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# **Measurement of gas pulsation in the discharge line of a compressor for HVAC: Method to calculate the progressive acoustic wave from the compressor and reflected wave from the installation.**

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## **ABSTRACT**

The pulsation produced by a compressor and measured in the downstream pipeline is an important characteristic for system design. Unfortunately, this is also a difficult parameter to measure. In fact, the level is depending on the interaction with the system and the amount of energy due to reflection.

Currently, they are 2 types of methods to achieve good measurements. One requires a large amount of hardware to create an anechoic termination. The measurement is done in the time domain recording peak-to-peak value. The other method is based on signal analysis in the frequency domain. This paper will propose a new numerical method for pulsation measurements of compressors using both the time and frequency domain.

The frequency domain is used to separate the incident and reflected wave based on the complex representation of acoustic waves travelling in the pipe.

The final target is to measure the progressive wave due to the compressor without the use of an extended length of the piping.

## **1. DESCRIPTION OF TEST BENCH OF REFRIGERATION COMPRESSOR**

### **1.1. Installation**

Refrigeration and air-conditioning compressors are tested on load stands during the development phase but also for production audit (figure 1). That equipment simulates a HVAC or refrigeration system and makes it possible to run the compressor at any condition inside the operating map. Suction and discharge pressures and temperatures define the condition. Those features can be adjusted by the operator using expansion and bypass valves, and water flow. The bench test is typically used to make sound, vibration and gas pulsation measurements (usually in the discharge line) of the compressor. The pulsation is measured at a distance equal to 10 times the pipe internal diameter using a dynamic pressure sensor giving the peak-to-peak value in the time domain. The piping is done in such a way as to minimize the acoustic wave reflection. The cross section is almost constant from the compressor towards the condenser, every change is made using a smooth convergent/divergent profile. The condenser is the first major change of section, followed by the bypass and expansion valves. These are the main contributors to the reflection of acoustic pressure waves back into the discharge line. A wide range of compressors are tested on the same equipment to avoid proliferation of the piping and to minimize the testing cost. The consequence is that the dimension of the piping from the compressor is generally not comparable to the dimension of the piping of the load stand. The compensation is done by cross section changes generating reflections. In other words, the stand is optimized for one model of compressor. Only testing other models will generate wave reflection.

The standard method is to extend the refrigerant line up to 12.2 m without cross sectional change or sharp bends[1]. With this configuration, very low attenuation in the refrigerant is assumed. However, this test arrangement is neither realistic nor practical.

Typically we do not experience severe reflection with the small capacity compressors because the flow and speed of the refrigerant are low. With large capacity compressors, the flow and velocity increase and every discontinuity of the piping creates a restriction, which induces more reflection and disturbs the pulsation measurement. The presence of the reflected wave can seriously corrupt the reading of the single pressure sensor. This is due to a standing wave created by the combination of the reflected wave with the incident wave. The reflection will depend on the operating condition of the system. In fact, the stand adjustment using the expansion and by pass valves will control the pressures, temperatures (that define the density and the speed of acoustic wave) into the system and also the quantity of refrigerant. The admittance of the system is linked to the position of the valves, density and temperature of the refrigerant and quantity of refrigerant in the load stand. All those arguments make the measurement difficult to perform with just one sensor, in the time domain. Our experience shows a large variability in the results that make this measurement unreliable.

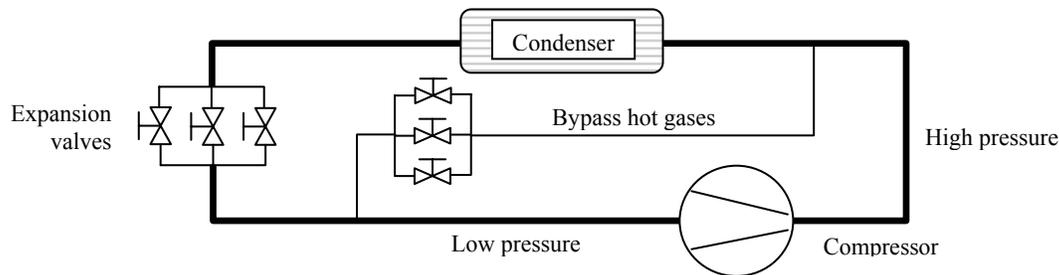


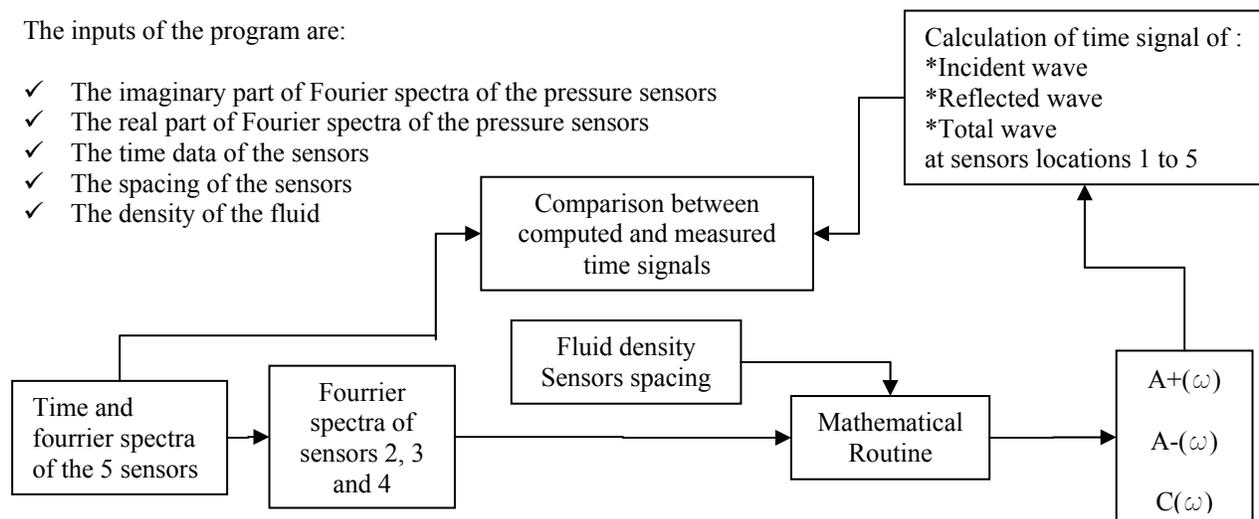
Figure 1

## 2. MATHEMATICAL POSITION OF THE PROBLEM

The time domain is not sufficient to define correctly the pulsation level. The separation of the reflected and incident waves can only be done numerically in the frequency domain. The development of the method considers 5 sensors equally spaced by 10 or 20 cm (see figure 5). The sensors at the second, third and fourth positions are used to calculate the wave parameters. The sensors Nr. 1 and 5 are used to verify that the model of the acoustic wave is acceptable. A complex FFT is performed with a Hanning time window over 800 Hz and using 1600 points which makes the resolution 0.5 Hz. We perform also a linear averaging of 100 samples. The sensor in the middle (nr. 3) is the space reference ( $x=0$ ).

The inputs of the program are:

- ✓ The imaginary part of Fourier spectra of the pressure sensors
- ✓ The real part of Fourier spectra of the pressure sensors
- ✓ The time data of the sensors
- ✓ The spacing of the sensors
- ✓ The density of the fluid



The analyser directly provides the frequency spectra and the time data. The spacing of pressure sensors is defined by the design of the test bench. The density of the fluid is calculated using a thermodynamic model [2]. That value has to be recalculated for every condition because it's a function of pressure and temperature.

The core of this new method is to combine the frequency and time domain to validate the model of wave propagation based on the frequency analysis but also to generate time data that are easy to be shared.

**2.1. Wave equation**

For every location along the discharge piping, the total acoustic pressure wave is the summation of the incident (coming from the source) and the reflected wave (from the system).  $A^+$  is the complex amplitude of the incident wave and  $A^-$  is the complex amplitude of the reflected wave and  $\tilde{c}$  the complex speed of sound.  $X$  represents the position of the pressure pick up along the discharge line of the system ( $x=0$  is the sensor 3) and  $\omega = 2 * \Pi * f$ .

$$P(x, \omega) = A^+(\omega) e^{-i\frac{\omega}{\tilde{c}}x} + A^-(\omega) e^{i\frac{\omega}{\tilde{c}}x} \tag{1}$$

The 3 unknowns are the amplitude of incident and reflected wave and the speed of sound. To solve that system, we will use the signal from the 3 central pressure sensors (sensors labelled 2, 3 and 4). The sensors 1 & 5 will be used to validate the model.

**2.2. Model**

The middle sensor is the reference ( $X=0$ ). The model can be expressed by a 3 x 3 matrix (3 inputs  $P_2, P_3$  and  $P_4$  and 3 outputs  $A^+, A^-$  and  $\tilde{c}$ ).

We apply the wave equation to the sensors 2, 3 & 4 for frequency  $f$  ( $\omega = 2 * \pi * f$ ).

$$\tilde{c} = \frac{(\omega \times a)}{\text{acos}\left(\frac{1}{2} \frac{P_2(x, \omega) + P_4(x, \omega)}{P_3(x, \omega)}\right)} \tag{2}$$

The speed of sound is considered to be the same for both incident and reflected wave. This is a complex number and the imaginary part represents the attenuation of the acoustic wave through the piping. If we assume negative exponential attenuation (rate  $-\alpha$ ):

$$P = Ae^{-\alpha x} e^{-ikx} \rightarrow P = Ae^{-i\frac{\omega - i\alpha c}{c}x} \quad P = Ae^{-i\omega\frac{1 - i\alpha c}{c}x} = Ae^{-i\omega\left[\frac{1}{c} - \frac{i\alpha}{\omega}\right]x} \approx Ae^{-i\omega\frac{1}{c\left(1 + i\frac{\alpha c}{\omega}\right)}x} \tag{3}$$

with  $\tilde{c} \approx c\left(1 + i\frac{\alpha c}{\omega}\right)$

That speed of sound is calculated first, which reduces the dimension of the matrix to a 2x2 system to solve the equations using the pressure at the points 3&4.

$$\begin{bmatrix} 1 & 1 \\ e^{-i\frac{\omega}{\tilde{c}}a} & e^{i\frac{\omega}{\tilde{c}}a} \end{bmatrix} \times \begin{pmatrix} A^+ \\ A^- \end{pmatrix} = \begin{pmatrix} P_3 \\ P_4 \end{pmatrix} \tag{4}$$

The matrix (4) has to be solved for every frequency. Additionally, the equivalent admittance of the piping can be calculated. This parameter represents mathematically the response of the system.

$$\text{Admittance}(x, \omega) = \frac{\text{Fluid Velocity}}{\text{Fluid Pressure}} \approx \frac{1}{\rho\tilde{c}} \frac{A^+(\omega)e^{-i\tilde{k}x} - A^-(\omega)e^{i\tilde{k}x}}{A^+(\omega)e^{-i\tilde{k}x} + A^-(\omega)e^{i\tilde{k}x}} \quad \text{with } \tilde{k} = \frac{\omega}{\tilde{c}} \tag{5}$$

The time signal can be recalculated from the frequency domain of the pressure field after the calculation of the fluid velocity, the amplitude of incident and reflected wave. The computation of the solver is programmed in MATLAB®.

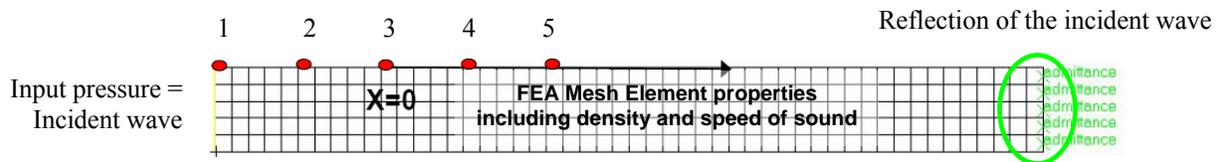
$$p(x,t) = \Re \left( \sum_{n=0}^{\infty} P(x, \omega_n) e^{i\omega_n t} \right) \tag{6}$$

### 3. VALIDATION OF THE MODEL

The validation consists of two steps. The first one is designed to verify that the mathematical routine provides correct results based on an ideal case establish with simulation tool. The concept is to calculate the incident, as well as the reflected wave and speed of sound and to verify that the value are the same as the simulation inputs. The second test considers a real experimental hardware supposed to reproduce a free field condition.

#### 3.1. FEA model with ACTRAN® software

To verify this method, we decided to conduct a numerical simulation to compare the results from the mathematical routine with the FEA simulation done with ACTRAN® software (from Free Field Technologies [3]). The idea is to generate a pressure field into a straight tube closed by a known admittance that will reflect a portion of the incident wave. The Pressure can be calculated at any location using the FEA solver in the frequency domain, taking into account the reflection due to the defined admittance. The chosen fluid for that validation is the air without attenuation (speed velocity is real and constant at 340 m/s).



The validation will be done using the total pulsation time signal calculated by ACTRAN® at the 3 sensors' location (P2, P3, P4). The routine will calculate the speed of sound, amplitudes of incident and reflected wave and admittance for every frequency. The validation is done if the amplitude of the incident wave, reflected waves and speed of sound calculated by the mathematical routine are similar to the FEA inputs and admittance of the termination (defining the reflected wave in the FEA). The frequency range of interest is 0 – 500 Hz. High frequency errors are due to large spacing between the sensors and low frequency limitation is due to very small phase shift making the calculation of the speed of sound numerically unstable.

The equation (2) shows that, if  $P2 \approx P3 \approx P4$ , the speed of sound tends to infinity. The sensor spacing which is small compared to the wavelength limits the accuracy in the low frequency range.

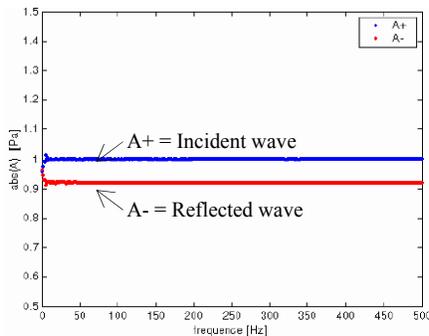


Figure 3: Amplitude of reflected and incident wave are similar to Actran simulation

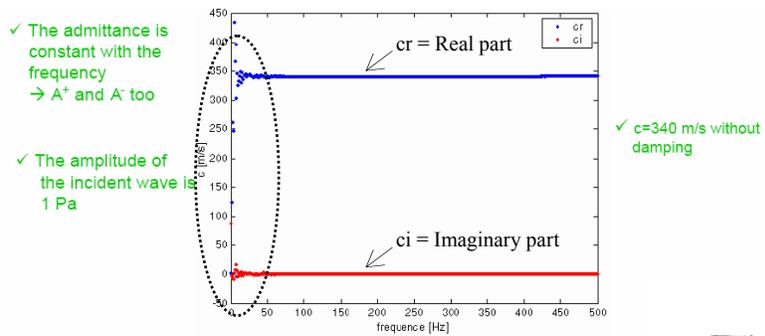


Figure 4: Speed of sound (real and imaginary part) small instability below 25 Hz

The speed of sound recalculated by the routine and based on the Fourier spectra from the FEA simulation corresponds to the inputs of the FEA. The speed of sound is only a real number. There is no attenuation in the medium. We notice less accuracy in the low frequency range (below 75 Hz) as expected due to the sensors' spacing. The divergence becomes large below 25 Hz. The limitation at higher frequency is not so problematic because the calculation seems to be acceptable up to 425 Hz. That frequency range covers the typical excitation from compressor.

### 3.2. Experimental Validation

The objective of the experimental validation is to demonstrate that the calculated incident wave is not affected by the presence of reflected wave and is equal to the measured pulsation in free field condition.

The free field condition is approximated by a long piping (13 m) without any sharp bend and discontinuity. With this configuration, the measured pulsation level is only due to the incident wave from the compressor and the level is the same along the piping since there is no standing wave due to reflection. The inner diameter stays the same all along the discharge piping length. The design of the tube allows introducing a perforated plate 2.5 m after the compressor rotalock fitting. The purpose is to create waves reflection and standing waves between the compressor and this plate. The validation is successful when the routine calculate incident wave values equal to the free field configuration.

The pipe inner diameter is 25 mm. The sensors are spaced by 100 or 200 mm. This special setup is design for the low and high frequency pulsation knowing that the theoretical value of speed of sound is 174 m/s.

The compressor (air-conditioning application - 43 m<sup>3</sup>/h) ran the reference condition ARI (7.22/54.4/18°C; discharge pressure = 22.1 BarA - Dew point) with R407C refrigerant, at 50Hz (power supply), which is about the maximum capability of the testing stand.

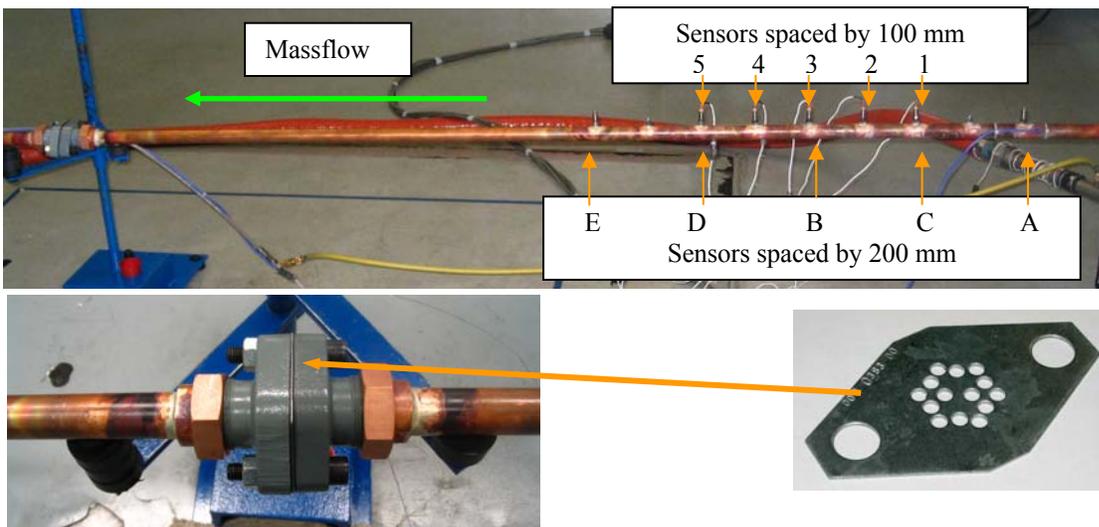


Figure 5: Installation of the sensors in the experimental test bench  
No cross section change when the muffler is not inserted

#### 3.2.1. Comparison between measured and calculated pulsations.

The first comment concerns the assumed anechoic termination by the long uniform piping (supposed to guarantee the absence of a reflected wave). This is obviously not the case. Standing waves are created by the recombination of the wave reflected by the system and the incident wave from the compressor, even without muffler. This simple measurement method is unreliable even if we try to use a good setup that minimizes the reflection by design.

The variation of the pulsation measurement along the array of sensors using the simple method based on the time signal of sensors is at least 19% (Spacing 100 mm, no muffler). This variation of the measured value can be 37% in the test done with the muffler and the sensor spacing of 200 mm.

For the same test, the incident wave calculated using the routine is only varying by maximum 3%. This mathematical method has the advantage to be less sensitive to the setup and gives much more consistent results.

The second comment concerns the effect of the muffler being inserted in the discharge line. It generates a strong cross section change and it increases the pulsation measurement in the portion of the piping between the compressor and itself due to wave reflection (+18% for 100 mm and +14% for 200 mm). The variation of the pulsation measurement along the piping increases by 8% (from 19 to 27%) and 6% (from 31% to 37%) when the muffler is inserted.

The results of calculated incident waves (in the time domain, based on the Fourier spectra) are more stable. The variation along piping is limited to maximum 3% (test with muffler and spacing 200 mm). The very important fact is the low sensitivity of the calculated incident wave regarding the presence of the muffler. The pulsation of the incident wave varies by 2% (from 111% to 109% - spacing 100 mm) and 4% (from 103% to 107% - spacing 200 mm) due to the reflection at the muffler. The variation is much lower compared to the time based method. This is expected since the incident wave is only depending on the compressor output which remains the same for the entire test.

Now considering the effect of position of the sensors and the reflection due to the muffler, we understand that the use of the presented routine may avoid to make strong under or overestimation of the real input of the compressor.

Table 1: Pulsation level compared to references (average of measured values without muffler)

<b>Spacing of 100 mm</b>		Testing without muffler						Reference
Distance from the compressor [mm]		300	400	500	600	700	Average	Maximum - Minimum
<b>Measurement</b>	<b>Total Wave</b>	<b>113%</b>	<b>100%</b>	<b>94%</b>	<b>94%</b>	<b>100%</b>	<b>100%</b>	<b>19%</b>
<i>Calculated</i>	<i>Total Wave</i>	117%	108%	101%	96%	102%	105%	21%
<b>Calculated</b>	<b>Incident Wave</b>	<b>110%</b>	<b>110%</b>	<b>111%</b>	<b>112%</b>	<b>112%</b>	<b>111%</b>	<b>2%</b>
		Testing with muffler						
Distance from the compressor [mm]		300	400	500	600	700	Average	Maximum - Minimum
<b>Measurement</b>	<b>Total Wave</b>	<b>130%</b>	<b>124%</b>	<b>121%</b>	<b>112%</b>	<b>104%</b>	<b>118%</b>	<b>27%</b>
<i>Calculated</i>	<i>Total Wave</i>	128%	117%	115%	108%	101%	114%	27%
<b>Calculated</b>	<b>Incident Wave</b>	<b>110%</b>	<b>109%</b>	<b>109%</b>	<b>108%</b>	<b>109%</b>	<b>109%</b>	<b>1%</b>
<b>Spacing of 200 mm</b>		Testing without muffler						Reference
Distance from the compressor [mm]		200	400	600	800	1000	Average	Maximum - Minimum
<b>Measurement</b>	<b>Total Wave</b>	<b>119%</b>	<b>106%</b>	<b>90%</b>	<b>97%</b>	<b>88%</b>	<b>100%</b>	<b>31%</b>
<i>Calculated</i>	<i>Total Wave</i>	116%	109%	93%	91%	85%	99%	31%
<b>Calculated</b>	<b>Incident Wave</b>	<b>103%</b>	<b>102%</b>	<b>103%</b>	<b>102%</b>	<b>102%</b>	<b>103%</b>	<b>1%</b>
		Testing with muffler						
Distance from the compressor [mm]		200	400	600	800	1000	Average	Maximum - Minimum
<b>Measurement</b>	<b>Total Wave</b>	<b>129%</b>	<b>127%</b>	<b>118%</b>	<b>104%</b>	<b>92%</b>	<b>114%</b>	<b>37%</b>
<i>Calculated</i>	<i>Total Wave</i>	131%	125%	115%	99%	96%	113%	35%
<b>Calculated</b>	<b>Incident Wave</b>	<b>108%</b>	<b>107%</b>	<b>107%</b>	<b>107%</b>	<b>105%</b>	<b>107%</b>	<b>3%</b>

### 3.2.2. Comparison between measured and calculated time signals

Another target of the method is to reproduce the original signal based on frequency analysis. The calculation of the total wave back in the time domain is based on combination of the calculated incident and reflected waves in the frequency domain.

We will compare the time signals from experimental measurement with the 100 spacing, without muffler and the mathematical model. The good correlation between the calculated and measured time signal is a part of the validation of the method (figure 6a&b).

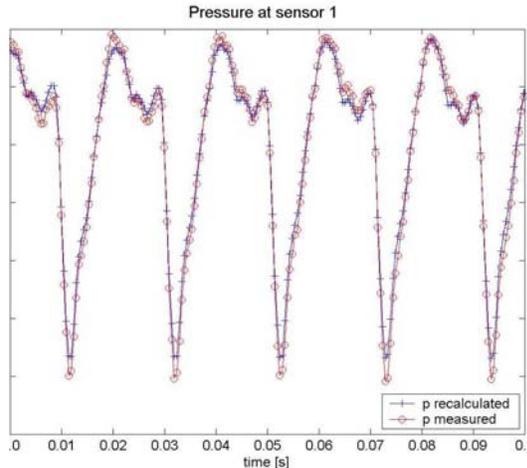


Figure 6a

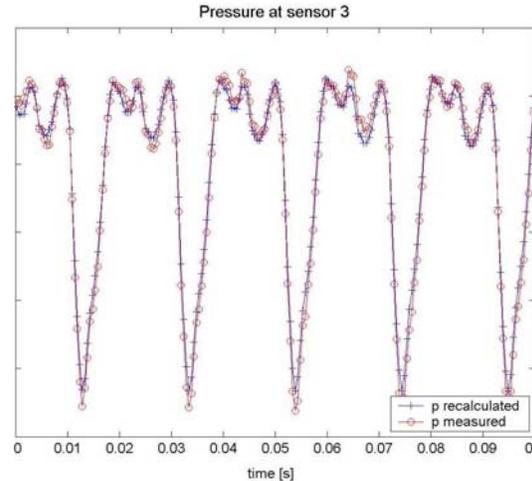


Figure 6b

### 3.2.3. High frequency limitation

When the distance between the sensors represents more than  $170^\circ$  of phase, the accuracy of the phase measurement is poor and the calculation of the speed of sound and wave parameters are no longer accurate. The 100 mm spacing stays consistent up to 700 Hz. The degradation in the low frequency is acceptable compared to that of the 200 mm spacing. We just notice that the variability is a little bit larger with the 100 mm spacing. The lowest excitation of the compressor is 50 Hz. If lower frequencies have to be considered, another spacing pattern is required.

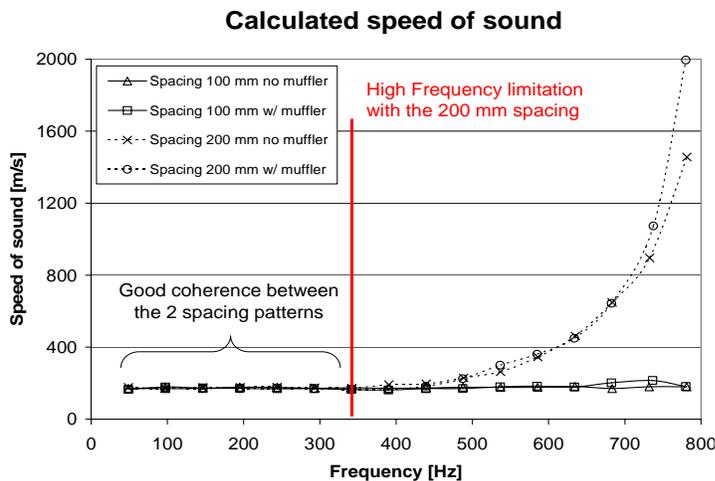


Figure 7

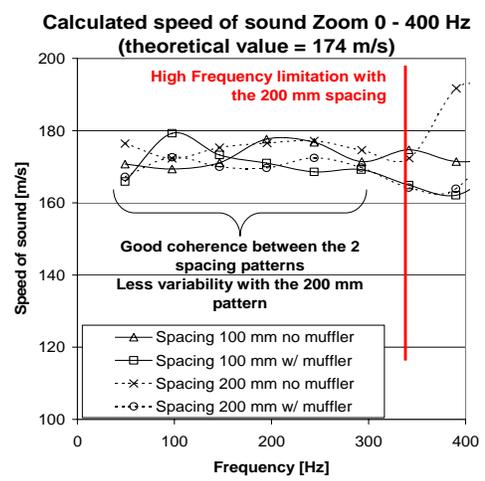


Figure 8

### 3.3. Conclusion of the validation

The routine used to separate the incident and the reflected pressure wave has been successfully validated in the frequency and time domains by comparison with experimentation. The incident wave is not affected by the reflected

wave and has a marginal sensitivity with the setup. The speed of sound is comparable to the theoretical value. The experimental and computed time signals are similar. Unfortunately, it was not possible to reproduce a free field condition to establish the baseline.

### 3.4. Remark

The frequency spectra can be recombined using the full frequency band. The summation will incorporate all the frequency components and the computer time can be very large. An alternative is to select only the components that are at the harmonics of the running speed. The calculation will be significantly shorter but that would have an impact on the accuracy of the time signal. Our recent experience with that method indicates that this approximation is acceptable.

## 4. CONCLUSIONS AND RECOMMENDATIONS

The advantages of the presented method are multiple. We have shown that it is possible to make correct measurements of the pulsation level of compressors using a limited hardware. Of course, more sophisticated analysis including frequency domain is required. This measurement method may be very attractive for the large compressors that require a very long uniform piping to approach the free field condition. For those tests, it would be really easier to incorporate sensor patterns in the existing tooling than to rebuild a complete pipe (expensive for large diameter) system just for the pulsation testing.

The validation meets most the addressed targets:

- Stability of the incident wave
- Correlation between experimental and computed time signals
- Calculated speed of sound

The method presented is a new approach to pulsation measurement. Of course, some activities are still required to complete the validation of the method:

- Repeat the tests in a true free field condition
- Evaluate the impact of the time weighting and averaging
- Optimise the data acquisition setup (frequency resolution)
- Repeatability analysis with different compressor size

Finally, we may imagine some improvements to the method:

- Calculation of the fluid density based on measurement instead of theoretical estimation
- Evaluation of oil circulation
- Special arrangement of the sensor spacing to better cover both high and low frequency

## NOMENCLATURE

$\omega$	$= 2\pi f$	angular speed	(rad/s)
x		distance from the reference	(m)
a		distance between pressure sensor (spacing)	(m)
$\alpha$		wave attenuation number	(m <sup>-1</sup> )
$\tilde{c}$		speed of sound (Complex)	(m/s)

## REFERENCES

- [1] ANSI/ARI 530-89, Method of measuring sound and vibration of refrigeration compressors  
 [2] REFLIB for Excel, version 2.1, ILK Dresden (D)  
 [3] ACTRAN, Free Field Technologies, Place de l'université, 16, B-1348, Louvain-la-Neuve, BELGIUM - [www.fft.be](http://www.fft.be)