NONSYNCHRONOUS FORCED VIBRATION IN CENTRIFUGAL COMPRESSORS

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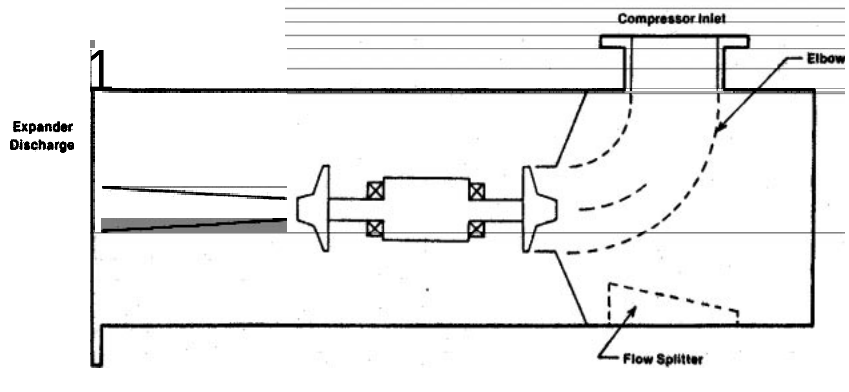
Flow instabilities in centrifugal compressors can produce low frequency turbulence and pulsations which can result in nonsynchronous rotor vibrations, vibrations of the case and attached piping, speed modulations, and reduced compressor performance. High frequency pulsations in rotating equipment typically occur at multiples of running speed such as blade, diffuser and nozzle passing frequencies and can excite blade natural frequencies and radial shell wall resonances. The piping lateral mechanical and acoustical natural frequencies are generally at lower frequencies and are not excited by these pulsations.

Compressor surge is commonly known to cause violent compressor and piping vibrations; however, a pre-surge condition caused by stage stall can exist which causes bounded rotor vibrations at subsynchronous frequencies.

Flow modulation is nearly always present in centrifugal and axial compressors and can cause aerodynamic excitation which is one of the main sources of subsynchronous vibration (Ref. 1, 2). Two types of subsynchronous vibrations have been observed: self-excited vibrations and forced vibrations. The self-excited vibrations generally occur near the first critical speed of the shaft and are controlled by the stability of the rotor and oil film. Forced vibrations of the rotor are caused by the aerodynamic excitation due to stage stall or full machine stall and



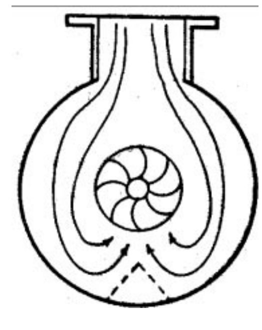


Fig. 1. Cutaway of **Turboexpander/Compressor** Illustrating Inlet **Modifications.**

are influenced by the acoustical response characteristics of the combined compressor and piping systems. Examples of rotor and piping vibrations of the latter type will be presented.

These forced vibrations have the following characteristics:

1. The subsynchronous vibrations occur at the lower flows negr surge and are bounded in amplitude (as opposed to unstable shaft vibrations which san increase until the shaft contacts stationary parts such as seals).

2. The asynchronous frequencies are lower than the running speed frequency.

3. The asynchronous amplitudes are a function of the tip speed and gas density.

4. The asynchronous shaft vibrations and pulsations are phase coherent.

5. The asynchronous pulsations generally occur on the discharge unless there are inlet flow distortions.

6. In multi-stage compressors the asynchronous pulsations are generally associated with the final stages.

7. 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.

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

SYMPTOMS OF FLOW INSTABILITIES

Often the most obvious indications of flow instabilities are low frequency piplng vibratioqs. The asynchronous 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 a ghaking force 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 ibs; 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.

Another indication of the occurrence of flow instabilities is Increased shaft vibrations caused by the pulsation forces on the rotor. These can become excessive if the acoustical response coincides with one of the damped natural frequencies of the rotqr.

A third indication is a loss of performance due to stage stall or surge of one or more of the impellers. 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 drastically reduces the performance' (Ref. 2).

Two ease histories are briefly reviewed to illustrate the effects and symptoms of subsynchronous pulsation and vibration in centrifugal equipment.

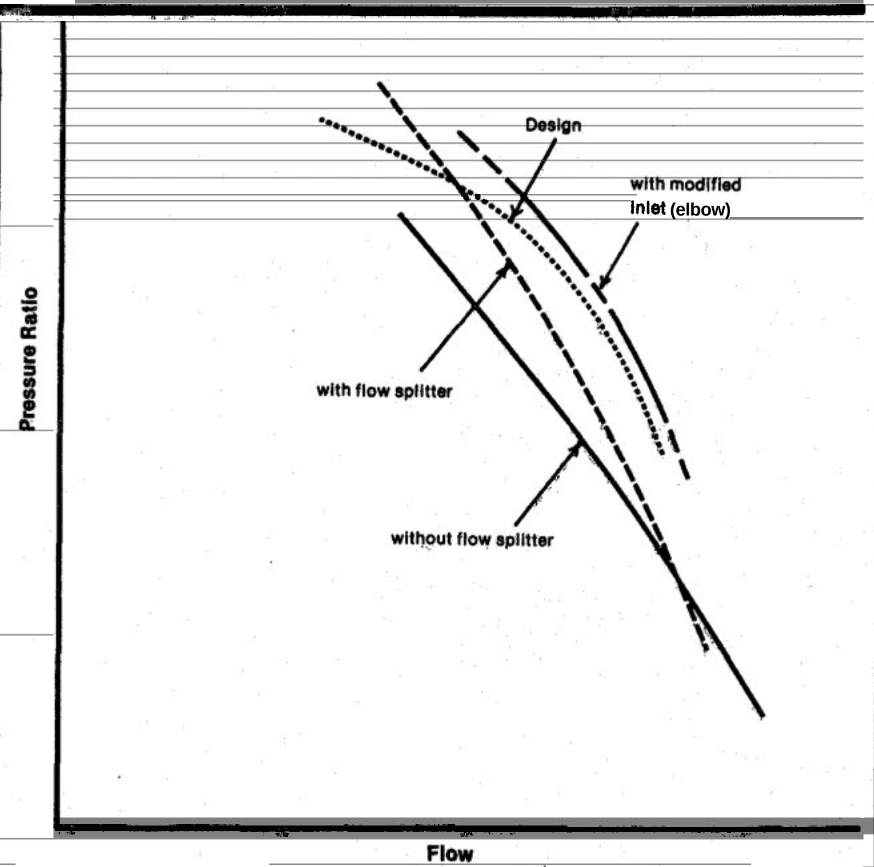


Fig. 2. Compressor Performance Test Results.

CASE A. TURBOEXPANDER/COMPRESSOR

A turboexpander/compressor unit installed in a gas processing plant experienced numerous mechanical failures and low frequency, high amplitude vibrations on fhe piping and housing. The performance of the unit was less than predicted. The unit operated from 11,000-13,500 rpm (183-225 Hz) and the piping vibrations were primarily near 12 Hz.

Tests ware made with special instrumentation Installed on the unit to identify the source of the excitation. Pressure transducers were required to determine the aerodynamic excitation; proximity probes and a torsiograph were installed to confirm the existence and to assess the severity of the resulting vibrations.

Pressure pulsations were measured in the expander 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 expander piping where the pulsations occurred primarily at multiples of running speed. Analysis of this data indicated that the low frequency excitation was. primarily associated with the compressor suction.

Vibrations of the expander/compressor shaft relative to the bearing housing were measured with proximity probes. Two probes were installed near each bearing 90 degrees apart ta 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 while the subsynchronous vibrations were approximately 3 mils. 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 were coherent which indicated that the shaft vibrations were forced and caused by the pulsations.

While the unit was operating near 13,000 rpm, speed modulations of 500 rpm at approximately 12 Hz were measured wlth a torsiograph. The speed modulation was obtained by analyzing the tachometer signal from a magnetic pickup with a frequency-to-voltage converter. The speed modulation was another indication that the loading was not constant which suggested a forced aerodynamic excitation on the system.

As shown in the sketch 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 iniet guide vanes or turning vanes in the compressor inlet chamber. It was determined that the problem was caused by turbulence occurring at the inlet of the compressor impeller. A flow splitter was fabricated on-site and installed in the inlet chamber directly in line with the suctlon inlet and the vibratlons and pulsations were stgnificantiy reduced and the compressor performance was improved. Similar flow splitters are used in induced draft fans to prevent inlet vortices which create rotating stall conditions (Ref. 3).

TABLE I

COMPARISON OF VIBRATIONS AND PULSATIONS WITH DIFFERENT INLET MODIFICATIONS

	Without Splitter	Wlth 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/Hz			
Compressor Suction Primary Amp/Freq	1.4/12	0.2/12	0.1/6, 0.2/12
Compressor Discharge Primary Amp/Freq	2.0/11	0.2/11	—
Piping Vibration mils peak-peaMHz			
Compressor Suction at Elbow North-South @ 13000 rpm	5.0/11	2.6/12	—

Based upon the data obtained wlth the fiow splitter, a compressor inlet modification was designed to further improve the compressor iniet fiow conditions. The modification used an eibow inside the compressor inlet chamber to direct the fiow into the impeller. A vertical fiow splitter was added to ensure that the fiow was properly distributed over the flow area of the elbow. The inlet modiflcatlon greatly improved the inlet flow conditions, reduced the subsynchronous shaft vibratlons and pulsations, lowered the speed modulations, virtually eliminated the low frequency piping and case vibrations, and improved the compressor performance (Figure 2). A comparison of the data in the original condition, with the fiow splitter, and with the elbow inside the case is shown in Table I.

CASE B. MULTISTAGE CENTRIFUGAL COMPRESSOR

This multistage centrifugal compressor was used in a gas lift service and operated near 10,000 rpm (166 Hz). At reduced flow rates the discharge piping vibrated excessively and the shaft vlbrations increased above the alarm levels on the proximity probe meters. The primary vibration frequency of the piping and shaft was 25 Hz. The manufacturer's performance curves indlcated that the compressor was operating far to the right of the predicted surge curve even at the reduced flow rates.

The compressor was operated at

several different conditions on the performance map to determine the cause of the excitation. During the testing, pulsations were recorded in the suction and discharge piping and the compressor shaft vibrations were measured with proximity probes. The compressor suction and discharge static pressure and flow was logged on a multi-channel strip chart recorder.

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 continously monitored as the flow rate was reduced while maintaining a constant speed. As the flow rate was reduced to a certain flow condition (Figure 3, 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 3, 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.

This problem appeared to be due to stage stall which prevents a stage from operating stably at reduced flow rates (Ref. 4). If the unstable stage reacts with the rest of the system then a surge condition will result. Modifications to the impeller or channel diffuser would be required to prevent the stage stall and allow the unit to operate at reduced flow rates without producing the subsynchronous pulsations and vibrations.

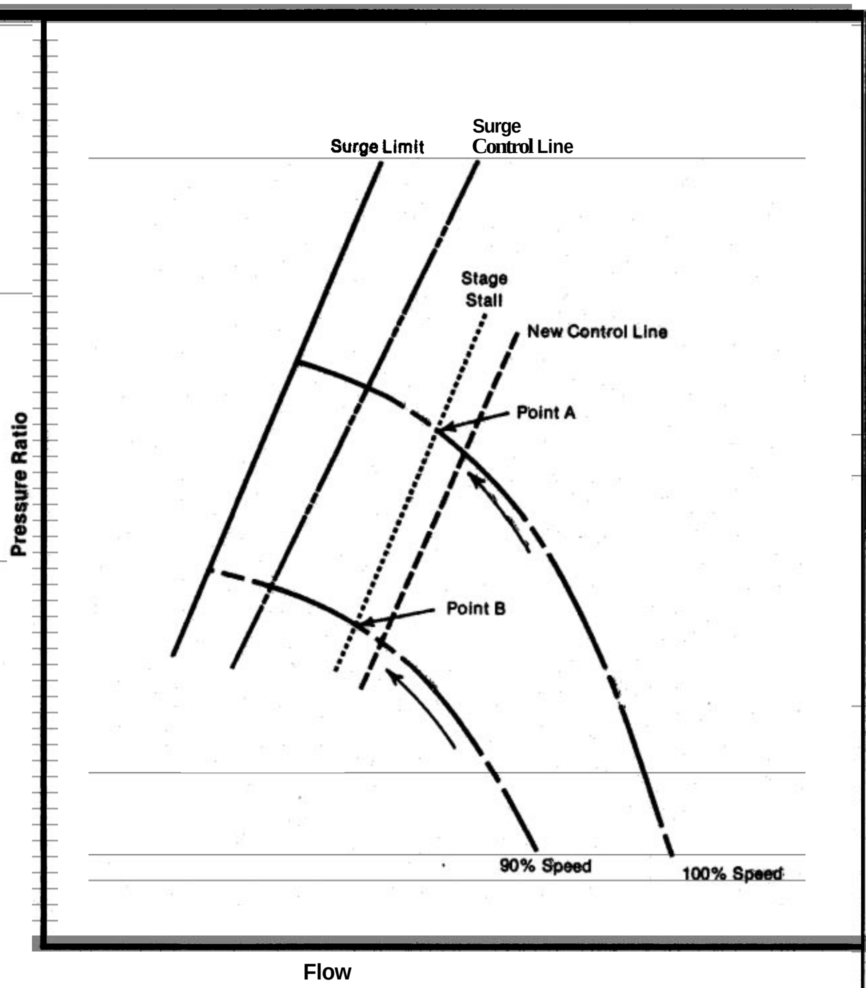


Fig. 3. Compressor Performance Surge Curve.

CONCLUSIONS

These two 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 on a test stand and then experience problems after it is installed in a piping system. Some units 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.

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