Compressor Noise Control

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INTRODUCTION

Compressors have been identified as major noise sources, and this is understandable. Large numbers of compressors, of all types, are used in refineries, chemical plants, generating stations, and other major industries. Certain types of compressors generate relatively high noise levels -- above those permitted by the Occupational Safety and Health Act -- and therefore need attention. Portable compressors produce some of the most objectionable noise on city construction projects, and for this reason, most city noise control codes set maximum permissible levels for compressors. Pending Federal legislation includes compressors in the list of products for which noise emission standards will be established. It is obvious that compressor sound control is needed, and this requires an understanding of the noise generating process. Various techniques have been found to be effective in reducing the noise of centrifugal, axial, axi and reciprocating compressors.

COMPRESSOR NOISE SOURCES

The noise radiated from a compressor is complex, and consists of components from many sources. In order to reduce the total noise, the various contributions must be identified and evaluated, and the largest ones worked on first. The ideal approach is to prevent the generation of noise by design, but this is not always the most economical solution. Noise reduction after a machine has been built is often the most practical procedure.

TURBULENCE

Consider turbulence for example. Turbulence is the most important source of noise in centrifugal compressors. This is really a combination of two effects, (a) Vortex shedding, and (b) upstream turbulence. The boundary layer over each blade is turbulent by the time it reaches the trailing edge. The turbulent layers on the top and bottom surfaces produce a fluctuation in the lift, and this fluctuation has a broad frequency spectrum. The application of a fluctuating force to a gas generates sound at the same frequency. Therefore, broad band noise is radiated. If the flow is turbulent when it enters a blade row, the turbulence is increased and the noise is greater.

Turbulence noise is radiated through the compressor casing, and it can be controlled by an acoustic enclosure -- after the compressor has been installed. It is almost impossible to eliminate the turbulence by design.

Piping noise, produced by the same source can be reduced effectively by lagging the pipes with 2 to 3 inches of Fiberglass, Ultracoustic, Rockwool, or similar material, with a density of about 4 pounds per cubic foot, and covering this with a jacket weighing about 1 pound per square foot. The jacket can be #24-gage steel, or the equivalent weight of aluminum, lead or leaded-vinyl.

To be most effective, enclosures and pipe lagging must be tight. Leaks in an enclosure greatly reduce its effectiveness. It is unfortunate that a small leak in a high quality enclosure is more damaging than the same leak in a poor enclosure. For example, an opening with an area of 1 percent of a wall whose transmission loss is 50 dB reduces the overall transmission loss to only 20 dB. A 1 percent leak in a 25 dB wall results in a final transmission loss of 19 dB. That is, the effectiveness of the 50 dB enclosure is reduced by 30 dB while the effectiveness of the 25dB enclosure is reduced by 6 dB. This shows that if you plan to have leaks in the enclosure there is no point in paying for a high quality one.

The installation of inlet and discharge silencers is another example of effective noise reduction after a compressor has been built. Silencers reduce the noise entering the inlet and discharge pipes and makes piping noise reduction easier.
In some instances, pipe lagging is not necessary, depending, of course, on the final noise level required. When silencers are used they should be as close as possible to the compressor inlet and discharge flanges. It should be noted that it is almost impossible to make a significant reduction in the noise from centrifugal compressor installations unless the piping is treated.

INTERACTION OF ROTATING AND STATIONARY VANES

On the other hand, centrifugal compressors have other major noise sources which can be reduced by design. An example of this is the noise produced by the interaction of rotating impeller blades with stationary vanes.

Every time a blade passes a given point, the air or fluid at that point receives an impulse. Therefore, that point will receive impulses at a frequency equal to the number of impeller blades times revolutions per second. In axial flow compressors the magnitude of this blade-passing frequency component is one of the largest in its generated sound spectrum. It is present also in centrifugal compressors with diffusers, but in most cases it is not as important as the blade-rate frequency. This is calculated as follows:

\[ f = \frac{N_r \times N_s \times \text{R.P.S.}}{K} \]

where
- \( f \) = Frequency in hertz
- \( N_r \) = Number of rotating (impeller) blades
- \( N_s \) = Number of stationary (diffuser) vanes
- \( K \) = Highest common factor of \( N_r \) and \( N_s \)

For example, let \( N_r = 6 \), \( N_s = 8 \), and the speed equal to 6000 RPM. Then,

\[ f = \frac{6 \times 8}{2} \times 100 = 2400 \text{ Hz} \]

With a combination of 4 impeller blades and 6 diffuser vanes,

\[ \frac{N_r \times N_s}{2} = \frac{4 \times 6}{2} = 12 \]

That is, there are 12 times in each revolution when impeller blades line up with diffuser vanes, and each time this happens, 2 impeller blades match 3 diffuser vanes. Therefore, the frequency will be 12 times RPS and the pulses will be of double strength.

When there are 6 impeller blades and 9 diffuser vanes, there are 18 times when impeller blades are in line with diffuser vanes. Each pulse is 3 times as strong as it would be with a single coincidence, because 3 rotating blades match 3 stationary vanes. The frequency is 18 times RPS.

It is obvious that combinations like 12 and 12 are not recommended because of the many points of coincidences, and the strength of the pulses.

In the case of 5 rotating blades and 9 stationary vanes, there are 45 points of coincidence, but each time only 1 rotating blade lines up with one stationary vane. This combination produces a frequency of 45 times RPS.

It can be seen that it is better to use unequal numbers of rotating and stationary vanes. Prime numbers are the best of all because they have no common factor. This produces high frequency pulses, which are easier to control than low frequency ones. Furthermore, when prime numbers are used, each pulse is only of single strength.

IMPELLER-DIFFUSER DISTANCE

Increasing the radial distance between impeller blades and diffusers reduces noise. It is particularly effective in reducing the blade-passing frequency and blade-rate components. Unfortunately this procedure also decreases performance, but for close initial spacing, the decrease in noise is greater than the decrease in performance. That is, the noise increases rapidly as the spacing becomes smaller and smaller.

EFFECT OF HORSEPOWER

Centrifugal compressor noise is affected by many operating parameters. There is a direct relation between horsepower and noise, but the relation is not the same for all types. On one particular class, the overall noise can be predicted quite accurately by

\[ \text{Increase in dB} = 17 \log \text{H. P. Ratio} \]

In most instances, doubling the horsepower results in an increase of about 4 to 5 dB in the overall noise.

EFFECT OF SPEED

Rotational speed has a definite effect on noise. For any particular design, the sound level will increase anywhere from 20 to 50 times the logarithm of the speed ratio. At lower speeds, centrifugal compressor noise will increase about 20 log RPM ratio. At high speeds, the increase
will more nearly equal 50 log RPM ratio. The increase in sound with speed applies to the overall noise and to the component of highest level -- usually blade-passing frequency or blade-rate frequency. The increase at other frequencies is not as great, and may be of the order of 10 to 15 times the log of the speed ratio.

These same relations apply to impeller tip speeds, but in general less noise will be produced with large diameter, slow speed units, than with small diameter, high speed machines, even though the impeller tip speeds are the same in both cases. There are several reasons for this:

(a) Not all the turbulence is produced by the impeller. Even though the tip speeds are the same, the slow speed machine will have lower velocities at the impeller. The slow speed machine will also have larger areas in internal passages, meaning lower velocities and less restriction.

Extra care in producing fine interior finish in centrifugal compressor casings to reduce noise is not justified. There is no detectable difference when passages are hand finished.

(b) Mechanical forces due to unbalance are proportional to the square of the speed, and, therefore, will produce less structural resonances when the speed is low.

NUMBER OF STAGES

The noise generated by centrifugal compressors can be reduced by decreasing the work per stage— that is, by increasing the number of stages.

EFFECT OF GAS MOLECULAR WEIGHT

The molecular weight of the gas in a compressor system has a pronounced effect on the generated noise. Very little test data is available on this and it is difficult to predict mathematically. It is certain though that more noise is produced with high molecular weight gas than with low molecular weight gas.

HEAD-CAPACITY OPERATING POINT

Mass flow and discharge pressure both have a profound effect on the noise produced by a compressor. As the mass flow is reduced the noise decreases until a point near surge is reached. Beyond this point the noise increases rapidly.

INLET AND DISCHARGE

The inlet to a centrifugal compressor plays an extremely important part in the generation of noise and pressure pulsations.

It is sometimes thought that if the inlet and discharge areas are increased, the gas velocities will be reduced and, therefore, less noise will be generated. On the contrary, if the inlet opening is increased, it may actually create more noise rather than reduce it. In order to obtain proper entrance, the gas should be as near as possible to the impeller center line. This, of course, indicates a small inlet. When the gas enters near the center line on its way to the impeller vane there will be relatively low shock and turbulence. If the suction opening is enlarged, the gas is admitted farther up on the impeller vane where the linear speed is higher. The sudden change from low velocity in the large inlet to high velocity part way up on the vane causes shock, turbulence and increased noise.

Reducing the flow velocity in discharge piping is an effective way to reduce piping noise. This should not be accomplished by enlarging the opening at the cutwater, but by using a properly designed pipe increaser between the correct discharge opening and the piping.

A Xi COMPRESSORS

Axi compressors are helical rotor, positive displacement compressors. Gas enters the intake ports and flows into a pocket formed between the rotors and the wall of the casing.

The pocket rotates away from the intake ports, and as the lobes and grooves roll into each other the pockets shorten and compress the gas. It moves axially and is carried toward the discharge end.

The major noise source in compressors of this type is the discharge. Next in importance is the inlet, and then the noise radiated from the compressor casing. The sound pressure level measured in the discharge of the larger, higher horsepower units may be in the range of 140 to 145 dB. Without silencers or an enclosure, the sound level 3 feet from the compressor may be about 132 dBA, and it consists of a fundamental frequency equal to the number of lobes times revolutions per second, plus higher order harmonics of this.

The amount of necessary sound control depends
on the final acoustic design criterion. For maximum noise reduction, both discharge and inlet silencers should be installed, as close to the compressor as possible. Inlet and discharge piping and inlet and discharge silencers should be lagged, and vibration isolators should be installed between the silencers and the piping. An acoustic enclosure should be installed on the compressor itself to reduce casing radiation.

**RECIPROCATING COMPRESSORS**

In general, the sound level of reciprocating compressors is not very high. It usually consists of multiples of the piston movement, and is generated by both aerodynamic and mechanical forces.

Inlet and discharge noises are major components, but the pulsating flow can be reduced effectively by a chamber-type silencer or snubber.

Inertia forces are major exciting factors causing vibration and noise in reciprocating compressors. These forces are due to the motion of the pistons and related parts, and the imbalance of the connecting rod and crank mechanism. The forces produced by unbalance masses also appear in the rotating parts of the machine, as both static and dynamic unbalance.

Impacts in the crank-connecting rod system, and the knocking of the pistons against the cylinder-liners during crossover, are major sources of noise in reciprocating compressors. During each revolution of the crankshaft, the piston shifts from one side to the other, several times, moving in the plane of the connecting rod motion. The gap between the piston and the cylinder liner permits the piston to move with a certain velocity in the transverse direction, impacting against the wall of the cylinder. These knocks produce an intense vibration of the cylinder walls at their resonant frequency.

This discussion is confined to some of the major noise sources in compressor systems, and does not include those attributed to motors, turbines, gears, or valves. It does not include foundation noise either, even though improper mounting on an inadequate foundation is often the cause of very high vibration and noise.

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