EXPERIENCES WITH NONSYNCHRONOUS FORCED VIBRATION IN CENTRIFUGAL COMPRESSORS

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When a compressor rotor experiences subsynchronous vibration, the problem is generally considered to be a shaft instability problem associated with the stability of the rotor on the bearing oil film. However, many times the high subsynchronous vibrations are forced vibrations caused by flow instabilities, such as stage stall. In these cases, modifications to improve the rotor stability by changing the bearings or seals will have little effect on the subsynchronous vibrations. It is therefore important to understand the differences between forced vibrations and self-excited vibrations so the problem can be properly diagnosed and corrected (References 1-4). The following is a list of characteristics of the two types of subsynchronous vibration.

Self-Excited

1. The vibrations generally occur near the first critical speed of the shaft.

2. The vibrations are controlled by the stability of the rotor and the oil film.

3. The vibration amplitudes can suddenly increase and become unbounded until the rotor contacts stationary parts, such as seals and labyrinths.

4. The rotor whirls at the subsynchronous frequency and the whirl direction can be in the direction of rotation (forward) or opposite the direction of rotation (backward).

Forced Vibration

1. The vibrations are caused by aerodynamic excitation (flow instabilites) and are influenced by the acoustical characteristics of the combined compressor and piping systems.

2. The subsynchronous vibrations occur at the lower flows near surge and are bounded in amplitude (as opposed to unstable shaft vibrations which can increase until the shaft contacts stationary parts).

3. The whirl direction is generally forward.

4. The subsynchronous vibration frequencies are usually 5 - 20% of the running speed frequency.

5. The subsynchronous amplitudes are a function of the impeller vane tip speed and gas density.

6. The subsynchronous shaft vibrations and pulsations are phase coherent.

7. The subsynchronous pulsations generally are higher amplitude on the discharge side and do not occur on the suction side unless there are inlet flow distortions.

8. In multi-stage compressors the subsynchronous pulsations are generally associated with the final stages.

9. The pulsation frequencies are determined by the acoustical responses of the entire system including the compressor internals and the piping. Many times there are multiple harmonics of some basic response frequency.

10. In centrifugal compressors, the excitations are often associated with stage stall in the diffuser or return channel.

SYMPTOMS OF FLOW INSTABILITIES

Subsynchronous forced vibrations are often an indication that the compressor is operating near a stage stall condition. Stage stall is a pre-surge condition which occurs when one of the final stages is unstable at reduced flow rates. If the unstable stage reacts with the rest of the system, then a surge condition may result. Therefore, the subsynchronous rotor vibrations can be an indication of incipient surge.

The stage stall or surge of one or more of the impellers can cause a loss of performance. Many times there will be a small drop in head as one particular frequency is excited. As the flow is further reduced, multiple frequency components are sometimes excited which can drastically reduce the performance.

Other indications of the flow instabilities, and often the most obvious, are low frequency vibrations of the attached piping. The piping vibration is due to low frequency pulsations at a fraction of the running speed frequency. This is in contrast to the normally occurring high frequency pulsations at multiples of running speed, such as blade and diffuser passing frequencies which do not normally excite the piping lateral mechanical natural frequencies.

The subsynchronous pulsations are generally less than 10 psi and seldom exceed 1% of line pressure on high pressure units. The pulsations couple at the piping elbows to produce shaking forces which can be significant in large diameter piping since the shaking force is approximately equal to the pipe cross sectional flow area multiplied by the pressure pulsations. For example, an 8 inch pipe with a pulsation of 4 psi could have a dynamic shaking force of approximately 200 lbs. Overhead piping can normally be clamped and restrained to withstand forces of 400-500 lbs; however, centrifugal piping systems typically have very few clamps due to the thermal flexibility requirements and thus the pulsation forces can produce high vibration amplitudes on the piping.

The following case histories were selected because they illustrate the effects and symptoms of subsynchronous pulsation and vibration in centrifugal equipment.

CASE A. TURBOEXPANDER/COMPRESSOR

A turboexpander/compressor unit installed in a gas processing plant experienced numerous mechanical failures, the performance was less than predicted, and the unit had high amplitude, low frequency piping vibrations. The unit operated from 11000 - 13500 rpm (183 - 225 Hz) and the piping and shaft vibrations were primarily near 12 Hz. As outlined above, these were all symptoms of forced vibrations.

Test Procedures

Field tests were made with special instrumentation to identify the source of the excitation. Dynamic pressure transducers were installed to measure the aerodynamic excitation in the turboexpander and compressor. Proximity probes were installed near the rotor bearings to confirm the existence and to assess the severity of the resulting shaft vibrations. Pressure pulsations were measured in the turboexpander inlet and discharge piping and in the compressor suction and discharge piping. Low frequency pulsations near 12 Hz were measured in the compressor suction and discharge piping. There was no indication of the low frequency pulsations in the turboexpander piping where the pulsations occurred primarily at multiples of running speed.

Vibrations of the turboexpander/compressor shaft relative to the bearing housing were measured with proximity probes. Two probes were installed near each bearing, 90 degrees apart, to obtain a shaft vibration orbit. The shaft vibration orbit showed total vibrations of approximately 4 mils peak-peak. The shaft vibration at the running speed frequency was only 1 mil peak-peak while the subsynchronous vibrations near 12 Hz were approximately 3 mils peak - peak. The shaft orbit was unsteady and similar to whirl phenomena experienced on shaft instability vibration problems (self-excited vibrations); however, the amplitude remained bounded. The shaft vibrations and suction pulsations near 12 Hz were phase coherent, which indicated that the shaft vibrations and the compressor pulsations were definitely related.

Although the data indicated that the shaft vibrations and compressor pulsations were related, it was not known which was the cause and which was the reaction. Experience has shown; however, that the shaft vibrations were probably due to the pulsations because it is difficult for the low amplitude subsynchronous shaft vibrations to produce high amplitude coherent pulsations in the gas stream. Therefore, it was felt that the shaft vibrations were forced vibrations and that modifications or balancing of the rotor would not reduce the vibrations.

While the unit was operating at a stable condition near 13000 rpm, speed modulations of 500 rpm near 12 Hz were measured. The speed modulation was obtained by analyzing the tachometer signal from a magnetic pickup with a frequency-to-voltage converter. The digital speed readout in the control room also indicated speed fluctuations, although to a lesser degree, because the signals were averaged for the readout. The speed modulation was another indication that the loading was not constant, which suggested a forced aerodynamic excitation on the system.

Solution

As shown in Figure 1, the suction piping was perpendicular to the compressor shaft and the gas flow had to make a sharp 90 degree turn to enter the compressor impeller. There were no inlet guide vanes or turning vanes in the compressor inlet chamber. It was felt that the problems were caused by turbulence occurring at the inlet of the compressor impeller. In an effort to improve the flow into the compressor, a flow splitter was fabricated on-site and installed in the inlet chamber directly in line with the suction inlet. After the flow splitter was installed, the subsynchronous vibrations, pulsations and speed modulations were significantly reduced and the performance was improved (Figure 2). Similar flow splitters have been used on induced draft fans to prevent inlet vortices which create rotating stall conditions (Ref 5).

Based upon the data obtained with the flow splitter, a compressor inlet modification was designed to further improve the compressor inlet flow conditions. The modification used an elbow inside the compressor inlet chamber to direct the flow into the impeller. A vertical flow splitter was added to ensure that the flow was properly distributed over the flow area of the elbow. Tests showed that the inlet modification greatly improved the inlet flow conditions, reduced the subsynchronous shaft vibrations and pulsations, lowered the speed modulations, virtually eliminated the low frequency piping and case vibrations and improved the compressor performance (Table 1). It is interesting to note that the rotor vibration amplitudes at the running speed (13000 rpm) were not affected by the flow instabilities. This unit has operated successfully for several years since the inlet was modified.

CASE B. CENTRIFUGAL COMPRESSOR

This compressor system operated satisfactorily for several years until the aftercooler was replaced and the flow rate was reduced. The aftercooler was replaced with a larger unit designed to increase the cooling capacity with a lower pressure drop. The flow rates were down because of the reduced demand for the product. After these changes were made to the operating system, the unit experienced piping and aftercooler vibrations and subsynchronous shaft vibrations.

Test Procedures

A field study was performed to determine the causes of the vibrations. Pulsations were measured in the compressor discharge piping and at the inlet and outlet of the aftercooler. The discharge piping vibrations were measured with accelerometers at the points of maximum vibration. The rotor vibrations were obtained with proximity probes installed near the bearings.

Frequency analyses of the compressor discharge pulsations, aftercooler inlet pulsations, discharge piping vibrations and compressor shaft vibrations are plotted for comparisons in Figure 3. It was found that pulsation and vibration amplitudes could be significantly changed by running the compressor at different operating conditions (Figure 4). The operating condition with minimum pulsation and vibration amplitudes was the condition with maximum flow rate and minimum pressure ratio. The piping vibrations and pressure pulsations were primarily at subsynchronous frequencies near 25 Hz and 75 Hz. The amplitude of the frequency components were beating, as can be seen from the time cascade spectral plots given in Figures 5-6.

During the testing, the suction absolute pressure, the discharge gage pressure, the compressor running speed and the total flow were logged every minute on the process computer in the control room. The test conditions were plotted on a performance map (Figure 7). As shown, the measured test conditions plotted on the performance map did not agree with the predicted curves displayed on the process computer. The source of the discrepancy was not known; however, it could have been caused by an instrumentation error or the actual performance curve was different from the curves stored in the computer.

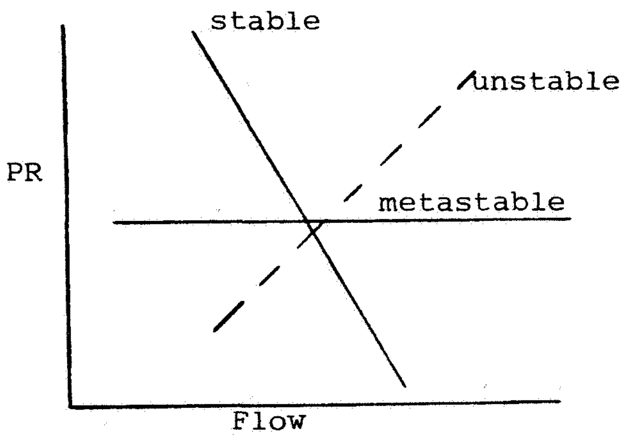
Although the measured performance data was different than the predicted performance curve, the data did show that the subsynchronous vibration and pulsation amplitudes were greater at operating conditions closer to the calculated surge line shown on the performance map. The surge line shown on the performance map is the surge line for the combined low and high pressure compressors. Although the compressors did not appear to be operating near the system surge line, the high pressure compressor was probably operating near surge or stage stall conditions for one or more of the final stage impellers.

The problems observed on this compressor appeared to be due to stage stall in which the operation of a particular stage is unstable at reduced flow rates (Refs. 3-4). Similar unsteady flow phenomena have been documented on other centrifugal compressors (Ref. 4). It has been observed that the limit of stable operation is where the pressure ratio (PR) vs. mass flow characteristics (m) is horizontal. Therefore, the criterion for stage stall may be:

$$\frac{\partial PR}{\partial m} < o \text{ stable}$$

$$\frac{\partial PR}{\partial m} = o \text{ metastable}$$

$$\frac{\partial PR}{\partial m} > o \text{ unstable}$$



As shown on Figure 7, the plot of the measured pressure ratio versus flow was much flatter than the predicted curves. This would indicate that the compressor was operating near the metastable region.

A stage can operate in an unstable condition for extended time periods without any damage or significant pulsations or noise, if the stage does not react acoustically with the rest of the system. However, if the system, including the inlet elements and discharge elements (such as the aftercooler), interacts with the unstable stage to create high pulsations, then the entire system may become unstable and a surge condition can result.

It has been observed on other units that when a compressor is operating near surge, the frequencies of the nonsynchronous compressor shaft vibrations are generally at the acoustical natural frequencies of the piping system. This helps to explain why changing the aftercooler changed the compressor discharge pulsations and increased the vibration amplitudes on the discharge piping. The discharge piping acoustical natural frequencies of the new aftercooler were considerably different from the original design. The new aftercooler had almost three times as many tubes in each section as the original design and the acoustic end conditions on either end of the U-bend pipe which connected both sections of the aftercooler were different. A comparison of the effective flow diameters for the original and new aftercoolers is shown in Figure 8. The flow area of the original aftercooler was approximately equal to the area of the discharge pipe. The flow area was much larger on the new aftercooler and appears acoustically as a volume-choke-volume which will respond as a Helmholtz resonator.

A Helmholtz resonator is a low pass filter which is typically used to filter or attenuate high frequency pulsations which are higher than the acoustical natural frequency of the Helmholtz resonator (referred to as the Helmholtz frequency). Pulsations at the Helmholtz frequency are amplified rather than attenuated. Using simplified equations, the Helmholtz frequency for the new aftercooler was calculated to be approximately 20 Hz which is near the measured fundamental pulsation frequency of 20-25 Hz.

The acoustical natural frequency of the choke tube (U bend between the aftercooler sections) was calculated to be approximately 72 Hz which was near the pulsation frequency of 62-75 Hz. This choke tube half-wave resonance is referred to as a pass band frequency and pulsations at this frequency can pass through the inlet and outlet of the filter. The pulsations near 75 Hz were undesirable because the first lateral critical speed of the high pressure compressor was also near 75 Hz. The pulsations from the aftercooler near 74 Hz increased the rotor subsynchronous vibrations because the rotor was sensitive to excitation at the first critical speed. The measured vibrations at 75 Hz were approximately 0.3 mils peak-peak at the bearings; however, the amplitudes could have been several times higher at the shaft midspan near the discharge flange. There was concern that the increased shaft vibrations could be damaging to the seals and bearings.

Solution

The measurements on the units illustrated several changes that could be implemented to reduce the subsynchronous vibrations.

1. The first step would be to operate the unit at high flow rates near the design point where the unit operated satisfactorily for several years. However, due to low product demand, the flow rates could not be increased.

2. The second modification was to operate the unit with more recycle flow which would allow the compressor to operate near the design point. The only disadvantage was that the unit was less efficient and required more horsepower for the same net flow. The recycle flow rate could easily be increased by redefining the surge control line on the process computer.

3. The third step would be to redesign the diffuser, impeller and return

channels on the last few stages to prevent the stage stall at reduced flow rates. This modification would be the most costly and may not be required if the second modification could be implemented.

4. A fourth possibility would be to change the acoustical response of the piping near the aftercooler. Orifices could be designed to reduce the pulsations without causing a significant pressure drop.

The stage stall phenomena exhibited on this compressor were similar to the symptoms shown on another centrifugal compressor which also experienced stage stall. Field data measured on the second compressor suggested that the problem could be corrected by increasing the recycle flow. Detailed tests were made to redefine the surge control valves.

The testing was begun with the compressor operating at high flow rates where the subsynchronous piping and shaft vibrations were not present. The data was continuously monitored as the flow rate was reduced while maintaining a constant speed. As the flow rate was reduced to a certain flow condition (Figure 9, Point A), subsynchronous discharge pulsations and shaft vibrations near 25 Hz would suddenly appear and the flow rate would simultaneously decrease. This flow condition was considerably to the right of the predicted surge line. This type of data was obtained at several different speed lines on the performance map (Figure 9, Point B). A line drawn through the points where the subsynchronous vibrations occurred paralleled the surge line. This line was considered to be due to stage stall or surge of one or more of the final stages.

The recycle control valve was adjusted to keep flow rates to the right of this new surge line and the compressor then operated satisfactorily without any subsynchronous pulsation or vibration. As shown, this line was considerably to the right of the manufacturer's surge line for the entire compressor. These stage-stall conditions are different from machine surge and should not be confused. The machine surge is usually much more violent compared to the surge for individual impellers.

CONCLUSIONS

These compressors exhibited subsynchronous vibrations which had characteristics similar to a shaft instability; however, these were forced nonsynchronous vibrations due to unstable flow conditions. These two compressor rotors were stable (vibrations were bounded) and modifications to the bearings and shafts would not have reduced the subsynchronous vibrations.

The stage stall and surge conditions are a function of the entire system which explains why a compressor can operate satisfactorily for several years and then become unstable after modifications are made to seemingly unrelated piping elements, such as heat exchangers or downstream receivers.

REFERENCES

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3. L. Bonciani, L. Terrinoni, A. Tesei, "Unsteady Flow Phenomena in Industrial Centrifugal Compressor Stage", Instability Workshop, Texas A&M University, NASA CP-2250, 1982, pp. 344-364.

4. David Japikse, "Stall, Stage Stall, and Surge", Proceedings of the Tenth Turbomachinery Symposium, Texas A&M University, December 1981.

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TABLE I. - COMPARISON OF VIBRATIONS AND PULSATIONS WITH DIFFERENT INLET MODIFICATIONS

	Without Splitter	With Splitter	Modified Inlet
Shaft Vibration Mils peak-peak			
Compressor - Orbit	3.5-4	2.5	1.2
Running Speed @ 13000 rpm	1.1	0.9	1.1
Expander - Orbit	1.5	1.0	0.7
Running Speed @ 13000 rpm	0.5		0.5
Torsional speed modulation, rpm			
Peak-Peak Speed Modulation	500	400	40
Primary Frequencies, Hz	1,6,9,11	1,3,5,6,11	6,12
Pulsation psi peak-peak/Hz			
Compressor Suction	1.4/12	0.2/11	0.16, 0.2/12
Compressor Discharge	2.0/11	0.2/11	-
Piping Vibration mils peak-peak/Hz			
Compressor Suction at Elbow North-South @ 13000 rpm	5.0/11	2.6/12	-

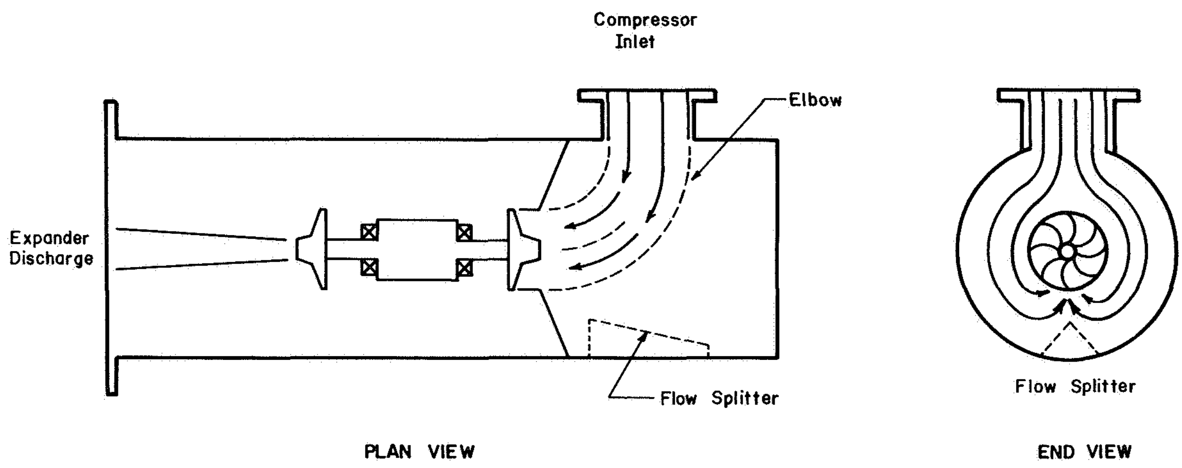

Figure 1. - Cutaway drawing of turboexpander/compressor, illustrating inlet modifications.

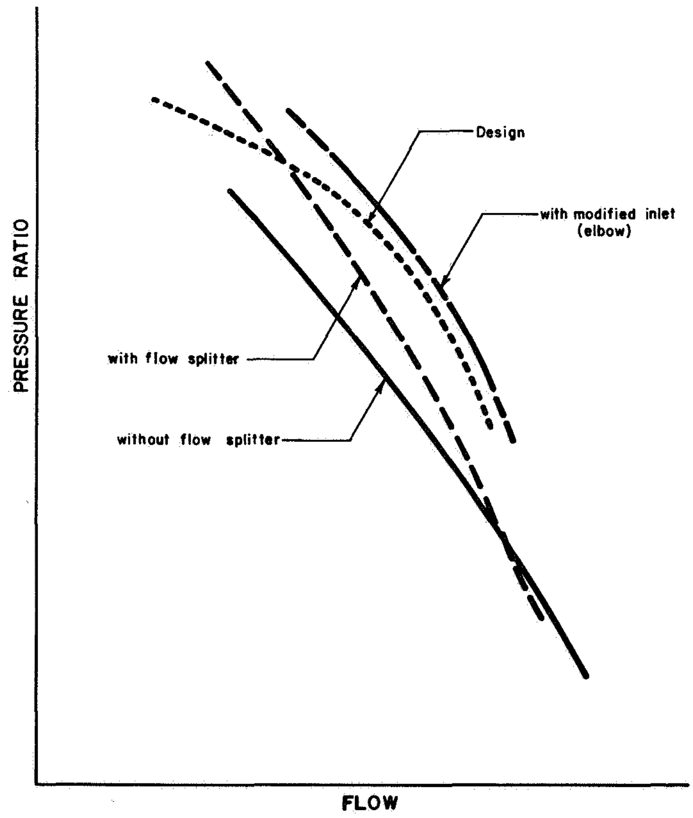

Figure 2. - Results of compressor performance test.

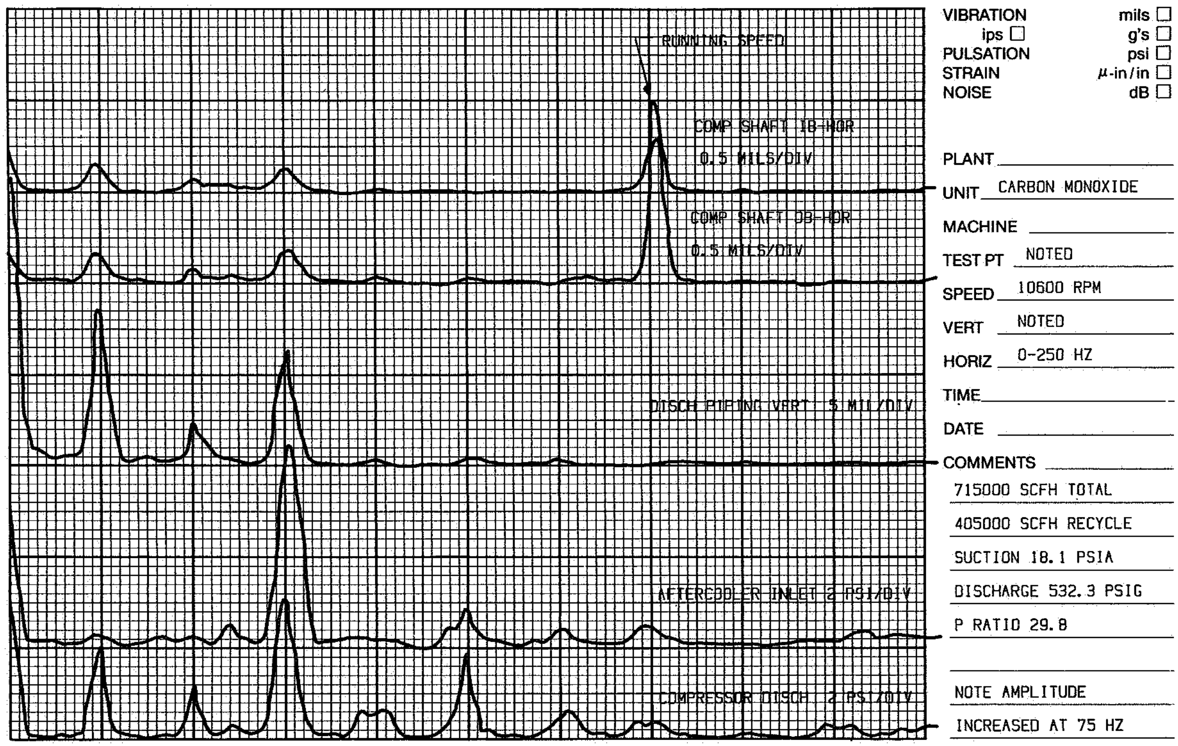

Figure 3. - Comparison of vibrations and pulsations with different inlet modifications.

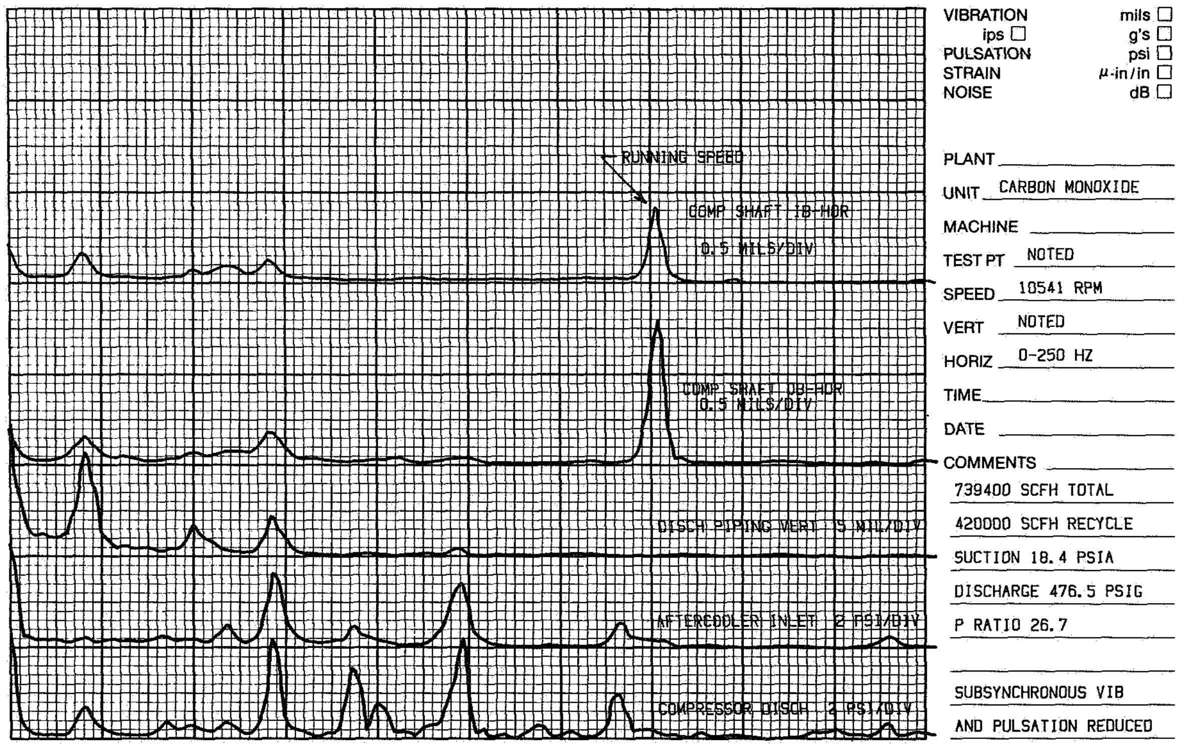

Figure 4. - Comparison of vibrations and pulsations at high flow rate and low pressure ratio.

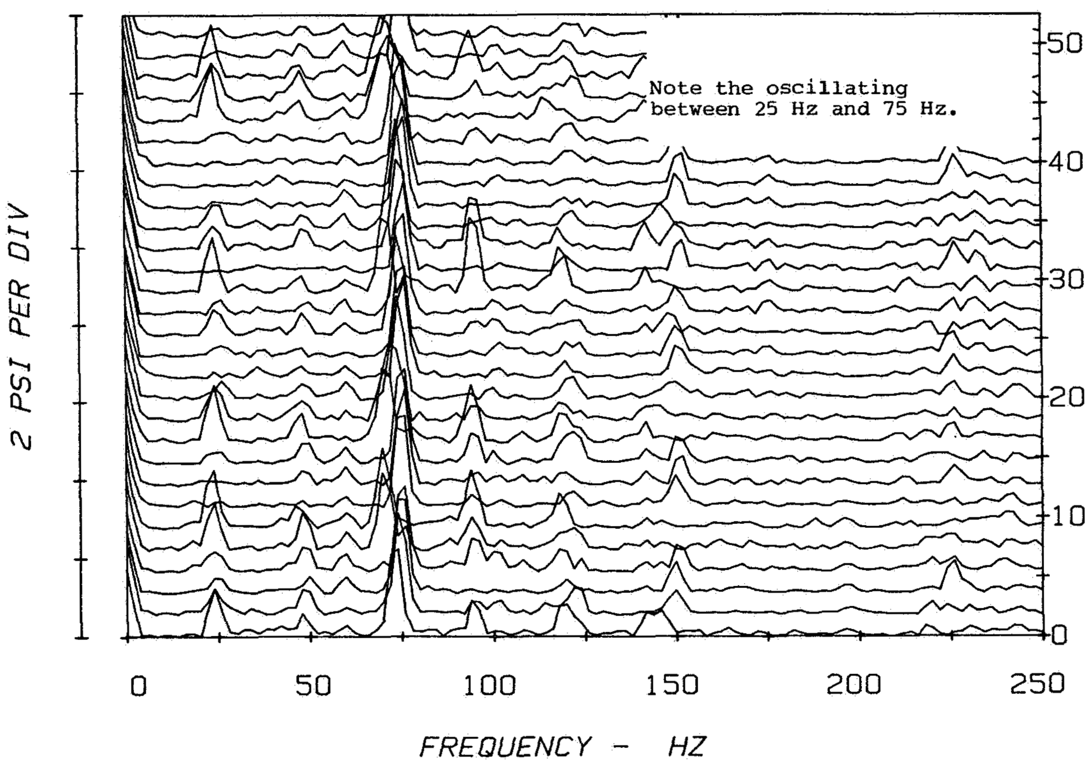

Figure 5. - Compressor discharge pulsations as a function of time.

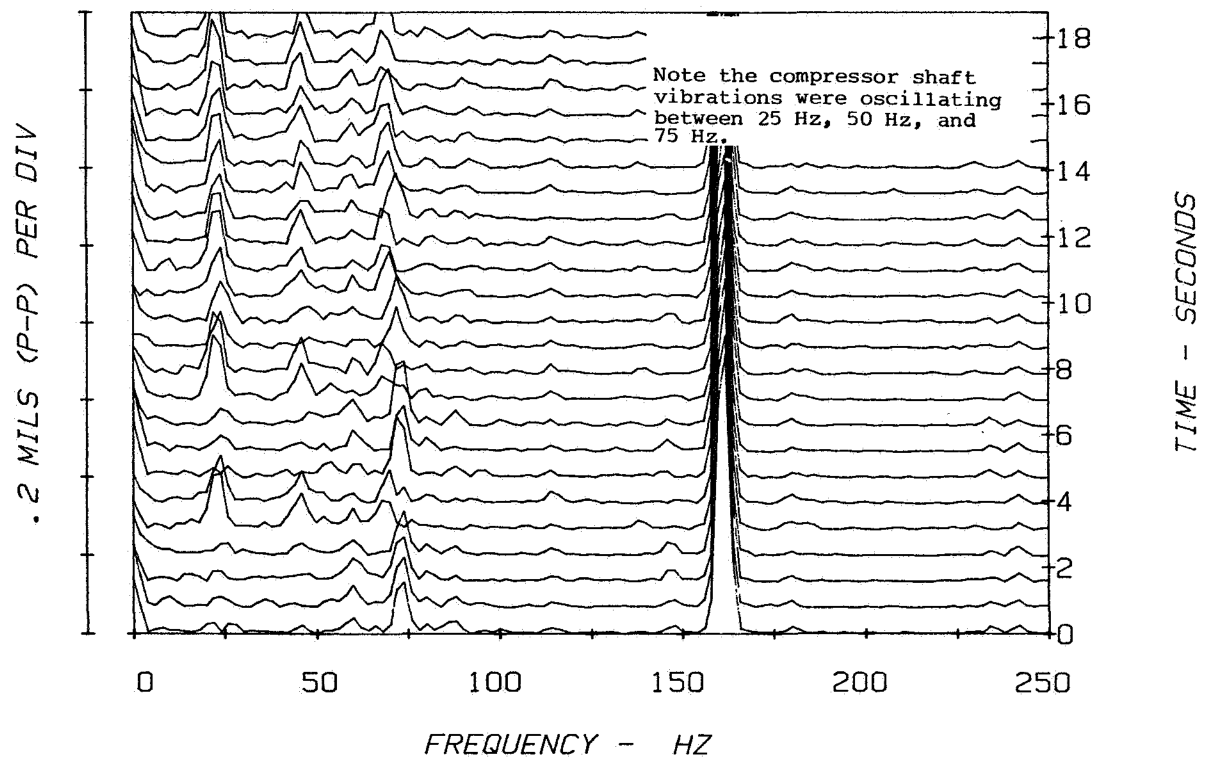

Figure 6. - Compressor shaft vibrations as a function of time.

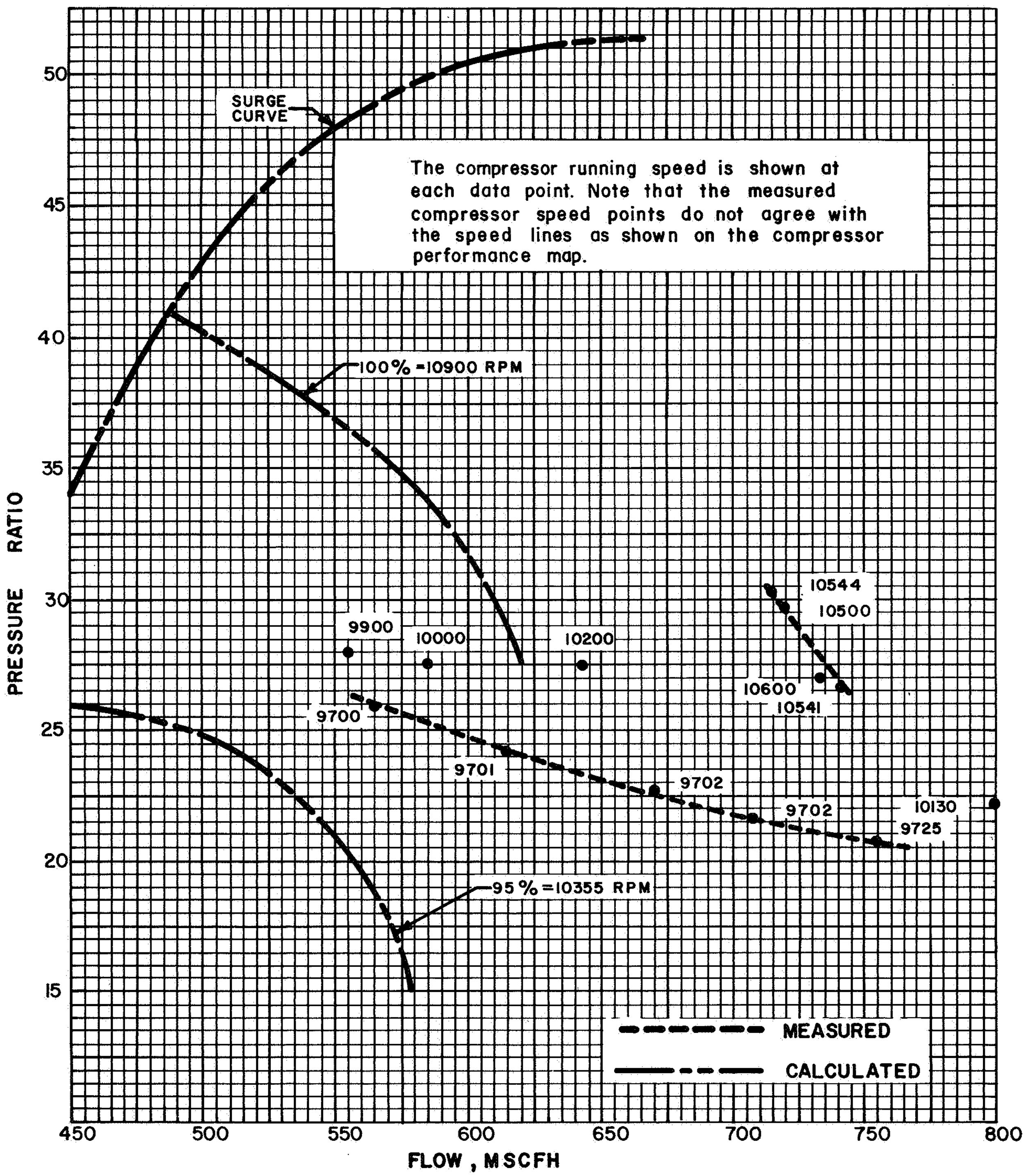

Figure 7. - Measured pressure ratio as a function of flow.

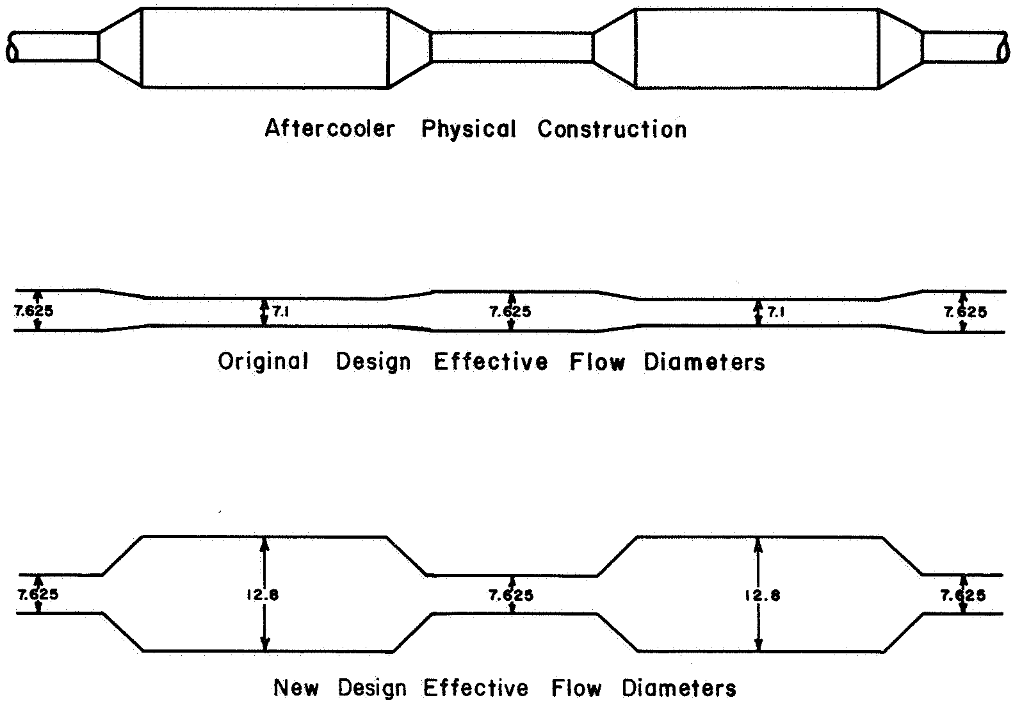

Figure 8. - Acoustical comparison of original and new aftercooler.

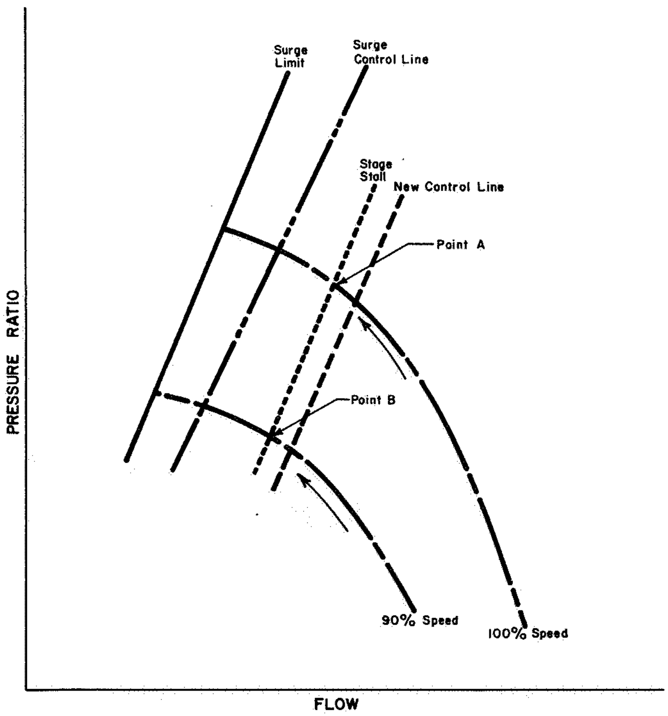

Figure 9. - Compressor performance surge curve.