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# Compressor Noise Control in Applications

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This paper is concerned with current techniques of compressor sound control in refrigeration applications. The compressors referred to are the hermetic, reciprocating type, with direct drive, two pole, induction motors. The operating speed range of these compressors is about 3300 to 3500 RPM. This generates a fundamental frequency of 55 to 58 Hertz. Two cylinder versions would have a pumping frequency of twice that, or 110 to 116 Hertz. The compressors are designed with internal mufflers at the intake and discharge, with internal spring suspension between the crankcase and outer housing and with a coiled, flexible steel tube to direct the discharge gas to the outside of the housing. The space within the hermetic case contains the oil reservoir and the return refrigerant.

The hermetic refrigeration compressor is never operated alone. It is one of several components in a refrigeration system that usually includes one or two air moving blowers, a condenser, an evaporator and some sort of cabinet that contains and protects all or part of these components. Sound that is generated by the compressor may be radiated directly from the steel case (case radiation) or indirectly by exciting other components of the system (structure-borne noise).

The compressor case radiation is independent of the cabinet, or enclosure, in which it is mounted. This sound is measurable in the laboratory and the quantity is expressed in decibels, sound power level. The sound produced indirectly is not measurable by any test on the compressor alone. The excitation can be measured as vibration or gas pressure pulse but these can give only a very general indication of the sound that will be produced in an application. For this reason, the final determination of the quantity and quality of the sound produced by a compressor must be done in the system in which it will be operating.

The major paths by which compressor noise is transmitted to the system are: (1) The air surrounding the compressor (case radiation), (2) The refrigerant within the tubing (discharge pressure pulse), (3) The tubing (discharge and suction lines), and (4) The mounting feet (vibration).

The control of compressor noise in the application is generally limited to the treatment of these transmission paths. On occasion the source may benefit from damping applied to a localized housing resonance or the addition of oil to the reservoir may help the case radiation. These cases are exceptions to the rule, however, and do not have universal application. The following techniques are applicable in all cases where noise reduction is required.

## Reducing Case Radiation

The case radiation is an important factor in the sound of residential, central air conditioners. It is also important, and may be the major sound in a household refrigerator. It may be of significance in window air conditioners.

The sound waves generated at the surface of the compressor housing are radiated outward and (1) reflected about, within the walls of the compartment and eventually are radiated out through openings in the compartment, or (2) transmit some energy to the compartment panels (causing vibration of the panels), or (3) lose some energy in the form of heat due to absorption.

Figure 1 tells something about the nature of this case radiation. The sound of a three ton air conditioning compressor has been analyzed in octave, one-third octave and narrow (6%) bandwidths. Note from the narrow band analysis that compressor case radiation is primarily line spectra. In the frequency range below about 1000 Hertz, the harmonics are spaced far enough apart to be readily distinguished. Above that, the constant percentage bandwidth analyzer begins to loose resolution and only prominent tones can be recognized. The one-third octave and octave analyzers provide decreasing resolution.

The chart of figure 2 is a composite of eight compressor tests. Its purpose is to show the variation in sound spectra of a number of compressors. It is constructed by plotting the levels in each band of all the compressors included. The maximum and minimum levels in each band are connected to form an envelope within which all of the compressor spectra fall. The compressors were all tested at the same load conditions. A wide range of load conditions would produce a further scatter in the data. An indication of this is shown by the "X" plots in the 315 and 400 Hertz bands which indicate the levels attained as the compressor speed was lowered to near stall point.

Where reduction of compressor case radiation is needed, it must be done within the enclosure before the sound is radiated out of the cabinet and is beyond control. This is accomplished by absorption. Acoustical absorbing material, such as fiber glass or an open cell plastic foam, is applied to the compressor side of the enclosure panels. To be effective, the sound must be directed into the material repeatedly. This is done by enclosing the compressor as much as possible. Baffles may be placed between the compressor and cabinet openings to reflect a portion of the sound back into the enclosure.

The effectiveness of the material used for sound absorption varies with the construction of the material with respect to density, fiber or pore size, its thickness and the method of application. It is also dependent on the frequency and the angle of incidence of the sound to be absorbed. The supplier of acoustical materials rates the absorption value on a scale of 0 to 1.0 called the sound absorption coefficient. The value of these materials in small enclosures is subject to some limitations:

(1) It is not possible, in a small compartment, to obtain significant absorption of low frequency sound because of space limitations. For sound of low frequency (long wavelength) absorption relies heavily on the reactive effect of the space between the surface of the absorbing material and the reflecting surface in back of it. For this reason, the absorption coefficients of a material will show improvement when the depth of the mounting is in the range of 1/4 wavelength of the sound. The available space in most enclosures limits the depth of material to about one inch or less. The absorption coefficients fall off rapidly for frequencies below about 1000 Hertz.

(2) The absorption coefficients given for the material are obtained in a diffuse sound field. This condition is approximated in an enclosure that is large in relation to the wavelength of the sound and in which the total absorption is not large. In the usual compressor enclosure this may be true in the frequency range above about 1000 Hertz.

(3) The primary limitation on the use of absorbing material is the total area to which the material can be applied. The total absorption within the enclosure is the product of the effective absorption coefficient times its area.

In applications where 10% or more of the cabinet panel area is open for ventilation, the reduction in compressor case radiation is limited to about 3 to 5 dB(A). If greater reductions in case radiation are needed, it may be possible to completely enclose the compressor. If the enclosure is carefully sealed, the reduction in case radiation is only limited by the transmission loss of the enclosure and the total sound absorption within it. Figure 3 is an example of the reduction in case radiation that was obtained with a sealed enclosure. The higher sound spectrum was produced by a compressor operating in the open. The lower spectrum is the same compressor enclosed in a sheet metal box having a layer of one inch thick fiber glass covering five of the six sides in the enclosure. Significant sound reduction was obtained at frequencies well below 1000 Hertz.

A disadvantage to completely enclosing the compressor is the loss in performance that results. The ambient compressor temperature increases which increases the temperature of the return refrigerant in the housing. The increased refrigerant temperature entering the cylinder results in less mass flow of refrigerant without an equal decrease in power consumption. The performance loss in the E.E.R. may range from about 1% to as much as 5%. There will also be a loss in "high-load, low voltage" performance. It may be possible to recover the performance loss by a change in the refrigeration system. An increase in condenser surface can reduce the discharge pressure, for example. This entails an increase in cost, however.

The compressor motor protection will be affected by the higher compressor temperatures. The effect is minimized where internal motor protectors are used since the internal compressor temperature will not increase as much as the ambient temperature. Many compressors in the window air conditioning sizes (below two ton capacity) employ externally located motor protectors. The increase in protector temperatures may prevent the system from meeting its full performance specification. The tendency would be to "de-rate" the protector causing overprotection of the motor. In some cases, this can be handled with a change in the motor protector.

Reduction Of Discharge Pulse

The "discharge pulse" is a pressure fluctuation in the superheated refrigerant leaving the compressor. The pulse is actually an acoustic wave that is generated by the large, rapid pressure changes that occur when the discharge valve opens. The pressure pulse is confined to the refrigerant gas in the discharge piping and ordinarily does not contribute directly to the case radiation. Its contribution to noise in the refrigeration system normally occurs because pressure waves in the refrigerant excite the condenser tubing causing vibration of the tubing and the attached unit base and cabinet panels.

Small window air conditioners appear to be more susceptible to this problem than most other applications. This follows from the design of these units as they utilize a common cabinet to house both the condenser and evaporator sections of the system.

Vibration induced in the condenser tubing is readily conducted to the indoor portion of the unit where the inside blower is quieter and the compressor case radiation is at least partially blocked by the weather wall of the unit.

The speed of the acoustic noise in the discharge gas is about 600 ft. per second in R22 and about 500 ft. per second in R12. It varies with the temperature and pressure of the refrigerant. An important characteristic of the discharge noise is its harmonic content. Frequencies as high as 1000 Hertz have been observed. The amplitude of the pressure wave may range up to 5 psi or more, under some conditions.

Reduction of the pressure pulse is accomplished by inserting a muffler in the discharge line between the compressor and the condenser. The best location for the muffler is usually as close to the compressor outlet as possible to minimize the excitation of the discharge tubing as well as the condenser tubing.

The most practical muffler, for this purpose, is the single expansion chamber. It has the advantages of reasonable cost, availability and a relatively broad frequency range of attenuation. This is a "reactive" type muffler and the greatest attenuation occurs at the "tuned" frequency and at odd multiples thereof: 1f, 3f, etc. The attenuation at 1/2f and 1 1/2f is only 3dB less, making the muffler suitable for the full range of compressor load conditions. The length of the muffler can be found from $l = 3c/f$; where $l$ is the length in inches, $c$ is the speed of sound in the refrigerant and $f$ is the frequency at which maximum attenuation is desired. The maximum attenuation is a function of the cross-sectional area ratio (m) of the muffler to the discharge line. As a rule, a muffler diameter of 4 to 5 times the tube diameter is sufficient.

## Reduction Of Line Transmission

The discharge and suction lines are a possible path for conducting compressor noise to the system. The transmission path, in this case, is the wall of the tube. The applications that are most susceptible to this noise are household refrigerators and freezers where steel, rather than copper, tubing is likely to be used. The desired result of the tube design is flexibility.

In some window air conditioners, the limitation on space is a problem. In these cases, the tubing can be lengthened by providing a coil in the horizontal plane of at least 270° around the compressor.

## Reduction Of Mounting Vibration

The transmission of noise through the compressor mounting feet to the unit base is common to all applications. From the smallest household refrigerator to the largest air conditioner, some type of isolator is used to reduce this noise.

There are three types of vibration associated with the compressor:

(1) Unbalance (rotational, reciprocating).

(2) Torque reaction (load generated).

(3) Flexural case vibration (audible frequency range).

The first two forms of vibration occur at low frequency and appear primarily at the first two harmonics. The motion originates at the compressor crankcase and the internal spring suspension is in the transfer path to the housing. The internal suspension is usually adequate to reduce the vibration to acceptable levels. The third type of vibration occurs at higher frequencies. It is this vibration that generates the case radiation discussed earlier. The compressor internal spring suspension is ineffective here as much of this energy is transferred to the housing through the refrigerant, the discharge tube, and the oil reservoir.

Synthetic rubber isolators are applied to the compressor mounting feet and are effective in reducing the transmission of high frequency noise to the unit base. In many cases, these isolators are soft enough to offer some reduction in the low frequency vibration also.

The choice of rubber isolator is based upon the following criteria:

1. Transmissibility - The degree of isolation required.
2. Life - The rate of deterioration due to temperature and stress.
3. Shipability - The ability to restrain compressor motion during shipment.
4. Physical size, shape and ease of assembly.
5. Cost.

The transmission loss of an isolator is related to the static deflection of the isolator under the weight of the vibrating mass. The allowable static deflection of a rubber isolator is limited to about 10% of its free (unloaded) height.

While the rubber isolator is the choice on nearly all applications, coil springs may be used to achieve greater transmission loss at low frequencies.

## Determining the Transmission Path

In the event that a sound test discloses an objectionable noise in the refrigeration system, it remains to identify the source and the major path. If a sound analysis of the system is available, it may give an indication of the type of compressor noise involved. All compressor noise is at one or more harmonics of the pumping rate so that frequency alone is not definitive. However, the noise from various causes does tend to be limited to certain frequency ranges, as follows:

1. Case radiation: 200 to 10,000 Hertz

2. Discharge noise: 100 to 1,000 Hertz

3. Tubing: 55 to 1,000 Hertz

4. Mounting: 55 to 350 Hertz

The frequency ranges shown are rough indications only and assume that normal techniques of compressor installation are used. If the objectionable noise is in the frequency range above 1,000 Hertz, it is likely that case radiation is the cause. If below about 200 Hertz the compressor mounting is likely to be the cause. In the frequency range between these, some investigation is required to separate the possibilities.

The best procedure is to attempt to operate the compressor, in the system, at the load conditions at which the noise is present and with all other sound sources eliminated. In addition, it should be possible to remove all the structure-borne paths for noise transmission so that the effect of case radiation can be measured alone.

To operate the compressor, alone, an external load source is required. This does not need to be a refrigeration system with condenser and evaporator.

It is sufficient to keep the refrigerant in the gaseous state and use a throttling valve to obtain the required discharge and suction pressures. In addition to the throttling valve, some means of cooling the return refrigerant to the compressor is needed. If the compressor discharge and suction pressure, return gas temperature and voltage supply is the same as the original test conditions, the case radiation will be duplicated.

It is seldom possible, of course, to separate the compressor structure-borne noise paths from the system without compromising the sound radiation to some extent. The refrigerant piping must be brought out through the cabinet. If there are openings in the cabinet, these may be used. If not, openings must be cut and patched. The compressor mounting is never easily eliminated. It may be enough to change from a rubber to a soft, spring mount in some cases. In any case, ingenuity is often required, and interpretation of the test data must take into account the alterations made to the system.

After the compressor case radiation has been measured, the structure-borne paths may be added so that their effect can be evaluated. The procedure might be:

1. Case radiation only

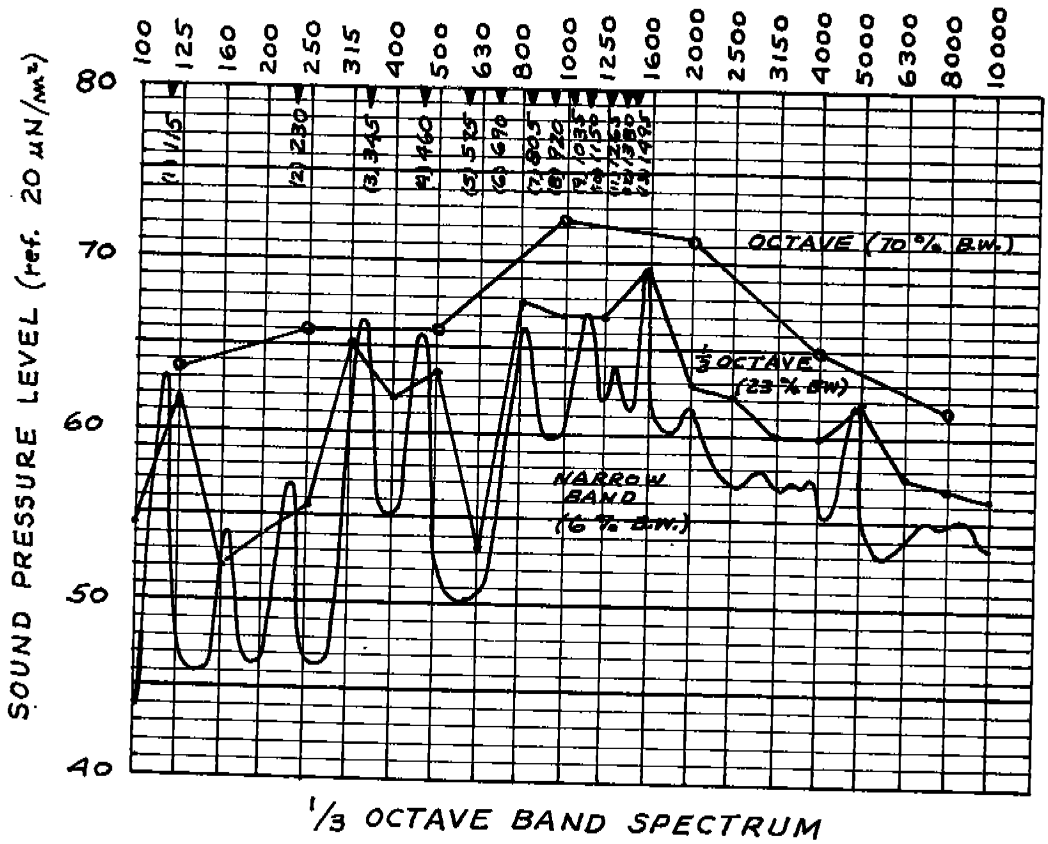
2. Case radiation and mounting

3. Case radiation, mounting and discharge.

4. Complete system.

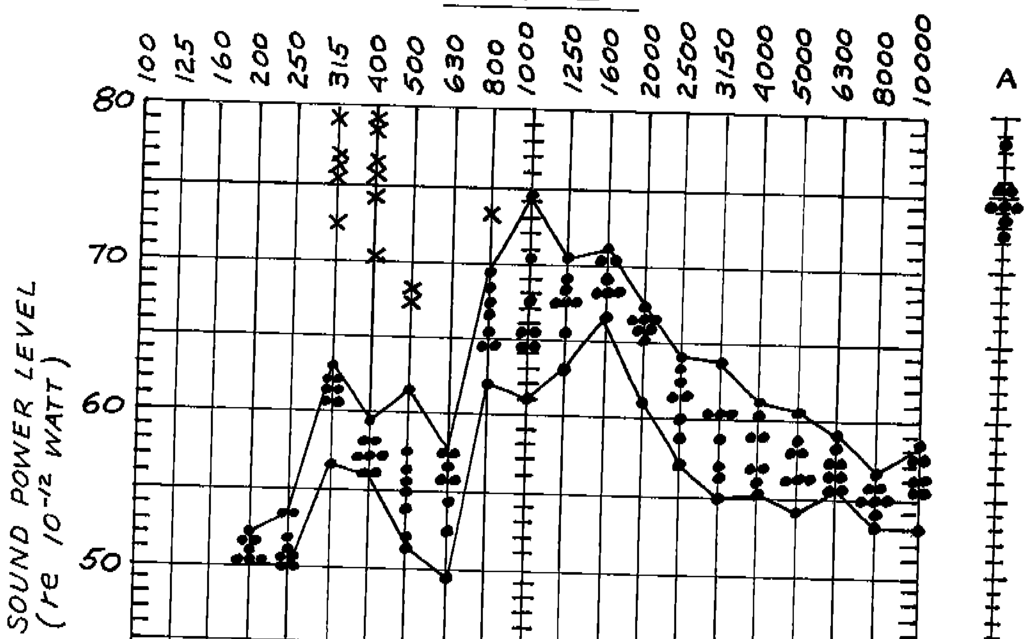
At any step in the program, the appropriate means of attenuation for that path may be inserted and evaluated before the complete system is re-tested.

FIG. 1



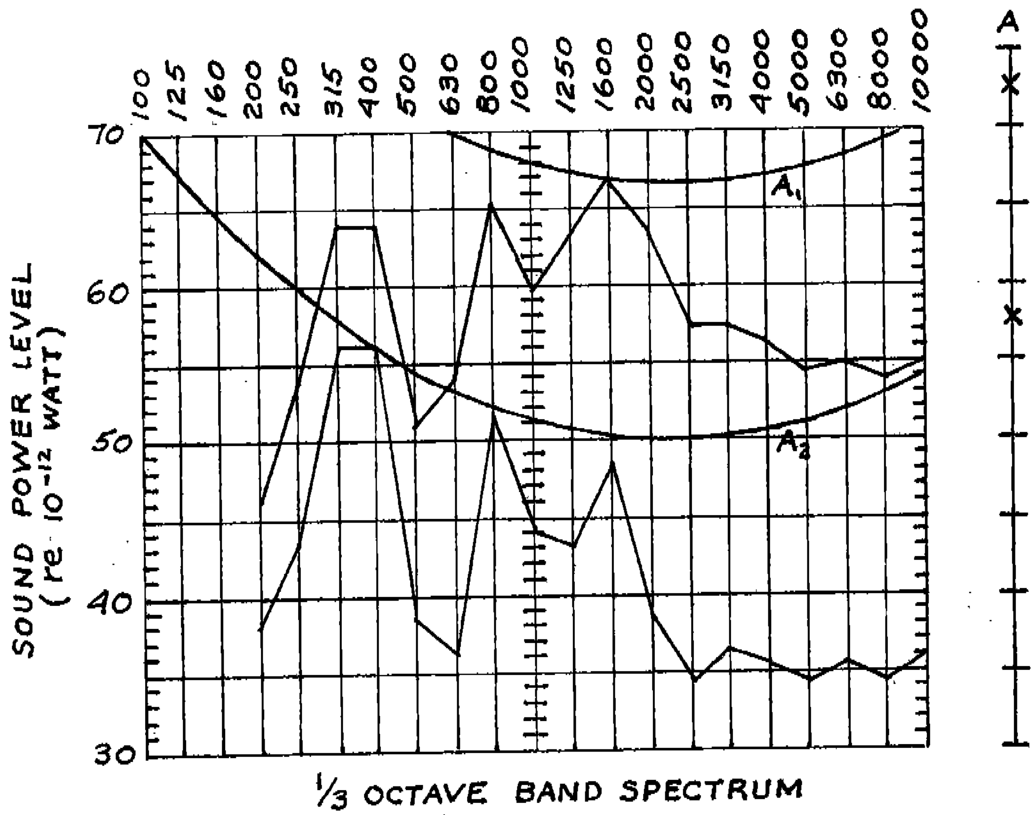
SOUND PRESSURE LEVEL (ref. 20 μN/m²)
40
50
60
70
80
100
125
160
200
250
315
400
500
630
800
1000
1250
1600
2000
2500
3150
4000
5000
6300
8000
10000
(1) 115
(2) 230
(3) 345
(4) 460
(5) 575
(6) 690
(7) 805
(8) 920
(9) 1035
(10) 1150
(11) 1265
(12) 1380
(13) 1495
OCTAVE (70% B.W.)
1/3 OCTAVE (23% B.W.)
NARROW BAND (6% B.W.)
1/3 OCTAVE BAND SPECTRUM


FIG. 2



SOUND POWER LEVEL (re 10⁻¹² WATT)
40
50
60
70
80
100
125
160
200
250
315
400
500
630
800
1000
1250
1600
2000
2500
3150
4000
5000
6300
8000
10000
A
1/3 OCTAVE BAND SPECTRUM

FIG. 3



100
125
160
200
250
315
400
500
630
800
1000
1250
1600
2000
2500
3150
4000
5000
6300
8000
10000
A
70
60
50
40
30
SOUND POWER LEVEL
(re 10^-12 WATT)
A1
A2
1/3 OCTAVE BAND SPECTRUM