OIL/GAS PIPELINING HANDBOOK

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Table of Contents

Table of Contents

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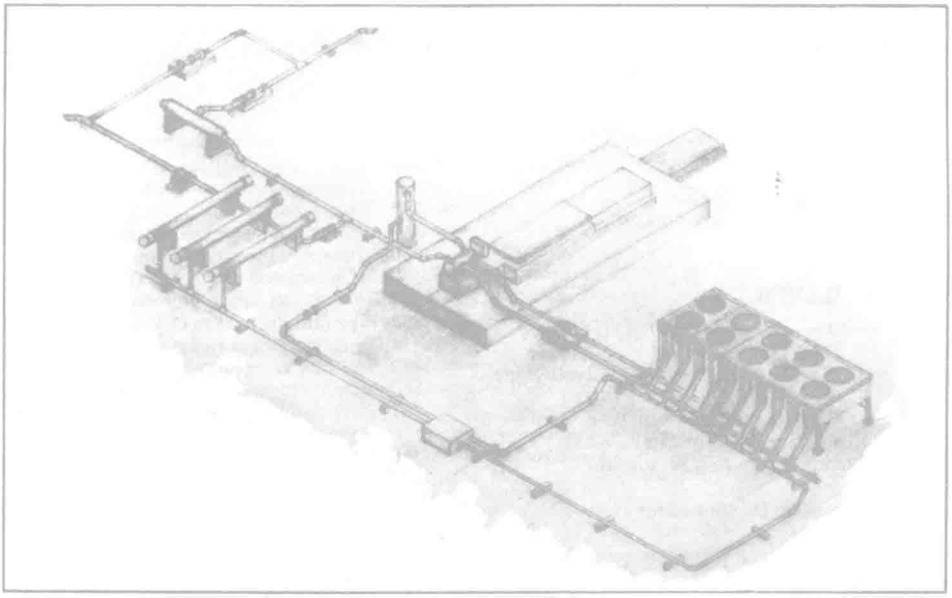

Thermal and pressure expansion can cause large pipe stresses and loadings on the compressor and cooling equipment. Flexible piping design is necessary to ensure smooth equipment operations.

New Ideas Are Evolving In Compressor Station Piping Design

by **Jerry L. Van Norman**, *Senior Engineer, Engineering Technology Analysts, Inc., Houston, Texas*

Natural gas pipeline designers and operators must adapt to a changing environment brought about by:

- Greater demand for natural gas
- Decline in the pressure and flows in existing fields, and
- The remote location of potential new natural gas sources (especially in the Arctic regions).

New methods are being developed to meet these new demands. Natural gas suppliers must now find answers to the engineering problems brought about by:

- Larger diameter piping systems (48 to 60-in. diameter pipe as opposed to the current 24, 36 and 42-in. systems),
- Higher operating pressures (in the order of 1,500 psi to 2,000 psi)
- Higher operating temperatures within compressor stations (in the order of 350 F to 400 F)
- Reduced noise criteria imposed by the Occupational Safety and Health Act
- Extremely severe weather conditions in Arctic regions

All of these engineering problems can team-up to make a piping designer's life uncomfortable.

In the course of solving piping-flexibility problems, it has become necessary to severely question the adequacy of traditional, tried and tested approaches to compressor station piping designs. As the gas transmission compressor stations are being operated at higher temperatures, capacities, and pressures, the traditional "rules of thumb" become marginal. A rethinking of older compressor station piping design principals is occurring, spurred on by the increasing use of high ratio stations.

By the use of actual case problems, we are able to demonstrate some of the design techniques and resulting benefits from utilizing proven computerized stress analysis programs to assist gas compressor station designers in providing timely, effective, reliable and less expensive compressor stations. Some new design guidelines have evolved from successfully solving each of the case problems.

The computer program used to provide this design assistance was developed by Engineering Technology Analysts, Inc. The Extended Piping Flexibility Analysts Program (ETA/EPFAP) was designed for analyses of piping stress and equipment loadings in the design of new installations, modification and/or reevaluation of plant, pump and compressor station piping. This program has been used to solve numerous problems in station piping, process piping, and steam line piping.

Pipe Supports Designed for Flexibility

Traditionally, designers have wanted to tie piping down thoroughly to restrain it against vibration and reduce expansion movements. Many compressor stations that have been operating safely and reliably for many years bear out the success of this approach. A pivotal point in such piping designs is the fact that for a given wall thickness and span, the stiffness of pipe goes up with the fourth power of its diameter. This means, then, that as the pipe gets larger, it becomes much more difficult to restrain it properly.

In practical terms larger pipe diameter means either much heavier, more expensive supports or accepting slight expansion movements in some directions at certain points. On its own, this isn't anything to worry about, except that with the higher pressures

(1500 to 2000 psi) and temperatures (to 400 F in some booster plants) being used more frequently now, the thermal and pressure expansion effects are much more significant than before and cause much higher stresses in the piping. It thus becomes much more difficult to restrain piping in order to minimize movement.

Vibration with its attendant problems of noise and fatigue must also be avoided. Vibration can be avoided through adequate design analysis. There is frequently no reason why piping can not be "freed up" considerably. If a piping system can be freed up, important benefits follow in the form of reduced stresses and reduced equipment loadings. There can also be valuable dollar savings, from elimination of selected pipe supports and simplification of others.

Case Problem No. 1: A 2900-hp booster compressor station in a larger interstate line required operation at up to 980 psig and 160 F. Piping sizes were 14 and 16-in. Use of centrifugal equipment and careful design checks indicated that vibration would not be a problem.

By deliberately freeing up the piping, maximum stress values were generally reduced, with reductions at certain locations of up to 75%. Mechanical loadings on the compressor from the piping were reduced to between 1/5 and 1/70 of their original values. All this was achieved by judicious removal and relaction of pipe supports. Nine concrete pipe support piers were eliminated — a worthwhile cost saving.

Piping Design and Mechanical Performance of Equipment

With the increased use of centrifugal units, static loadings on compressor flanges now can become increasingly important. This is best illustrated by a specific example:

Case Problem No. 2: A 12,000-hp compressor was to be installed at a station that had its yard and station piping laid out in a pattern similar to smaller, earlier designs that had operated successfully for many years. The main difference lay in the increased temperatures existing in the piping for the new station. A second group of recently installed stations with higher operating temperatures similar to the proposed unit had had mechanical problems during start up.

Field engineers attributed these mechanical problems to loadings from pipe expansion. Construction of a computer model to ~~simulate the~~ piping showed that compressor flange loadings were on the order of 70,000-lb, combined with moments of up to 120,000 ft-lb. These were much greater than those occurring in any of the earlier designs and considerably exceeded the compressor manufacturer's criteria.

The effects of various modifications were tried out in the computer model. They demonstrated that by altering support conditions for the piping, compressor flange loadings could be reduced to 1/30 of their previous values. As a by-product, two pipe supports were eliminated and stresss reduced to 1/16 of their previous values. The modifications were incorporated in the design and station start up was free of mechanical problems.

Higher Temperatures

Compressor stations where wellhead pressures are declining and the pressure ratios are high, frequently have compressor discharge temperatures in the 300 to 400 F range. The problems of connecting the coolers, scrubbers and compressor equipment makes the piping design a tough problem since gas flow pressure drops must be avoided — yet the geometry of connecting all this equipment is not easy. On top of this, thermal expansion problems become serious and sometimes dictate how the piping should be routed.

Case Problem No. 3: A new booster station in a pressure declining field had a maximum gas temperature of 390 F. Interstage and after cooling required complex piping. Gas flow problems were compounded by the storage of available space in which to route the piping. Initial flexibility analyses showed that expansion stresses exceeded ANSI B31.8 allowables. Equipment loadings were also excessive.

After several subsequent analyses and discussions with the equipment manufacturers a number of changes were made:

a) Some equipment was relocated
b) Design changes were made by the cooler manufacturer for the cooler/piping connections
c) Piping was rerouted and stress loops were installed in certain locations to alleviate stress and equipment loading problems.

Major rethinking of the piping design was necessary here since the behavior of the piping was no longer like that of other, earlier designs. Temperatures and pressures were similar to those in refineries or in steam lines. Thus piping design had to resemble piping systems commonly seen in refinery process piping or in steam lines. A trade-off was required to match mechanical requirements with desirable gas glow characteristics.

Vibration and Shock Problems

One interesting by-product of these static analyses has been that once a system is modified to satisfy the static criteria, the final solution is usually very close to satisfying vibration or shock loading requirements. This tends to be especially true if the engineer is aware that dynamic criteria might have to be satisfied when he makes the static analyses.

Case Problem No. 4: A reciprocating unit had a failure at the 8 in. connection leaving the pulsation dampening bottle on the discharge side of the unit. The weld quality was not good to start with, but fatigue appeared to be the mechanism that led to the failure.

Static flexibility analyses indicated high loadings at this connection and pointed the way to several simple piping support modifications that reduced the loadings at this connection to a fraction of their formal level.

A dynamic analysis performed by ETA showed that dynamic behavior was also improved. The modifications were made, the system returned to satisfactory service.

Effects of Fabrication Tolerances on Equipment Loading

Compressor station piping system can not be adequately designed without considering both the compressor flange allowable force and moment values and field fabrication practices. This can be illustrated in a case problem.

Case Problem No. 5. a proposed compressor station was to have a centrifugal compressor unit installed instead of a reciprocal unit of a type used in previous, similar installations. One design difference was that compressor manufacturer specified much smaller maximum allowable loadings from the piping acting at the suction and discharge flanges.

A flexibility analysis of the piping revealed that piping alterations were necessary to reduce the compressor flange loadings. After further analyses, it became obvious that although flange loading criteria could now be satisfied, the modifications necessary could easily be negated by the effects of loose fabrication and construction tolerances. There was also concern over the increased cost of modifying piping and adding spring hangers to adjust the piping to satisfy the flange loading criteria.

Consequently, discussions were held with the manufacturer to see whether these allowable loadings were perhaps on the conservative side. The loading criteria were eventually raised (by a factor of 10 to 50) after the manufacturer was convinced of the thoroughness and validity of the analyses (maybe they were bluffing all the time).

The question on the effects of piping tolerances still remained. If the piping misalignment was too much, what would it do to flange loadings? Further flexibility analyses showed that misalignment had a critical effect in certain directions. In addition, the allowable piping misalignment tolerances specified by the compressor manufacturer turned out to be inconsistent with the maximum flange loadings allowed. The solution was to revise fabrication tolerances specified for this particular station's design which would be small enough to avoid overloading the compressor.

Other Problem Areas

Although not illustrated in a case problem, the ETA/EPFAP program has been used to analyze the stress levels occurring in extremely cold (−259 F) LNG systems and to analyze stress and expansion problems resulting from assembly of the system in Arctic regions (generally −20 F) and subsequently operating at considerably higher temperatures. ETA has also developed computer programs to handle sliding (friction) supports, complex piping supports, extreme weights and expansion forces, extreme lateral forces, soil-pipe friction, large deflections and curvatures, earthquake shearing actions and station static and dynamic loadings.

Many of these problems are non-standard structural analysis problems requiring recent "state of the art" non-linear computer programs. ETA has developed, in addition to ETA/EPFAP, ETA/PIPLAY (for marine and below ground piping analysis) and ETA/DYNPIP (for dynamic piping analysis). These are all non-linear structural analysis programs, to solve specific piping problems.

Satsifying D.O.T. Part 192 Requirements

An important by-product of these flexibility analyses has been that they can provide clear proof that the stress criteria of the current version of ANSI B31.8 are satisfied. Inspectors from the Office of Pipeline Safety can have access to clear records that show that every element of the piping has been checked against Department of Transportation records Part 192 stress criteria. Many transmission companies in different parts of the United States have had such analyses run to provide this type of thorough, conclusive evidence that their compressor stations satisfy all D.O.T. stress requirements.

Observations and Conclusions

As a result of the engineering knowledge gained from solving numerous compressor station piping and equipment loading problems, several observations can be made:

a) Higher temperatures and pressures can radically change the loads induced on equipment by piping. These effects can necessitate major redesign of compressor station piping.
b) Much of the traditionally designed compressor station piping is over-supported. Supports can be judiciously removed to provide lower stress levels and lower mechanical loadings on equipment.
c) Equipment manufacturers frequently are very conservative in the loadings that they say can be accepted by their equiment. Thorough analysis of the piping connecting to their equipment and specification of careful piping
d) Piping misalignment can be critical for smooth equipment operation. The piping design should allow sufficient flexibility to allow for misalignment tolerances.

From an analysis of many gas compressor station design problems, ETA has developed a set of six rules that can simplify a piping designer's stress problems.

The Dont's

1. Don't put large equipment pieces (scrubber, compressor, collers, etc.) in a straight line configuration.
2. Don't over-support the pipe. This is *not* an effective way to minimize stress problems.
3. Don't try to anchor larger diameter pipe near the compressor. In most cases it will not work to your advantage.

The Do's

4. Do allow for pipe expansion by allowing for pipe movement.
5. Do make sure that field fabrication tolerances match the design prameters.
6. Do let the computer assist you (at the proper stage) in verifying the integrity of your design.

The methods and case histories described above are derived from numerous analyses that have been made on compressor station piping designs for installations in many parts of the United States and Canada. In each case, the project or design engineer responsible for the design was able to quickly work out whatever modifications were necessary through the use of computerized flexibility analyses.

The ETA Extended Piping Flexibility Analysis Program (ETA/EPFAP) computer program has been developed from experience with many practical design problems in various types of compressor stations. It has proven to be an extremely useful tool for this type of work.

Past "rules of thumb" are no longer adequate with larger lines and larger, more efficient compressor stations since scaling up has changed many of the ground rules. ETA, realizing what's at stake in large gas compressor station piping, and appreciating the cost of failures, has recognized the need to put a really sharp pencil to the problems in its approach to the solution. **P&GJ**

About the Author

Jerry L. Van Norman is a senior engineer specializing in piping flexibility analysis with Engineering Technology Analysts, Inc. He has made numerous contributions to turbine loading and piping overstress problems. Jerry has extensive experience in facilities planning and manufacturing engineering. He received his BS degree in Industrial Engineering and MS degree in Operations Research from Arizona State University. Affiliations include membership in TSPE, NSPE, and Mensa. He is a registered professional engineer in Texas.

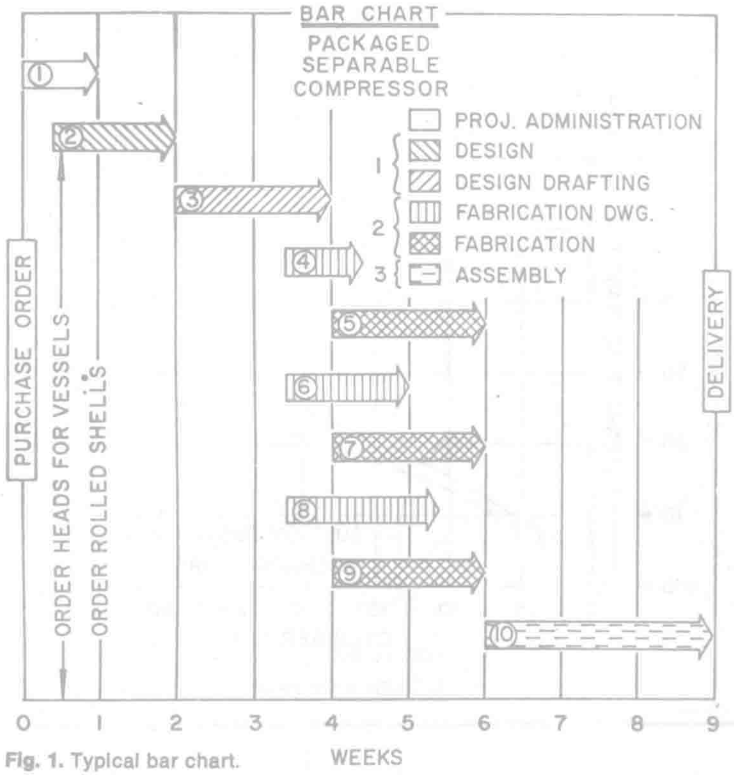

Fig. 1. Typical bar chart.

Gas Piping Design For High Speed Reciprocating Compressor Units

by **Marvin L. Barta,** *Creole Production Services, Inc., Houston, Texas*
and
Thomas P. Bass, *C-B Southern, Houston, Texas*

In many areas of the gas industry, steps are being taken to reduce cost and increase efficiency in an effort to improve profitability. One such area is the construction of facilities to produce and transport gas to market, particularly from remote areas. To reduce onsite construction cost, the industry has gone to prefabricated, or packaged, construction and, in particular, to preassembled production and compression modules. This concept is being extended to complete prefabricated (packaged) plants.

Because of this need for lower cost, the packaging organization of today evolved from a simple fabricator to a highly specialized organization offering custom-engineered, preassembled, and automated compressor modules for installation anywhere in the world with total responsibility from design to delivery and with performance guarantees.

A packager of gas compression plants is defined as an organization whose primary function is to produce a complete compressor plant as a module which can be transported and installed at job site by conventional means. The organization must have the ability to design, select, cost, price, sell, procure, fabricate, and assemble in a short period of time a unit which must perform satisfactorily upon start-up.

A "packaged gas compressor" contains all necessary equipment to raise the pressure of a gas from one level to another. The equipment is arranged as a module whose size and weight allows for conventional transportation to the job site. The module contains equipment to scrub, compress, cool and dehydrate gas to the design conditions with the required auxiliary equipment for unattended operation.

The shortage of skilled craftsmen and the high expense of transporting, housing, and feeding personnel offshore and in remote areas have brought increasing demands from the industry for equipment designed for assembly into a unitized packaged requiring minimum handling and personnel for erection, start-up, and operation.

Other factors that have promoted the packaging concept include:

1) Reduction in time from purchase order to start-up;

2) Reduction in size to minimize space, particularly on offshore platforms;

3) Increased specialization required to design, fabricate and assemble today's plants; and

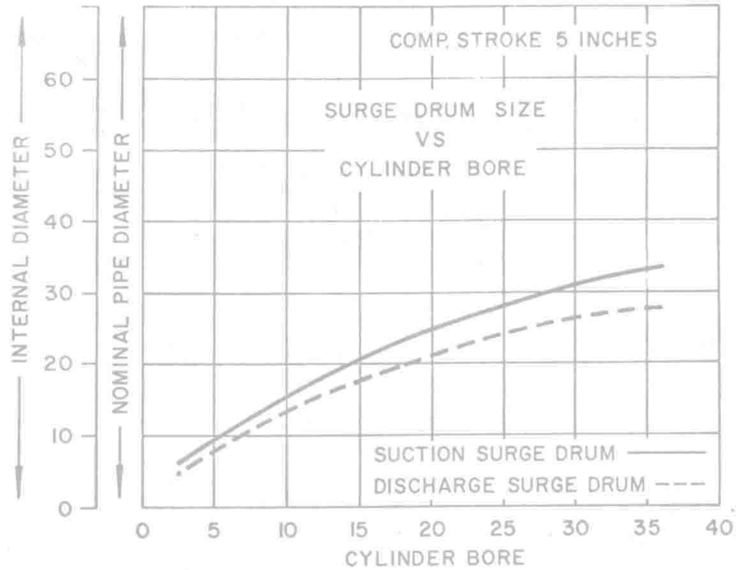

Fig. 2. Surge drum sizes.

4) Loss of production due to conventional onsite construction and completion methods.

Producing A Compressor Module Today

The job begins with a need for gas compression, which is processed through a local sales representative of a compressor manufacturer. The sales representative, who is a sales engineer, is aware of the operating company's needs, conditions, and requirements and performs as its extension. From this information a unit is sized and selected to best meet the needs of the job. This involves a preliminary equipment selection by the sales engineer with final selection for optimum sizing made with the aid of a computer complex staffed with experienced personnel.

Next, a bar chart is developed — from purchase to delivery — using such input data as the unit's delivery date, major equipment delivery dates, work loads, etc.

The greatest part of the time required to produce a packaged unit can be divided into three main categories — design, fabrication, and assembly. The various functions, where possible, are dovetailed to reduce the overall time.

The actual work in design begins with a preliminary study to establish long-delivery items such as gas coolers, cooler drives, scrubber elements, building, cranes, or special piping, so that they can be placed on order immediately to maintain the schedule. The actual design begins with developing the mechanical flow diagrams, which fall into two major categories:

utility flow, and

process flow schematic.

At this point, it should be emphasized that experience is the most important single element in establishing which items must be selected and ordered for arrival in time to assemble into the module. These items must be compatible with the overall design yet to be established, thus requiring a considerable amount of experience and insight. This can be accomplished with success only by a specialist who deals with this problem daily.

Referring to the bar chart (Fig. 1) and concentrating on the time allowed to design a unit and, in particular, the time to design the gas piping, let us begin our detailed analysis. It takes approximately a week from the date of the purchase order to obtain sufficient information to begin design. The time varies somewhat from job to job, but usually by the time engineering has this information the job is into its first week, and very often the heads and shells must now be ordered for the material to arrive in time to begin fabrication.

This presents a problem, and it brings us to the purpose of this article — to describe a method available to us for accomplishing the desired result in the time allowed by the industry.

It is possible, and very effective, to incorporate acoustical filtering in gas piping designs for high speed reciprocating compression equipment. This is due fundamentally to two basic characteristics of this equipment, both of which are attributable to engine speed. They are:

1) Relatively high frequency of the first engine order, and

2) Spread in frequency between engine orders.

High speed equipment is defined as separable equipment in the 750 to 1000 rpm range. For example, a 900 rpm compressor unit has a fundamental engine order at 15 cps, with a frequency spread between engine orders of 15 cps.

Now, if it is required to place a major acoustic response between engine orders which are 15 cps apart, the accuracy of computations becomes less significant than for lower speed equipment. Also to set the cut-off frequency of an acoustical filter equal to the first engine order when it is 15 cps it becomes economically feasible in most cases we encounter.

As speed is reduced, the frequency spread is reduced, thus requiring greater accuracy in computation. Also, as speed is reduced it becomes more and more difficult and expensive to provide for acoustical filtering where the cut-off frequency of the filter is set equal to the first engine order. These points will

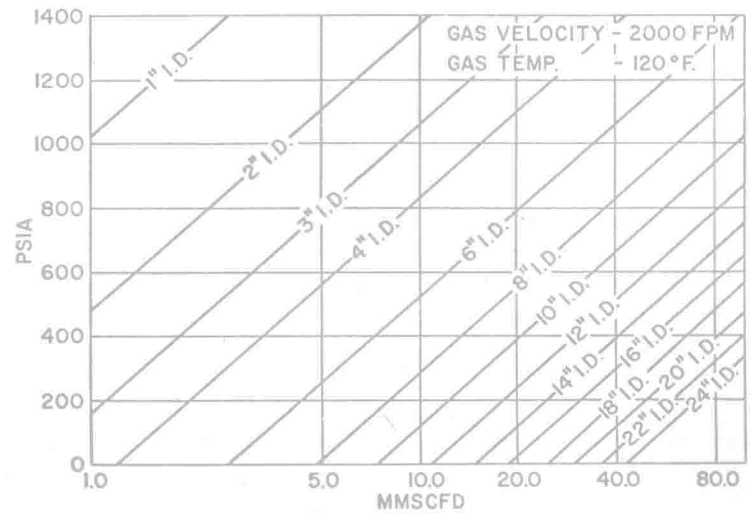

Fig. 3. Line sizes based on 2000 fpm at 120 F.

be brought out as we proceed with the calculation procedure and as we go through several examples.

Gas Piping Design

Surge Drum Sizes. Over a period of years many standard methods have been developed to size surge drums. Most of these are empirical — that is, based on someone's experiences — while others are theoretically derived — based on assumptions and simplified so that the method can be used by designers in day-to-day applications.

Experience in this area indicates that a good starting point for determining the size of a surge drum is as follows:

Suction Drum: The volume of the suction drum should be equal to twelve (12) times the head end swept volume. To determine the diameter, set the length equal to twice the diameter ($L = 2D$). Actual length will be determined by layout considerations.

Discharge Drum: The volume of the discharge drum should be equal to eight (8) times head end swept volume. To determine the diameter, set the length equal to twice the diameter. Actual length will be determined by layout considerations.

For applications where more than one compressor cylinder is required for a given stage of compression, the diameter of the surge drums is determined as in the case of a single cylinder using the swept volume of one compressor cylinder and a length two (2) times the diameter. Once the diameter is determined in this manner, the actual drum length then will be the centerline-to-centerline dimension of the outermost compressor cylinder plus two feet, or as required by layout considerations.

Suction Surge Volume (V_S) = [Twelve (12)] · [Head End Swept Volume]

and,

Suction Surge Drum Length = [Two (2)] [Diameter] and solving for the diameter we have,

Suction Surge Drum Length D_S

$$= \sqrt[3]{[6] \cdot [\text{Bore}]^2[\text{Stroke}]} \quad \text{(Eq. 1)}$$

Now on discharge,

Discharge Surge Volume (V_D) = [Eight (8)] · [Head End Swept Volume]

and,

Discharge Surge Drum Length = [Two (2)] · [Diameter] and solving for the diameter we have,

Discharge Surge Drum Diameter (D_D)

$$= \sqrt[3]{[4] \cdot [\text{Bore}]^2[\text{Stroke}]} \quad \text{(Eq. 2)}$$

Line Sizes. The industry's accepted method for sizing gas lines is based on 2000 fpm gas velocity. This is considered to be a good place to start but each individual situation should be evaluated from the standpoint of pressure drop and horsepower. There is no question that the most meaningful method is to convert all line pressure drops into horsepower losses, but it is also the most difficult, requiring the accuracy of computers.

Now let us look at line sizes based on gas velocity, although it is the least meaningful, but the simplest method and a good starting point. From the basic equation, where capacity is equal to area times velocity, it can be shown that:

Line Size (d)

$$= \sqrt{\frac{(1.8)\ (\overline{\text{M}}\text{SCF/D})\ (T_A)}{(P_A)}} \quad \text{(Eq. 3)}$$

where:

d = internal line diameter based on 2000 fpm velocity
$\overline{\text{M}}$SCF/D = flowing capacity
T_A = absolute temperature
P_A = absolute pressure

Acoustical Filter Sizing

In sizing acoustical low pass wave filters, some basic guidelines must be established. First we must decide where we are to place the so-called cutoff frequency of the filter; in reality, however, if we are to provide effective filtering, we have but two choices.

One choice is to place the cut-off frequency equal to the first engine order (see Fig. 4); the other is to set it equal to the third engine order (see Fig. 5). To set the cut-off equal to the first engine order is the ultimate advance in acoustical filtering, and is thus the desired system. Only when conditions are such that it is uneconomical or physically impractical should we consider the high cut-off frequency system—that is, only when pressure drop is very critical, as in the case of a very low suction pressure or when space is limited (see Fig. 6).

Next, we must establish the maximum allowable pressure drop for the filter, which will dictate the size of the choke tube. The smaller the choke tube, the most economical the filter system. The pressure drop is determined on an individual basis; however, the value usually falls in the range of 1/2 to 1 percent of line pressure per acoustic filter.

Now, with these basic ground rules in mind, let us begin our acoustical filter sizing. The cut-off frequency equation will not be developed here, but it can be shown briefly that for a symmetrical low pass acoustic wave filter (see Fig. 7) the cut-off frequency

Fc (cut-off frequency, cps) equals

$$\frac{\sqrt{2}\cdot\alpha\,(\text{vel. of sound, fps})}{\pi} \times \frac{d\,(\text{diameter choke, in.})}{D\,(\text{diameter volume, in.})\cdot l\ (\text{length, ft})} \quad (4)$$

Now, simplifying further and setting the cut-off frequency equal to the first engine order, we have the following equation:

$$l = 27.0\left(\frac{\alpha}{\text{RPM}}\right)\left(\frac{d}{D}\right) \quad (5)$$

where:

l = length of choke, ft
α = velocity of sound, fps
RPM = revolutions per minute
d = diameter of choke, in.
D = diameter of volumes, in.

for a symmetrical filter system (see Fig. 7). The first engine order is defined as RPM divided by 60.

Now by setting the cut-off frequency equal to the third engine order (3·RPM/60) the equation becomes

$$l = 9.0\left(\frac{\alpha}{\text{RPM}}\right)\left(\frac{d}{D}\right) \quad (6)$$

also it can be shown that for a nonsymmetrical acoustic wave filter (see Fig. 8) the equation becomes

$$Fc = \frac{\alpha\cdot d}{\pi}\sqrt{\frac{1}{l}\left[\frac{1}{D_1^2L_1} + \frac{1}{D_2^2L_2}\right]} \quad (7)$$

where:

Fc = cut-off frequency, CPS
α = velocity of sound, fps
d = diameter of choke, in.
l = length of choke, ft

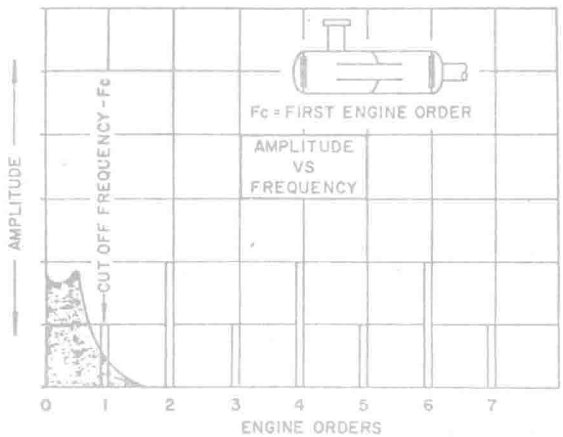

Fig. 4. Acoustical response when cut-off frequency equals the first engine order.

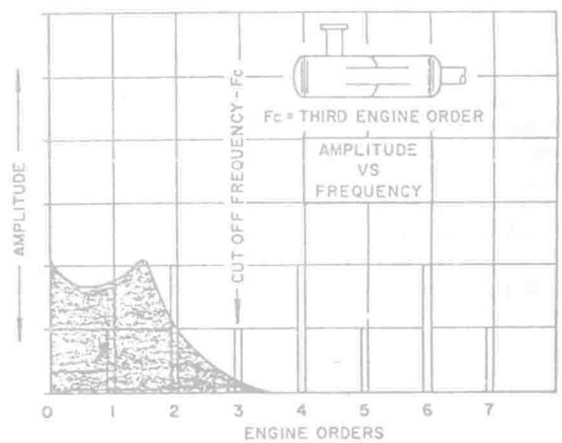

Fig. 5. Acoustical response when cut-off frequency equals the third engine order.

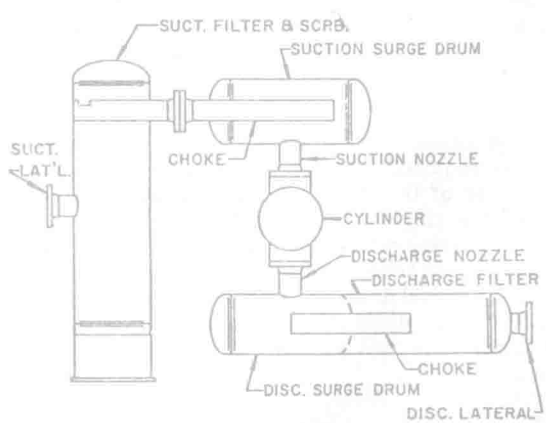

Fig. 6. Nomenclature.

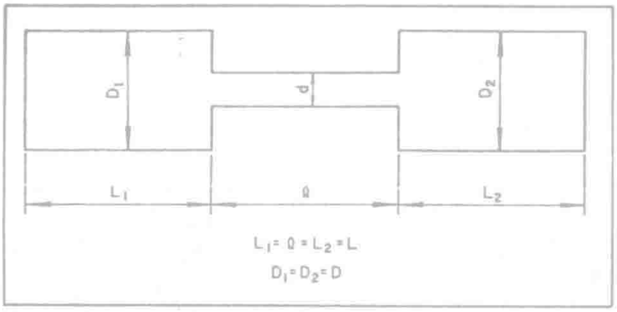

Fig. 7. Symmetrical acoustic filter.

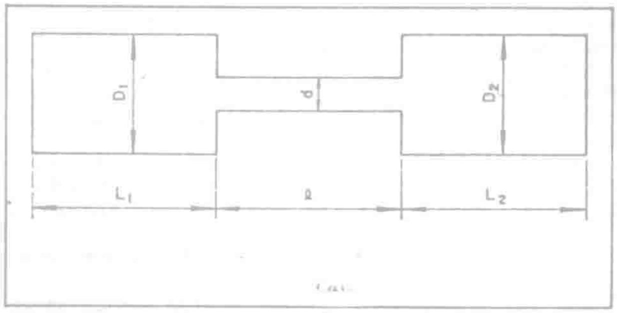

Fig. 8. Nonsymmetrical acoustic filter.

D_1 = diameter of surge drum, in.
L_1 = length of surge drum, ft
D_2 = diameter of fiber bottle, in.
L_2 = length of filter bottle, ft

Now, at this point we have a preliminary surge drum size and a choke tube size based on an allowable pressure drop (including entrance and exit losses). Thus, we are ready to proceed with the acoustic filter sizing.

The first try at sizing will give us a general idea of the sizes and lengths of the filter elements required to satisfy our design objectives. This first try will also reveal the practicality of our approach through such considerations as: whether it is practical to place the cut-off frequency equal to the first engine order; whether this frequency must be set equal to the third engine order; or even whether it is practical to filter at all.

Next, assuming that we have ascertained that filtering is required and feasible, we can refine our filter (size and length) and incorporate the use of the acoustic junction theory to eliminate transmission of pass band into yard piping and to reduce bottle unbalance where possible. All other design considerations should be incorporated at this point, such as the mechanical (layout), thermal, and fabrication (bolt-up) characteristics. To demonstrate the above sizing procedure let us now go through several actual examples.

Conclusions

It is possible to design effective gas piping systems incorporating an acoustic filter analytically for high speed reciprocating compressors in a short period of time, if some basic decisions are made and the basic relationships are known. The following basic guidelines should be used:

1. The acoustic filter will be incorporated with the cut-off frequency set equal to the first engine order; and only when it is not economically feasible or is physically impractical should consideration be given to setting the cut-off frequency equal to the third engine order.
2. Maximum pressure drop per filter will be determined on an individual basis; however, the value will usually fall in a range from 1/2 to 1 percent of line pressure.
3. Vibration levels of the major gas piping will not exceed 10 mils peak-to-peak for frequencies up to, and including the second engine order and 5 mils for high frequencies.
4. Capacity guarantee plus or minus 3 percent.
5. Incorporate the use of the acoustic junction theory to reduce acoustic pass band transmission and bottle unbalance.
6. Consider the mechanical characteristics of the filter system to reduce the coincidences of acoustical and mechanical resonances.
7. Consider stresses due to thermal expansion.
8. Consider fabrication (bolt-up) stresses.

For any design to be successful, all design parameters must be considered before an optimum piping design is realized. However, since it is not the purpose of this paper to consider all the design aspects, except one—the acoustical characteristics—no treatment will be given these other considerations.

The author has no argument from a design point of view against the use of an SGA analog facility for high speed equipment, if the time is available and the additional cost is not prohibited. If these are not major considerations, then certainly the SGA analog should be used. It is important, however, to remember that the analog is only a tool which must be used properly and with the overall design in mind before full benefit is realized. A much more important question to ask is, "Who is analogging the unit," rather than "Is it being analogged?" If an analog study is to be performed, it should be considered to be a part of the design procedure, not a separate study. The above procedure provides an excellent starting point for an analog study if one is to be conducted and can be used for that purpose.

Finally, at this point another design consideration should be discussed, and that is the use of orifice plates in the compressor flanges. It has been found through experience that with high speed equipment using acoustic filters, the acoustic resonance of the compressor's internal passages and nozzles often coincides with the acoustic response of the filter elements. The orifice plate often eliminates this coincidence, thus improving the performance. An orifice plate with a Beta ratio of 0.5 is recommended as a starting point. I consider the use of the orifice plate to be part of a starting procedure, such as the use of suction screen. Once the unit is completed, all supports are secured and adjusted properly, and the engine is tuned; then the orifice plates can be removed. If any adverse effects are noticed after the removal of the orifice plates, they should be re-installed while the problem is evaluated and a permanent solution is determined.

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Example No. 1

Required information:

engine speed	— 1000 rpm
comp. cyl. bore	— 5.125 in.
stroke	— 5.0 in.
suction pressure	— 700.0 psig
suction temperature	— 100 deg F
discharge pressure	— 1400.0 psig
discharge temperature	— 162.0 deg F
specific gravity of gas	— 0.67
flow	— 11.9 $\overline{M}scf/d$
No. of comp. cyls.	— 2.0

Suction. In considering an acoustical filter for the suction of a compressor unit, it is a good idea to utilize the suction scrubber wherever possible. The vessel can serve as a gas scrubber as well as a filter bottle, thus eliminating the need for an additional bottle.

Step No. 1: Determine the size vessel required to scrub the gas using a standard scrubber sizing procedure. In our case, the vessel size was calculated to be 30-in. ID by 8 ft seam-to-seam.

Step No. 2: Determine the smallest diameter choke tube using an appropriate pressure drop equation and the allowable pressure drop across the filter. In our example, a size of 2.624-in. ID was determined to be the minimum diameter choke to which we can go from the standpoint of pressure drop.

Step No. 3: Calculate the size of the suction surge drum using equation (1) or Fig. 2.

$$D_s\,(\text{suction drum})\ \sqrt[3]{(6)\ (5.125)^2(5)} = 9.236 \text{ in. ID} \qquad (8)$$

At this point select a nominal pipe size for the design pressure which will result in an internal diameter near the calculated ID—that is, using appropriate piping codes, the wall thickness must be determined.

Step No.4: Sketch the piping system to determine an optimum layout using the basic sizes of the filter elements determined in the above steps (see Fig. 9).

Step No. 5: Determine the velocity of sound in the gas stream at the operating conditions using appropriate thermodynamic data for the gas under consideration. In our case, the value was determined to be 1332.2 fps.

Step No. 6: Determine the length of choke tube by using equation (7) for a nonsymmetrical filter and setting the cut-off frequency equal to the first engine order. Therefore,

$$1000\ 60 = \frac{(1332.2)\,(2.624)}{\pi}\sqrt{\frac{1}{l}\left[\frac{1}{(30)^2(8)} + \frac{1}{(10.02)^2(6)}\right]} \qquad (9)$$

$$l = 8.01 \text{ ft}$$

This length represents a cylinder with square ends; it must be converted, therefore, to a seam-to-seam dimension, considering the volume of the heads.

Step No. 7: Update and refine the layout, incorporating the above sizes and lengths, and determine if there are any layout problems. In our case, an 8-ft long choke tube is a good length from a layout standpoint; therefore, we have a filter design for the suction satisfying all design considerations. If the length is determined to be too long or too short to fit the layout, then some adjustments must be made to the filter elements.

Discharge.

Step No. 1.: Determine the size of the discharge surge drum using equation (2) or Fig. 2.

$$D_d\,(\text{discharge drum}) = \sqrt[3]{(4)\ (5.125)^2(5)} = 8.068 \text{ in. ID} \qquad (10)$$

Select the nearest nominal pipe size which will result in an internal diameter near the calculated value, considering the design pressure. In this example we must go to a 10-in. nominal pipe with an internal diameter of 9.134 in., which is to say a standard 10-in. schedule 100 pipe.

Step No. 2: Determine the minimum choke tube size using the maximum allowable pressure drop across the filter. In this example it can be shown, using an appropriate pressure drop equation, that the minimum choke size is a 2-in. extra strong pipe with an internal diameter of 1.939 in.

Step No. 3: Sketch the piping system using the above sizes to determine a desirable layout. In our case, we chose to use a combination surge drum-filter bottle; therefore, the filter bottle is the same size as the surge bottle (see Fig. 10).

Step No. 4: Determine the velocity of sound in the gas stream at the operating conditions using appropriate thermodynamic data for the gas under consideration. In our case, the value is calculated to be 1429.6 fps.

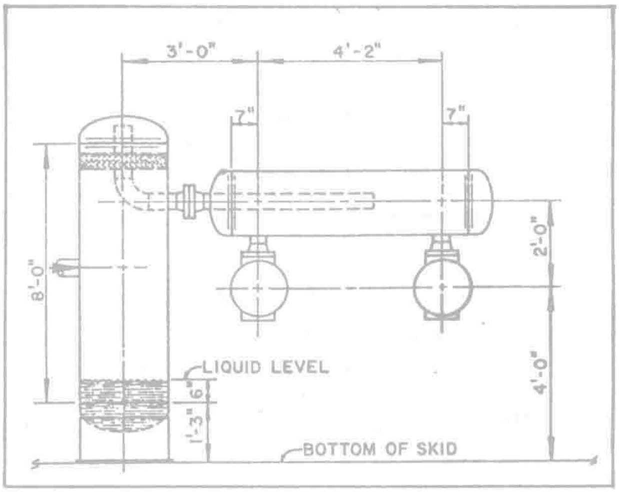


Fig. 9.

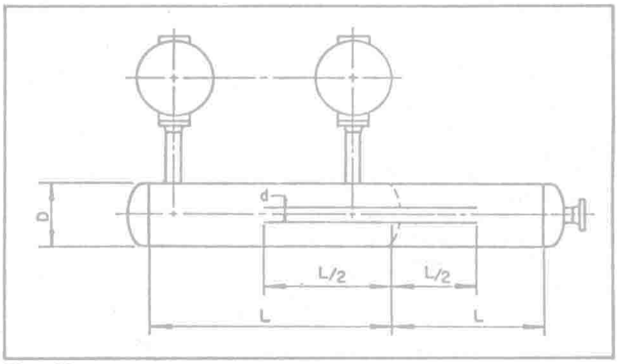


Fig. 10.

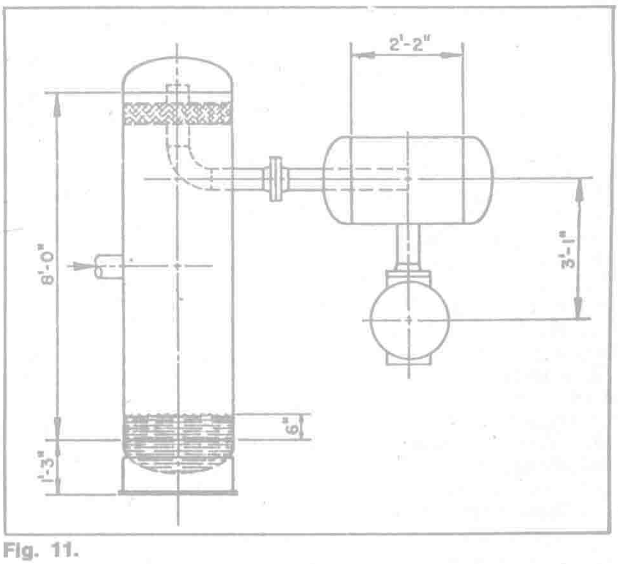


Fig. 11.

Step No. 5: Determine the length of the filter elements. From Fig. 10 it can be seen that the surge drum portion of the bottle is the minimum length required to span the cylinders, and that the filter portion of the bottle is set equal in length to the surge drum portion in order to have a symmetrical filter system. Therefore,

$$l = 27.0\left(\frac{1429.6}{1000}\right)\left(\frac{1.939}{9.314}\right) = 8.03 \text{ ft} \qquad (11)$$

Step No. 6: Investigate the effects of this length to our layout and determine any adverse characteristics. In this

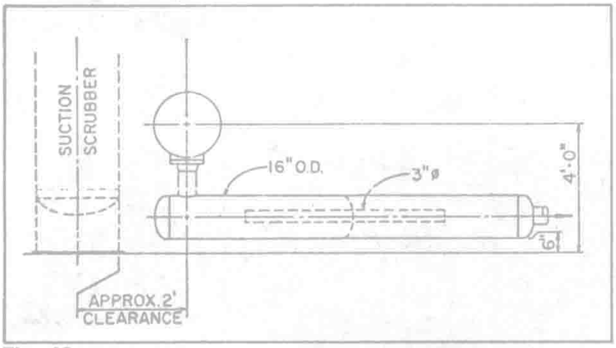


Fig. 12.

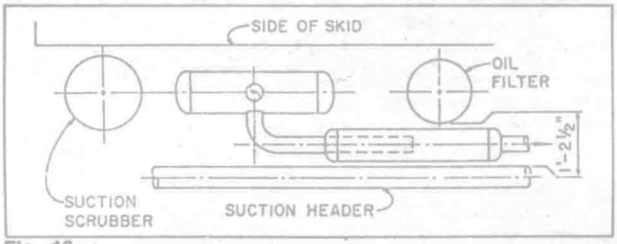


Fig. 13.

case, it is desirable to reduce the overall length of the combination bottle to 12 ft as determined from our layout sketch. Substituting this length into equation (5) and solving for a new diameter, we get a 12.5-in. ID. This requires a nominal pipe size of 14 in., thus requiring us to raise the centerline of the engine to accommodate this larger bottle. Evaluating the overall effects of this larger diameter/shorter length bottle and the cost of the system, it is decided that in this case the use of the longer-smaller diameter filter is the better solution.

Example No. 2

Required information:

engine speed	— 900 rpm
compt. cyl. bore	— 13.25 in.
stroke	— 5.0 in.
suction pressure	— 58.8 psia
suction temperature	— 130.0 deg F
discharge pressure	— 127.8 psia
discharge temperature	— 197.0 deg F
specific gravity of gas	— 0.72
flow	— 3.15 $\bar{M}$scf/d
No. of comp. cyls.	— 1.0

Suction.

Step No. 1: Determine if the suction scrubber is to be used as part of the acoustic filter and, if so, determine the size vessel required to satisfy the gas scrubbing sizing procedure. In our example, this size calculated to be a 23-in. internal diameter by 8-ft seam-to-seam gas scrubber.

Step No. 2: Determine the minimum choke tube size using the maximum allowable pressure drop across the filter. In this example the minimum choke tube size calculated to be 4-in. schedule 40 pipe.

Step No. 3: Size the suction surge drum using equation (1) or Fig. 2

$$D_s\,(\text{suction drum}) = \sqrt[3]{(6)\,(13.25)^2(5)} = 17.39 \text{ in. ID} \quad (12)$$

Again as in Example No. 1, select a nominal pipe size good for the design pressure which will result in an internal diameter near the calculated ID. In our case, we can use an 18-in. nominal pipe with a wall thickness of 0.312-in. (17.375-in. ID).

Step No. 4: Sketch the piping system to determine an optimum layout, using the basic sizes of the filter elements as determined in the above steps (see Fig. 11).

Step No. 5: Determine the velocity of sound in the gas stream at the operating conditions, using appropriate thermodynamic data for the gas under consideration. In our case, the value is determined to be 1342.2 fps.

Step No. 6: Next, determine the length of choke tube by using equation (7) for a nonsymmetrical filter and setting the cut-off frequency equal to the first engine order.

We have

$$900/60 = \frac{(1342.4)\,(4.026)}{\pi}\sqrt{\frac{1}{l}\left[\frac{1}{(23^2)\,(8)} + \frac{1}{(17.375)^2(3)}\right]} \quad (13)$$

$$l = 25.2 \text{ ft}$$

The length of choke required to obtain a cut-off frequency equal to the first engine order in this case is obviously not realistic, so we must investigate other approaches. A thorough analysis indicates that it is impractical to set the cut-off frequency equal to the first order; therefore, our next choice is to set it equal to the third engine order.

Therefore, using equation (7) again, we have

$$(3)\,(900)\,/\,(60) = \frac{(1342.4)\,(4.026)}{\pi}\sqrt{\frac{1}{l}\left[\frac{1}{(23)^2(8)} + \frac{1}{(17.375)^2(3)}\right]} \quad (14)$$

and solving for l we have a length of 8.4 ft. This length fits our layout and can be installed quite reasonably.

Step No. 7: Refine and update the sketch with the new length choke, and double check the pressure drop, making sure all losses are considered, including entrance and exit.

Discharge.

Step No. 1: Determine the size of dischage surge drum using equation (2) or Fig. 2

$$D_d\,(\text{discharge drum} = \sqrt[3]{(4)\,(13.25)^2(5)} = 15.2 \text{ in.} \quad (15)$$

Select the nearest nominal pipe size which will result in an internal diameter near the calculated value, considering design pressure. In this example, we must use 16 in. nominal pipe with an internal diameter of 15.375 in., which is 16 in. schedule 20 pipe.

Step No. 2: Determine the minimum choke tube size using maximum allowable pressure drop across the filter. In this example it can be shown, using an appropriate pressure drop equation, that the minimum choke size is a 3 in. schedule 80 pipe (2.9 in. ID).

Step No. 3: Sketch the piping system, using the above sizes to determine a desirable layout. In our case, we chose to use a combination surge drum-filter bottle; therefore, the filter bottle is the same size as the surge drum.

Step No. 4: Determine the velocity of sound in the gas stream at the operating conditions, using appropriate thermodynamic data for the gas under consideration. In our case, the value was calculated to be 1409.4 fps.

Step. No. 5: Determine the length of the filter elements. From Fig. 12 it can be seen that the surge drum portion of the bottle is set equal in length to the surge drum portion in order to have a symmetrical filter system. Therefore,

$$l = (27.0)\left(\frac{1409.4}{900}\right)\left(\frac{2.9}{15.375}\right) = 7.975 \text{ ft} \quad (16)$$

It can be seen from this that the overall length of the bottle will be approximately 16 ft.

Step No. 6: Investigating the effects of this length on our layout, it is determined that this length will not clear our oil filter. Therefore, sketch a plan view of the layout, including any equipment whose location is fixed, in order to determine where a filter may be placed.

Step No. 7: It can be seen from the sketch (Fig. 13) that a 12¾ in. OD bottle is the largest size that will fit our layout. Applying the proper vessel code it is determined that a 12.0 in. ID bottle will be adequate for this design pressure. Therefore, applying this diameter to equation (7), letting the length of choke equal the length of filter ($l = L_T$), and solving for l, we have

$$15 = \frac{1409.4\ 2.9}{\pi}\sqrt{\frac{1}{l}\left[\frac{1}{(15.375)^2(2.5)} + \frac{1}{(12)^2 l}\right]} \quad (17)$$

$$l = 16 \text{ ft}$$

How To Automate Existing Compressor Stations

by **R. F. Cook,** *Superintendent, Technical Operations,*

and **J. L. Williams,** *Chief Instrumentation Engineer, El Paso Natural Gas Company, El Paso, Texas*

Fig. 1. Typical gas turbine station has three machines installed for series operation. Turbines are two-shaft, regenerative cycle driving centrifugal compressors and were designed for future automation.

In December 1971, El Paso Natural Gas Co. completed the automation of 12 turbine stations on its existing Southern System, which extends from the Pecos River in West Texas across New Mexico and Arizona to the California border.

In 1951 the reciprocating compressor stations on the Southern System were located approximately 100 miles apart, as shown in Fig. 2. From 1951 through 1953, 28 gas turbines were purchased and installed in 10 new compressor stations for mainline gas transmission service. They were the first turbines used for this type of service. These machines were installed for series operation, three at each station, except for two locations which only had two. The turbine stations were located approximately 35 miles apart between the existing reciprocating stations, as shown in Fig. 3.

These turbines are G. E. frame three, two-shaft, regenerative cycle, heavy-duty type with variable nozzle and variable speed controls. The controls were originally designed for possible future automation. All of these turbine stations when completed were manned with one plant superintendent, one repairman, and five operators.

In 1957 and 1958 two more stations, each with two G. E. turbines, were built with the same design and manning philosophy as above.

In 1959, two plants were built utilizing 5000 HP electric motors to drive centrifugal compressors. These were our first remote operated compressor units. One of these was located at an existing reciprocating station and the other one at a new location. Both of these stations were remote controlled from existing reciprocating stations. With the experience gained from the operation of these plants, and from other companies' experience with remote operation of gas turbines, it was felt that all future mainline gas turbine stations should be remote operated.

The opportunity to remote control our first gas turbine station did not arise until 1964 with the construction of our Dilkon Station. This station, consisting of one G.E. single-shaft, frame 5, simple-cycle, 12,000-hp sea-level-rated turbine, was placed in service and remote controlled from an existing reciprocating station approximately 40 miles downstream. Later in 1964 a 5500-hp reciprocating plant was placed in operation which was also controlled from this location. These plants were manned with two operating personnel as compared with seven operating personnel at the original turbine stations.

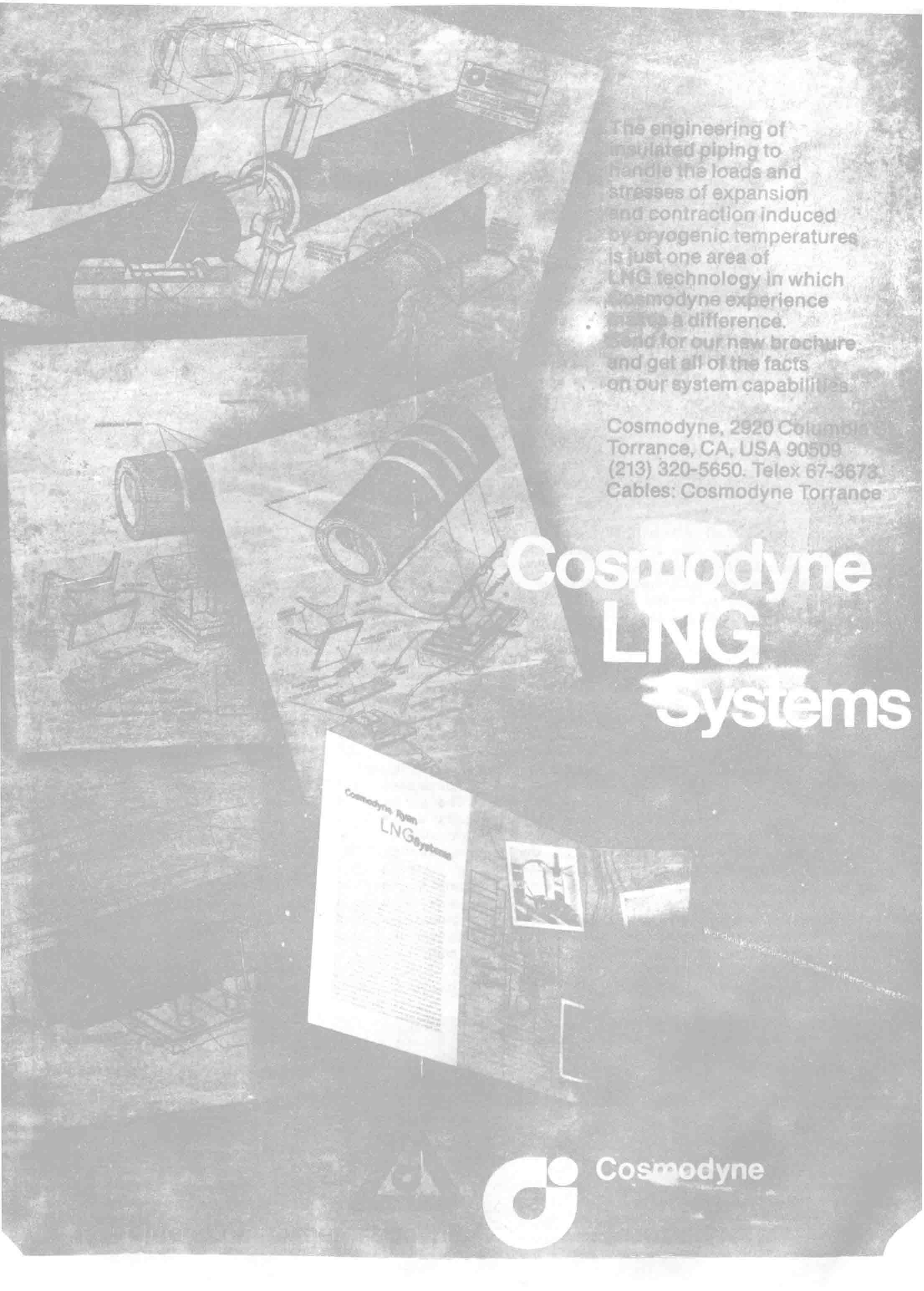
Several more remote controlled turbine plants were placed in operation over the next few years with the same philosophy of remote control with two operating personnel on site.

In 1967 we began giving serious consideration to automating the ten original turbine stations due to our experience and confidence in operating the new stations and the attraction of the apparent reduced operating costs.

A committee was established for this purpose, consisting of Engineering Design and Operating Division personnel. Numerous studies were made to determine if it was economical to automate these stations. Due to the foresight in 1952, in designing the compressors for automatic operation, this project appeared feasible. A second round

Fig. 2. Location of reciprocating compressor stations in 1951 — about 100 miles apart.

Fig. 3. Ten new gas turbine compressor stations were placed about 35 miles apart.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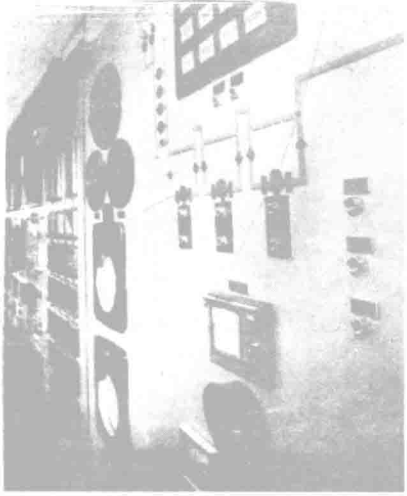

Fig. 4. Front view of new control panel.

Fig. 5. Inside of new control panel.

Fig. 6. Electricians and instrument men revamping existing panels.

of studies was made to determine the best method to control the stations.

Three basic plans were evaluated:

1. manned satellite control from existing reciprocating station;

2. control of all units from one central location with a mimic console; and

3. central control of all units with a computer.

The manned satellite concept was felt to be the most desirable operation. This was due to having a responsible superintendent located within approximately 35 miles of the station, operating personnel who had turbine experience and a feel for their operation, and our past experience with satellite operation. Once this decision had been made, then the main office design personnel became responsible for system design and the Division Technical Services Group for the installation and construction.

The first step was to design, build, and install a test panel for automatic control, in order to obtain the confidence of our operating personnel in this project. In the meantime, all of the drawings were updated so that the design could proceed. Our operating personnel were assured that there would be no loss of jobs due to this project.

To absorb the displaced operating personnel into other phases and locations of our operation, the project was scheduled to be completed over a two-to-three-year period using existing technicians for the construction.

As the design proceeded, many meetings were held with operating personnel at all locations to seek out and avoid any oversights. Numerous items that could have created control problems were brought to our attention.

At a manned station many controls that do not operate exactly as designed can be overlooked because of the ability of the man to perform certain tasks; but all of the systems at a remote operated station must work correctly at all times.

Several systems were added, since the stations would be unmanned 16 hours each day. Fire detection and hazardous atmosphere detection were installed as continuous monitoring systems. The air washers were automated using ambient air temperature and humidity controls. When the relative humidity increases to 50% or when the ambient air temperature decreases to 50 F, the air washer operation is discontinued. The humidity control is to prevent carryover of water and fouling of the compressor. The temperature control is to prevent the formation of ice on the inlet air screen and possible subsequent damage to the turbine. All of the turbines are installed with shaft-driven generators, and most of the stations are connected to purchased power. Controls were added to automatically switch from one source of power to another to obtain the most reliable operation.

Other control systems were revised as needed. The cooling systems were further automated to control the oil cooling and nozzle cooling water temperature. The original reliable A. C. power supplied from the motor generator set was supplemented with a new static inverter to supply power for all of the instruments, supervisory and telemetering systems. The old variable nozzle control system was completely redesigned using updated equipment. This was necessary to maintain maximum efficiency.

Because spare parts were no longer available for the old system, and because the equipment was not compatible with the telemetering system, a new discharge pressure control with related pressure recorders was installed. A new vibration system was installed on each unit to *replace* existing systems. (Two velocity pickups on the turbine and a shaft nonrider on the compressor.) This was done to telemeter this signal to the remote control room at the complex station. Numerous pressure, level, temperature, and limit switches were replaced due to obsolescence and in order to obtain required dependability. Also, transducers were added to telemeter exhaust temperature, turbine speeds and gas discharge temperature.

During the automation program, we investigated the need to install surge control; but due to the pipeline conditions under which we were operating, it was determined unessential at that time. Pipeline con-

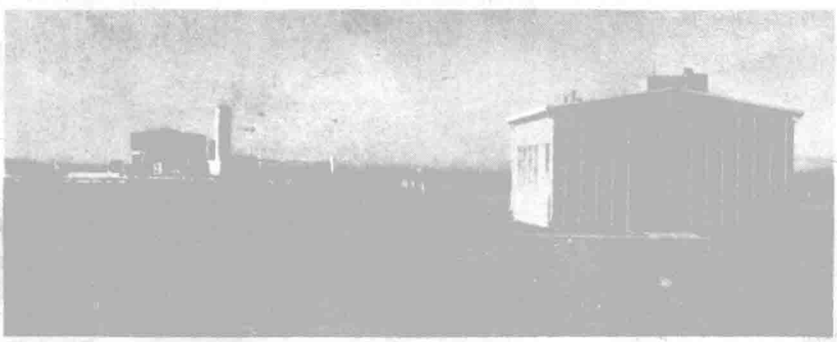

Fig. 7. Typical remote control building at a reciprocating compressor station.