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On Noise Generation of Air Compressor Automatic Reed Valves

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INTRODUCTION

One of the important noise sources of a single stage air compressor is the suction valve. In the following, some experimental results are presented for a 0.5 hp, two cylinder air compressor with automatic reed valves. This may be of some interest to the compressor designer.

Earlier investigators [1] showed that the location of circular orifices relative to beam type reed valves is important from the viewpoint of vibration response and dynamic stresses. Valve port locations, sizes and shapes were varied as allowed by basic geometrical design constraints. The plate type reed valves were left unchanged. The noise reductions were achieved without streamlining the port holes.

EXPERIMENTAL ARRANGEMENT

The experimental arrangement used is shown in Figure 1. The two cylinder, single stage air compressor was suspended from the ceiling of Herrick Laboratories' anechoic chamber. During the tests the suction valve remained opened. A one inch Bruel & Kjaer condenser microphone was positioned four feet from the exposed suction valve. Sound pressure level measurements were taken in one third octave bands using a Bruel & Kjaer spectrum analyzer. The exhaust port was sealed and discharge line was attached to the check valve of the air compressor and positioned outside of the anechoic chamber. A Hoke needle valve and an orifice plate were positioned for capacity measurements. Pressure measurements were taken using two 150 psig Heise pressure gauges. The first was positioned at the check valve and the second was positioned upstream of the orifice plate.

SUCTION AND DISCHARGE VALVE SYSTEM CONFIGURATIONS

The suction and discharge valve system of the air compressor consisted of a stainless steel ring plate of high tensile strength riveted to a valve assembly block. During the compression and expansion cycle air is forced through a set of ports located in the valve assembly block and directed against the valve element. The location, size and configuration of these ports are critical from a noise and performance point of view. In addition to the geometry of the ports, the valves' overlay area is also an important factor. A large number of port configurations were investigated with the valve element unchanged. Some of these configurations are shown in Figures 2 and 3 which consist of circular and slotted ports. The original suction valve port configuration, as shown in Figure 3, consisted of six circular holes, three of which were symmetrically aligned about a centerline intersecting the rivet hole location. Four circular holes were used for the exhaust valve system. Flow areas of 0.5, 0.75, 1, and 1.5 times the flow area of the original valve were considered for the slotted and circular designs. All ports were of the sharp square edge type.

The purpose of investigating the different port configurations was to determine a configuration which would yield a suitable reduction in noise for flow performance equal to, or greater than, that of the original valve system.

FLOW MODULATION NOISE

One type of noise radiation in an air compressor is caused by the dynamics of the valve element and the flow. In certain valve designs a fluid elastic interaction may occur which will cause the flow and...
valve element to oscillate during the suction and discharge cycle. The feedback of energy associated with this type of valve motion depends largely on the flow which may remove or add energy to the oscillating valve. This oscillation occurs at a frequency close to the mechanical valve system fundamental natural frequency. In most air compressor designs, this frequency of oscillation is different from the fundamental pumping frequency of the air compressor. One possible mechanism for flow modulated noise is the transition from attached to separated flow which occurs between the valve and seat. The time duration of this transition is important insofar as it influences the rise time of the valve from the seat. Stiffness and mass of the mechanical valve and port configuration are all interrelated.

MEASUREMENT OF COMPRESSOR CAPACITY & SPL

The Hoke needle valve was adjusted such that a mean pressure of 75 psig was maintained at the check valve in the discharge plenum. Mean pressure readings were taken upstream of the orifice plate and used as an indicator of flow performance. Third octave band spectrums were taken at the valve motion, thus causing high peak-to-peak first modes valve bending oscillations. During the discharge cycle the cavity acts to store energy therefore delaying the opening of the discharge valve. While all this is occurring the suction valve remains opened over a longer period of time.

The result of the measurements can be summarized in a plot of the "A-Scale" sound pressure level as a function of the mean pressure upstream of the orifice. See Figure 4. As suspected, the overall sound pressure level increased when the mean upstream orifice pressure was increased. At higher mean flows the overall sound pressure level tends to be proportional to the dynamic pressure. From the measurements taken there appears to exist a lower bound to the possible noise reduction which can be achieved.

Figures 5 and 6 show noise spectrums for valve port configurations exhibiting both high and low flow performance. In the high flow performance region the noise seems to be dominated by flow modulation. This type of noise is illustrated in Figure 5 by the high sound pressure level peak at approximately 500 Hz which corresponds to the fundamental bending frequency of the ring valve. In the low flow performance region, Figure 6, modulated flow noise is dominated by other sources of sound radiation in the high frequency bands. These higher frequency noise sources are suspected to be caused by turbulence and structural resonance.

Two valve port configurations are shown in Figure 7 which exhibit approximately the same flow performance. In the first configuration some material was removed from the original discharge assembly block resulting in a small cavity adjacent to the circular ports. The suction valve remained unchanged. The second configuration consisted of two slotted ports in the assembly block. Sound pressure level spectrums are shown in Figures 5 and 6 respectively for both configurations. Comparing the sound spectrum in Figures 5 and 6, a 10 db overall noise reduction was achieved for the second configuration. Likewise a 13 db noise reduction was achieved at the valve-flow modulated frequency. High sound output of the first configuration is possibly attributable to the enlarged cavity in the discharge valve assembly. As the air begins to flow through the suction valve some of the flow energy is transferred to the cavity causing the flow rate to increase. The rate of energy flow from the cavity depends on the compressibility, cavity size and the velocity of the piston. The initial rise time of the suction valve is therefore lowered and the transition to separated flow occurs over a longer fraction of piston motion. This situation also further contributes to an unstable valve motion, thus causing high peak-to-peak first modes valve bending oscillations. During the discharge cycle the cavity acts to store energy therefore delaying the opening of the discharge valve. While all this is occurring the suction valve remains opened over a longer period of time.

Each oscillatory peak of valve motion can be thought of as a discrete source of acoustical energy. For the first configuration shown in Figure 1 high noise radiation resulted due to the longer opening time and larger peak-to-peak valve oscillatory amplitudes. The slot location and the small overlay area of the valve and seat seemed to minimize the radiated noise.

EFFECT OF PORT GEOMETRY UPON SOUND OUTPUT AND FLOW PERFORMANCE

The effect of different ports is illustrated by the sound pressure level spectrums shown in Figures 8 and 9. Two port configurations were considered. The first consists of slotted ports which are indicated by B in Figure 2. Circular ports were considered for the second configuration. The arrangement of the circular port configuration for the suction valve system were quite similar to that shown in Figure 3. The two ports located furthest from the rivets were blocked off. Two circular ports were then placed between the first set of ports and rivets. The circular discharge port geometry was left unchanged. A 0.003 inch stainless steel ring valve was used in both designs. The
third octave noise spectrum is nearly flat when the slotted ports were used with an overall sound pressure level of 71 db as opposed to 76 db for the circular hole configuration. Pressures measured upstream of the orifice were similar for both designs, thus indicating similar flow performance.

Certain circular ports were eliminated in the suction valve assembly block and acoustic measurements were taken. The discharge valve system was left unchanged. Three configurations were considered. In the first the circular ports next to the rivet holes were blocked off. The middle and remaining two ports were blocked off in the second and third configuration. When the two circular ports furthest from the rivets were blocked off, low flow performance was observed. The noise spectrum is shown in Figure 6. This configuration also showed a lower third octave sound pressure level at the natural frequency of the mechanical valve system. However, higher frequency excitation predominated resulting in a higher "A-Scale" reading. When the circular ports adjacent to the rivet holes were blocked off valve dynamics began to dominate the noise. Thus a higher "A-Scale" sound pressure level and flow performance were observed. The best port configuration which seems to trade off flow performance for noise reduction consists of eliminating the middle port. The measured noise spectra is shown in Figure 10. The measured "A-Scale" sound pressure level did not differ significantly from the "A-Scale" sound pressure level measured for the original port configuration and good flow performance was achieved.

**EFFECT OF DISCHARGE PRESSURE**

One third octave band spectrum measurements were taken showing the effect of discharge cavity pressure upon air compressor noise radiation. The 0.005 inch stainless steel multiport ring valve system shown in Figure 2 was used. Two noise spectra are shown in Figures 11 and 12, for mean discharge cavity pressure of 20 and 80 psig respectively. Comparing both sound spectra, the sound pressure levels at approximately 60 Hz decreased as the discharge cavity pressure was increased.

**SUMMARY**

The predominant source of air compressor noise investigated here was caused by the vibratory motion of the valve coupled with air flow. Means of noise reduction was explored by considering changes in the design of the air compressor valve system. From the experimental study conducted here, significant reduction in air compressor noise can be achieved with appropriate valve port designs. Noise due to valve vibration becomes more significant with increasing mass flow.

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**REFERENCES**

AIR COMPRQIIOIII
ANECHOIC CHAMER
1/3 OCTAYIE
NOISE
ANALYUR

FIGURE 1 EXPERIMENTAL ARRANGEMENT

FIGURE 2 SLOTTED PORTS

FIGURE 3 CIRCULAR PORTS

FIGURE 4 OVERALL SOUND PRESSURE LEVEL VS UPSTREAM ORIFICE PRESSURE

FIGURE 5 HIGH PERFORMANCE VALVE

FIGURE 6 LOW FLOW PERFORMANCE VALVE
PORT CONFIGURATION 1

FIGURE 7

PORT CONFIGURATION 2

FIGURE 8 SLOTTED PORTS

FIGURE 9 CIRCULAR PORTS

FIGURE 10 MIDDLE PORT BLOCKED

FIGURE 11 MULTIPORT RING VALVE

FIGURE 12 MULTIPORT RING VALVE