   1.7128E-28  3.2738E-31 -1.9342E-29 -2.7487E-29 -2.9735E-28
0     5 = SEQ          6.9516E+01 = FREQ (CPS) *
0 AMPLITUDE =     1.3202E+00
   8.7605E-29  7.0595E-29  2.3386E-29 -5.3524E-30  3.7848E-30
  -5.8589F-29  2.6214E-30  4.4669E-29 -2.6119E-29 -2.6107E-29
  -1.5419E-30  1.7513E-31  1.4581E-29  2.5469E-29 -2.8384E-01
   5.8949E-01 -1.4420E-02 -8.0944E-03 -1.2931E-01 -1.0000E+00
0     6 = SEQ          9.2186E+01 = FREQ (CPS) *
0 AMPLITUDE =     1.6105E+00
   1.0174E-28  8.2328E-29  1.9269E-29 -4.6157E-30  2.6474E-30
```

Table 2

A dynamic structural 3-dimensional computer program has been developed by SwRI based on the use of Eigenmatrix, Eigenvalue techniques. This program is very flexible and permits the analysis of systems with 100 and more masses. It also includes forced vibration solutions, permitting excitation of the system at the points where acoustical shaking forces are present and calculating the corresponding vibration amplitudes throughout the piping system.

The most complex and critical portion of a compressor installation is the response of the compressor-manifold system. The pioneering efforts and present capabilities of the available evaluation techniques have been developed over a period of some 10 years by SwRI for the Pipeline and Compressor Research Council. Starting with simple mathematical modeling, a mathematical model of the elastic and mass properties of a compressor-manifold system has been developed which simulates the response of the physical systems. Such system elements as nozzles and crosshead guides are represented as flexural, axial and torsional springs while the cylinders and bottles are masses. The natural frequencies and modes shapes of the systems are then obtained by solving the differential equations of motion for the masses of the system.

The efforts to use the developed technology were met with the problem of defining the flexibility of the branch connections such as cylinder nozzle-bottle junctions. Since the needed information was not available, an experimental program was initiated several years ago to define the bending and torsional flexibility and stress intensification factors of typical branch connections. The experimental facility is shown in Figure 9 . Basically,

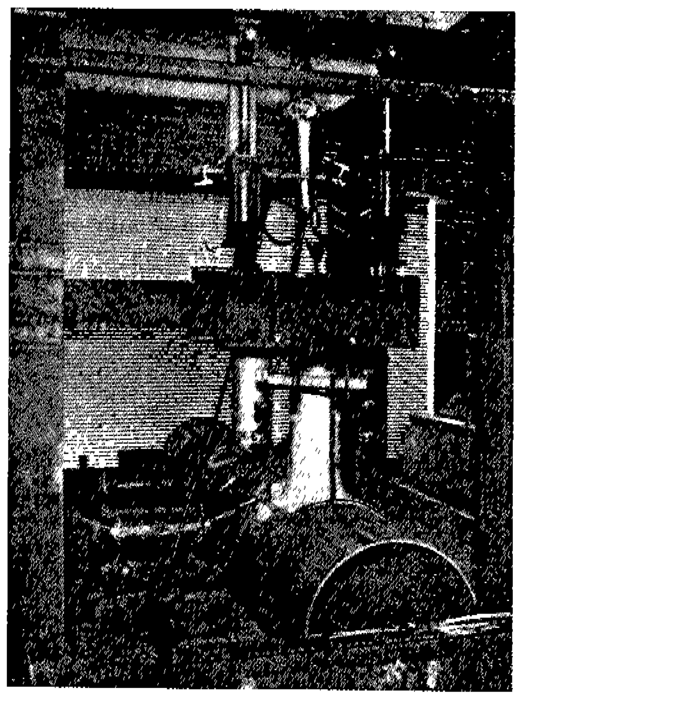

Figure 9

when external forces are applied to the branch connections, the bottle shell does not remain fixed, but rather it experiences some local bending which manifests itself as a rotation of the shell. This is illustrated in Figure 10 which explains why accurate calculations of mechanical natural

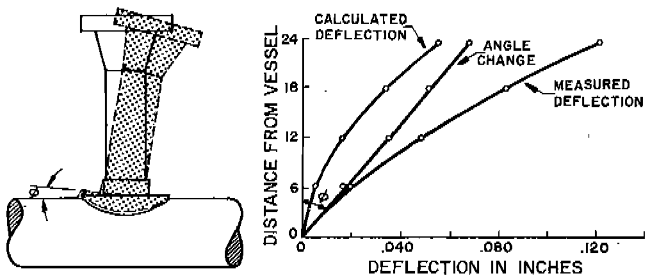

Figure 10

frequencies are not possible without an ability to account for this shell rotation. It is also not surprising that a considerable disagreement usually exists between the calculated and actual thermal stress levels without accounting for the branch connection flexibility. As a result of this experimental program, empirical equations were developed and incorporated into a compressor-manifold program.

While the ability to calculate the mechanical natural frequencies of a compressor-manifold system is essential in compressor installation design, it is not sufficient since the vibration levels and corresponding dynamic stresses in compressor cylinder nozzles must also be known if the reliability of the design is to be assured. The efforts initiated by SwRI several years ago resulted in the development of a computer program using the Eigenvector method which determines the mechanical natural frequencies and mode shapes of compressor-manifold systems and obtains the forced vibration response of the system using various forcing functions at different mass locations.

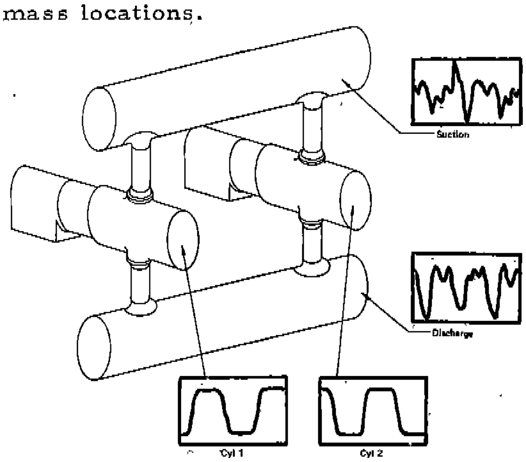

Figure 11

A Fourier expansion of any complex forcing function, including phasing, can be applied at each mass location. Typical excitation of a compressor-manifold system will consist of complex waves of unbalanced forces in suction and discharge bottles and the dynamic compressor loadings as illustrated in Figure 11. This force input data is obtained in an acoustical system study on the SGA-Analog Simulator.

The dynamic displacements resulting from the applied forcing functions of all the masses can then be calculated. A somewhat different approach has been adopted; however, where the vibration amplitudes are calculated for a constant force input of 1000 lbs. p-p. As shown in Figure 12, the computer plots the forced vibration response of

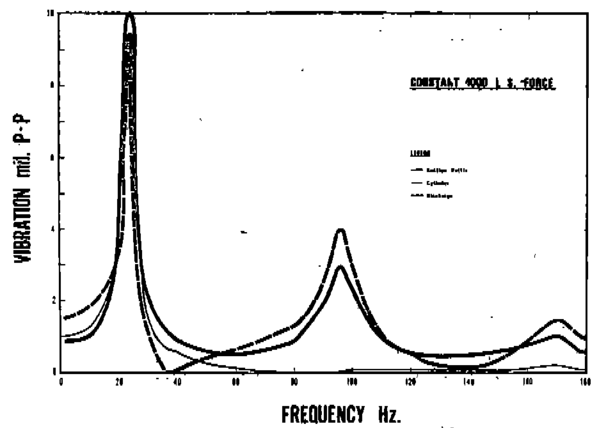

Figure 12

the compressor cylinder and manifolding suction and discharge bottles. The vibrations due to the actual acoustical shaking forces can then be determined for the unbalanced force components of interest.

A further computer program calculates the stress values based upon the relative deflections between connecting members for each harmonic of excitation frequency. Flexural bending and torsional stresses in the nozzles are calculated for each type of motion and then combined into the resultant maximum shearing and principal stresses by the combined stress equations. Similarly, to the prediction of vibration levels, a computer program calculating dynamic stress levels in compressor cylinder nozzles for a constant force input of 1000 lbs. has been written. A computer plot of the same system as shown in Figure 12 is illustrated in Figure 13. Applying the actual unbalanced force levels determined in the SGA-Analog Simulator study and applicable stress concentration factors the reliability of the design is determined in compliance with the plant reliability and performance assurance criteria outlined in Part I of this paper.

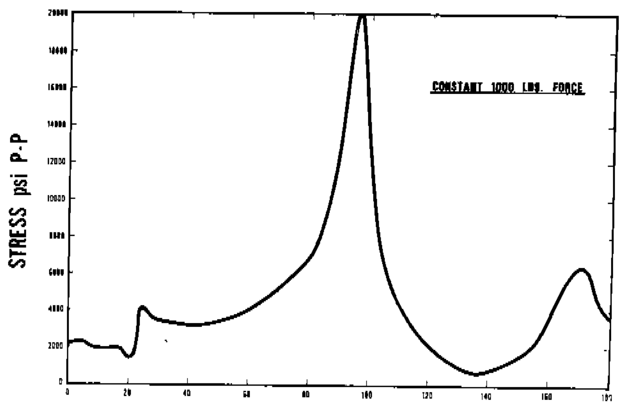

Figure 13

PERFORMANCE AND OTHER DESIGN TECHNOLOGY

In the section on pulsation control it was illustrated how the effect of pulsation on compressor PV card can be determined by superimposing the ideal and actual PV cards. Instrumentation has been developed to measure the percent change in compressor cylinder horsepower (change in PV card area) due to pulsations. Similar techniques have been developed to measure percent change in mass flow due to pulsation effects on compressor cylinder. From this data both the cylinder overloads and cylinder efficiency loss due to pulsation are readily determined. The SGA Analog Simulator is the only available design tool capable of accurate determination of pulsation effects on compressor cylinder efficiency.

The typical pressure drop evaluation techniques still in use today fail to account for the additional loss due to dynamic flow components in the flow of gas and are not capable to account for the differences in physical configurations such as side entrance to a vessel versus end cap entrance nor can special low pressure drop elements be evaluated such as bell mouth entrance and diffusers.However, such problems have been solved and pressure drop technology available today permits unrealistic evaluation of system losses under actual dynamic flow condition and accounting for the actual construction of system elements.

The difficulty in calculating the correct velocity of sound in the system lies in the fact that the operating pressures and temperatures are at times beyond the range for which a given equation of state has been derived. On other occasions, the equation of state constants may not be available for some of the components in the gas mixture. While the problem is not completely resolved, a computer program has been developed based on BWR, Redlich - Kwong and API Equations of State which calculates all three equations, indicates which equation is most valid for the given gas composition and operating conditions, and uses two different methods to calculate the same variable as a built-in consistency check.

The pulsation and vibration design technology, the methods described above and other technology such as dynamic foundation analysis and noise study normally used in the design of a reciprocating compressor installation are shown in Figure 14.

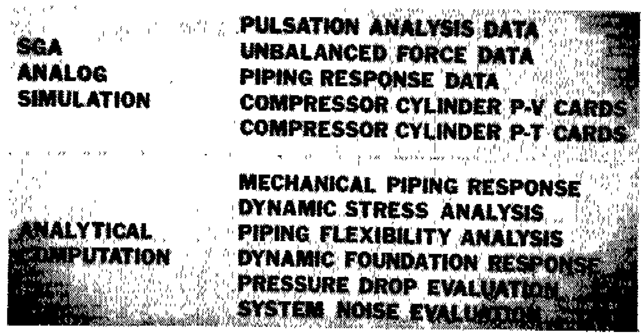

Figure 14

FIELD VERIFICATION OF PRESENTED DESIGN TECHNIQUES

Over the years the validity and reliability of the SGA-Analog Simulator has been verified in dozens of field tests which included independent verification tests by a large number of user companies in the U.S. and abroad. A meaningful example of simulation accuracy was obtained at a gas processing plant in West Texas (Figure 15). Chosen for tests was the fourth stage discharge of a 3000 HP compressor unit operating in the speed range of 400 -

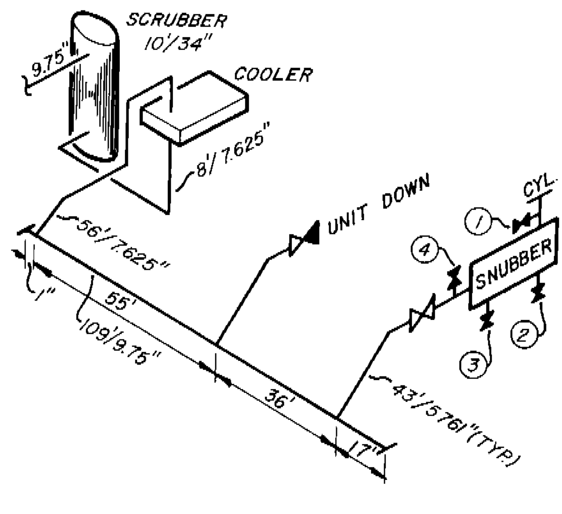

Figure 15

600 RPM and incorporating a complex commercial snubber with several perforated internal elements. To make the comparison more meaningful, four test point locations were chosen: (1) at compressor cylinder flange, (2) and (3) in the snubber, and (4) in the lateral just beyond the line flange of snubber.

Using a real time analyzer in the peak value retention mode, the envelope of pulsation buildup at each discrete frequency was obtained while the compressor speed was varied from 400 - 600 RPM. The system was then simulated on the SGA Analog Simulator using normal established techniques for compressor and piping representation including standard representation of perforated elements and pressure drop. The simulated compressor speed was varied again from 400 - 600 RPM and the envelope of harmonic pulsation spectrum was recorded in the same manner as field data. The results of this comparison test, shown in Figure 16, speak for themselves. Similar correlations have also been observed under extreme operating conditions such as vacuum systems, hyper compressors in polyethylene plants, liquid pump systems studies, etc.

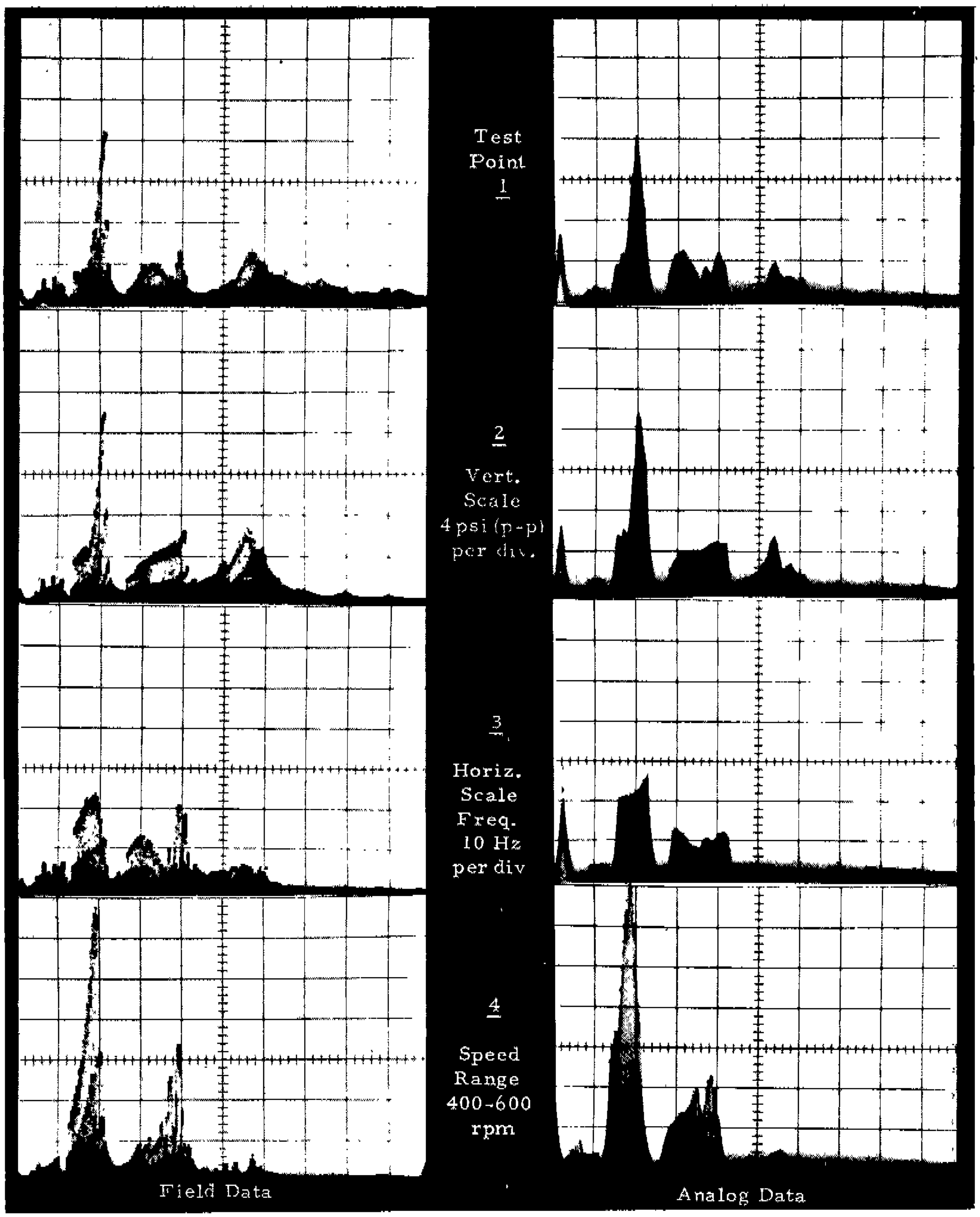

Figure 16

The compressor-manifold system evaluation technology breaks down into calculation of basic mechanical natural frequencies and mode shapes of the system and into a forced vibration response and dynamic stress computer programs computing vibration levels for a given force input and the corresponding stresses at compressor cylinder nozzles.

The accuracy and reliability of compressor - manifold computer program, COMPRE, to calculate the five basic modes have been verified in a great number of comparisons with field data and, of these, four are shown in Table 3 . In general, both in-plane and out-of-plane vibration modes

		Frequencies	
Case	Mode	Calculated	Measured
A	Low	25.1	--
	Rotary	67.2	73.0
	Cylinder Resonance	38.5	39.0
	Suction Cantilever	27.8	29.0
	Discharge Cantilever	18.6	19.6
B	Low	18.7	15.0
	Rotary	76.5	72.0
	Cylinder Resonance	48.7	--
	Suction Cantilever	32.5	--
	Discharge Cantilever	33.0	31.0
C	Low	25.5	24.0
	Rotary	--	--
	Cylinder Resonance	58.4	58.0
	Suction Cantilever	42.8	38.0
D	Low	30.1	29.4
	Rotary	42.5	42.7
	Cylinder Resonance	90.8	93.0
	Suction Cantilever	24.5	28.8
	Discharge Cantilever	18.3	19.7

Table 3

correlate with 3 Hz. Refinements to the computer program since these comparison tests were made two years ago (particularly in the consideration of vertical stiffnesses and their effect on cross - coupling between the modes) permit even closer correlation between calculated and actual measured system natural frequencies.

The application and reliability of the developed technology for calculation of vibration levels and dynamic stresses in compressor-manifold system can be illustrated in an actual case of a compressor plant where repeated discharge nozzle failures occurred after a change in operating conditions.

Since pulsations were expected to be the exciting force, the system was first represented on the SGA - Analog Simulator. As shown in Figure 17, the

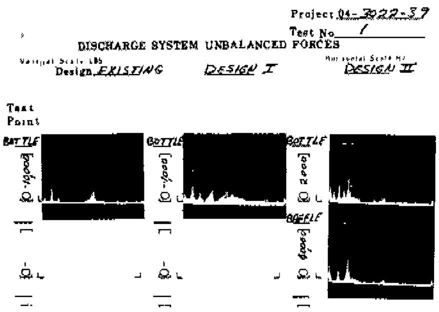

Figure 17

acoustical shaking forces in the existing system were some 7000 lbs. p-p at the 10th engine harmonic (50 Hz) at the rated unit speed of 300 RPM. While this shaking force was suspect, the question of failure cause could not be positively answered without also establishing vibration amplitudes and corresponding cyclic stress levels.

The next step in the analysis, therefore, was to establish the forced vibratory response of the system for a nominal, 1000 lb. shaking force in the discharge bottle. This computed system response is illustrated in Figure 18. Substituting the actual

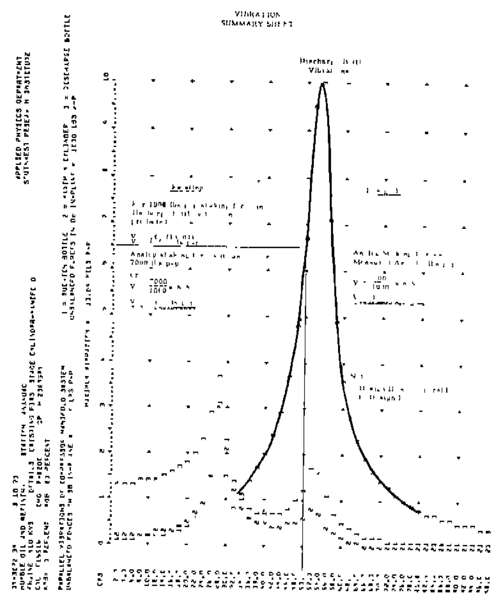

Figure 18

shaking force of 7000 lb. p-p at 50 Hz predicts 61 mils p-p vibration level at this frequency. This level is excessive and would be expected to cause failures. However, the confirmation of this suspicion is obtained in the final evaluation step, a cyclic stress analysis.

The basic program developed by SwRI calculates dynamic stresses without stress concentration factors for a shaking force of 1000 lbs. p-p in the manifolding bottle and nominal compressor cylinder stretch. The computer plot for this sample case is presented in Figure 19. Even without the

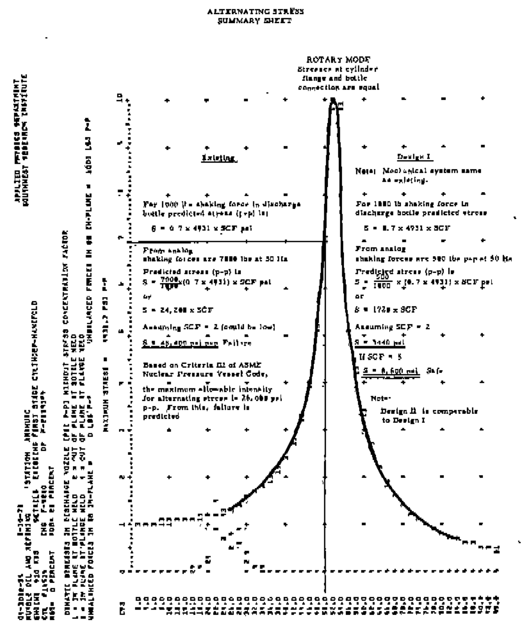

Figure 19

stress concentration factor, the predicted stress at 50 Hz is 24, 164 psi p-p which is close to the maximum safe design level of 26,000 psi p-p as defined in the ASME Code. Considering, however, that in typical well designed and constructed systems, the stress concentration factor is between 2 and 3, the actual dynamic stress in the discharge nozzle is between 48,328 and 72,492 psi p-p indicating failure at the discharge nozzle.

The acoustical system modifications designated as Design I and Design II in Figure 17 reduced the acoustical shaking forces by a factor of 10 to 1 and, when installed (Design II), eliminated further nozzle failures.

CONCLUSIONS

The shortcomings of presently used design criteria prompted the development of the presented plant reliability and performance assurance criteria based on maximum allowable dynamic stress levels and performance safeguards. The application of proposed criteria will assure safe and efficient plant design and is within the capability of design technology available today.

The major elements of such technology include latest generation of the SGA Analog Simulator which offers the most reliable, accurate, and efficient technique for the analysis and control of pulsations in compressor and piping systems. Also included are analytical computer programs for mechanical system analysis capable of calculating not only the mechanical natural frequencies and mode shapes in compressor-manifold and piping systems but also the vibration amplitudes and corresponding dynamic stress levels. Finally, the availability of more reliable computer programs for evaluation of thermophysical properties and new improved techniques for calculation of pressure losses in the system extend the design technology beyond what was available only a few years back.

Even though significant advancements in design technology have been made, the research and engineering effort should be continued to permit reduction of safety margins in compressor installation design without reduction in plant safety and efficiency through further advancement of design technology.

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