Noise and Vibration Characteristic Studies of Twin Screw Compressor in Different Operating Conditions

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ABSTRACT

The tonal noise or vibration of air-cooled or water-cooled chillers with rotary twin screw compressors is crucial for environmental concerns. This work aims to perform the receiver tests regarding the radiated noise and structural vibration due to the compressor in different operating conditions. The sound pressure near the compressor is recorded by the one-third octave and narrow band frequency analyzers, respectively. For environmental concern, the one-third octave band spectrum is frequently adopted to evaluate the radiated noise, while the narrow band spectrum can be used to characterize the noise sources. The axial, vertical and horizontal accelerations on the compressor shell at the motor and oil-injection locations are also monitored to correlate the structure-borne noise. Other than the comparison of spectral content between radiated noise and compressor shell vibration, the frequency response functions and the coherence functions are examined to further characterize the relation between noise and vibration. This work lays out the general steps and measurement procedures for noise and vibration evaluation in primary stage for receiver tests considering different load capacities. The vibration energy transmission path tests as well as the refrigerant pulsation effects on the compressor noise and vibration can then be carried out accordingly.

1. INTRODUCTION

Twin screw compressors are frequently used for air conditioning systems in buildings. The noise and vibration from the compressor may be annoyance and of interest for environmental concerns. Paulauskis (1999) indicated the pure tone noise effect in 1/3 octave band for the air-cooled rotary screw chiller is a critical noise problem. The pure tone effect can be characterized by examining the difference of sound pressure levels between any two adjacent 1/3 octave bands. The rules of thumb to identify noticeable pure tone effects are 5dB for between 500Hz and 10,000Hz, 8dB for between 160Hz and 400Hz, and 15dB for between 25Hz and 125Hz.

Three types of noise and vibration sources can be categorized for oil-injected twin screw compressors, including compression mechanisms, geometry and assembly of screw rotors, and operating conditions. Fujiwara *et al.* (2011) showed an effective method to detect lobe mesh vibration problems in oil-injected twin-screw compressors by comparing generated different waveforms for the identification of phenomena such as eccentricity and unbalance of drive rotor, and pitch error or damage on rotor lobes. The harmonics of lobe mesh frequency with respect to the rotating frequency can be characterized for different types of defects for twin screw compressors.

Lee *et al.* (1994) examined the noise sources for a rotary compressor that may come from the pressure pulsation due to the refrigerant gas flow, and shell vibration of compressor case excited by pressure pulsation. They performed path tests on the compressor to measure system transfer functions and coherence to correlate the noise and vibration path transmission. They found that the low frequency in outside of the compressor is affected by the refrigerant gas pulsation, and the high frequency (3-4kHz) is due to the compressor shell vibration excited by inside noise and other sources. Woo *et al.* (2008) investigated four categories of major noise sources of two-cylinder rotary compressors and discussed possible design modifications for practical implement so as to reduce the noise levels for each noise source effectively. The reduction of noise level is approximately 6dB by implementing several modifications such as increasing the natural frequency of rotor shaft, fixing the rotor-pump unit to the outer shell more tightly, improving the motor core, optimizing the shape of stator core, and altering the control scheme for motors.

Wang (1994) reviewed and adopted several sound quality parameters such as loudness, roughness, sharpness, tonality, and fluctuation strength to study noises characteristics for different types of compressors. Kim *et al.* (2000) studied the reciprocating compressor by using Frequency Modulated Noise (FMN) plot in time-frequency space and revealed the main source of noise coming from the higher mode as a local cavity resonance of the compressor due to the oil surface pattern.

Silveira (2004) presented the reduction of noise and vibration for reciprocating commercial compressors by three steps, i.e. identifying predominant source, characterizing the main path, and redesigning the compressor shells. The noise contribution of the components in the compressor in spectral bands are identified and used for selecting the redesign strategy. Oh *et al.* (1994) conducted experimental modal testing on the compressor shell to measure frequency response function and coherence function to correlate the noise characteristics of the refrigerant compressor. They showed the compressor shell vibration is strongly correlated with the radiated sound in certain frequency bands for the running compressor.

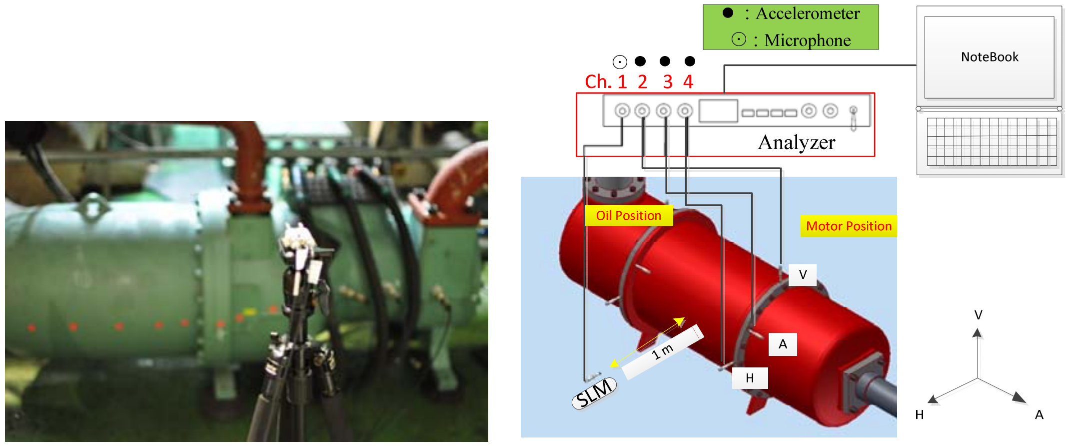
In examining the noise and vibration problems for machineries such as the oil-injected twin screw compressor studied in this work, both the receiver test and path test can be conducted, respectively, for identifying sources and transmission path effects. This work presents the primary investigation for noise and vibration evaluation on the oil-injected twin screw compressor operating in different loading capacity conditions by performing receiver tests. The 1/3 octave band frequency analysis is first shown to calibrate the major contributed bands for compressor noise as well as the overall A-weighted sound pressure level (SPL). The narrow band or linear frequency analysis is also adopted to characterize the relation between the emission noise and compressor shell vibration. The physical interpretation of spectral contents for both noise and vibration is shown. The harmonic frequencies with respect to the male screw shaft rotating speed are found dominating the noise emission. The path test on performing experimental modal testing on the compressor structure to calibrate the resonances is suggested for future works.

2. EXPERIMENTAL METHODS

Figure 1(a) shows the picture of the oil-injected twin screw compressor that has the tooth ratio 5:6 between the male and female screws. The operating conditions of the compressor for this study are set to the vaporized temperature 5℃ and the condense temperature 36℃. The electrical motor is running at the speed of 3600 rpm (60Hz) to drive the male screw shaft. This work aims to measure the noise and vibration response for the compressor operated at loading capacities from 25% to 100%.

Figure 1(b) shows the experimental setup. The precision sound level meter (CEL593.C1) with 1/3 octave band function is applied to measure the noise emission level. The microphone is located at 1m apart from the compressor as shown in Figure 1(b). The four-channel FFT analyzer (SigLab Model 20-42) is also applied to measure the sound response by the microphone (PCB-130D20) connected to Channel 1 and the vibration response for other channels in the axial (A), vertical (V) and horizontal (H) directions by three accelerometers (WR-786A) at the motor and oil injection locations on the compressor shell as shown in Figure 1(b), respectively.

The measurement setup for performing FFT in obtaining the auto power spectral density (PSD) function or simply auto spectrum is as follows: the effective cut-off frequency 5000Hz, effective spectral lines 3200, the frequency resolution 1.667Hz, the time frame length 0.64sec. The auto spectrum is calculated with the average number of 10 and applied with the Hanning window for every time frame. The frequency response functions between noise and vibration as well as the coherence function are also recorded. The slide valve of the compressor is controlled for different load capacities to be 25%, 50%, 75% and 100%.



(a) picture of compressor (b) measurement setup

Figure 1: Experimental setup of noise and vibration measurement.

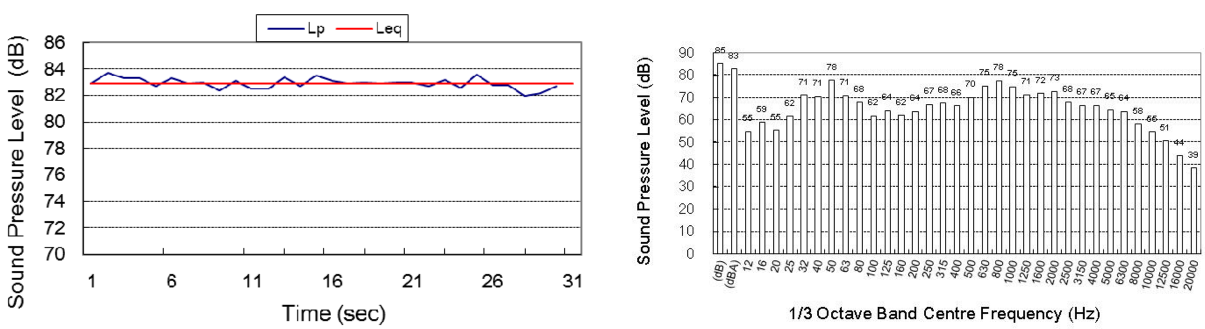
3. RESULTS AND DISCUSSIONS

This section will show the measured noise and vibration response for different operating conditions of the compressor. Both the 1/3 octave frequency band analysis and linear frequency band analysis will be discussed to show the correlation between the radiated sound pressure and the structural vibration on the compressor shell in the axial, vertical and horizontal directions.

3.1 1/3 Octave Band Analysis

Figure 2(a) and 2(b) show the sound pressure level (SPL) for 100% loading condition in both the time and 1/3 octave band frequency domains, respectively. That the SPL in time domain varies within 2 dB indicates the steady response for the compressor. The un-weighted SPL is shown in Figure 2(b). The peak amplitudes can be identified at the bands of 63 Hz, 800 Hz and 2000 Hz.

Figure 3 shows the A-weighted SPL for different loading conditions. The amplitudes in low frequency ranges are adjusted and do not contribute much to the overall A-weighted SPL. The 1/3 octave bands at 800Hz and 2000 Hz are the most dominant frequencies. As indicated in (Paulauskis, 1999) by comparing the relative SPL difference between adjacent bands less than 5 dB, there is not much tonal effect for the compressor in all loading conditions. As shown in Figure 4, the A-weighted SPLs are generally increased with the higher loading capacities. The SPLs at the central frequency bands of 800 Hz and 2000 Hz that are strongly correlated to the A-weighted SPL for different loading conditions are the major sources of noise bands.



(a) time domain (b) 1/3 octave band

Figure 2: Sound pressure level for 100% loading condition in time and 1/3 octave frequency domains.

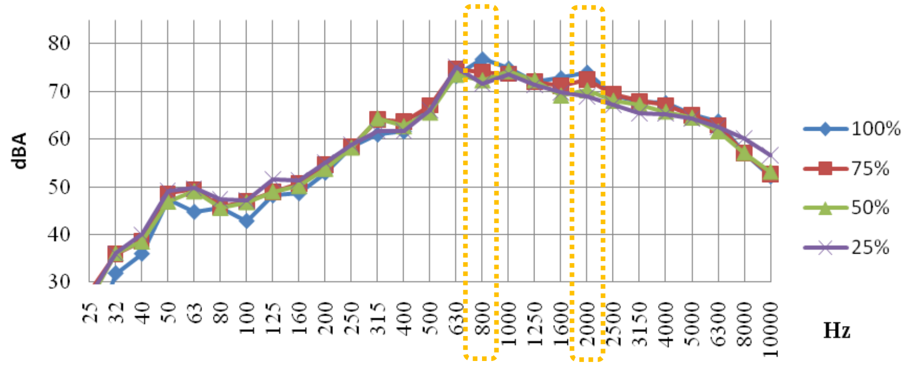


Figure 3: A-weighted sound pressure levels for 25-100% loading conditions in 1/3 octave band.

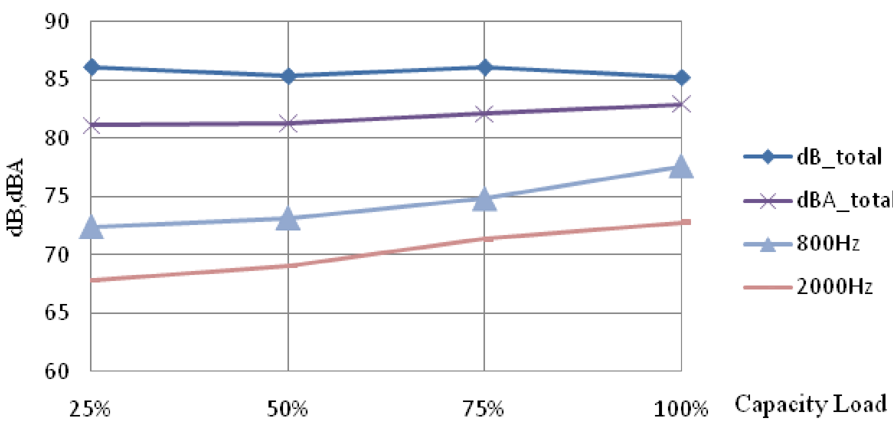
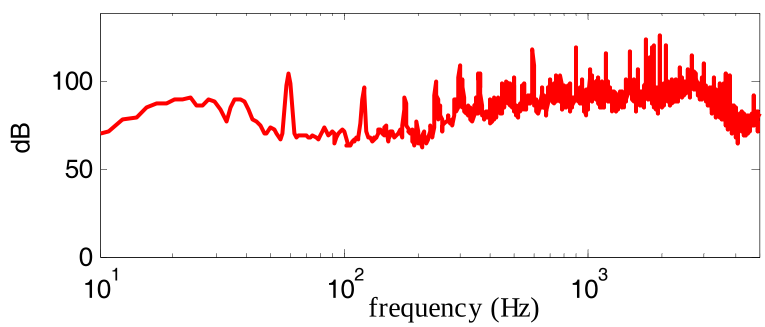
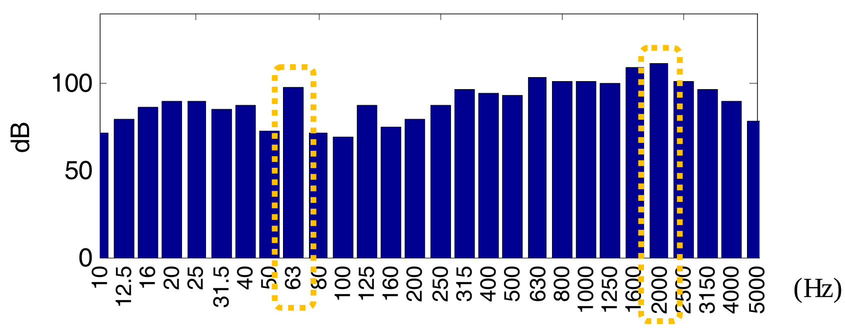


Figure 4: Comparison of sound pressure levels for different loading conditions in 1/3 octave bands.

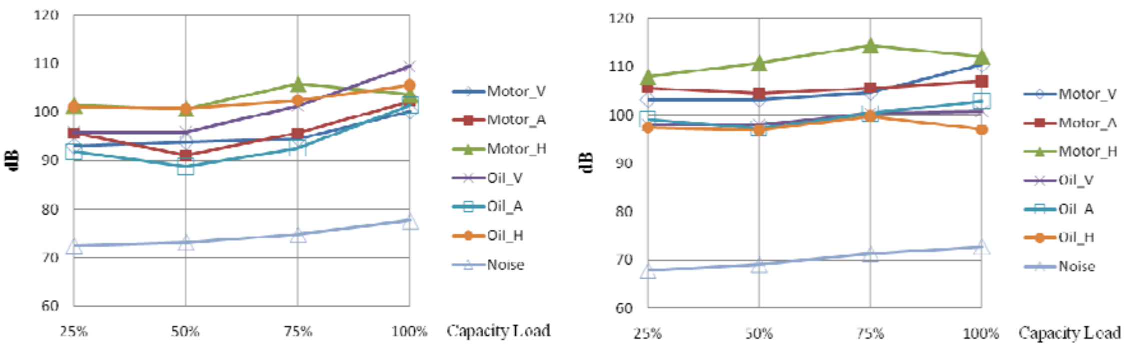


(a) Linear Spectrum



(b) 1/3 Octave Band

Figure 5: Vibration spectrum for 100% loading condition in the vertical direction at motor location.



(a) 800 Hz (b) 2000Hz

Figure 6: Comparison of noise and vibration spectrum at central frequencies 800Hz and 2000Hz for different loading conditions.

Figure 5(a) and 5(b) show the vibration spectrum in the linear and 1/3 octave bands, respectively, for the acceleration level in the vertical direction at the motor location. The relatively high peak amplitudes can also be observed at those bands, such as 63Hz and 2000Hz, where the SPLs also reveal higher values. As shown in Figure 6, the vibration levels for different loading capacities in different directions in terms of A(axial), V(vertical) and H(horizontal) at the Motor and Oil injection locations can be observed generally complying with the noise level at the bands of 800 Hz and 2000Hz. The compressor noise level can be shown directly associated with the vibration levels of the structural shell vibration of the compressors. This may suggest that the reduction of vibration levels on the structure can decrease the noise emission levels for the compressor.

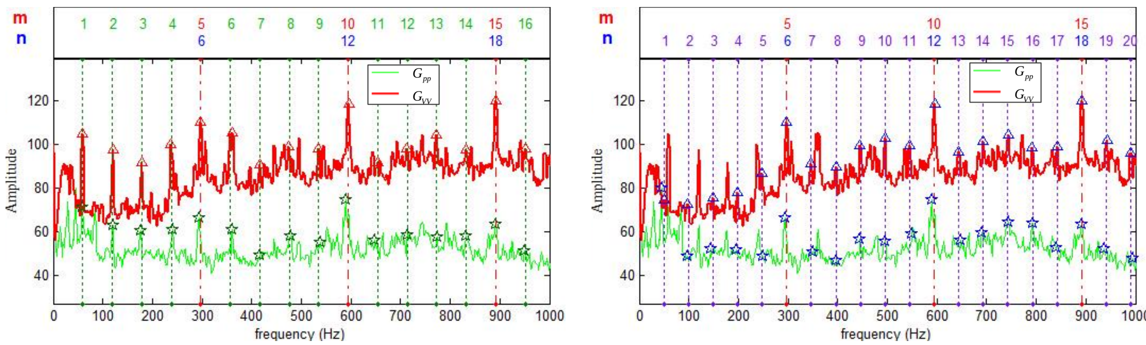
3.2 Linear Spectrum Analysis

The 1/3 octave band analysis is generally suitable for noise evaluation related to human hearing mechanism. The linear or narrow band spectrum analysis is required to identify the sources of noise and vibration. This section will show those linear spectra for both noise and vibration levels in different operating conditions of loading capacities.

Figure 7 shows the comparison of noise and vibration spectra for 100% loading condition below 1000 Hz to calibrate those peaks coming from. As shown in Figure 7(a), the integer multiply values of harmonics with respect to the rotating speed 3000 rpm, i.e. 60Hz, driving the male screw shaft is specified as ***m*** on the top of the plot. As one can observe, noise and vibration levels are much higher at the ***m***–th harmonic frequency with respect to the male screw shaft rotating speed. Both noise and vibration levels reveal the similar pattern and correspond to each others, especially at those harmonic frequencies. Also, higher noise and vibration levels can be seen at the harmonic numbers of ***m***=5, 10 and 15 that are multiply values of 5 due to the tooth number of male screw. The coincident female screw rotating harmonic numbers are the multiply values of ***n***=6 for the female screw teeth being 6. It is interesting to note that as shown in Figure 7(b), the harmonic number ***n***, which is the integer value of frequency ratio with respect to the female screw shaft rotating speed, is shown on the top of the plot. The peaks of noise and vibration levels are not at the harmonic number ***n*** except for ***n***=6, 12 and 18 where are coincident with ***m***=5, 10 and 15. The most dominant peaks of harmonic frequencies come from the male screw shaft effects. And, the coincident male and female screw shafts harmonics dominantly contribute to the noise and vibration of the compressor.

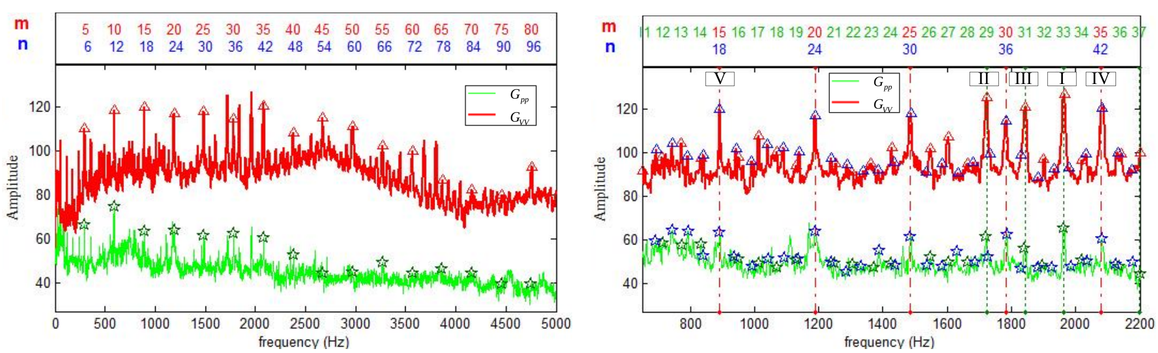
As shown in Figure 3, the frequency above 5000Hz in 1/3 octave bands contribute less to the overall noise level. Figure 8(a) shows the comparison of linear spectra below 5000Hz frequency. The noise levels below 2500Hz are most annoyance bands. The high noise levels generally come from the coincident male and female screw shafts harmonics as discussed in Figure 7. It is interesting to examine the frequency ranges between 800Hz and 2200Hz as shown in Figure 8(b). The order of peak amplitudes is specified by roman numerals in sequence. Other than those peaks at ***m***=15, 20, 25, 30 and 35 that are the coincident male and female screw shafts harmonics, there exists three peaks at ***m***=29, 31 and 33 near the frequency of 1800Hz that results in the higher peak amplitude of SPL at the central frequency 2000Hz as shown in Figure 3. The unexpected high peak values at ***m***=29, 31 and 33 are not clear at the moment. However, the structural resonance effects can be suspected and needed to perform path test, such as the experimental modal testing on the compressor, to calibrate those unexpected peaks.

For examining the vibration levels in A, V and H directions in comparison to the noise level, Figure 9(a) and 9(b) present the auto spectra and coherence function plots, respectively. The coherence function indicates the correlation between two signals. For the coherence value close to 1, the two signals are strongly correlated. On the opposite, the two signals are rarely correlated for the coherence value close to zero. In Figure 9(a), at the frequency band of 2000Hz, the vibration levels in three directions are about the same. The peaks of sound spectrum are correlated to those of vibration spectra in three directions. Also shown in Figure 9(b), the coherence values are larger than 0.8 at ***m***=29, 30, 31 and 33. This again indicates the direct relation between noise and vibration, in particular for those peaks. For the concern of noise reduction for the compressor, the frequency band of the central frequency at1800Hz is the major contributions that come from those peaks at ***m***=30, 31 and 33. The structural resonances are presumed and will be investigated.



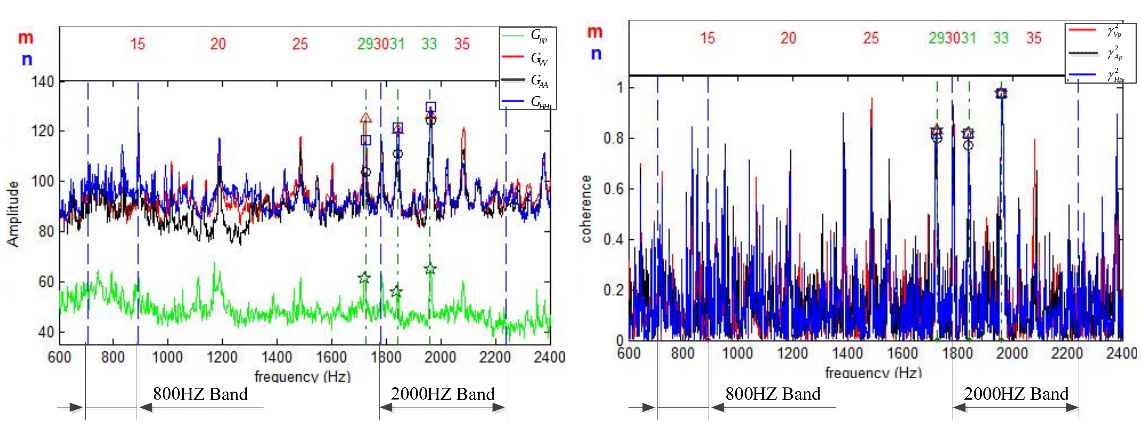
(a) harmonics for male screw frequency (b) harmonics for female screw frequency

Figure 7: Comparison of noise and vibration spectrum for 100% loading condition below 1,000Hz.



(a) 0-5000Hz (b) 800-2200Hz

Figure 8: Comparison of noise and vibration spectrum for 100% loading condition.



(a) auto spectrum (b) coherence function

Figure 9: Comparison of noise and vibration spectrum and coherence functions for 100% loading condition.

4. CONCLUSIONS

This work investigates the noise and vibration characteristics of the twin screw compressor running at different loading capacity conditions from 25% to 100%. Both the 1/3 octave band and linear spectra are measured to calibrate the relation between noise and vibration of the compressor. Some observations are summarized as follows:

- The overall A-weighted SPL of the compressor increases with the loading capacity. The dominant frequency bands are identified at the central frequencies 800Hz and 2000Hz in 1/3 octave bands.
- The most dominant frequency bands in SPL spectra are also the higher vibration levels in the same frequency bands. The linear vibration spectrum can be clearly interpreted and dominated by the harmonics with respect to the male screw shaft rotating speed frequency.
- The noise and vibration spectra for the compressor are also shown strongly correlated, especially for those peaks at some harmonics with respect to the coincident male and female screw shafts speed frequency.
- In the critical frequency band of 2000Hz where is the major source of noise and vibration for the compressor, other than the coincident male and female screw shafts speed frequency three unexpected peaks are found and needed to further investigate.

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