ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

NOISE AND VIBRATION IN RECIPROCATING AND ROTARY COMPRESSORS - A REVIEW

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Abstract

Compressors are in wide use throughout the world in household appliances, air-conditioning systems in buildings, vehicles and industry. There are a large number of quite different designs. The compressor design adopted for each application depends upon various factors, including the gas or working fluid which must be compressed, and the discharge pressure and flow rates that need to be achieved. Compressors can be classified into two main types: 1) positive displacement compressors including reciprocating piston, and rotary types, and 2) dynamic compressors including axial and centrifugal types. The noise generated by the piston type depends upon several factors, the most important being the reciprocating frequency and multiples, number of pistons, valve dynamics, and acoustical and structural resonances. The noise produced by the rotary types depends upon rotational frequency and multiples, numbers of rotating elements, flow capacity and other flow factors. This paper provides descriptions of the operation of the main types of positive displacement compressor in use and some examples of how noise and vibration problems have been overcome in practice.

INTRODUCTION

Compressors are usually classified as either (1) *positive displacement* or (2) *dynamic* machinery types. Positive displacement compressors work on the principle of trapping a volume of gas and then, through the mechanical action of the machinery, reducing its volume and thus increasing its pressure. Dynamic compressors, on the other hand, work on the principle of using bladed impellors on continuously flowing gas to increase its kinetic energy, which is eventually converted into potential energy and gas of higher pressure. Positive displacement compressors can be further subdivided into 1) reciprocating types: piston, diaphragm or membrane, and 2) rotary types: screw, vane, or lobe. Positive displacement compressors are normally used for small flow rate capacity requirements such as in household refrigerators or room air conditioners. For higher flow rate capacities, valve and seal leakage, mechanical friction, and flow effects quickly decrease the efficiency of positive displacement compressors. This paper will concentrate on the noise of positive displacement compressors.

POSITIVE DISPLACEMENT COMPRESSORS

Reciprocating Piston Compressors

The reciprocating compressor was the first type designed for commercial use. It sees service in a wide variety of industrial and household applications such as refrigerators and heat pumps and it remains the most versatile compressor design. It can operate economically to produce very small pressure changes in the deep vacuum range up to very high pressures. The operation of a reciprocating piston compressor is in many ways similar to that of an internal combustion engine, although the design of such small compressors is simpler. The mechanical system of a typical small refrigerator compressor is comprised of an electric motor driving a reciprocating piston pump mechanism. Two thin metal “reed” valves are provided. As the piston moves to compress the working gas during the compression stroke, the suction valve closes and the discharge valve opens. After the piston has reached top dead center, and it begins the suction stroke, the suction valve opens and the discharge valve closes.

Diaphragm Compressors

Diaphragm compressors are a form of piston compressor. The diaphragm separates the gas undergoing compression on one side from the hydraulic working fluid on the other side. A piston is provided to force the hydraulic fluid upwards; it is commonly driven by an electric motor via a connecting rod, which is eccentrically connected to the motor drive shaft. As the piston moves up, it displaces the incompressible hydraulic fluid upwards making the diaphragm move up also. The membrane is sandwiched between two perforated metal plates which allow hydraulic fluid to flow through the perforations in the lower plate and gas to flow through the perforations in the upper plate. When the piston is at top dead center, the diaphragm is pressed hard against the underside of the top plate by the hydraulic fluid and the discharge valve has already opened, but is ready to start closing. On the piston down stroke, the diaphragm is drawn downwards thus allowing the intake valve to open and a fresh charge of gas to enter above the diaphragm ready to be compressed on the next upward stroke of the piston.

Screw Compressors

Screw compressors are formed by the intermeshing action of two helical rotors. The rotors are comprised of two types: male and female. The male rotors have convex lobes and the female rotors have convex flutes. The gas to be compressed enters through the inlet port and is trapped by the rotors which continually reduce the volume available to the gas until it is expelled through the discharge port. A typical screw compressor has four lobes on the male rotor and six flutes on the female rotor. In such an arrangement the compressor has six compression cycles during each revolution of the female rotor, which is operated at two thirds of the male rotor speed. Screw compressors have the advantages that they are 1) lighter and more compact than reciprocating compressors, 2) that they do not have reciprocating masses requiring expensive vibration isolation, and 3) that the rotors can be operated in a dry condition without the need for oil lubrication. Their main disadvantages include the fact that they have rapid rotor wear when operated in a dry state and that they are inherently very noisy. Oil is sometimes used as a lubricant to reduce wear. The use of water as a lubricant is under development.

Lobe or “Roots” Compressors

One of the oldest and simplest designs of compressor is known as a straight lobe or “Roots” compressor. This type of compressor normally employs two identical cast iron rotors. Each rotor has a figure eight shape with two rounded lobes. As the rotors turn they sweep the gas into a constant volume between the rotors and the compressor case wall. Compression takes places as the discharge port becomes uncovered. Initially backflow occurs from the discharge line into the casing cavity, until the cavity pressure reaches the compressed gas pressure. The gas flow then reverses direction and further rotation of the rotors causes increasing gas pressure with a reducing gas volume as the gas is then swept into the discharge line. Lobe or “Roots” compressors have the advantage of being low cost and needing low maintenance. They have the disadvantage that they are 1) less efficient than screw or centrifugal compressors, 2) they only achieve low pressure increases, and 3) they are inherently noisy because of the high frequency flow reversal which occurs at the discharge port.

Sliding Vane Compressors

The sliding vane compressor consists of a rotor mounted in an eccentric casing. Non-metallic sliding vanes are fitted to the rotor in slots. The vanes are held in contact with the casing by centrifugal force. The gas is taken in from the suction inlet and discharged through the port. The gas is trapped and sucked into volumes which increase with vane rotation up to top dead center. The trapped gas is then compressed as the trapped gas volume continually decreases after top dead center (TDC). There are no inlet and discharge valves. The times at which the inlet and discharge ports are open are determined by the time when the vanes are located over the ports. The inlet port is designed to admit gas until the gas "pocket" between the two vanes is largest. The port closes when the second vane of that pocket passes the inlet port. The gas pocket volume decreases until the vanes have passed TDC. Compression of the gas continues until the discharge port opens when the leading vane of the pocket passes over the discharge port opening. The discharge port closes when the second valve passes the end of the port.

Rolling Piston Compressors

Rolling piston rotary compressors are widely used because they are small in size, lightweight and efficient. Small rolling piston rotary compressors are often driven by electric motors. The rolling piston is contained in a cylinder and the piston is connected to a crankshaft eccentrically mounted to the drive shaft of the motor. The stator of the electric motor is normally fixed to the interior of a hermetic shell. A spring mounted sliding vane is provided. As the piston rotates inside the cylinder, the volume of gas trapped ahead of the piston, between the piston, cylinder and vane is reduced and the gas is expelled through the discharge. Simultaneously gas is sucked into the increasing volume following the piston. After the piston has passed top dead center and the inlet, the volume of trapped gas ahead of the piston is decreased again as the piston moves further towards the discharge valve and the compression cycle is repeated.

Orbital Compressors

So-called orbital compressors have many good characteristics such as high efficiency, good reliability and low noise and vibration. A common type of orbital compressor is the scroll compressor which uses two interlocking, spiral-shaped scroll members to compress refrigerant vapor. Such compressors are in common use in residential and industrial buildings for air-conditioning and heat-pump systems and also for automotive air-conditioners. They have high efficiency and low noise, but have poor performance if operated at low suction pressures. They also need good lubrication. Scroll compressors normally have a pair of matched interlocking parts, one of which is held fixed and the other made to perform an orbital path. Contact between the two scrolls happens along the flanks of the scrolls and in the process a pocket of gas is trapped and progressively reduced in volume during the rotary motion until it is expelled through the discharge port. Most scroll compressors are hermetically sealed inside a shell casing. The so-called trochoidal type is another type of orbital compressor. The Wankel design has a three-sided epitrochoidal piston with a two-envelope cylinder casing.

NOISE CONTROL OF POSITIVE DISPLACEMENT COMPRESSORS

Noise Control of Small Reciprocating Piston Compressor

Webb was one of the first to write about noise control of small reciprocating refrigeration compressors in 1957.[1] Since then many other authors have discussed their noise and vibration sources and methods of noise control. All of the main sources of noise in a small reciprocating compressor originate from the compression process. The sources include: 1) gas flow pulsations through the inlet and discharge valves and pipes, 2) gas flow fluctuations in the shell cavity, which excite the cavity and shell modes, 3) turbulent eddy formation in the shell cavity and inlet and exhaust pipes, 4) vibrations caused by the mechanical system rotation of the drive shaft and out-of balance reciprocating motion of the piston, connecting rod, and 5) impulsive motion of the valves and impacts they cause. Electric motors are the normal power sources. The noise and vibration are transmitted in four main ways: 1) refrigerant gas path, 2) discharge tube path, 3) suspension system path, and 4) lubricating oil path. All four paths lead directly or indirectly to the compressor shell, which after its modes are excited into vibration, radiates noise. Figure 1 gives a detailed cut-away drawing of a typical reciprocating piston compressor.

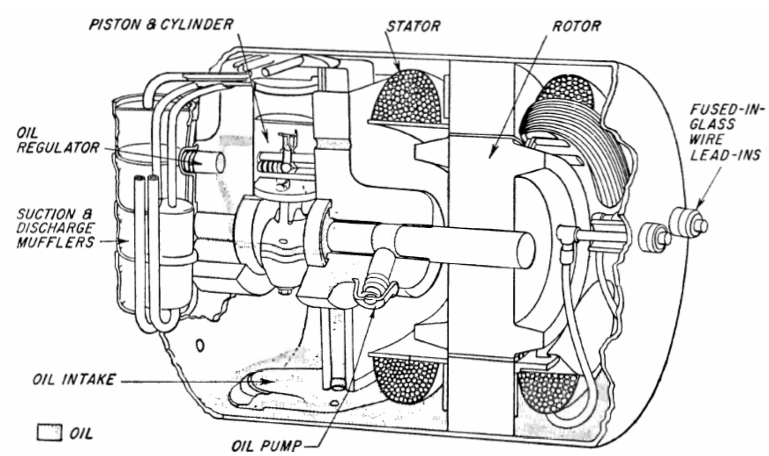

Figure 1 – Cut-away diagram of a typical reciprocating piston compressor.[1]

With such a reciprocating piston system, impulsive noise is created by mechanical impacts caused by rapid closure of the suction and discharge valves. In addition, since the fit of the piston in its cylinder is not perfect, and a small amount of clearance must be provided, the gas forces on it caused by compression make it “rock” from side to side resulting in impacts known as “piston slap.” This is another potential source of radiated noise. Blow-by noise caused by the piston/cylinder clearance can sometimes also be important. Although steady non-turbulent flow, in principle, does not cause the creation of sound waves, fluctuating flow does, and impulsive flow changes caused by the rapid opening and closing of the suction and discharge valves is responsible for the creation of sound waves which propagate throughout the inlet and discharge pipework. The mechanical system is normally hermetically sealed in a compressor shell. Such compressors are expected to have a long operating life of ten years or more.

Figure 2 presents a schematic of the main noise and vibration sources in a reciprocating piston compressor used in household refrigerators, air conditioners or heat pumps. In many such compressors, the noise and vibration sources are strongly correlated (inter-related) and it is difficult to separate them.[1-2] In a typical household refrigerator, besides the airborne noise radiated from the compressor shell, airborne noise is also produced by the cooling fan, flow-induced noise of the refrigerator, and structure-borne noise caused by all of these sources, which is then radiated as airborne noise by the refrigerator itself. Thus, in order to study the compressor noise experimentally, it is necessary to remove the compressor from the refrigerator and mount it in a load stand which provides the compressor with the correct refrigerator and pressure conditions. The load stand noise sources are separated from the compressor noise stand in well designed experiments.[1-2]

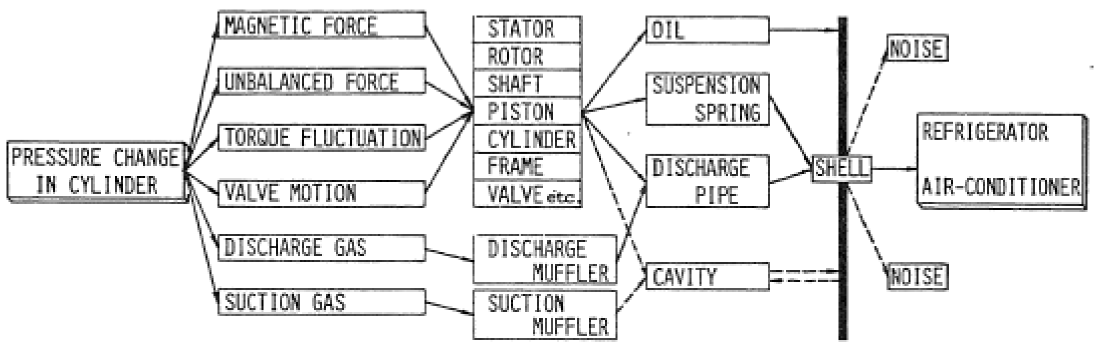

Figure 2 – Schematic noise generation mechanisms in a compressor driven by an electric motor[2]

Vibration and noise measurements on reciprocating piston compressors

Figure 3 presents measured time history results obtained on a small reciprocating piston compressor.[3] It is observed that there is no obvious close correlation between the vibration of the body vibration (V1) and the low frequency sound pressure (noise) (P4). The compressor working fluid has a discrete frequency component of 240Hz in the discharge pressure and of 480Hz in the suction pressure. In such a compressor, modification of the fluid path volumes and pipe diameters to ensure that none of these frequency components match with the shell cavity volume natural frequency normally helps to reduce the low frequency compressor noise in the range of 25-1000 Hz. The fundamental acoustic natural frequency of the cavity depends on its temperature of operation and will always be excited momentarily if the excitation frequency passes through this natural frequency during compressor start-up and/or shut-down.

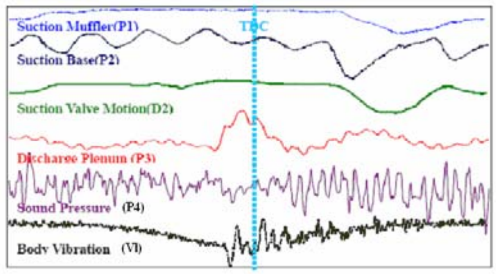

Figure 3 – Experimental results for compressor discharge pressures, valve suction motion and body vibrations, and sound pressure; P1-suction muffler inlet pressure, P2-suction outlet base pressure, P3-cylinder head (discharge plenum) pressure.[3]

Improved design of suction muffler

Other methods of noise control include improved suction muffler design.[4] In this design, the compressor pump unit consists of a piston-cylinder block mounted on top of an electric motor. The compressor pump-motor unit is enclosed in a 3 mm thick hermetic steel shell, which, together with the suction and discharge lines, connects the unit with the appliance.[4] A cut through view of the muffler is shown in Fig. 4(a) and of a BEM model of it in Fig. 4(b).

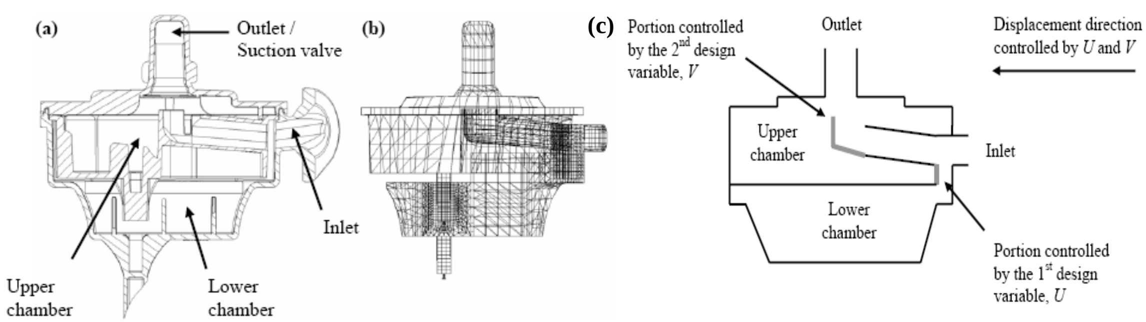

Figure 4 – Suction muffler: (a) cut-through drawing and (b) boundary element model (BEM)[4] *(c) Diagram of original muffler. Gray thick lines show which structural items were modified geometrically*[4]

When the compressor was operated under appliance conditions, it was observed that the sound power increased in the 800 Hz, 3.2 kHz and 4 kHz one-third octave bands. Separate experiments on the compressor showed that the dominating source of noise in these bands is caused by the suction valve.[4] Pressure pulsations near to the inlet of the suction valve were thought to excite cavity modes. The lowest cavity modal resonance frequencies are at about 620 and 720 and they have associated sound pressure distributions which are favorable at exciting shell deformed (breathing) modes of the hermetic shell. Unfortunately these shell vibrations have rather high radiation efficiencies. These cavity resonances are assumed to be responsible for the relatively high sound power levels particularly in the 630 and 800 one-third octave bands.

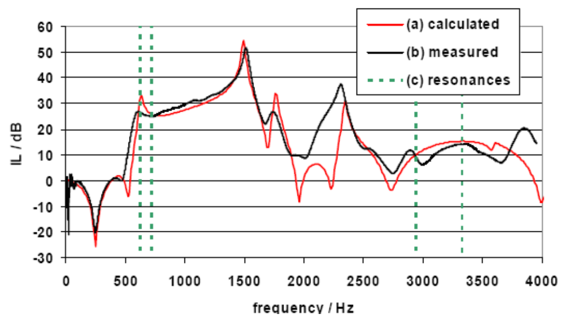

Figure 5a – Calculated and measured insertion loss of original muffler (a) measured and (b) calculated, together with (c) resonance frequencies[4]

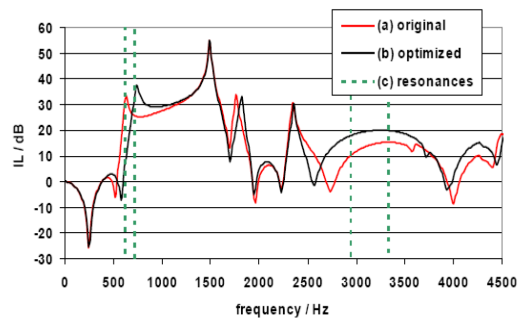

Figure 5b – Calculated insertion loss of (a) original and (b) improved muffler together with (c) resonance frequencies[4]

Two other resonance frequencies were found to be very important with this compressor. These are the shell vibration natural frequencies of 2970 Hz and 3330 Hz. These presumably are responsible for the high sound power levels in the 3.2 kHz one-third octave band. The original suction muffler used in this compressor possesses two chambers connected in series by the inlet and the flow guide tube. See Figs. 4a, and 4b. Figure 4c shows a schematic diagram of the model which was used to analyze the insertion loss of the suction muffler system.

The insertion losses measured and that predicted using a BEM model are shown in Fig. 5a. It is observed that there is very good agreement up to a frequency of almost 2000 Hz. Above that frequency, the prediction is not so accurate, presumably because the BEM mesh size used was not small enough. The BEM program used to predict the IL was run changing two variables U and V (see Fig. 4c). By increasing the slit between the inlet suction tube and the flow guide tube from 2.4 mm to 4.8 mm and moving the bending portion of the flow guide tube 1.4 mm in the direction of the arrow (see Fig. 4c) BEM predictions showed that the muffler insertion loss was improved. This is shown in the predictions in Fig. 5b. The sound power radiated at the four resonances 620 Hz, 720 Hz, 2970 Hz, and 3300 Hz is reduced.[4]

Reed valve vibration and noise

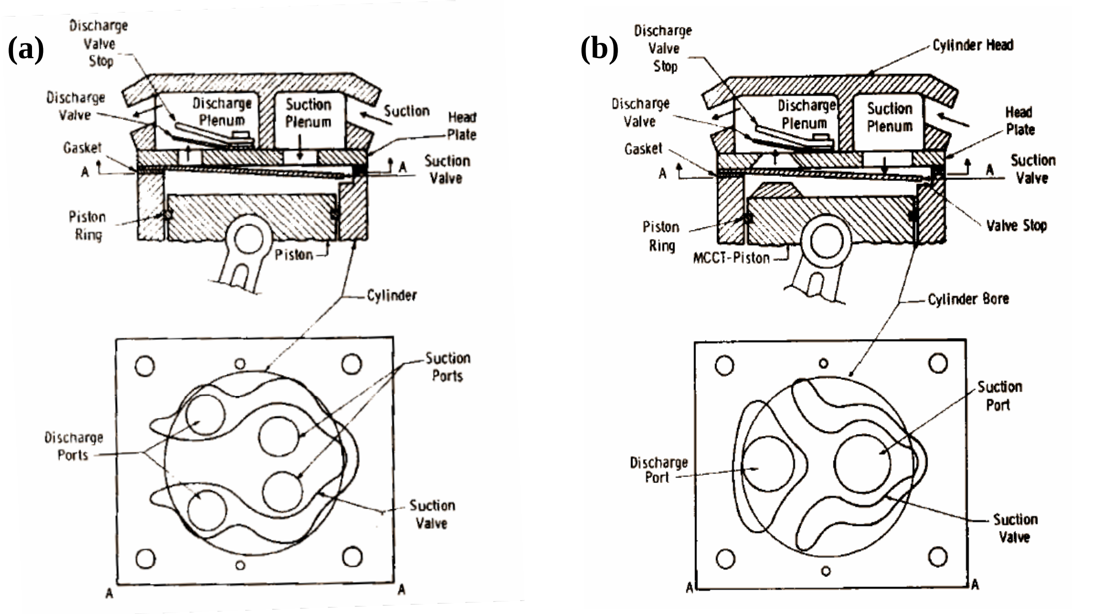

Figure 6 – Hermetic reciprocating refrigeration compressor. a) Side view (top) and plan view (bottom) of original compressor. b) Side view (top) and plan view (bottom) after modification of compressor.[5]

Noise reduction has also been achieved on a reciprocating piston compressor by modification of the piston cylinder head and valves.[5] Figure 6a shows a schematic of a standard compressor cylinder head, piston and valves before modification, and Fig. 6b shows the same compressor parts after modification. The modified compressor had a special piston and suction valve as shown. This piston is seen to have a small “tap” attached to its upper surface which is made to fit into the discharge port when the piston reaches top dead center of its stroke. With the use of the “tap” the new piston assembly reduces the clearance volume when the piston is at top dead center and this prevents back flow occurring during the suction stage, thus permitting the use of thinner suction and discharge “reed” valves. Use of the thinner reed valves changes the suction and discharge process and reduces the valve impact excitation and resulting compressor vibration response. It was found that these changes produced reductions of 3 dB in both the suction and discharge space-averaged externally radiated A-weighted sound pressure levels.

Shell vibration

The noise radiated by the compressor of a household refrigerator is mostly caused by the noise radiated by the compressor shell. Many attempts have been made to study and understand compressor shell radiation from small compressors.[6] In one small refrigerator compressor, modal analysis tests, sound intensity contour plots and sound power frequency spectra were measured to try to identify sources and paths of vibration/noise energy transmission.[6] Results show that the sound power radiated is dominant in two one third octave bands at 800 Hz and 3150 Hz. Further investigation with excitation by a calibrated impact hammer and use of modal analysis software revealed that two modes of vibration at 2810 Hz and 3080 Hz were responsible for the intense sound generated in the 3150 Hz one-third octave frequency band. The modal analysis contour plots and the mode shapes show that for this compressor the intense sound in the 3150 Hz one-third octave band is radiated predominantly by the 2810 Hz and 3080 Hz modes from the bottom of the compressor shell. The intense noise radiated in the 800 Hz one-third octave was found to be related to forces fed through the compressor spring mounts to the shell resulting in shell sound radiation.[6]

Research has also been conducted on compressor shell vibration using theoretical models. Most small compressor shells have a cylindrical shape, of either circular or elliptical cross section with doomed end caps or plates at each end of the cylinder. The shell modes of vibration can be grouped into three main classes: 1) *cylindrical modes* in which large deflections of the cylindrical part of the shell occur, but the end plates remain essentially undeflected, 2) *top-bottom modes* in which large deflections of the end plates occur leaving the cylindrical part largely unaffected, and 3) *mixed modes,* in which both the cylindrical and end plates undergo deflections simultaneously. Cossalter et al. studied the vibrations of a shell system to the main excitation forces: a) discharge pipe force, and b) spring suspension forces.[7] They showed that, with the elliptical cylinder shell studied, for the same force amplitude, the discharge pipe force excites more modes and the seventh mode having a natural frequency of 2676 Hz with the greatest vibration amplitude. Figure 7 shows the location and direction of the discharge pipe force and Fig. 8 presents the response in the seventh mode which is the most excited mode.

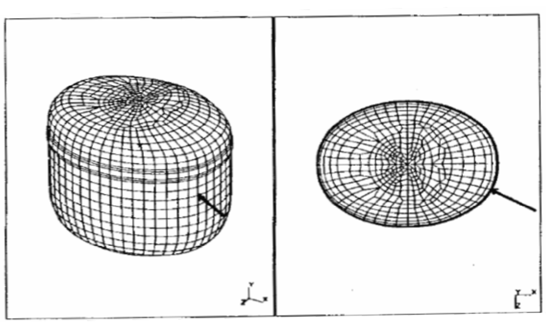

Figure 7- Location and direction of the discharge pipe force.[7]

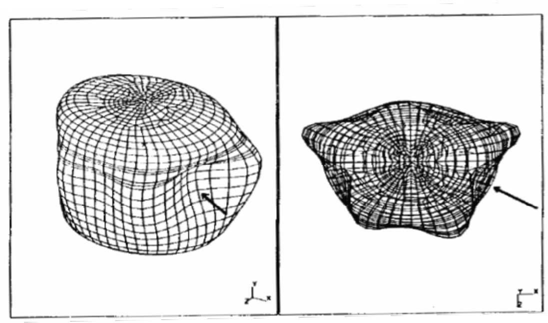

Figure 8 – Deformed shape of the shell when the 7th mode is excited.[7]

Using a different number of spring support systems, moving the location of the spring supports relative to the discharge pipe location, ensuring that the compressor shell natural frequencies are not close to any internal forcing frequencies, and increasing the shell damping can also all be effective in reducing the compressor shell radiated noise.

Noise Control of Rotary Compressors

Case histories of noise control on rolling piston and scroll compressors are described here. Wang et al have conducted noise control studies on a hermetic shell rotary compressor.[8] Modal analysis and finite element analysis were carried out in parallel. The experiments were conducted first with the completely disassembled unit and then with the unit built up step by step in order to understand the complicated dynamics of the complete compressor-shell structure. In this compressor, the stator and cylinder block are welded at three points to the shell. This makes the shell stiffer, but allows vibrations of the cylinder assembly to be transmitted directly to the shell. It was found that the shell has its first structural natural frequency at about 600 Hz. Table 1 gives a comparison of the first five natural frequencies of the shell on its own calculated by 1) finite element (FE) analysis, and 2) measured by the modal analysis tests.

Table 1 – Comparison of the finite element (FE) predictions with measured natural frequencies of the compressor shell.[8]

Mode number	1st	2nd	3rd	4th	5th
FE analysis	616Hz	1712Hz	2465Hz	2813Hz	3153Hz
Modal test	635Hz	1742Hz	2464Hz	2731Hz	3132Hz
Error(%)	3.0	1.8	0.	2.9	0.7

Wang et al. also studied the complete built-up compressor system using both FE analysis and modal testing.[8] The conclusion was that two main compressor resonances occurred. In the 1.5 kHz region, vibration of the cylinder block system excited the shell in a rigid body mode, while in the 3.5 kHz frequency region, the cylinder block vibration excited the shell in an elastic bending mode. Modal testing gave a frequency of 3368 Hz for the latter elastic mode, while FE modeling predicted a frequency of 3512 Hz for this mode. It was concluded that, for the elastic bending mode in the 3500 Hz region, the nodal points were close to the weld points. It was thought that most of the vibration energy was transferred to the shell from the cylinder assembly at these weld locations. The rotation of the rotor is supported by the motor and pump bearings and it was believed that these are the main sources of excitation for the shell vibration in the 3500 Hz frequency region. To reduce the excitation, the hub length of the motor bearing was increased to try to make the shaft rotation more stable.[8]

Rolling piston compressors

In a rolling piston compressor, the interaction between the suction and discharge pressure pulsations, mechanical forces, reciprocating motion of the sliding vane, roller motion, roller driving forces, and electromagnetic forces in the electric motor are very complicated. Refrigerant gas pulsations take place on both the low and high pressure sides of the rolling piston system. During the compressor operation, the rolling piston and sliding vane divide the gas into variable volume suction chamber and discharge gas chamber volumes. It is difficult to model the forced vibration and noise system as a whole. Experimental approaches to reduce vibration and noise radiation are normally used.

In one study, the vibration magnitudes were mapped over the shell surface.[9] High levels of vibration at different frequencies were recorded on the shell above and below the electric motor stator, on the accumulator strap, and near to the wire welds and suction line. Three main methods of reducing the compressor vibration and noise were applied successfully: 1) A vibration damper consisting of wire loops wound around the compressor housing near to the regions of maximum vibration magnitude resulted in an A-weighted sound pressure level reduction of 2.5 dB, 2) modification of the suction inlet passageway to provide a smoother inlet passage, a narrow smaller cross section passage to act as a diffuser throat and a more symmetric inlet passage in the cylinder sidewall to connect with the cylinder suction volume produced an A-weighted drop in sound pressure level radiated of about 2.0 dB, and 3) a redesigned rotor and crankshaft thrust bearing made of low friction polyamide material gave a further A-weighted sound pressure level reduction of about 2.0 dB.

Scroll compressors

The noise characteristics of a scroll compressor vary considerably with load. Measurements must be conducted under real load conditions to investigate the operating noise characteristics of the compressor. It is difficult to carry out noise source identification under load conditions. In one study by Kim and Lee, identification of noise sources on a scroll compressor and re-design of its structure were performed.[10] An Array of 15 microphones was used to identify the noise sources on the compressor. Since the noise generated depends considerably on the load, the noise source identification was conducted under load. It was found that the noise was predominant in the 1600 Hz and 2500 Hz one-third octave frequency bands. Structural resonances of the upper frame and fixed scroll were found to be at 1458 and 1478 Hz, respectively. It was observed from holograph measurements that the 1600 Hz one-third octave band noise is related to impacts between the fixed scroll and the upper frame, while the noise in the 2500 one-third octave band is related to the sound radiated from the upper chamber.[10]

For the reduction of the impact noise, damping material that had good characteristics at high operating temperatures was used. The thickness of the material chosen was 1mm. The material was applied to the upper frame. As a result, the A-weighted sound pressure level was reduced from 68.1 dB, when the original fixed scroll was used, to 56.2 dB when the modified fixed scroll was used, and a further noise level reduction of 3.6 dB resulted. By inserting a 0.5 mm thick copper sheet between the interchamber and the upper frame, transmission of impact energy was reduced resulting in a further 3.3 dB reduction. See Fig. 9

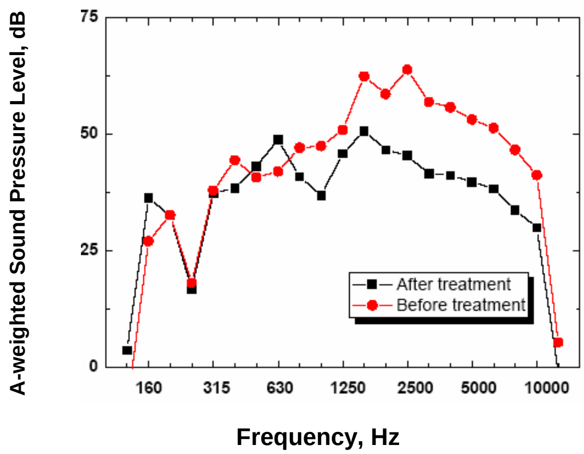

Figure 9 – Comparison of sound pressure level before and after treatment.[10]

As a result, an A-weighted sound pressure level reduction of about 12 dB was achieved by the use of damping material and of 3.6 dB by modification of the upper frame and inter-chamber, respectively. However, modification of the upper chamber and the fixed scroll were not put into practice because of manufacturing difficulties. The sound pressure level of this compressor was higher than usual because of the use of a compressor type in which the internal components can be changed easily. The overall noise level reduction achieved by the use of the damping material on its own was about 12 dB.

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