ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

IDENTIFICATION OF SOURCES AND PROPAGATION PATHS OF NOISE AND VIBRATION IN ROTARY COMPRESSORS

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Abstract. *This paper presents an experimental methodology for identifying the major sources of noise generated by internal and external components of rotary compressor. The identification becomes a very complex task due to the kinematics compression process, coupled with the large area of the accumulator and the housing. This study has identified that the low frequency noise are controlled by the kinematics of the compression engine, electric motor and vane compressor, the middle frequencies by gas flow, discharge valve, shaft and roller compressor, and finally the high frequencies by friction between the roller and its components. The experimental results of this work are presented and discussed.*

Keywords: *rotary compressors, noise, vibration, identification*

1. INTRODUCTION

The rotary compressors as a group can be found practically anywhere. They are the hearts of refrigerators, air conditioners and heat pumps. They supply compressed air to machine tools, construction equipment such as jack hammers, medical devices such as dentist's drills and diving equipment. They pump natural gas through pipelines and compress oxygen and nitrogen. Industrial applications are numerous; size range from a fractional-horsepower compressor in a household refrigerator to thousand and more horsepower compressors to pump natural gas. As group, they are not only one of the largest energy users, but also one of the largest noise pollutes (Werner, 2006).

The noise generate many undesirable effects, such as high level enough, can cause hearing loos and increased 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). The noise also influences the decision making of the consumer, when he chooses a noiseless product (Bistafa, 2006).

A pure and strong tone in 4 kHz is usually observed in rotary compressors (Bagepalli, 1989). Kim *et al.* 2004, the noise and compression process occurs due to pulsation of fluid during the compression process and due to unbalanced dynamic forces. Malcolm, 2007, mentions that the sound generated by rotary compressors depends on the rotation frequency and its multiples, numbers rotating elements and flow capacity. Ling (2008) noted that due to the accumulator be a large volume of components, it effectively contributes to the noise level.

Thus, it is clear the importance of the compressors in modern society; however the noise emitted by this machine beyond complex is worrying. Bearing in mind that sound interferes in various ways in human, the present paper is justified in identifying the major sources and propagation paths of noise and vibration in rotary compressors.

2. ROTARY COMPRESSOR

The compressor is an equipment designed to increase the pressure of a fluid in gaseous state. In a refrigeration system he's responsible for receiving the refrigerant at low pressure which is coming from the evaporator and raising his pressure. Fig. 1 shows an example of a rotary compressor with major parts labeled.

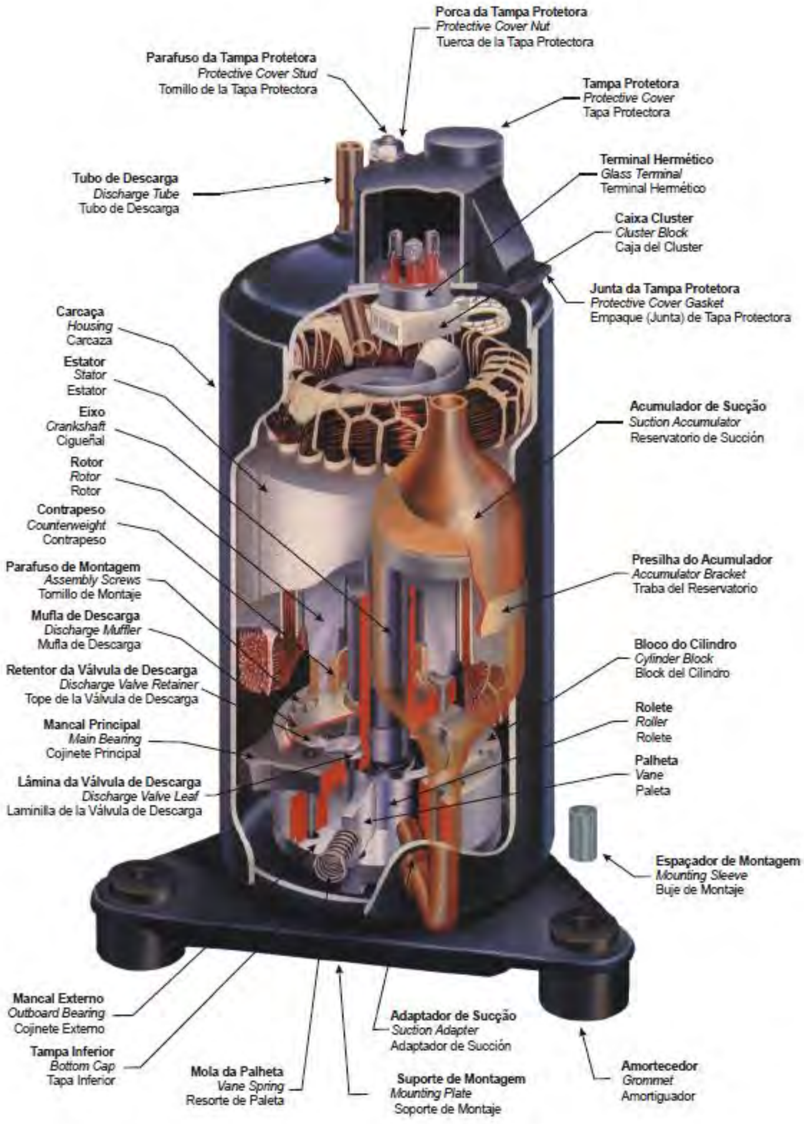

Figure 1. Rotary compressor, model RG – Tecumseh.

In addition, due to the compression process can be cited as major sources of noise:

- Pressure fluctuations
- Vane
- Roler
- Housing
- Valves
- Accumulator
- Friction
- Electric motor

3. METHODOLOGY

The experimental procedure consists of three groups of test.

ISSN 2176-5480

3.1 Repeatability analysis

Once all methods used in this study involve spectral analysis of experimental data and it isn't always possible to carry out various experiments in the same compressor, the first analysis is based on the variability of sound spectra, rotation frequency and frequency response function.

Initially are calculated sound pressure levels (NWS) from the average sound pressure levels (SPL) of 10 microphone positions for a sample of 15 compressors. The microphones, sensors responsible for the acquisition of sound pressure, are positioned as Fig. 2. All test are performed in the semi-anechoic chamber of Tecumseh, in accordance with ISO 3744 – Acoustics – Determination of Sound Power Levels of Noise Sources Using Pressure – Method in Engineering an Essential Free Field over a Reflecting Plane.

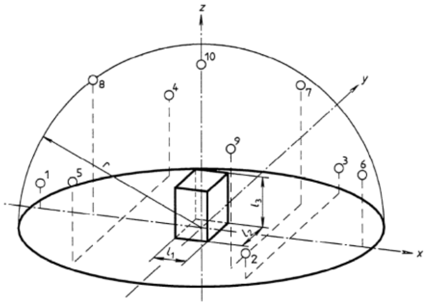

Figure 2. Microphone positions recommend by ISO 3744 on the surface of a hemisphere.

Adjusts the acquisition for 25 seconds with an acquisition frequency of 25640 Hz and a resolution of 0.7825 Hz where the response signals are collected by the microphones PCB model 377B02 then a signal conditioner NEXUS B&K is used to provide gain of the signal, then a board A/D National Instruments NI 9233 is necessary to convert analog signal into digital. Finally, the data is saved in a notebook through computational tool Virtual Lab®. Thus, the measurement chain is illustrated in Fig. 3.

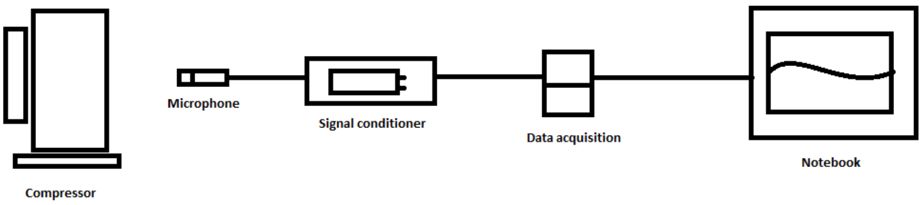

Figure 3. Measurement chain, sound pressure acquisition.

Then was analyzed the variability of the rotation frequency of 203 compressors. Where the input signal is the compressor itself in activity and output signal is collected with an accelerometer positioned at the welding point of Fig. 4.

Figure 4. Position of accelerometer in welding point near the region of the accumulator.

The signals are acquired for 10 seconds to scan a frequency of 25640 Hz acquisition and resolution of 0.7825 Hz. The measurement chain is formed by accelerometer model 4371 B&K, which passes through a signal conditioner PCB Model 482A20 then into A/D board from National Instruments NI 9233, and finally saved with a notebook by Virtual Lab®. The whole process is illustrated in Fig. 5.

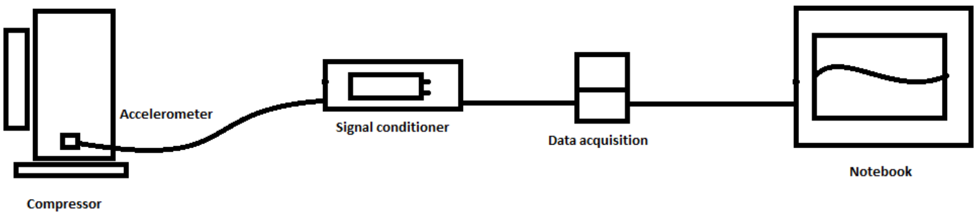

Figure 5. Measurement chain, vibration acquisition.

It is important to mention that the acquisition are realized when the system is in steady state, suction pressure of 57 psi and discharge pressure of 226 psi. Once made the acquisition to a compressor, the compressor is exchanged and the whole procedure is repeated until all compressors are measured.

Finally it is estimated the Frequency Response Function (FRF) of the 15 compressors used in the test for determining the NWS, but disconnected. An excitation via impact hammer is used as an input signal to the system, and the response is acquired via accelerometer positioned at the welding point. The H1 estimator is used for calculating the FRF, also the FRF is determined from 16 impulse responses and rectangular window for an acquisition of 10 seconds with a resolution of 0.7825 Hz and acquisition frequency 25640 Hz.

Was used an impact hammer model 8204 B&K, where the load cell signal acts as a trigger to collect vibration data and a accelerometer. Both signals are amplified in 482A20 signal conditioner PCB, then converted from analog to digital in the A/D board from National Instruments NI 9233 and finally saved into notebook via Virtual Lab®. The entire chain is shown in Fig. 6.

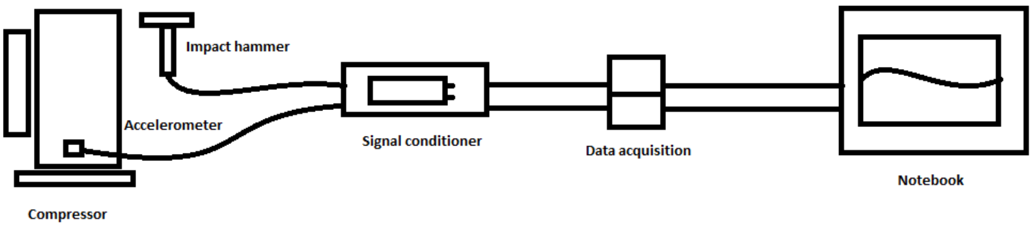

Figure 6. Measurement chain, FRF acquisition.

The location of the accelerometer is the same as defined into previous test, welding point near accumulator. For impulsive excitation is given three points of impact, first at protective cover, second at housing and third at accumulator.

3.2 Identification of the influence of the electric motor, dynamic behavior of the roller and roller- vane interaction noise levels generated

The experimental procedure consists in tests for measuring vibration and noise for three groups denominated respectively: Standard (4 RG randomly selected compressors on the production line), Without Vane (5 RG assembled without vane) and Without Vane and Roller (4 RG assembled without vane and roller).

To minimize the influence of gas, all compressors are initially tested in vacuum. But due to the impacts occurring between vane and roller, testing is repeated for vacuum conditions in the suction and discharge pressure variable, 30 psi suction and discharge pressure variable. Table 1 shows the test conditions for these three sets of compressors.

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Table 1. Test conditions of the three sets of compressors in the influence of the electric motor, dynamic roller and roller-vane interaction.

Set	Number of compressors tested	Suction pressure	Discharge pressure
Without Vane and Roller	4	Vacuum	Vacuum
Without Vane	5	Vacuum	Vacuum
Standard	4	Vacuum	Vacuum
	4	Vacuum	125 psi
	4	30 psi	125 psi
	4	Vacuum	50, 100, 150, 200, 250 and 300 psi
	2	30 psi	

The microphone is positioned 1 meter from the compressor and accelerometer sensors used in this experimental procedure are at the kit welding point, next to the accumulator, and at protective cover.

With operation compressor the acceleration signals are acquired for 10 seconds and scanned in resolution to 0.7825 Hz sampling frequency of 25640 Hz. The measurement chain is the same described in the vibration acquisition for the analysis of repeatability. It is analyzed also the NWS of each group, with accordance to the ISO 3744, where the sound pressure is acquired for 25 seconds at a resolution of 0.7825 Hz and sampling frequency of 25640 Hz, the measurement chain used is the same in the purchase of sound pressure analysis repeatability.

3.3 Identification of the influence of spring, discharge valve, muffle, roller and axial clearance between roller and bearing at generated noise levels

The experimental procedure consists of tests for measuring vibration and noise with five groups of compressors called respectively: Standard, Without Spring, Modified Roller and Without Valve.

Into Without Spring the spring from kit are removed to evaluate the influence of the interaction between spring-roller, 3 compressors are mounted for this test.

The Modified Roller group comprises 4 compressors where the inertia was relieved of the roller with a reduction of 32% of its mass, as shown in Fig. 8. The purpose of the change is to reduce the friction roller-kit and lower inertial forces involved at roller-kit impact.

Figure 8. Modified roller.

The last group Without Valve comprises 3 compressos mounted without the discharge blade. In this test the maximum pressure is 200 psi, since above this value there is return gas to the compressor chamber, which prevents its operation. The purpose of this assembly is to evaluate internal shocks in the system compressor.

In the acquisition of vibration are 5 points for location of the accelerometer, the accelerometer, the first into weldgin region of kit, next to the accumulator, the second into protective cover, the third at electric motor region, the fourth at 105° anticlockwise from the accumulator (impact region of the roller) and the last at accumulator. A

These data are obtained by measuring chain present in Fig. 5 and saved for acquisition time of 10 seconds with a sampling frequency of 25640 Hz and a resolution of 0.7825 Hz. Also a second acquisition with 2 second, resolution of 0,5 Hz and sampling frequency of 120 kHz was realized for a more detailed analysis of the impacts.

It is analyzed also the NWS of each group, with accordance to the ISO 3744, where the sound pressure is acquired for 25 seconds at a resolution of 0.7825 Hz and sampling frequency of 25640 Hz, the measurement chain used is the same in the purchase of sound pressure analysis repeatability.

Table 2 shown the number of compressors tested in each experimental procedure. Note that the Standard group is tested with two different discharge pressures, 226 and 200 psi. The latter is exclusive for comparison with the test group Without Valve, and the first is used for the other groups.

Table 2. Test conditions of the four sets of compressors in the influence of spring, discharge valve, muffle, roller and axial clearance between roller-bearing.

Set	Number of compressors tested	Suction pressure	Discharge pressure
Without Spring	3	57 psi	226 psi
Modified Roller	3	57 psi	226 psi
Without Valve	4	57 psi	200 psi
Standard	4	57 psi	226 psi
	3	57 psi	200 psi

4. RESULTS AND ANALYSIS

4.1 Dispersion analysis

Figure 10 shown a typical spectrum obtained experimentally. For reasons of industrial and since the test are qualitative values are not displayed in the y-axis on all graphs presented in this experimental work.

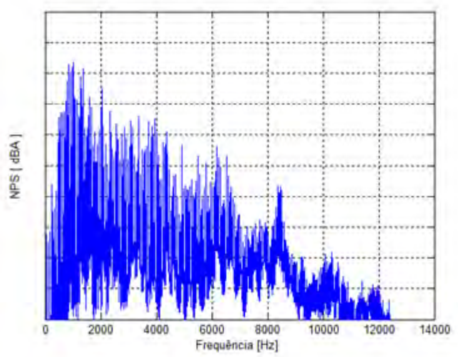

Figure 10. Typical NPS frequency spectrum of rotary compressors.

It is observed a spectrum rich in harmonics, which features a sharp discontinuity signal in the time domain, as can be seen in Figure 11, which is shown a curve of acceleration of a rotary compressor. There is a discontinuity in the motion system, which explains the large amount of harmonics present in the spectrum.

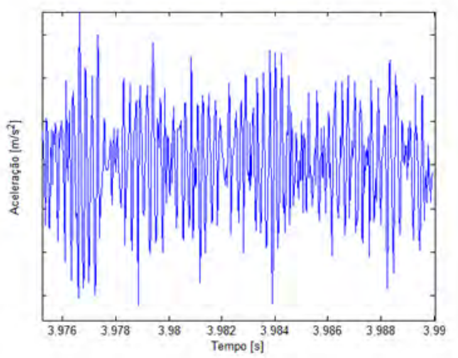

Figure 11. Acceleration time curve.

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Another fact observed is a phase modulation, or pulse, responsible for the sidebands present in any region of the spectrum. Further, it is said that this pulse is inherent kinematic rotor and is derived from the difference in speed between the shaft and roller.

With respect to this variation Fig. 12 shows the histogram of occurrences percentage by rotational frequency and the histogram with the number of occurrences of the natural frequency of the housing.

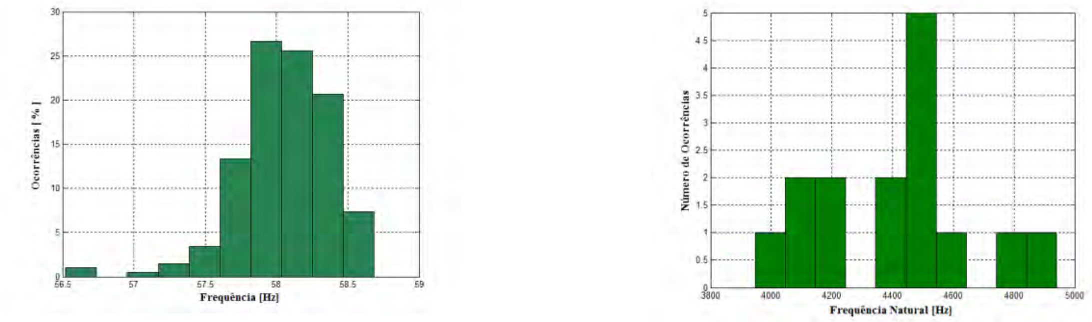

Figure 12. Histrogram of rotational frequency and natural frequency.

The variability in the frequency of rotation seems to be quite low, but it should be noted that the critical frequencies of compressor noise are above the 50^{th} harmonic. Can be observed that FRF has variation of plus or minus 10% from the fashion value

Therefore, due to the great variability of noise levels generated by theses compressors, the analysis of experimental results will be made based on noise and vibration mean calculated.

4.2 Identification of noise sources

4.2.1 Above 6000 Hz

Figure 13 shows the graphs of SPL per frequency to a Standard (blue) and a Without Valve (red).

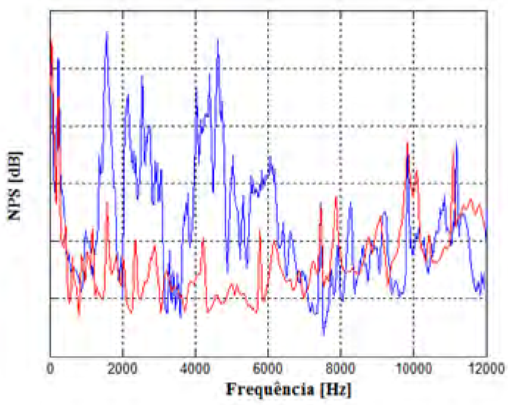

Figure 13.SPL to a Standard (blue) and a Without Valve (red).

Note that there is a great reduction of the amplitudes for the frequency range between 1000 and 6000 Hz, and since the operation of a compressor operating without vale is not expected to occur internal shocks may be credited noise levels measured above 6000 Hz to the frictional forces between the various mechanisms of the compressor and the low internal shocks.

4.2.2 Between 3000 to 6000 Hz

Figure 14 shows the graphs NWS per frequency to a Standard (blue) and a Modified Roller (red) and SPL per frequency to a Standard (blue) and Without Spring (red).

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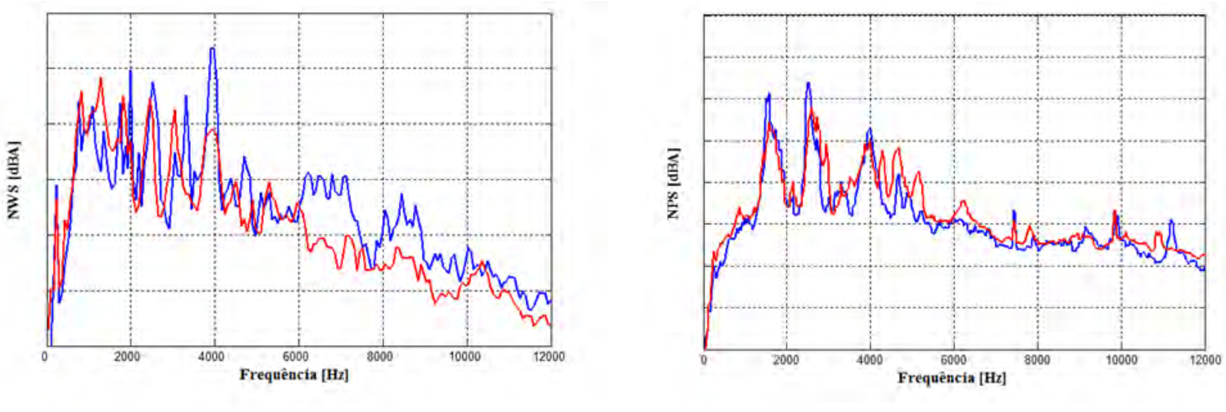
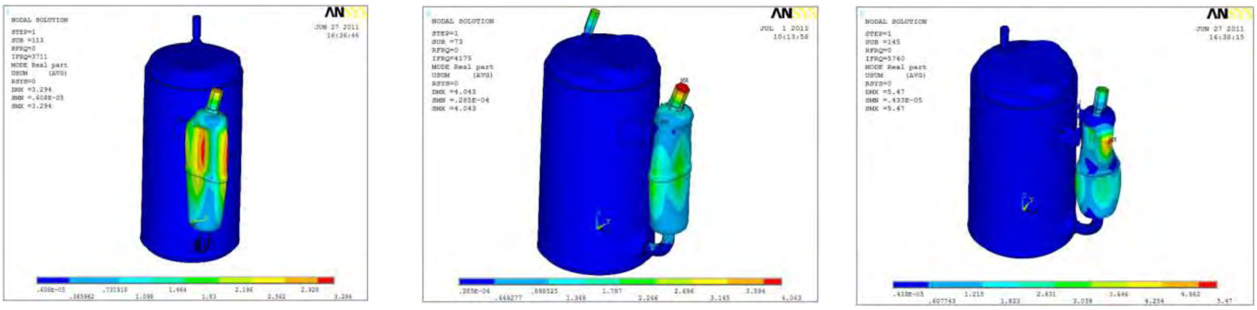


Figure 14.NWS to a Standard (blue) and a Modified Roller (red) and SPL to Standard (blue) and Without Spring (red).

Analyzing the data from Fig. 14 it is conclude that the interaction mechanisms roller-kit are mainly responsible for noise levels in the band 3000-6000 Hz. Also can be seen the importance of the control spring noise generated in the band from 3000 to 6000 Hz and below 2000 Hz, where without spring the impacts inherent in the operation have far more energy.

Another important aspect concerning this region of the spectrum is to the accumulator, although not a primary source of noise, it contributes significantly to the levels of radiated noise. Figure 15 shows some eigenmodes of vibration.



3711 Hz 4175 Hz 5740 Hz

Figure 15. Vibrate mode of the accumulator calculated via FEM.

4.2.3 Between 2000 to 3000 Hz

Figure 16 shown a FRF acceleration curve at a point on the shaft of a rotary compressor, where there is a strong resonance at the frequency of 2765 Hz, whereas under load conditions the resonance frequency tends to decrease (increase of weight) can be inferred that the noise levels in this region are due to the resonance of the compressor shaft.

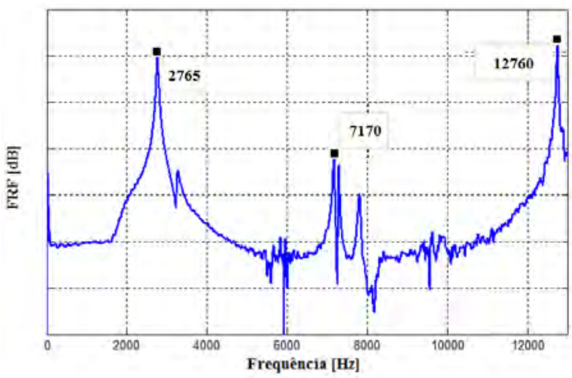


Figure 16. FRF acceleration of a rotary shaft compressor.

ISSN 2176-5480

4.2.4 Between 1000 to 2000 Hz

From the analysis of data from all treatments, it is believed that noise is predominantly derived from the impacts of the valve and the discharge mechanism of the compressor. Two analyzes results illustrate this hypothesis. At first a strong resonance in this frequency range can be observed in FRF acceleration of a point with respect to the protective cover shown in Fig. 17.

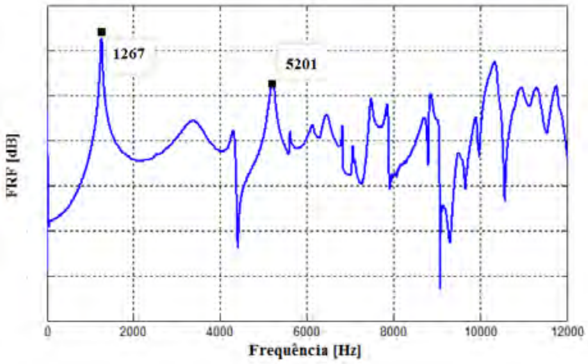


Figure 17. FRF acceleration of a protective cover.

The other hypothesis is based on the comparison os average NPS of compressors with discharge pressure variable and 30 psi suction.

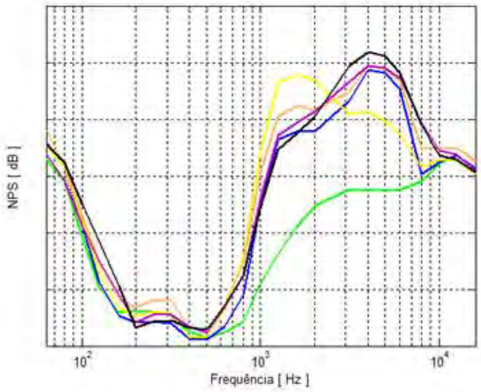
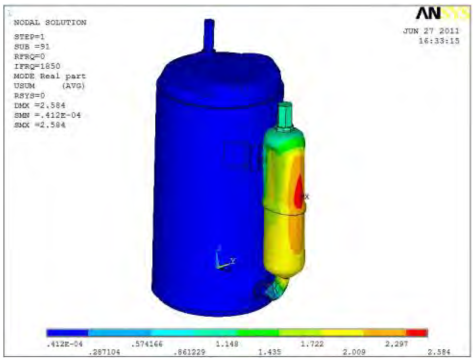
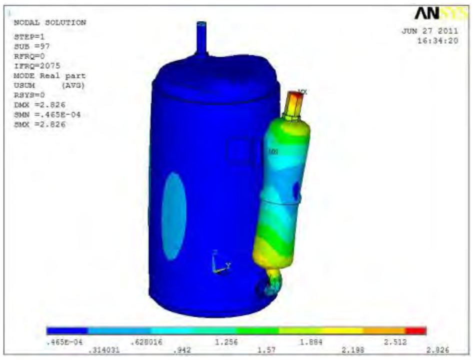


Figure 18. NPS (1/3 octave) to 30 psi suction and discharge of 50 (green), 100 (blue), 150 (yellow), 200 (orange), 250 (red) and 300 (black) psi.

It also mentions that in this frequency range, the accumulator also has an effect of noise amplifier. Figure 19 shows a strong resonance near 2000 Hz whose mode shapes.



1850 Hz



2075 Hz

Figure 19. Vibrate mode of the accumulator calculated via FEM.

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4.2.5 Between 600 to 1000 Hz

In this frequency band the dominant generation mechanism is derived from the fluid flow pulse. This statement can be readily confirmed by analysis of Fig. 20, which shows that the band 630-1000 Hz the noise levels generated by treatment with suction and discharge pressure (red) are larger than the total vacuum conditions (green) and suction vacuum and 125 psi discharge 125 psi (blue).

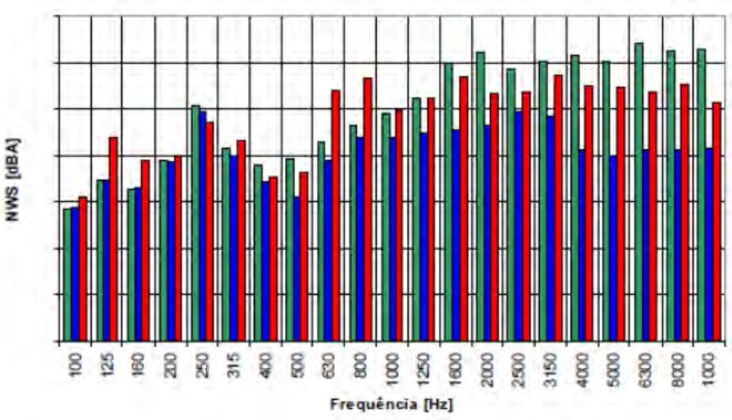


Figure 20. NWS band values in 1/3 octave for the treatment in vacuum (green), suction vacuum and discharge 125 psi (blue) and suction 30 psi and discharge 125 psi (red).

4.2.6 Between 250 to 600 Hz

Analyzing this frequency band in Fig. 21, it is observed that increasing the discharge pressure to treat with vacuum suction noise levels decrease. Since the processing suction vacuum and discharge pressure varies, there isn't the effect of fluid flow either vibration valve, it is concluded that the main mechanism of generation of noise is the interaction of spring-roller-vane.

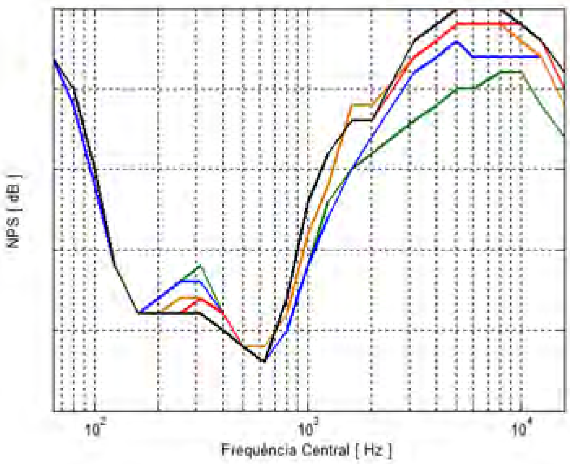


Figure 21. NPS (1/3 octave) to vacuum suction and discharge of 100 (green), 150 (blue), 200 (orange), 250 (red) and 300 (black) psi.

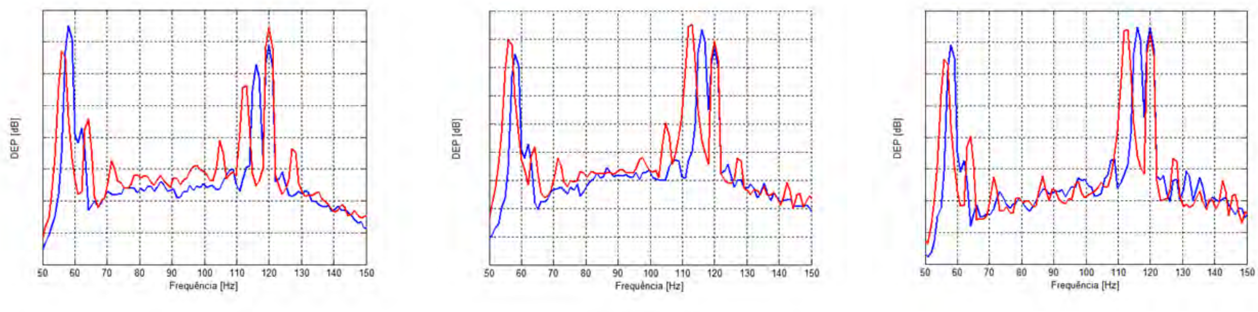
Still with reference to Fig. 20, it should be noted that the values of the NWS to full vacuum treatment is higher than for other treatments in all bands, which is a strong indication of the occurrence of internal shocks.

4.2.7 Between 60 to 250 Hz

The main sources of noise in this region of the spectrum are the origins of mechanical and electrical. In Fig. 20 is observed that in this frequency range the noise level is dependent on the pressure of the compressor, particularly in the

ISSN 2176-5480

band of 125 Hz. In Fig. 22 are shown centered at 100 Hz the acceleration spectra measured in the welding point, protective cover and housing.



Welding point   Protective cover   Housing

Figure 22. PSD of acceleration, 30 psi suction and 150 psi discharge (blue) and 30 psi suction and 300 psi discharge (red).

Although there was an increase in the levels of vibration on the entire frequency band studied with the increase in the discharge pressure were the most significant harmonics bands occurred in low frequency.

Importantly, in housing PSD the vibration amplitude of the second harmonic network (120 Hz) is high, ie, source noise on the electric motor.

5. CONCLUSION

Based on the presented results we can conclude that:

- Due to the great variability of noise levels generated by theses compressors, the analysis of experimental results was made based on noise and vibration mean calculated.
- A high amount of harmonic is attributed to multiple frequency of shaft rotation, approximately 60 Hz. The phase modulation or pulse is credited to the difference in angular velocity between roller and shaft.
- The experiments performed at Tecumseh's semi-anechoic chamber, proved useful in identifying the major sources of noise rotary compressor. It can also determine which characteristics determine each region of the frequency spectrum, where:
  - Mechanical: Due to the forces of unbalance, misalignment and electrical motor, region 60 – 250 Hz.
  - Vane: spring-vane-structure interaction, region 250 – 600 Hz.
  - Fluid: fluid flow pulse and cavitation modes, region 600 – 1000 Hz.
  - Valve: engine compressor discharge (valve and muffle), region 1000 – 2000 Hz.
  - Shaft: resonance and efforts in the bearings, region 2000 – 3000 Hz.
  - Roller: roller-vane-structure interaction, region 3000 – 6000 Hz.
  - Friction: arising from friction due to frictional forces, above 6000 Hz.
- The accumulator presents itself as an important source of radiating noise at bands centered in 2000, 4000 and 6000 Hz, and therefore its shape optimization imply improvements in noise rotary compressor.

6. ACKNOWLEDGEMENTS

The authors would like to thank CAPES (Coordination for the Improvement of Higher Education) FAPEMIG (Research Support of Minas Gerais State Foundation), FAU (University Support Foundation), FEMEC (School of Mechanical Engineering) of UFU (Federal University of Uberlândia) for their financial support and TECUMSEH for their base data and rotary compressor.

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