PULSATION PHENOMENA
In Gas Compression Systems

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WHEN gases are compressed by any of the usual methods now employed commercially, it is frequently found that several undesirable effects of pressure variation appear in the system associated with the compressor or in the compressor itself. All of these undesirable effects are connected in some manner with pulsation phenomena which appear as a result of the reciprocating action of the compressor. In certain systems where rotary or other types of compressors are employed, as contrasted with the reciprocating type, similar pulsation phenomena are evident. However, the present discussion will be confined to the phenomena which appear in systems utilizing reciprocating compressors, and in particular those which are employed in the compression of natural gas for transmission in pipelines, or for other purposes such as recycling or repressuring operations.

One of the most important difficulties which arise from pulsation phenomena is the effect of vibration which appears in the piping system or other equipment downstream of compressor plants. In such cases it frequently happens that pipelines, heat exchangers, scrubbers, processing vessels and even buildings are caused to vibrate to an extent which may have very serious consequences. These vibrations have been known to tear away anchor bolts and, in some instances, pipelines have been broken at welded joints. A frequent occurrence is that of breaking screwed fittings on small lines such as instrument air systems. Under severe conditions of pulsating flow the vibration is frequently accompanied by considerable noise, which appears to radiate from the piping itself and will be heard to travel long distances through the system. This is sometimes described as being an effect similar to that which would be produced by "rocks rolling through the pipe." Of course, there is also the accompanying rattle of piping and other metallic surfaces striking each other. In general, these effects occur with the most spectacular results in systems operating at relatively high pressure. For example, certain compressor installations whose purpose is to compress natural gas for recycling operations in the production of petroleum may operate at pressures as high as 4000 lb/in.² In cases where large quantities of gas are being compressed to high pressures the destructive effects of vibration may be considerably more pronounced than would be the case in systems operating

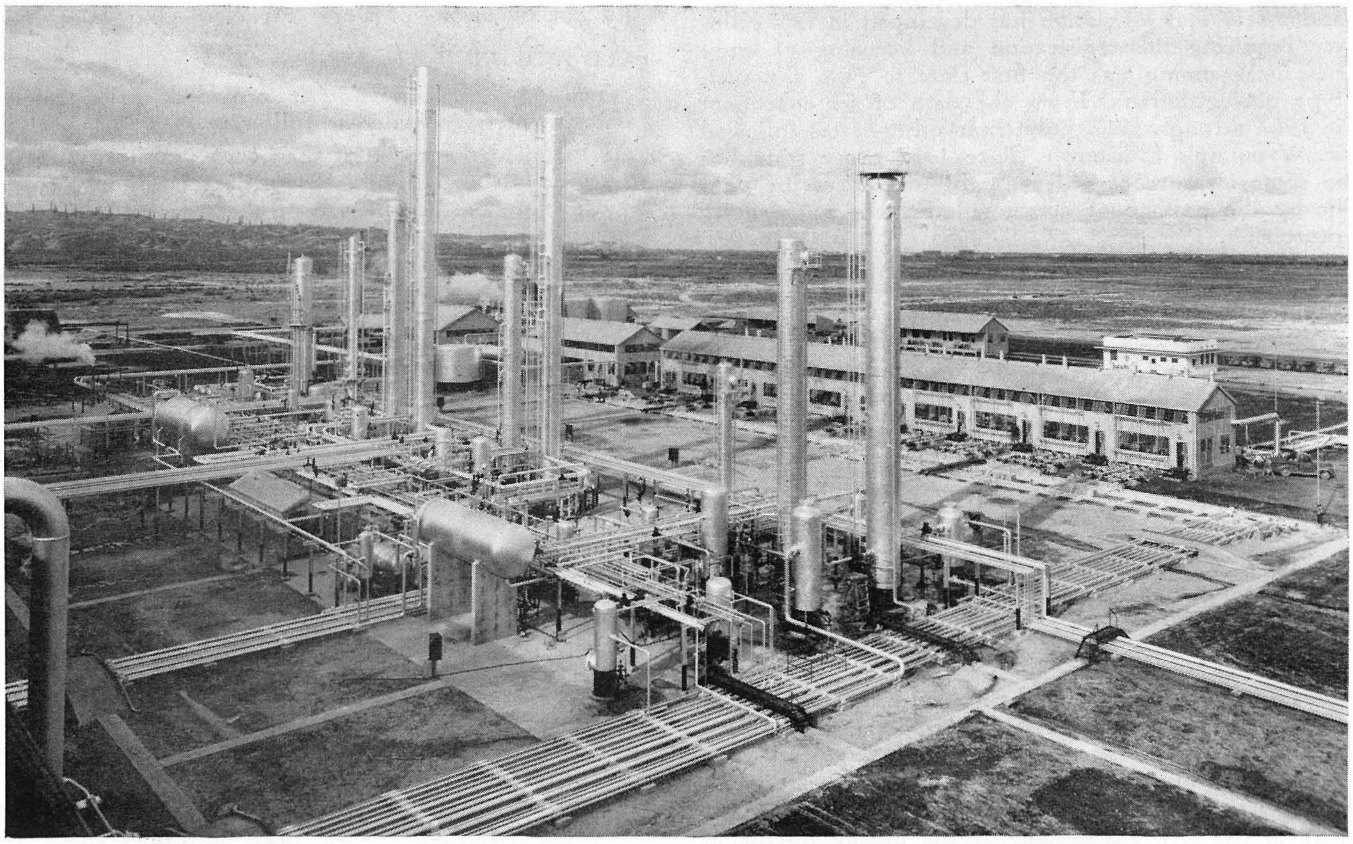


Figure 1. Modern recycling plant showing compressor building housing units which compress injection gas to 3700 lb/in.²

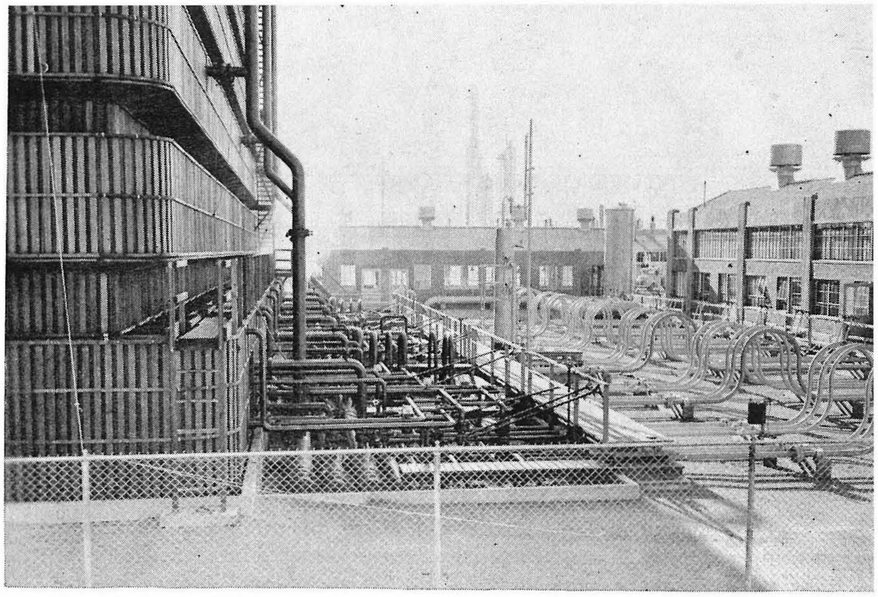


Figure 2. Typical compressor piping system employing vertical expansion bends in both suction and discharge laterals.

at low pressures. However, many systems have been observed where the effect of vibration is remarkably pronounced, even though the discharge pressure is quite low. It has also been common to find that the suction system of a compressor plant operating at low pressures will manifest extremely dangerous conditions of vibration. These effects have been studied to a considerable extent in the past, and many efforts have been made to design piping systems which will withstand the stresses imposed upon them by forces originating in the pulsating gas flow.

Arnold* has presented a study of these effects which gives a basis for determining whether they may be of a destructive nature. Even though vibration phenomena, which result from pulsative flow in certain instances, may not be of a destructive nature, many operators consider them to be a psychological hazard which could well be eliminated for the purpose of producing better working conditions for employees.

A second undesirable situation which may be engendered by pulsation phenomena is the necessity for oversizing pipelines to take care of increased pressure drops which may result from pulsative flow. It is also necessary to select heavier pipe when such flow exists, in order to insure safety in its operation. Both of these effects require the use of a large amount of steel which would be unnecessary if the gas flow were not pulsating.

A third result of extreme conditions of pulsative flow is concerned with the efficiency of the compressor itself. When pulsations are present in the piping downstream of a compressor cylinder, it is possible to have a standing wave of such nature that the cylinder is discharging under conditions whereby the internal pressure of the cylinder rises considerably above the average system pressure for a considerable portion of the discharge part of the cycle. This causes the consumption of power which is not effective in transferring compresesd gas to the lines, and hence the efficiency of the cylinder is adversely affected.

Another difficulty which may be traced to pulsative flow in gas lines is the well-known effect which becomes evident in orifice meters when the flow through the orifice plate is not uniform. In such cases the recorder pen will be caused to indicate a broad band of vibrating lines, rather than a single trace. When this condition exists accurate metering is not possible without considerable attention being given to the manner of analyzing the chart. There have been many discussions of this problem and much disagreement exists, even at the present time, as to the proper correction to be made to meters operating under these conditions.

Many measurements have been made in compressor plants operating in the field. These have resulted in a collection of data which reveal the nature of pulsations to be expected from various types of compressors, as well as the effects which are produced by different types of piping and other equipment which may be associated with the compressor. Data have also been secured from relatively small commercial compressors operating under laboratory conditions which allow a further study of pulsation phenomena. It is now possible practically to eliminate pulsations in piping systems by applying pulsation dampeners which have been developed for this specific purpose. Since this equipment usually reduces the peak-to-peak pulse pressure to a point where less than 10 per cent of the original pulse remains, it is possible to show the effects produced upon a piping system when the pulsation phenomena are substantially removed.

Although it is not the purpose of this paper to discuss the details of the design of pulsation dampeners which may be used for the service indicated above, it may be of some interest to outline briefly the manner in which these devices operate. A dampener of this type is a device constructed of piping usually somewhat larger than the line in which the dampener is inserted. Generally speaking, the dampener proper is included between two flanges in the form of a spool which is located in the lateral, which can be either the discharge or suction piping to the compressor cylinder. This unit includes a filtering device consisting of suitably proportioned inertance and capacitance passages which serve to smooth the pulsations by a process analogous to phase shifting. Such an apparatus requires no moving parts and hence may have a construction based entirely upon welded units. Because the operation is contingent upon a phase shifting process, it is possible to minimize any pressure loss which results from the insertion of this equipment in a pipe line. At the same time, a high degree of reduction of pulse amplitude can be achieved; for example, amplitude reductions of greater than 90 per cent are quite possible, with pressure drops of less than ½ of 1 per cent of the static pressure of the gas being transported in the line.

*Arnold, M. L. Vibration in Compressor Plant Piping. California Oil World, First issue, December, 1944.

NATURE OF PULSATIONS

When attempts are made to analyze the conditions which exist in pulsating systems of the type found in installations under discussion here, it is seen immediately that the problem is extremely complex. Compressor systems, which usually consist of several units, ordinarily discharge gas into pipelines in such a manner that pulsations of very complex wave forms are developed. The complexity of these waves indicates that a great many frequencies are present and hence the pulsating gas has a pressure variation which is the resultant of all these frequencies. These waves are primarily the result of the pulse produced by the stroke of the compressor cylinder, which is in turn modified by the action of the intake or discharge valve. It is found that the "fundamental" pulse repetition rate is determined by the speed of the compressor. However, in most cases, the wave form is shaped by the action of the valve, and is partially modified by the characteristics of the piping system downstream of the valve. Some compressors operate at such speeds that the fundamental discharge frequency may be less than 3 cycles/sec. The more modern high-speed double-acting cylinders found in present-day compressors may exhibit fundamental frequencies in the vicinity of 10 cycles/sec.

Because of the wide variety of frequencies higher than this fundamental which may be present in the complex wave, it is always likely that some portion of a plant downstream from the compressors will have such dimensions that it may be in resonance with at least one of the frequencies present in the pressure wave. In such a case this particular portion of the plant will be excited readily by the pulsating flow and will be maintained in a state of vibration which may involve considerable amounts of energy. It is to be noted that any particular pipeline or piece of equipment may have two possible media in which vibratory motion may be set up. The gas within the system may carry a pulsating wave whose wave length is determined by the speed of sound in the gas. Similarly the metal shell of the line or vessel may carry a vibratory wave of the same frequency but having a wave length considerably shorter because of the greater speed of sound in metal. The ratio of the speed of sound in steel to that of hydrocarbon gases at relatively high pressures is approximately 10 to 1. Hence, it may be seen that for the average type of construction used in compressor plants, there is a relatively great possibility that structural features will be such that a large number of portions of the plant will be excited by different frequencies in the pulsating wave. Therefore, the plant as a whole may exhibit a large number of vibrations of different frequencies throughout its various portions. When such conditions exist, a plant may be considered as a very complex mechanical oscillator which is being excited by the pulsating energy being emitted by the compressors.

A very important phenomenon associated with pulsating flow arises from the appearance of beat frequencies in the system. These beats result from the combining of waves from two or more compressors with the production of relatively long wave length beats. These are usually evident in headers and those portions of the plant beyond the point where the discharge streams of several compressors are combined. These beats are probably primarily responsible for the very pronounced vibrations of large amplitude which occur in lines of considerable length. They may also be the source of excitation for vessels of large dimensions.

FIELD DATA

Fig. 4a shows the pressure-time charts secured at a point immediately downstream of the cylinder in the discharge piping of a common type of compressor, the Cooper-Bessemer Type 19. Fig 4b is the same type of chart for transfer header which receives gas from six compressors. All of these records were made while the plant was operating under normal conditions with a suction pressure of approximately 275 lb/in.^{2} gauge and a transfer header pressure of 480 lb/in.^{2} gauge at a compressor speed of 172 rev/min. It is to be noted that both the suction and discharge systems immediately adjacent to the cylinder show pressure-time curves which are very smooth, and almost approach a sine wave in form. This is found to be more or less typical of low-speed compressors operating at pressures less than 700 lb. However, in both the suction and transfer headers the wave form becomes much more complex and shows evidence of higher frequencies having been superimposed

Figure 3. Pulsation dampener installation on 1500 and 3700 lb/in.^{2} discharge laterals feeding into header system.

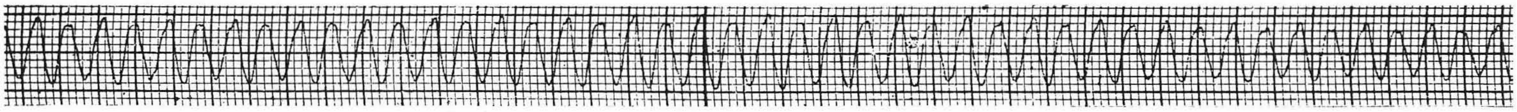


Figure 4a. Pressure-time curve for discharge piping immediately downstream of cylinder.

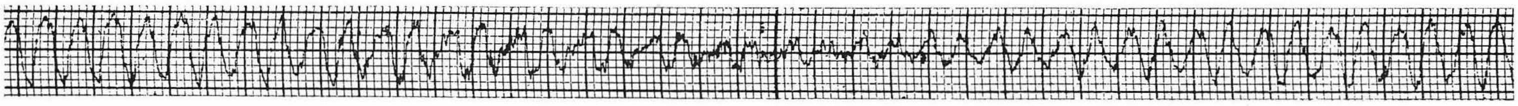


Figure 4b. Curve for discharge header connecting six compressors.

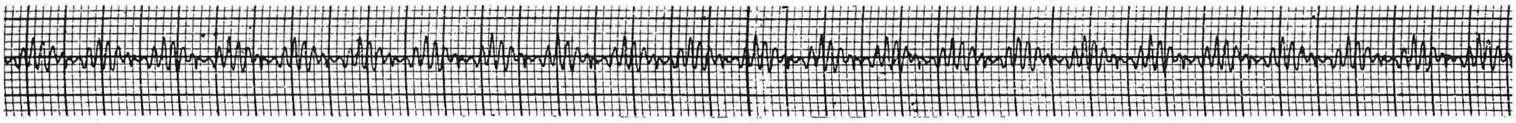


Figure 5a. Pressure-time curve for same point as Fig. 4a after installation of pulsation dampeners.

Figure 5b. Curve secured from discharge header at same point as Fig. 4b after installation of dampeners.

upon the original wave from a single compressor. Fig. 4b shows marked evidence of the presence of a beat frequency which occurs with a period of about 7 seconds. The charts shown here should be used for purposes of wave form comparisons only. Since they are not taken with the same amplification factor in the recording apparatus, they are not comparable with respect to amplitude. This compressor plant did not show evidence of serious vibration phenomena. However, because of the marked pressure changes taking place in the piping, it was desirable to eliminate the pulsations as much as possible for other reasons, such as saving horsepower in the compressors themselves. Therefore, pulsation dampeners were designed and installed in each compressor discharge lateral.

Fig 5a shows the pressure-time curve secured at the same point as the curve of Fig. 4a. Fig. 5b corresponds to Fig. 4a in that the curve was taken in the transfer header after the installation of dampeners on all six compressors in the plant. A comparison of Figs. 4b and 5b reveals that the peak-to-peak pressure in the header was reduced from 17 lb/in.^{2} to 1.0 lb/in.^{2} when the pulsations were removed from the system. It was not possible to make measurements in this plant at a point immediately downstream of the dampeners in the discharge laterals; hence, no comparison may be made of the reduction of pressure pulsation at this point. However, an investigation of Figs. 4a and 5a indicates a peak-to-peak pressure reduction from 97 to 28 lb/in.^{2} at a point between the cylinder and the dampener. From other information which is available and from indications found in this plant it is to be expected that the peak-to-peak pulse pressure found downstream of the dampener would have a value of less than 10 lb/in.^{2}

It is of interest to note that pulsations in the discharge piping immediately downstream of the compressor were reduced, even though this point is above the entrance to the dampener. As has been mentioned above, pressure pulsations at this point may be reflected in the pressure developed within the cylinder itself, and hence may contribute to inefficiency in compression. On the cylinder involved in Figs. 4 and 5, indicator card data showed a reduction of horsepower loss from 10.3 to 2.5 per cent at the head end and from 12.4 to 5.8 per cent at the crank end.

Although vibration problems were not of great significance in this plant, it was found that the acceleration on the suction pipeline immediately adjacent to the cylinder was reduced from 115 in./sec^2 to 12 in./sec^2 when the pulsations were reduced as indicated above.

Another type of compressor which may be considered typical of those found on gas transmission lines is the Cooper-Bessemer Type GMV. The following data were secured from a 1000 hp unit of this type operating with three double-acting single stage cylinders. The speed of the unit was 300 rev/min with a suction header pressure of 322 lb/in.^{2} gauge and a discharge header pressure of 773 lb/in.^{2} gauge. Fig. 6a is a pressure-time curve taken in the discharge piping immediately downstream of a single compressor. In this case three cylinders discharge into each lateral. The maximum peak-to-peak pressure was measured at 77 lb/in.^{2} Fig. 6b is a similar chart recorded in the discharge header which receives gas from 15 identical units. At this point, the peak-to-peak pulsation pressure was 17 lb/in.^{2} The long period wave shown on this chart is not a characteristic of the pulsating flow but is caused by voltage variation in the measuring instrument. This plant was subject to severe vibration which induced stresses in the piping system sufficient to cause damage to anchors and clamps used on the piping system. Vibration measurements showed values of acceleration between 250 and 400 in./sec^2 at critical points where damage had occurred.

It has previously been mentioned that the combination of pulsations in a piping system may result in beats which are of considerable magnitude and which may contribute markedly to damage which is traceable to pulsating flow. One of the most outstanding cases

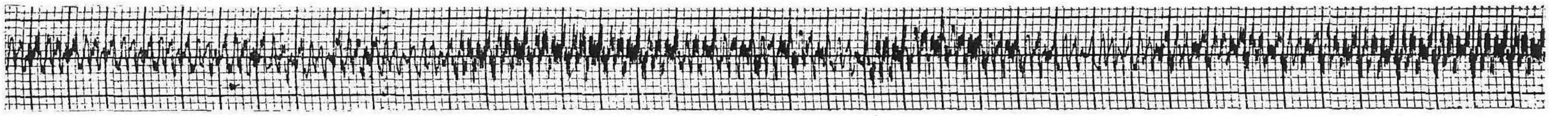


Figure 6a. Pressure-time curve taken in discharge lateral of a single compressor.

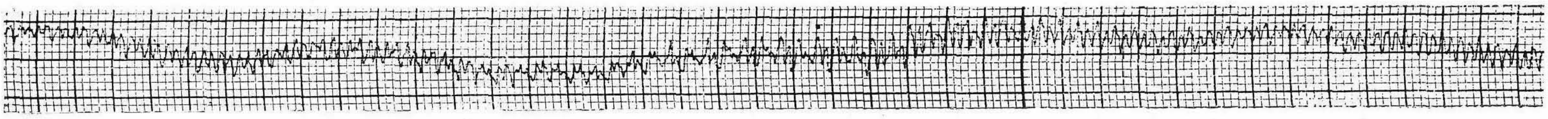


Figure 6b. Curve for discharge header receiving gas from 15 compressors.

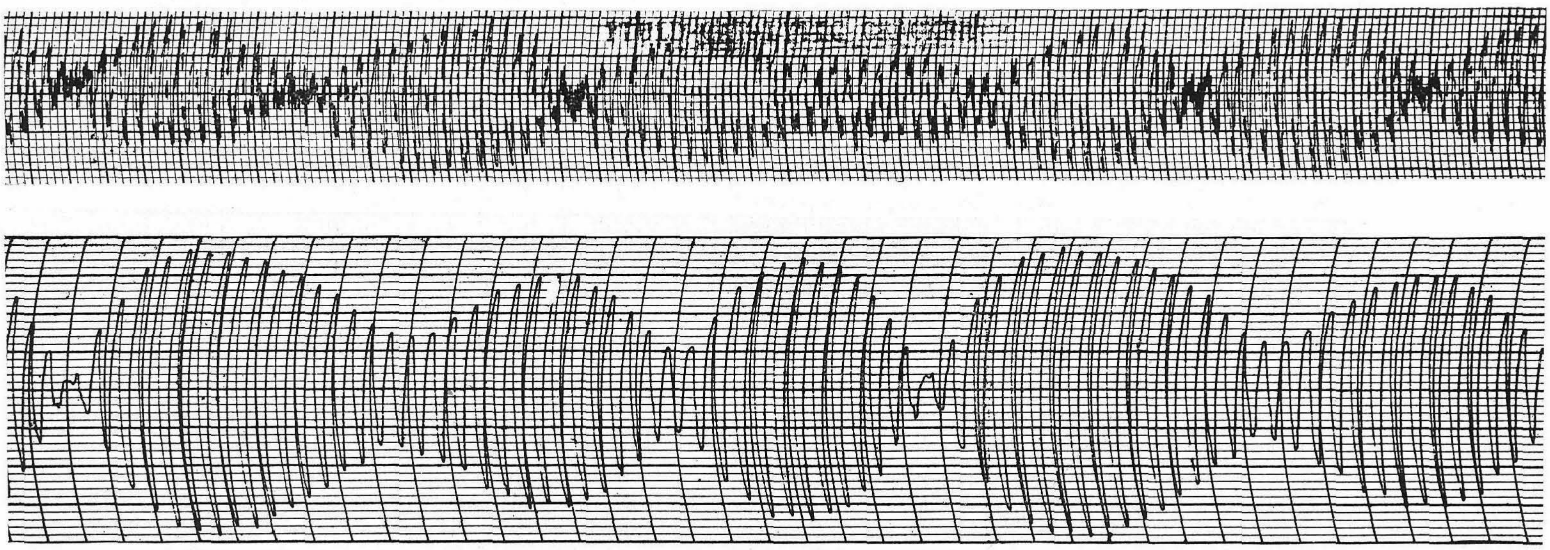


Figure 7. Pressure-time curves showing extreme conditions of beats encountered in suction systems of typical compressor installations.

of beats which has been measured to date is shown in Fig. 7. The top chart was recorded at the suction of a Clark RA-8 two-stage double-acting compressor. The measurement was made in the suction "bottle" of the second stage. The suction pressure of this unit was 42 lb/in.^{2} gauge and the maximum peak-to-peak pulse pressure was 8.5 lb/in.^{2} At this time the operating speed was 325 rev/min. It is to be noted that these beats have a period of approximately 1.25 seconds. The vibratory motion of the piping system at this point was severe enough to cause fracture of welded joints. Acceleration measurements made at this point gave values of 450 in./sec^2. Since the equipment included in this area is quite massive, it is believed that the presence of relatively long period beats was primarily responsible for the high stresses which were evident.

One of the most interesting investigations made concerned itself with measurements on a piping system in a plant where pulsation dampeners were in-

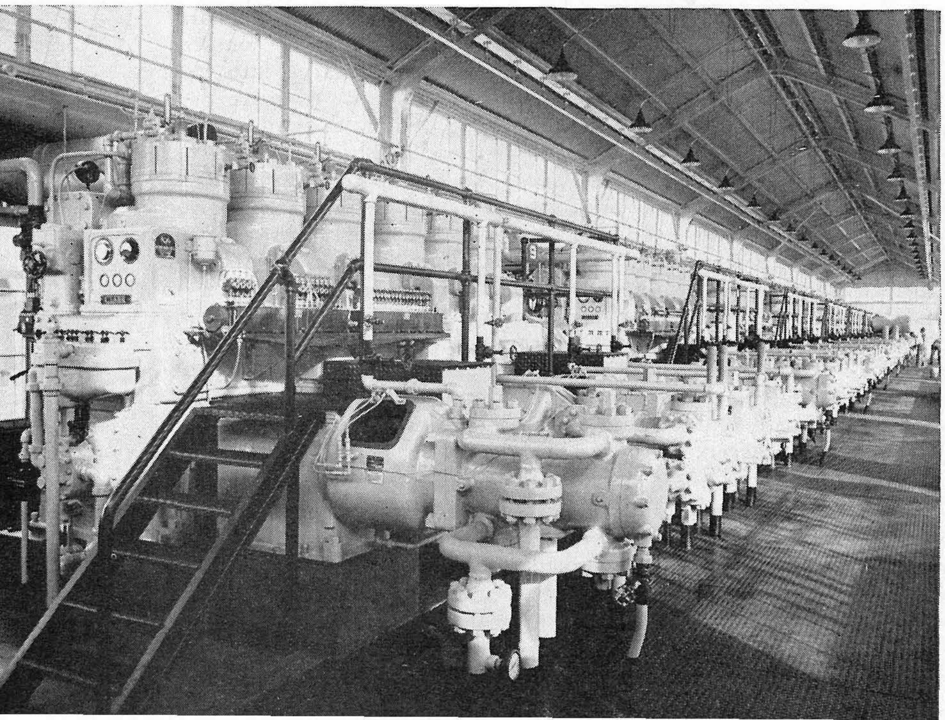
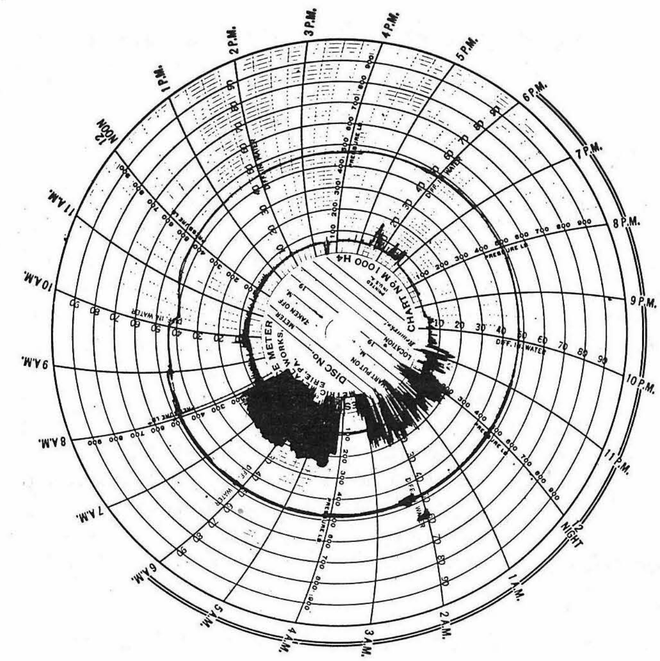
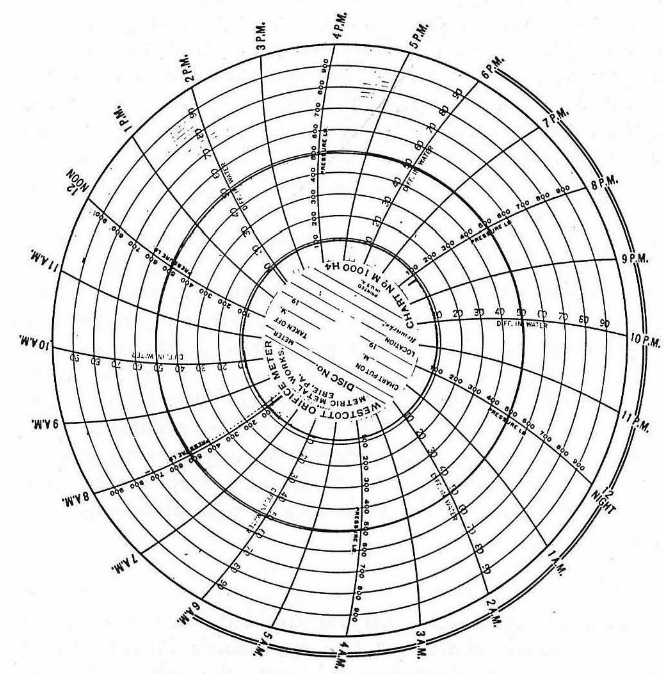


Figure 8. Recent installation of nine high-speed angle type compressors.

Figure 9. (Upper) Orifice meter chart from instrument located on the discharge side of gas compressor.
Figure 10. (Lower) Chart taken from same meter as that shown in Fig. 9 with dampener in system.





cluded in the original design. These dampeners were installed at the time the plant was constructed, and hence no data could be secured under conditions where dampeners were not in the system. This plant was equipped with nine 600 hp Clark Type RA-6 units (See Figs. 1 and 8). Two of the services being handled by these units were equipped with dampeners. The high-pressure service delivered gas into the header at 3620 lb/in.^{2} gauge. The speed of the compressor for which data are presented here was 302 rev/min. The peak-to-peak pulse pressure was 127 lb/in.^{2} at a point immediately downstream of the pulsation dampener a pulse pressure of 29 lb/in.^{2} was measured. Measurements made in the discharge header gave a pulse pressure calculated to be 15 lb/in.^{2} Although the discharge pressure of this system is quite high and the pulse pressures immediately below the cylinder are greater than those indicated above, it was found that the vibration measurements reveal very low values of acceleration throughout the entire piping system. Acceleration readings of a magnitude of 6 to 70 in./sec^2 were found in the system below the dampeners.

It has previously been indicated that pressure pulsations may have a very adverse effect upon orifice meter recorders. An outstanding example of the effect produced is shown in Fig. 9, which is a reproduction of a meter chart taken from an instrument on the discharge side of a commercial compressor system. When one of the compressors was started, the wide band of vibrating lines was recorded by the meter. After the installation of a pulsation dampener in this system, the recorder pen was no longer subjected to this pulsating condition and charts such as that shown in Fig. 10 were produced.

All of the pressure-time charts shown above are run with a horizontal speed of 25 mm/sec. The vertical or pressure deflections are not comparable within the series of charts presented here, since the amplification used was different in the several cases shown. However, since actual peak-to-peak pulse pressures are mentioned for each case, it is possible to arrive at a comparison of the data shown in the charts. The recordings shown here were made with a group of special instruments which have been developed particularly for the purpose of measuring pressure pulsation phenomena in piping and other equipment. This equipment has been especially designed to be responsive with good fidelity down to frequencies as low as 1 cycle/sec. Recordings are made with practically uniform response up to 120 cycles/sec.

The examples used in the above discussion were selected from many which have been encountered in making measurements in the field. They were purposely selected to illustrate the four effects of pulsation phenomena indicated in the beginning of this paper. The data have been drawn from measurements made on certain types of compressors. This does not imply that only these types produce pulsation phenomena in their associated piping systems. It has been found that all compressors produce some type of compressor may be found to give results of both systems. From the smallest units used in the laboratory, **e.g.,** ½ hp refrigeration compressors, up to the 1000 hp units discussed above, it has been found that pulsations are a common characteristic of all systems. In some systems the pulse pressure is so small as to be insignificant, while in others it is of extreme severity, and hence is of importance as a destructive factor in the system. The same type of compressor may be found to give results of both extremes, depending upon the piping system and other equipment associated with the unit.