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Noise Study of Fractional Horsepower, Rotary Vane, Refrigerant Compressors

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INTRODUCTION

Most pumps and compressors are common sources of noise today. Because of increased interest in reducing noise in refrigeration and air-conditioning apparatus, there exists a need to understand their sound production and transmission mechanisms to enable designers to develop quieter machines. This paper describes a series of experiments designed to identify the contribution to the total sound field of a fractional horsepower, rotary vane, refrigerant compressor produced by the time varying forces exerted on the internal parts of the compressor by the refrigerant during the compression process.

The problem of noise generation in small hermetic refrigerant compressors has received little attention in published literature. Nevertheless some attempts to explain the nature of compressor noise have been reported. Binder [1]* investigated noise generated by a fractional horsepower reciprocating compressor and concluded that noise transmission through the metallic path from the pump to the hermetic shell was stronger than the path through the contained gas. Lowery [2] concluded that the discharge valve and booster reed of a fractional horsepower reciprocating refrigerant compressor are significant vibration-noise sources. Webb [3] and Hanson [4] describe significant reductions in noise with a change in valve port locations. Ingalls [5] gives an overall description of the sources and transmission paths of hermetic reciprocating compressor noise.

Rotary vane compressor noise has not been examined the same extent that reciprocating compressor noise has. Rembold and Lo [6] reported on the development of a double-lobed rotary vane type compressor in which the sound levels showed a pattern of increasing with frequency.

This paper describes more fully the reasons why the discharge valve and valve port parameters detailed by Lowery, Webb and Hanson contribute so significantly to compressor noise.

ROTARY VANE COMPRESSOR NOISE

Figure 1 shows a cutaway view of the rotary vane compressor. The hermetic shell housing the pump is the principal radiator of noise from the typical compressor. Vibration excitation of the housing takes place in two ways: a distributed excitation consisting of the gas pressures acting over the inner surface of the housing and point excitations at those locations where the spring suspended pumping mechanism makes contact with the cylindrical housing. These contact points are the points where the mounting springs attach to the shell structure and where the suction and discharge tubes exit.

Compressor sound measurements were made in an anechoic chamber whose inside dimensions are a 12 feet cube having a cut off frequency of 100 Hz. A one inch Bruel and Kjaer condenser microphone was employed at a distance of 18 inches from the compressor shell. Narrow band instrumentation included a Bruel and Kjaer microphone amplifier a Spectral Dynamics Dynamic Analyzer, a Spectral Dynamics Sweep Oscillator, a Hewlett-Packard Log Converter and an Electro Instruments X-Y Recorder. A Bruel and Kjaer pistonphone was used for calibration. Sufficiently slow filter sweep rates were employed to insure an accurate reproduction of the spectra. A more complete description of the experimental setup as well as an analysis of the repeatability of compressor sound spectra are given in reference [7].

*Numbers in brackets refer to listings in the bibliography.
The compressor sound spectrum from 20 Hz to 5 KHz shown in Figure 2, in addition to the broadband noise, consists of a large number of pure tone components which constitute: harmonics of the pumping frequency (115 to 118 Hz); the lower harmonics of twice the electrical frequency (120 Hz); and to a lesser extent harmonics of the rotational frequency (57.5 to 59 Hz). A comparison with the narrowband background spectrum shows that all peaks above 10 dB are attributed to the compressor. Those harmonics having the highest sound pressure amplitudes (1,800 to 2,400 Hz) are examined by comparing changes in the level of the sound output of a particular harmonics with changes in the suspected source. If the changes can be related and explained in terms of acoustic and vibration theory, the source will be identified.

The source of compressor shell excitation due to structure borne noise carried by the mounting springs and inlet and discharge tubes is stator vibration.

The forces acting on the rotor and stator (front head) enumerated in the simplified system diagram of Figure 3 consist primarily of centrifugal forces due to rotor unbalance, magnetic field forces from the motor, and the forces due to the gas pressure acting on the rotor and stator respectively.

Centrifugal force due to rotor unbalance is very nearly sinusoidal and is therefore unlikely to have large harmonics above the rotational frequency. The magnetic forces create noise at the fundamental frequency of 120 Hz and its higher harmonics.

Another force arises from unbalanced pressure forces within the discharge muffler due to pressure wave effects and also due to the impact effect of the sudden closing of the discharge valve when the rotor vane uncovers the discharge port. Photographs of oscilloscope traces of valve motion show that valve excursions upon closing never exceed .002 inches so valve closure impact force is assumed to be negligible.

Two internal forces act on the stator: gas pressure exerted on the cylinder projected area and the journal bearing film pressure force. Figure 4 is a simplified free-body diagram of the rotor and stator assemblies illustrating forces tending to cause angular motions about axes perpendicular to the shaft centerline. Torques about the shaft axis due to gas pressures are an order of magnitude smaller than these because of the smaller areas exposed to the gas and can be neglected. The cylinder pressure force can be characterized as either a force whose magnitude and direction change with time or fixed orthogonal components whose amplitudes are time varying. The bearing pressure force magnitude, direction and location along the bearing are time functions. The total force system is resolved into a force acting through the center of gravity and a couple. This force and couple produce stator vibrations which couple to the shell through the mounting springs and the suction and discharge tubes. In addition, stator motions disturb the gas surrounding the stator creating a distributed shell forcing function.

**STATOR ACCELERATION AND SOUND SPECTRA COMPARISON**

Sound spectra of the compressor exhibits an unexplained, but repeatable sensitivity to suction pressure at a frequency slightly above 2 KHz. This sensitivity consists of a phenomenon not unlike resonance in that while the suction pressure is varied from 2.5 psig to 7.5 psig the 1.8 to 2.4 KHz sound pressure levels increase by as much as 15 dB. At its highest amplitude, the 1.8 to 2.4 KHz frequency range is by far the most annoying sound produced by the compressor. The suction pressure sensitivity is used to trace the path of the sound to the source.

Vertical stator acceleration measurements 90° apart (see Figure 4) were made to determine if stator motion and radiated sound were related. Accelerometer locations were selected on the following bases:

1. No significant force acts in the axial direction hence the accelerometer positions selected measure the angular accelerations produced by an applied couple.

2. Since the forcing function consists of a force perpendicular to the axis through the center of gravity and a couple, measurement of transverse accelerations would have to be made in a horizontal plane through the center of gravity to exclude rocking accelerations. This location was not accessible without major modification to the housing.

A measured acceleration spectrum is shown in Figure 5. Practically all harmonics of the pumping frequency up to and including the 43rd are clearly identifiable. Of the frequency ranges 600-900 Hz, 1,500-2,500 Hz, and 3,100-3,700 Hz which rise significantly above surrounding levels the 1,500-2,500 Hz range is of greatest importance because this is the frequency range where sensitivity to suction pressure is great, sound levels are high, and human hearing sensitivity is strong. In order to concentrate on this problem, the frequency range from 1,500 Hz to 3,000 Hz is examined more closely.
Figure 6 shows the effect of suction pressure on the two acceleration measurements and the sound pressure level averaged at four perpendicular microphone locations for the 18th through the 21st harmonic of the pumping frequency. Other harmonics show similar trends. More significantly, however, the variations (especially pronounced in the 18th and 20th harmonics) show that in this frequency range there is a close agreement between the change in acceleration and the change in the averaged sound pressure level. This agreement substantiates the hypothesis that stator vibrations are the largest contributor to sound in this frequency range.

**Cylinder Pressure**

Figure 7 is a typical trace of cylinder pressure and is representative of one-half rotor revolution. Starting at the left, the cylinder pressure increases by compression until the valve opens at the peak in the pressure curve. As the gas flows through the open discharge valve, the cylinder pressure drops until a steadily decreasing flow takes place because of the decreasing swept volume as the vane approaches the transfer slot. Pressure oscillations are seen following valve opening. These oscillations take place in the compression chamber as a result of the sudden decrease in pressure across one of the chamber's boundaries sending an expansion wave down the length of the chamber. As the rotor continues to turn, the sudden sharp drop in pressure occurs when a vane passes the pressure transducer tap exposing it to the following chamber.

At the point where a vane reaches the transfer slot, a sudden burst of gas into the next chamber causes conditions which are not unlike those existing in a closed shock tube. After the oscillations dampen sufficiently they become nearly sinusoidal and increase in frequency as a result of the shortening of the chamber length as the rotor turns.

**Rotord-Cylinder Force**

The rotor-cylinder force is calculated by integrating the cylinder pressures over the whole cylinder area. An accurate prediction of rotor-cylinder force requires precise cylinder pressures as a function of time and space. The pressure during compression is easily calculated, but the pressure fluctuations, believed to contribute highly to the 1,500-2,500 Hz sound levels, are difficult to analyze.

The horizontal component of the rotor-cylinder force can be expressed as:

\[ F_h(t) = W_c R_c \int_{\theta=0}^{\theta=2\pi} P_c(\theta,t) \sin 4\theta d\theta \]

where:

- \( F_h(t) \) = Horizontal component of rotor-cylinder force.
- \( W_c \) = Cylinder width.
- \( R_c \) = Cylinder radius.
- \( \Theta \) = Cylinder angle.
- \( P_c(\theta,t) \) = Cylinder pressure.

An accurate analytical prediction of is rather difficult due to effects such as interaction of the leaf type check valve and the fluid and moving boundaries.

A first approximation to the cylinder force time history is obtained by measuring compression chamber pressure at three locations and converting to a force by multiplying by the appropriate areas assuming uniform pressure.

The assumptions made in computing the cylinder force are:

1. The rotor turns at constant speed, i.e. no speed changes due to loading occur during the compression cycle.
2. The vanes have no thickness.
3. No pressure drop occurs at the suction ports.
4. No leakage occurs from the high pressure side of the cylinder to the low pressure side.

Pressure tap locations and uniform pressure area domains are delineated in Figure 8.

The general scheme used to calculate rotor cylinder force consist of applying \( P_1 \) over the area bounded by the angle \( \alpha_1 \); \( P_2 \) over \( \alpha_2 \); \( P_3 \) over \( \alpha_3 \), suction pressure, over \( \alpha_0 \). Some alteration of this plan is necessary when a vane passed over the pressure taps, however.

After the horizontal, vertical and total force magnitudes are computed as a function of rotor angle (or time), the Fourier coefficients \( C_n \) are computed using a digital computer. These coefficients represent the amplitudes of harmonic functions, which, when combined using the proper phase relationships, represent the periodic forcing function as follows:

\[ F(t) = \sum_{n=-\infty}^{\infty} C_n \sin\omega t \]

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An orthogonal transformation is used to determine the force components at a counterclockwise angle from the horizontal and vertical:

\[
\begin{align*}
F_1 &= \begin{bmatrix} \cos \gamma & \sin \gamma \end{bmatrix} F_x \\
F_2 &= \begin{bmatrix} -\sin \gamma & \cos \gamma \end{bmatrix} F_y
\end{align*}
\]

The choice of \( \gamma \) depends upon the coordinate system used to set up the dynamic equations. The purpose of this study, however, is to identify the forcing function and make recommendations for the reduction of its amplitude. To accomplish this, the magnitudes of the amplitudes of the harmonics as given by the Fourier Series of the forcing functions \( |cn| \) are computed for \( F_1, F_1, F_2 \) for \( \gamma = -30^\circ \) degrees. This angle is selected because the \( F_2 \) direction lies directly between the areas exposed to \( P_1 \) and \( P_2 \) where the pressure oscillations are the highest magnitude, to intensify the amplitudes of the higher frequency harmonics.

THE EFFECT OF CYLINDER MODIFICATIONS

The computer forcing functions and their line spectra in the frequency domain are of little use unless the system transfer function is known. However, a comparison of changes in output with input at the same frequency can often be helpful in identifying a vibration or sound source. Computed force and measured acceleration spectra are compared before and after a modification designed to significantly alter the high frequency harmonic content in order to relate the forcing functions \( F_1, F_1, F_2 \) with stator accelerations in the frequency range 1.5 to 3 KHz. The modification consists of an increase in plenum volume (Figure 8) designed to reduce cylinder pressure fluctuations and a groove to permit a more gradual release of plenum gas when the vane passes the transfer slot. The volume increase causes the plenum pressure to change more gradually as the discharge valve opens and thereby reduces the high frequency excitation of the compression chamber gas.

Figure 9 is a comparison of the three cylinder pressures before and after modification. Lower frequency oscillations can be seen in the clearance volume pressure (trace \( P_2 \)) just after the valve opens. In addition, the duration of the over-compression peak is longer, evident in the \( P_1 \) trace of the modified cylinder due to the enlarged plenum volume. Another characteristic of the pressure traces of the modified design is the reduction almost to the point of disappearance of the pressure oscillations within the cylinder measured by \( P_2 \) due to the gradual release of the plenum gas. The dimensions selected for the modification were arrived at on the basis of the size of existing volumes and port dimensions, and are by no means representative of the optimum design for noise reduction of the compressor.

The modification produces two noticeable effects on the periodic force shown in Figure 10. First, the force oscillations appear to be attenuated and, second, the force peak for the modified cylinder is not quite as sharp as for the standard design. In general, there is a rounding of the sharper areas.

The results of computing the force harmonics for force \( F_1 \) are shown in Figure 11. Since the calculated periodic forces are only approximations to the true periodic force because of the averaging of the angular variations in pressure in the computations, only the following observations are made in regard to the harmonic content:

1. The first ten harmonics are virtually unchanged.

2. There is a reduction in the amplitudes of the harmonics from the tenth on up.

Acceleration spectra were recorded with and without a modified cylinder at suction pressures ranging from -5 psig to +20 psig and discharge pressure of 180 psig. A typical result is shown in Figure 12 at a suction pressure of 7.5 psig. Careful examination of both spectra reveals that no reduction is evident in the first ten harmonics while the higher harmonics show a decrease all the way to the 43rd harmonic. This is the same result obtained by computing the harmonics of the gas pressure forces acting on the pump rotor and cylinder.

Stator acceleration and sound pressure harmonic amplitudes in the 1.5 KHz to 3KHz frequency range reveal a strong agreement. It is expected that the sound levels would be lower. Of particular interest is the 18th and 20th harmonics because their effect on the overall compressor sound level is more significant than all others. Comparisons of 1.5 to 3.0 KHz sound traces for a suction pressure of 7.5 psig show that the cylinder modification results in a significant reduction of the sound level as revealed in Figure 13.

Substantial improvement can be made in reducing noise levels of rotating (and reciprocating) machinery whose sound spectra consists of a series of harmonics. The problem reduces to one of identifying the most prominent periodic forcing function and studying the character of the spectrum to determine ways of reducing all harmonic amplitudes by making the forcing function approach a true sine wave. No significant reduction in overall levels would be achieved by the elimination of
every other harmonic, for example. Hence, the only sure method is to smooth the function enough to lower amplitudes over a wide frequency band.

CONCLUSIONS

Significant reductions of amplitude of certain harmonics in the rotary vane compressor sound spectrum were achieved by modifying the compressor to reduce the pressure fluctuations in the rotor cylinder area during the compression process. A similar reduction in amplitude of these harmonics in the computed pressure force spectrum was noted. It is concluded, therefore, that a major contributor to rotary vane compressor noise is the cyclic cylinder pressure acting on the rotor and cylinder during compression.

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Fig. 1 Fractional horsepower rotary vane compressor

Fig. 2 Five-Hz bandwidth compressor sound spectrum

Fig. 3 System diagram

Fig. 4 Free body diagram of rotor and stator
Fig. 5 Typical 20-Hz bandwidth stator acceleration

Fig. 6 Variation in harmonics of sound pressure level averaged at four locations with suction pressure: standard configuration

Fig. 7 Typical trace of cylinder pressure — $P_2$

Fig. 8 Rotor-cylinder geometry showing pressure transducer locations
Fig. 9  Standard and modified cylinder pressures

Fig. 10  Rotor forces — standard and modified cylinders

Fig. 11  Computed rotor force harmonics of standard and modified cylinders

Fig. 12  20-Hz front head acceleration spectra
Fig. 13 20-Hz sound spectra before and after modification