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Identification of Sources and Propagation Paths of Noise and Vibration in Rotary Compressors

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***Abstract* — this paper presents an experimental methodology, based on statistical analysis of repeatability, for identification of the major sources of noise generated by internal and external components of rotary compressor. The identification is a very complex task due to the kinematics of the compression process, coupled with the large areas of the accumulator and the housing. This study identified that at low frequencies the noise is controlled by kinematics of the compressor, electric motor and compressor vane. The gas flow, discharge valve, shaft and roller of compressor actuate at middle frequencies. Finally, the high frequencies are controlled by friction between the roller and its components.**

***Index Terms*-Rotary Compressor, Noise and Vibration, Source Identification, Statistical Analysis of Repeatability.**

I. INTRODUCTION

The rotary compressor can be found practically everywhere. They are in refrigerators, air conditioners and heat pumps, etc. Industrial applications are numerous, with various configurations. As a group, they are not the largest energy users, but certainly one of the largest noise polluters [16]. of the factors which determine human comfort, noise and vibration are the two factors which most affect human perception and were intensely by Harris (1957), Crocker (1975), Yerges (1978), Beranek (1971), Foreman (1990), Erwins (1984) e Allemang (1984) apud Kim, Cho, and Chou [10]. Noise generates many undesirable effects. High noise level can cause: hearing loss, increase blood pressure (physiological effects); annoyance (psychological effects), for example, sleep disturbance, stress, tension; reduced performance: interference with oral communication, which in turn causes irritation and may cause damage and structural failure (mechanical effect). Noise also influences the decision making of the consumer, when he chooses a more silent product [2, 15]. A strong pure tune in 4 kHz is usually observed in rotary compressor [1]. According to Kim et al (2004), from the noise viewpoint, the hermetic rotary compressor is one of the most important components in a refrigeration unit, once it has great influence on performance, noise and vibration of the system. Noise and vibration occur due to fluid pulsation during compression and unbalanced dynamics forces [11]. Malcolm, 2007, mentions that the sound generated by rotary compressors depends on the rotational frequency and its multiple, number of rotating elements, flow capacity and other flow related factors [12]. Ling (2008) noted that due to the accumulator’s large volume, it effectively contributes to the noise level [8]. Okur and Sinan, 2011, proposed the substitution of the vane with spring by an articulated vane as at higher operation frequencies the articulated vane has isentropic efficiency [14]. Lee, et al (1994) by the signal processing technique of acceleration and sound pressure observed that gas pressure pulsation and vibration of the housing are sources of noise in the 3 to 4 kHz band [7]. Kim, et al (1998) mentioned that although the technique of signal processing is a powerful tool, it is not quite appropriate to identify sources in complex structures such as rotary compressors, because it shows a strong correlation between various sources. An analysis by sound intensity and vibration of the surface was proposed. It was concluded that the suction accumulator is a source of noise and after some modifications a reduction of 20 dB at 2 kHz was obtained [9]. Kim, Cho and Chou (2000) noted that due to the complex nature of the generation and transmission of noise and vibration of rotary compressors it is essential to apply various techniques to identify the various sources and transmission paths. Their paper proposed a technique of analyses by finite elements, experimentation and signal processing. The gas pressure and the housing are the main sources of noise in 5 kHz [10]. In 2008, Huang et al, and later in 2012, Gu, Zhang and Xu, evaluated by spectral analysis the critical moment of noise generation based on the angular position of the compression system for various operating frequencies and found that at discharge valve opening (210° to 360°) larger amplitudes occurred [3,4]. A constant search and study to identify sources and transmission paths, as well as solutions to minimize the noise and vibration of rotary compressor was observed. Such studies focused only on the primary source of noise for a given frequency. But it is known that a reduction of the frequency spectrum is achieved reducing various components of frequency, and, therefore, it is necessary to know which component of the compressor is dominant in which frequency band/range. Another problem observed is related to the reliability of the data. Every manufacturing process, despite all modern control and quality control, is subject to variations. Thus, its propose the identification of sources based on the statistical repeatability analysis. Therefore, the importance of the rotary compressor in modern society is clear. However, the noise emitted by these machines besides complex is worrying. Bearing in mind that noise interferes in various ways in our life, the present paper identifying the contributions of the various components in each frequency band is justified. The compressor is equipment designed to increase the pressure of a fluid in gaseous state. Fig. 1 shows an example of a rotary compressor with the major components labelled.

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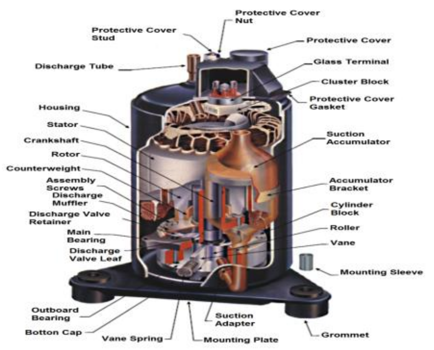

Fig 1. Rotary compressor, adapted from [17].

Based on the studies of Kim, et al, 2004, Soedel, 2006, Malcolm, 2007, and HVAC, 2008, the major sources of noise in rotary compressors are: Pressure, Vane, Roller, Valves, Accumulator, Housing, Friction and Electric Motor [5, 11, 12, 16].

II. METHOD AND EXPERIMENTS

To choose between experimental or numerical analyses procedure, first a statistical analysis of repeatability of the sound spectra, rotational frequency and natural frequencies of the compressors was carried out. The Sound Power Level (SWL) was calculated for a sample of 15 compressors in the semi-anechoic, in accordance to ISO 3744 [6]. The 10 microphones positions are shown in Fig. 2.

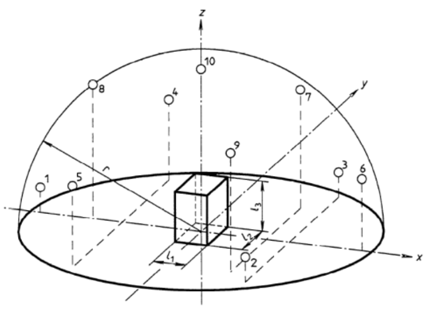

Fig 2. Microphone position as recommended by ISO 3744 on a hemispherical surface.

The signals were acquired during 25 seconds with sampling frequency of 25640 Hz. The response signals were acquired using PCB model 377B02 microphones, a NEXUS B&K signal conditioner and a National Instruments A/D 9233 board. The acquisitions were registered in a notebook using the Virtual Lab® computational tool. The experimental set-up is shown in Fig. 3.

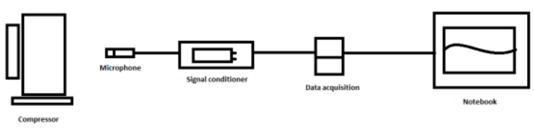

Fig 3. Sound pressure acquisition set-up.

The variability of the rotational frequency of 203 compressors was analyzed. The input signal is the compressor in activity and the output signal was obtained with an accelerometer positioned at the weld point shown in Fig. 4.

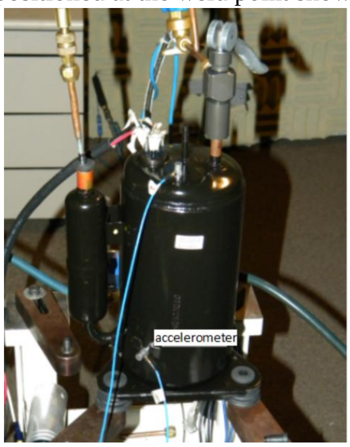

Fig 4. Position of accelerometer at reference weld point close the region of the accumulator.

The experimental set-up (Fig. 5) consists of an accelerometer (B&K model 4371), a signal conditioner (PCB model 482A20) and an A/D board (National Instruments model NI 9233) and Virtual Lab®. The signals were acquired during 10 s, digitalized using a sampling frequency of 25640 Hz. The acquisition was realized with the compressor in steady state, suction pressure of 57 psi and discharge pressure of 226 psi.

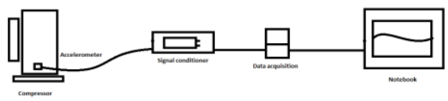

Fig 5. Vibration acquisition set-up.

Finally the Frequency Response Function (FRF) for natural frequencies of the 15 compressors were estimated using an impact hammer and a accelerometer. The H1 estimator was used for 16 impulse responses with rectangular window, during an acquisition period of 10 s with sampling frequency of 25640 Hz. The experimental apparatus consisted of a B&K model 8204 impact hammer a PCB 482A20 signal conditioner, a National Instruments A/D NI 9233 board and Virtual Lab®. The set-up is shown in Fig. 6.

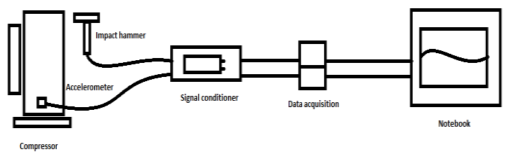

Fig 6. FRF acquisition set-up.

The location of the accelerometer is the same as in the previous test, the reference weld point (close to the accumulator). An impulsive excitation, in three points of impact: first at the protective cover, second at the housing (close to the stator) and third at the accumulator was used. Fig. 7 shows a typical SPL spectrum of the compressors

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studied. Due to industrial confidentiality the y-axis will be not displayed for some graphs. In its place a scale for comparison purpose will be shown.

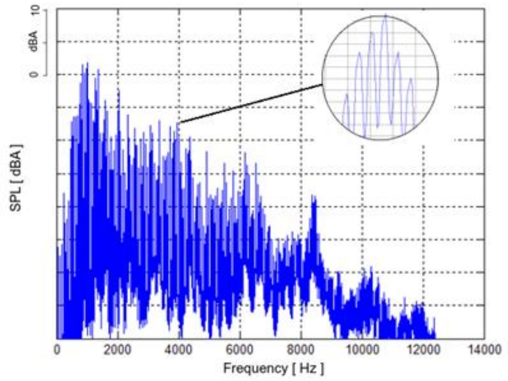


Fig 7. Typical SPL frequency spectrum of rotary compressors.

It shows a spectrum rich in harmonics, which characterizes a strong discontinuity of the signal in the time domain as can be observed in Fig. 8 of an acceleration curve of a rotary compressor measured at the reference weld point.

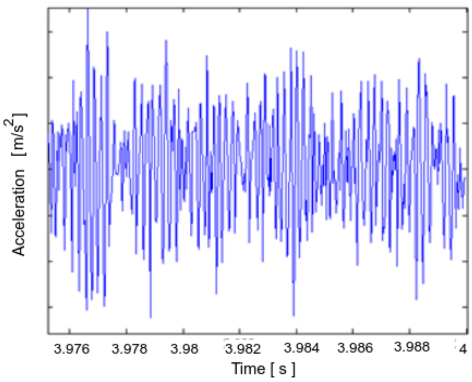
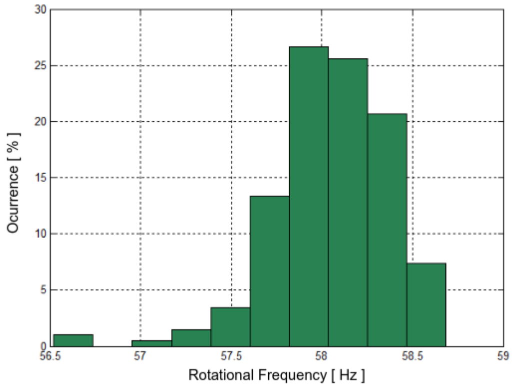


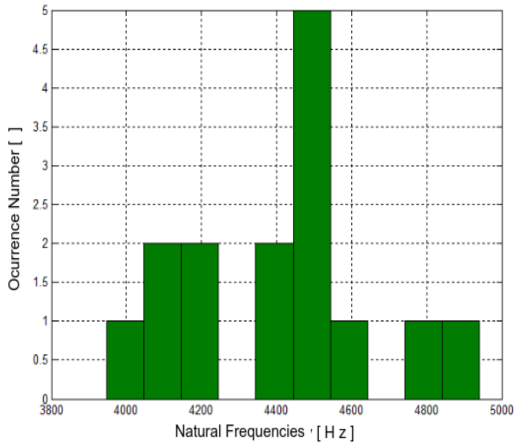
Fig 8. Acceleration against time curve.

Another fact observed in Fig. 8 is a phase modulation or beating responsible for the side bands presents in all regions of the spectrum shown in Fig. 7. This modulation is inherent of the rotor kinematic and is due to the difference in rotational speed of the shaft and the roller. To explain the variability of noise levels generated by these compressor, Fig. 9.a and Fig. 9.b show, respectively, the histograms of rotational frequency (percentage) and number of occurrences of natural frequencies of the housing (from 3 to 5 kHz) of the compressors studied.

Despite of the small dispersion observed in rotational speed, such variation combined with a variation of 10 % in natural frequencies explains the wide variability of the noise levels generated by the compressors once the noise generated is influenced by the coincidence or not of high order harmonics (above the 50th) with natural frequencies of the system. Therefore, due to the great variability of noise levels generated by these compressors, an experimental approach was used. Student's t-test was used to compare a baseline compressors group with a group of modified structural ones, since it is not practical to disassemble a hermetic compressor to make structural changes [13]. In analyzes without structural modifications (variations of operational conditions: discharge and suction pressures) the paired t-test was used [13]. Samples with at least six compressors of each group were used – when possible - and results are presented for a confidence of 10%. In resume: the methodology is based on experimental procedures where constructive changes in groups of compressor were made in order to observe the influence of the various components on generated noise levels.



a. Histogram of rotational frequency.



b. Histogram of natural frequency.

Fig 9. Histogram of rotational and natural frequency.

A. Identification of the influence of the electric motor, dynamics behavior of the roller and roller vane interaction on noise levels generated.

The experimental procedure consists in test for measuring vibration and sound pressure for three groups of compressor denominated respectively: Standard (8 randomly selected RG compressors of the production line), Without Vane (10 RG assembled without vane) and Without Vane and Roller (8 RG assembled without vane and roller). To minimize of the gas, all compressor were initially tested in vacuum. However due to the impacts occurring between vane and roller, testing was repeated with vacuum at suction and various discharge pressures and also 30 psi at suction and various discharge pressures in order to evaluate the influence of the discharge pressure during the compression process. Table I shows the test conditions for these three groups of compressors.

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Table I. Test conditions of the three groups of compressor for the influence of electric motor, roller and roller-vane interaction.

Set	Number of compressors tested	Suction pressure [psi]	Discharge pressure [psi]
Without Vane and Roller	8	Vacuum	Vacuum
Without Vane	10	Vacuum	Vacuum
Standard	8	Vacuum	Vacuum
	8	Vacuum	125
	8	30	125
	8	Vacuum	50, 100, 150, 200, 250 and 300
	4	30	

The microphones were positioned 1 meter from the compressor for the acquisition of Sound Pressure Level (SPL) and the accelerometers are at the reference weld point and at the protective cover. The experimental procedure is the same as in the repeatability analysis.

B. Identification of the influence of spring, discharge valve, roller and axial clearance between roller and bearing on generated noise levels.

The experimental procedure consists of tests for measuring vibration and noise in five groups of compressors denominated: Standard, Without Spring, Modified Roller, With Clearance and Without Valve. In the Without Spring group, the spring from the kit was removed to evaluate the influence of the spring on vane roller interaction. Eight compressors were mounted for this test. The Modified Roller group comprises of eight compressors, where the roller had a reduction of 32 % of its mass, as shown in Fig. 10. The purpose of this alteration is to reduce the roller-kit friction and to reduce inertial forces involved in roller-kit impact.

Fig 10. Modified roller with 32% mass reduction.

The With Clearance group consists of six compressors where the axial clearance between the roller and the bearing was approximately 0.029 mm. The objective of this clearance is to reduce the friction forces between the roller and the bearing. The Without Valve group consists of eight compressors mounted without the discharge leaf valve. In this test the maximum discharge pressure was 200 psi, since above this value there is return of gas to the compressor chamber, making it difficult to operate. The purpose of this group is to evaluate internal shocks in the compressor system. In the acquisition of signals of vibration, 5 positions of accelerometer were used: the first at the reference weld point (close to the accumulator), the second at the protective cover, the third at the electric motor region, the fourth at 105° anticlockwise from the accumulator (impact region of the roller) and the last at the accumulator. In parallel, SPL was obtained with 5 microphones positioned at 1 meter from the 5 accelerometers. These data where obtained using the set-up shown in Figure 5 with an acquisition of 10 s at a sample frequency of 25640 Hz. A second acquisition of 2 s with a sample frequency of 120 kHz was also made. This test is for a more detailed analysis of the impacts. Table II shows the number of compressor tested in each experimental procedure. Note that the Standard group was tested with two different discharge pressures, 226 and 200 psi. The second condition was exclusively for comparison with the Without Valve group, and the first was used for the other groups.

Table II. Test conditions of the five groups of compressor to study the influence of spring, discharge valve, roller and axial clearance between roller and bearing.

Set	Number of compressors tested	Suction pressure [psi]	Discharge pressure [psi]
Without Spring	8	57	226
Modified Roller	8	57	226
With Clearance	6	57	226
Without Valve	8	57	200
Standard	8	57	226
	6	57	200

II. RESULTS AND DISCUSSION

A. Identification of noise sources

1) Above 6000 Hz

Fig. 11 shows the graph of SPL against frequency for the groups: Standard (Blue) and Without Valve (Red).

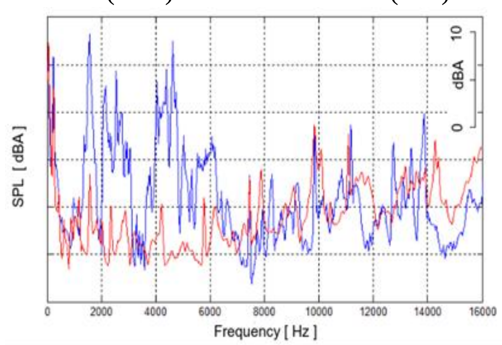

Fig 11. Comparative SPL for Standard (Blue) compressor and Without Valve (Red) compressor.

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Observe that there is a great reduction of amplitude for the frequency range between 1000 and 6000 Hz. Once in the operation of a compressor Without Valve internal shocks do not occur, the noise levels measured above 6000 Hz are credited to frictional forces between the various mechanisms of the compressor. Below 1000 Hz the reduction is credited to internal mechanical shocks. These results are representative of various analyses, such as, for example, the comparison between SWL of the Standard (blue) compressor and the With Clearance (red) compressors, as shown in Fig. 12.

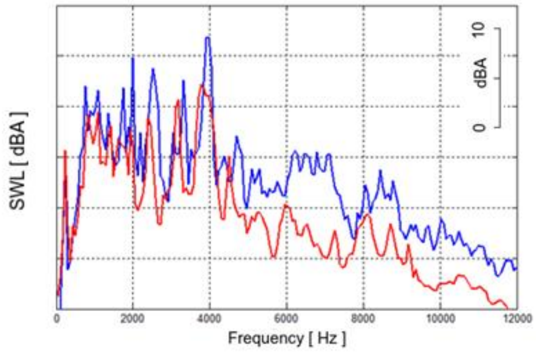
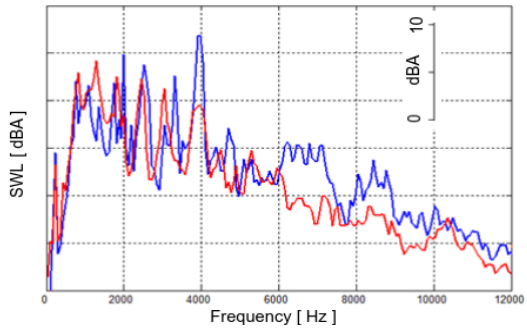


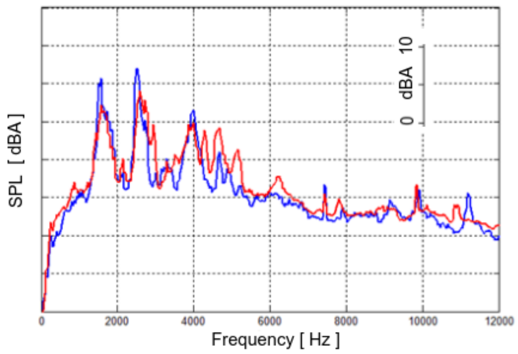
Fig 12. Comparative SWL for Standard (Blue) compressor and With Clearance (Red) compressor

2) 3000 and 6000 Hz band

Fig. 13.a and Fig. 13.b show the SWL against frequency graph for the groups Standard (blue) and Modified Roller (red) and the SPL against frequency for Standard (blue) and Without Spring (red), respectively.



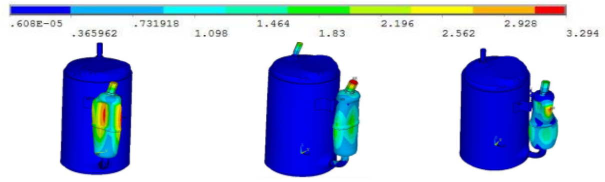
a. Comparative SWL for Standard (Blue) and Modified Roller (Red).



b. Comparative SPL for Standard (Blue) and Without Spring (Red).

Fig 13. Comparative for Standard and Modified Roller and Standard and Without Spring.

Fig. 13.a shows that the interaction of the mechanisms in the roller-kit is principally responsible for noise levels in the 3000 to 6000 Hz band. Fig. 13.b shows the importance of the spring on control of noise generated in the 3000 to 6000 Hz band and below 2000 Hz. Without the spring, the impacts inherent during operation have far more energy. Another important aspect concerning this region of the spectrum is the accumulator. Although it is not a primary source of noise, it contributes significantly to the levels of radiated noise. Fig. 14 shows the modal amplitudes of three natural frequencies for the frequency band analyzed where it is possible to observe the influence of the accumulator on these modes. The modal analysis was carried out using Finite Element Method (FEM) at the commercial finite element analysis software ANSYS®.



- 3711 Hz
- 4175 Hz
- 5740 Hz

Fig 14. Vibration modes of the accumulator calculated by FEM.

3) 2000 and 3000 Hz band

Fig. 15 shows the FRF acceleration curve at a point on the shaft of a rotary compressor, a Strong resonance at 2765 Hz is observed. Considering that under conditions of load the resonance frequency tends to diminish (as the mass increases), it can be inferred that the noise levels in this region are due to the resonance of the compressor shaft.

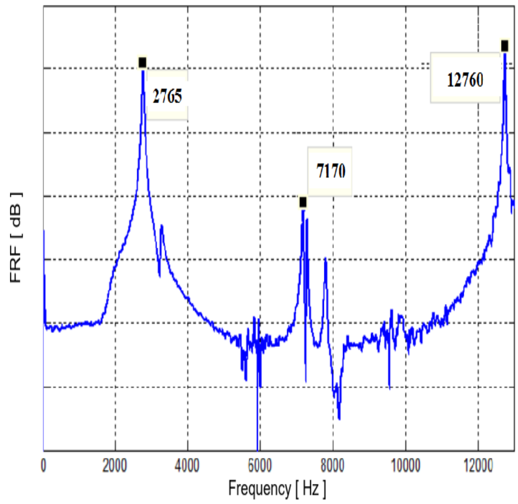


Fig 15. FRF (acceleration) of the compressor shaft

4) 1000 and 2000 Hz band

In this frequency range the generated noise is predominantly due to valve impacts and discharge mechanism. Two results illustrate this hypothesis. A strong resonance is observed on the FRF curve at a point close to the valve retainer in relation to the reference weld point in Fig. 16 and variation of the average SPL values with different discharge pressures in Fig. 17.

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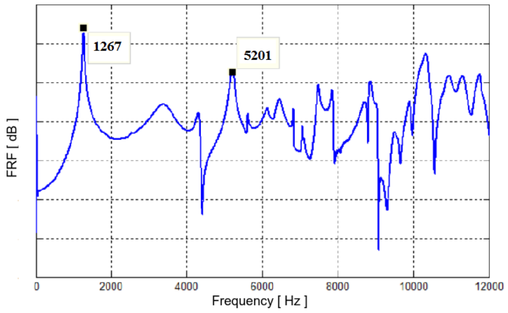

Fig 16. FRF between valve retainer and reference weld point.

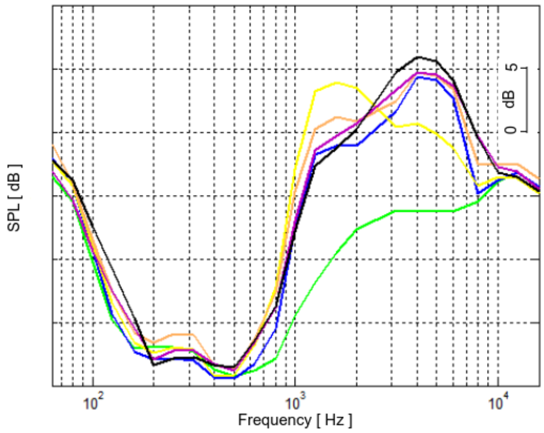

Fig 17. SPL (1/3 octave), suction pressure of 30 psi and discharge pressure of 50 (Green), 100 (Blue), 150 (Yellow), 200 (Orange), 250 (Red) and 300 (Black) psi.

Also the accumulator has effect of noise amplifier due to the presence of two natural frequencies in this frequency range shown in Fig. 18.

1850 Hz   2075 Hz

Fig 18. Vibration modes of the accumulator calculated by FEM

5) *600 and 1000 Hz band*

In this frequency band the dominant generation mechanism is due to the fluid flow and its pulsation. This statement can be readily confirmed by analysis of Fig. 19, which shows that in the 630 to 1000 Hz frequency band, the noise levels generated by tests with 30 psi suction and 125 psi discharge pressure (red) are higher than the total vacuum (green) and vacuum at suction and 125 psi discharge (blue) conditions.

Note that above 600 Hz, the SWL of fluid flow for the 30 psi suction and 125 psi discharge test is larger than the other tests. However in the range of 600 to 1000 Hz, the noise level of the vacuum at suction and discharge test exceeds that of the vacuum at suction and 125 psi discharge test. Thus, although the fluid flow acts in the frequency range above 600 Hz, it is between 600 and 1000 Hz that it has the highest impact.

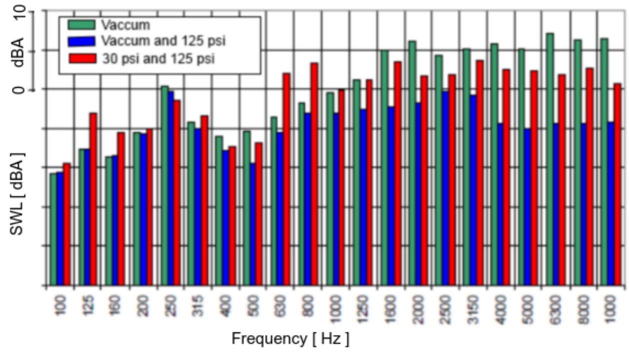

Fig 19. SWL (1/3 octave) for tests in vacuum (Green), vacuum at suction and discharge 125 psi (Blue) and suction 30 psi and discharge 125 psi (Red).

6) *200 and 600 Hz band*

Fig. 20 shows that increasing the discharge pressure, in the tests with vacuum at suction, the noise levels decreases in this frequency band. Since in the tests with vacuum at suction and variable discharge pressure, there is no effect of fluid flow and valve vibrations, it is concluded that the main mechanism of noise generation is the spring-roller-vane interaction. Note that the forces on the vane depend on the discharge pressure.

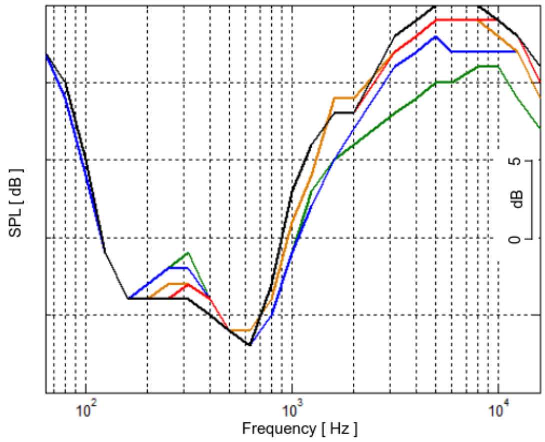

Fig 20. SPL (1/3 octave), for vacuum at suction and discharge pressure of 100 (Green), 150 (Blue), 200 (Orange), 250 (Red) and 300 (Black) psi.

Also in Fig. 19, the values of SPL for full vacuum tests are higher than for the other tests in all 1/3 octave bands, which is a strong indication of the occurrence of internal shocks.

7) *200 and 600 Hz band*

The main sources of noise in this region of the spectrum are of mechanical and electrical origins. In Fig. 19 in this frequency range the noise level is dependent on the suction and discharge pressures, principally in the 125 Hz band. Fig. 21 shows the acceleration Power Spectrum Density (PSD), centered at 100 Hz, measured at the reference weld point, at protective cover and at the center of the housing, for two operational conditions: 30 psi suction pressure with 150 psi discharge pressure (blue) and 30 psi suction pressure with 300 psi discharge pressure (red).

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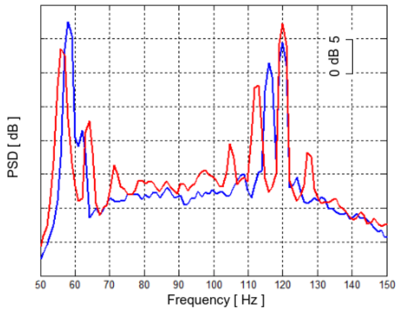
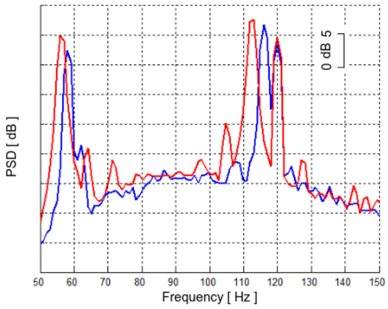
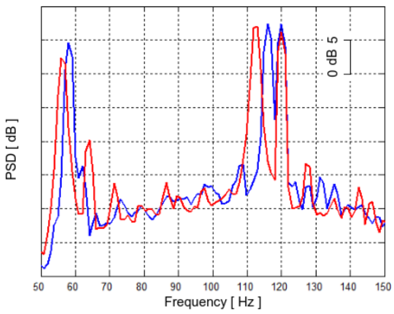


Welding point



Protective cover



Housing

Fig 21. PSD of acceleration for 30 psi suction with 150 psi discharge pressure (Blue) and 30 psi suction pressure with 300 psi discharge pressure (Red).

When the discharge pressure increased, the vibration levels also increased. The more significant values occur for the lateral harmonics of low frequencies. It is important to emphasize that the second harmonic of 120 Hz is high, that is, noise originated from the electric motor which operates at 60 Hz.

C. Sensitivity

1) Mechanisms of noise generation

The sensitivity analysis consists of making a 20 dB reduction in SWL in the region of the spectrum corresponding to the mechanism of noise generation being analyzed and evaluating the effect of the change on the overall SWL in dB(A). Fig. 22 shows the results of sensitivity analysis of the SWL in relation to na attenuation of 20 dB of the amplitudes in the frequency bands of the noise generating mechanisms. The mechanisms are:

- Mechanical and electric motor: 60 to 200 Hz;
- Vane: 200 to 600 Hz;
- Fluid: 600 to 1000 Hz;
- Valve: 1000 to 2000 Hz;
- Shaft: 2000 to 3000 Hz;
- Roller: 3000 to 6000 Hz;
- Friction: above 6000 Hz.

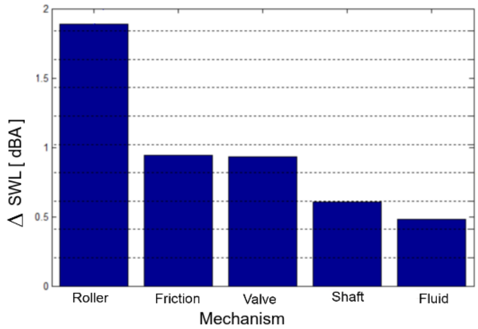


Fig 22. SWL sensitivity analysis of the noise generation mechanisms

The noise contribution from vane, mechanical sources and electrical motor are missing in Fig. 22, because were not significant to the overall gain achieved with as a control measure (less than 1 % each). Fig. 22 shows that the best control measure, with respect to overall levels of sound power, would be to mitigate the noise levels generated by the interaction of roller with the system, in the band of 3000 to 6000 Hz. Another observation is that the gain achieved with only one control measure on only one noise generation mechanism is small and can be easily obscured by the manufacturing variability. The combined effect of the control measures is shown in Table III, the largest gains in reduction of SWL occur for frequency regions above 2000 Hz.

Table III. SWL sensitivity analysis of combinations of noise generation mechanisms

Mechanisms							ΔSWL
Mechanical	Vane	Fluid	Valve	Shaft	Roller	Friction	
					X	X	1,8
				X	X		2,9
			X	X			1,7
		X	X				1,7
	X	X					0,5
X	X						0,1
				X	X	X	4,8
			X	X	X		4,9
		X	X	X			2,4
	X	X	X				1,6
X	X	X					0,6
			X	X	X	X	4,9
		X	X	X	X	X	8,7

2) Control of specific frequencies

To evaluate the influence of harmonics and resonance regions on the noise levels generated by rotary compressor, a sensitivity analysis which consisted in identifying the 15 harmonics of acoustic energy of the characteristic spectrum.

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Reduction of 20 dB in SWL of the levels of these harmonics and their sidebands (± 12 Hz) was made and the impact on the overall SWL (dB (A)) was evaluated. Fig. 23 shows the results of this analysis.

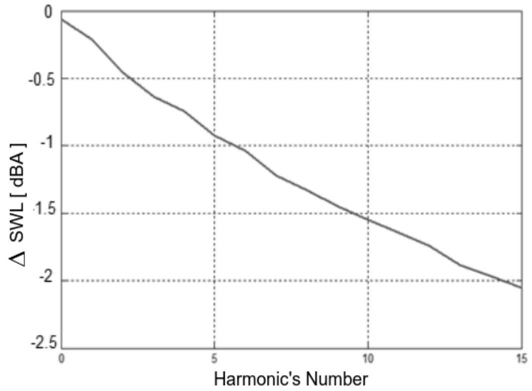

Fig 23. Sensitivity analysis of over wall SWL values (dB (A)) in relation to the 15 most significant harmonics of characteristic spectrum.

Analysis of Fig. 23 showed that the reduction of SWL from the 15 harmonics (and their sidebands) resulted in a gain of approximately 2 dB (A). This result is a strong indication that the problem of controlling the noise generated by these compressors is not a problem of a specific mechanism, but rather a more general problem, which involves the control of noise levels generated in the region above 2000 Hz.

III. CONCLUSIONS

Based on the present results it is possible to conclude that:

- Due to the great variability of noise levels generated by these compressor, an experimental approach was used;
- Student's t-test was used to compare a baseline compressors group with a group of modified structural ones, since it is not practical to disassemble a hermetic compressor to make structural changes. In analyzes without structural modifications (variations of operational conditions: discharge and suction pressures) the paired t-test was used.
- The high amount of harmonics is attributed to multiples of the shaft rotational frequency, approximately 60 Hz. The phase modulation or beating is due to the difference in angular velocity between roller and shaft;
- The experiments performed in the semi-anechoic chamber, were useful to identify the major sources of rotary compressor noise. It was also possible to determine the predominant mechanisms in each region of the frequency spectrum, as follows:
  - Mechanical: Due to the unbalance of forces, misalignment and electrical motor, region 60 – 200 Hz;
  - Vane: spring-vane-structure interaction, region 200 – 600 Hz;
  - Fluid: Fluid flow and cavitation modes, region 600 – 1000 Hz;
  - Valve: discharge mechanism of the compressor, region 1000 – 2000 Hz;
  - Shaft: resonance and forces at the bearings, region 2000 – 3000 Hz;
  - Roller: roller-vane-structure interaction, region 3000 – 6000 Hz;
  - Friction – resultant frictional forces, above 6000 Hz.
- The accumulator is an important source of radiated noise at bands centred at 2000, 4000 and 6000 Hz, and therefore its shape optimization imply in significant improvements in rotary compressor noise;
- From sensitivity analysis, the roller shows best results on overall SWL reduction, followed by the regions of friction, valve, shaft and fluid for the spectrum between 2000 to 6000 Hz. Vane, mechanical sources and electric motor were not significant, being less than 1 % each;
- The gain achieved with one control measure of only one mechanism is small (maximum of 2 dB(A)) and can be easily obscured by manufacturing variability;
- A reduction of 20 dB of the largest harmonics and their respective sidebands of the characteristic SWL spectrum resulted in a reduction of approximately 2 dB (A) in the overall SWL. These results shows that the problem of controlling the noise generated by these compressors is not a problem of a specific component, but a more general problem, which involves of noise levels generated in the region above 2000 Hz.

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