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Sound Quality improvement through tonal noise reduction of a screw air compressor with an extended bandwidth tuned resonator

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ABSTRACT

Pressure pulsations are considered to be a major cause of concerns in positive displacement machineries, more specifically to compressors. Low frequency pressure pulsations often results in mechanical vibrations of the piping systems causing reliability concerns like welding failures and joint leakages resulting in significant energy losses. High frequency pressure pulsations (> ~ 1kHz) often result in radiated noise with a tonal radiated noise behaviour, resulting in irritating perception causing concerns to perceived sound quality of compressors. In this paper, a tonal noise generated by the pressure pulsation excitations from an oil free screw compressor is contained by acoustically tuned resonator designed based on the transfer matrix approach and subsequently optimized using numerical analysis. The higher order modal behaviour of the resonator is exploited to enhance the bandwidth of the attenuation. The experimentally validated resonator on implementation in the compressor package reduces the tonal noise contribution by more than 20dB in the near field of the compressor in the frequency range of interest. This subsequently results in a reduction of the overall sound power of the compressor package by 9dB. This paper details the modelling through analytical and numerical approach, the subsequent experimental validation results at a component and system level of the compressor.

KEY WORDS

Pressure pulsations, Noise, Tonal, Resonator, Impedance, Sound Quality Transmission Loss.

1.0 INTRODUCTION

The increasing level of competition in the compressor industry and the increasing level of customer expectations has led a general trend towards quieter and more reliable compressors. Pressure pulsations in positive displacement compressors, in particularly screw compressors is a major source of concern for noise and vibrations. The pulsations are generated inherently by its principle of being positive displacement, but can be optimized by well-designed discharge ports. These pressure pulsation levels will further be modified by the acoustic system comprising of reactive and dissipative elements in the downstream of the compressor, which includes valves, pipes, orifices, etc as well as the source and radiation impedances of the boundaries. These compressors, while operating at its designed internal volumetric ratio generates optimal pulsations. But, while operating either at under pressure or at over pressure conditions, the pulsations will be significantly higher due to the flow interactions between the discharge chamber and the outlet housing [1]. In this paper, the high frequency pressure pulsations generated by an oil free screw air compressor is reduced significantly by the design and installation of an acoustic resonator with extended bandwidth.

Acoustic resonators are largely used in ducted noise reduction applications, primarily in the inlet and outlet of pulsating sources [2]. They are particularly useful when propagating pressure waves have a narrow frequency band, wherein they can be tuned to the specific frequencies of interest. Acoustic resonators are typically of reactive type and creates impedance mismatch, thereby causing reflection of the incident acoustic energy so that the transmitted pulsations are significantly reduced. Typically Helmholtz and Quarter wave resonators are widely used in this category and are often effective in containing low frequencies of interest. It is often possible to combine different acoustic elements to optimise a design specifically for a narrow band of frequency of interest. The transmitted pulsations often excite the downstream system and reradiates as noise, depending upon its vibroacoustic dynamic behaviour.

The compressor under analysis is a two stage oil-free screw air compressor of 110kW power rating with a flow capacity of 650cfm and an overall dimension of 4.3mx1.65mx1.85m. The airend (core element, where the compression takes place) comprises of two stages with inter-stage cooling and are driven by a two pole three phase induction motor coupled through a torsionally tuned flexible disc coupling and a speed increasing gearbox as shown in figure 1.

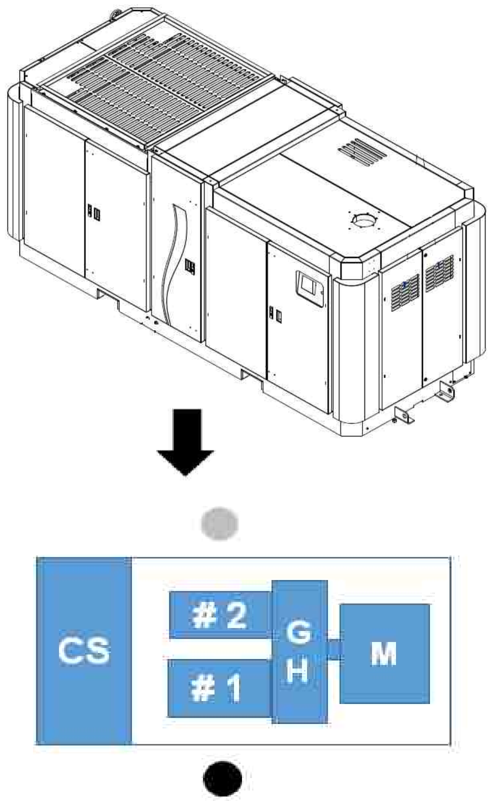

Figure 1 Schematic of the oil-free screw compressor (M-Motor, GH-Gear Housing, CS-Cooling system, #1, #2– Airend stages)

This compressor is of air-cooled type, and the first stage air outlet passes through an air-cooled intercooler before it gets into the second stage inlet for further compression to the intended overall system pressure. The first stage discharge air temperature typically varies in the range of 150 - 250deg.C depending on the operating inter-stage pressure ratio and thus modifies the propagating wave speed of the pulsations in the installation. The compressor at its operating speed radiates a highly tonal noise creating significant level of annoyance to the personnel. Preliminary acoustic source location techniques and frequency analysis with the drive train kinematics relates this annoying frequency to the first harmonic of the lobe meshing frequency of the first stage of the two-stage compressor. Acoustic measurements made in the near-field of the compressor as shown in figure 2 supports the cause definition. It can be observed that the compressor radiates a highly tonal noise dominated at a frequency of 950Hz.This tonal frequency is of concern, since at 1000 Hz, the human ears are more sensitive and results in considerable annoyance. Hence it is required to arrive at design solutions to eliminate the tonality to enhance sound quality. Tuned resonators are selected due to their inherent ability to have higher attenuation at its tuned frequency.

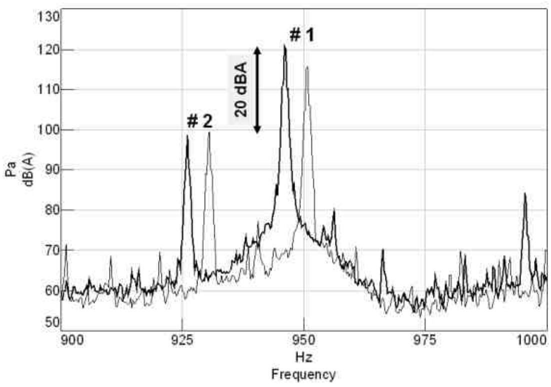

Figure 2 Near-field sound pressure measurements of the compressor

2.0 ANALYTICAL SOLUTION

The acoustic wave equation forms the basis for the wave propagation analysis of ducted systems. The linearized wave equation for the propagation of one dimensional acoustical waves is given by

$$\frac{\partial^2 p}{\partial x^2} = \frac{1}{{c_0}^2}\frac{\partial^2 p}{\partial t^2} \tag{1}$$

where p is the acoustic pressure, c_0 is the speed of sound in the medium and x and t are the space and time variables [3]. The solution to the wave equation is given in Cartesian coordinates as

$$p(x,t) = f(c_0 t - x) + g(c_0 t + x) \tag{2}$$

where f and g are continuous functions which are determined by initial conditions. It can be seen that the first term of the general solution represents an acoustic disturbance of shape $f(x)$ travelling undistorted and unattenuated in the positive x-direction with the speed c_0.The second term is a similar wave that travels in the opposite direction. For harmonic wave propagations, the wave equation transforms into the Helmholtz equation, which can be expressed as

$$\frac{\partial^2 p}{\partial x^2} + k^2 p = 0 \tag{3}$$

where k is the acoustic wavenumber, which is expressed as (ω/c_0), with ω being the angular frequency. The general solution to the above equation can be written as

$$p(x) = Ae^{-jkx} + Be^{jkx} \tag{4}$$

where A and B are complex amplitudes of the forward and backward travelling waves respectively.

The transfer matrix of a simple circular duct of length l, can be expressed as

$$\begin{bmatrix} p_r \\ Q_r \end{bmatrix} = \begin{bmatrix} \cos(kl) & jY_r \sin(kl) \\ \frac{j}{Y_r}\sin(kl) & \cos(kl) \end{bmatrix} \begin{bmatrix} p_{r-1} \\ Q_{r-1} \end{bmatrix} \tag{5}$$

where p and Q denote the acoustic pressure and volume velocity respectively and Y represents the acoustic impedance. The transfer matrix relates the above parameters between any two locations of the straight duct, say r and $r\text{-}1$ in the above representation [3]. Likewise, the transfer matrix of various acoustic elements are well documented [4] and can be combined to form the overall system transfer matrix as shown below.

$$\begin{bmatrix} p_{inlet} \\ Q_{inlet} \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p_{outlet} \\ Q_{outlet} \end{bmatrix} \tag{6}$$

where T_{11}, T_{12}, T_{21} and T_{22} are referred to as the four poles of the combined acoustical system. The transmission loss is defined as the ratio of the incident acoustic energy to the transmitted energy and is the measure for the acoustic behaviour of a system. The transmission loss (TL) of the combined acoustical system can be calculated using the expression [4].

$$TL = 20\log\left\{ \left| \frac{T_{11} + T_{12}/Y_I + T_{21}Y_A + T_{22}(Y_A/Y_I)}{2} \right| \right\} \tag{7}$$

In this work, a reactive resonator is configured by a combination of quarter wave resonators and an expansion chamber of length to diameter ratio of 1.2 within the space constraints in the installation. The quarter wave resonators are achieved through an intrusion and extrusion as shown in figure 3 and are designed to target the specific frequency of interest with the required bandwidth. The 1-D plane wave based estimations through transfer matrix approach also incorporate the Karal correction factors to account for the extended acoustic length due to the sudden expansions and contractions of the acoustic wave propagations [5].

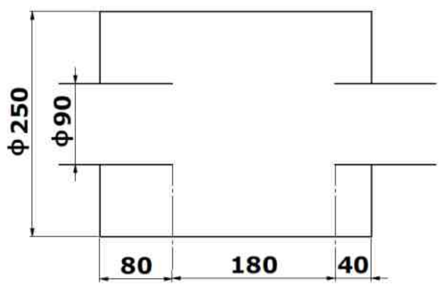

Figure 3 Base geometry of resonator
(All dimensions in mm)

However the transmission loss predictions using transfer matrix approach was accurate only till 790Hz for the plane wave limit due to the geometry constraints, whereas the frequency of interest remains at 950Hz. The prediction due to plane wave based transfer function approach is inadequate as higher order modes start to cut on. Hence it becomes imperative to identify alternate methods like numerical analysis. Also it is highly desirable that the designed resonator be able to accommodate the variations in the frequency due to the changes in speed, load and temperature in the installation. In addition, a resonator with extended bandwidth would be desired so as to standardize the resonator for the family of compressors within a limited range of operating speeds. This requires the resonator to have adequate bandwidth to account for the variations of pressure wave propagating speed and excitation frequency. Apart from this, the resonator also demands a constant transmission loss in the frequency range of interest. The transmission loss predicted using the analytical approach for the base geometry is shown in figure 4.

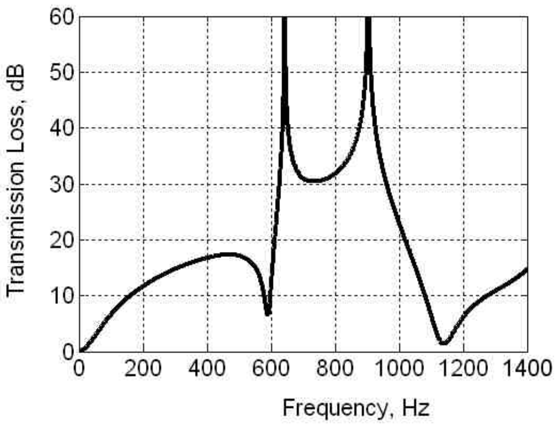

Figure 4 TL prediction – Analytical

3.0 NUMERICAL ANALYSIS

Owing to the plane wave limit of the base design, transmission loss was predicted through a finite element based multi-physics software COMSOL. The software focuses on the coupling of different physics together and it allows the user to program his own differential equations [6]. The resonator geometry was analysed for transmission loss predictions in the Pressure acoustic module with the necessary discretisation and boundary conditions. The predicted transmission loss behaviour of the base geometry using finite element analysis is shown in figure 5. The predictions of the transmission loss between the transfer matrix approach and numerical approach are accurate in the low frequency range, till 790Hz, which is the plane wave limit. The deviations of the transmission loss from the transfer matrix approach beyond the plane wave limit illustrates the inadequacy of the one dimensional transfer matrix approach to incorporate the effect of higher order modes.

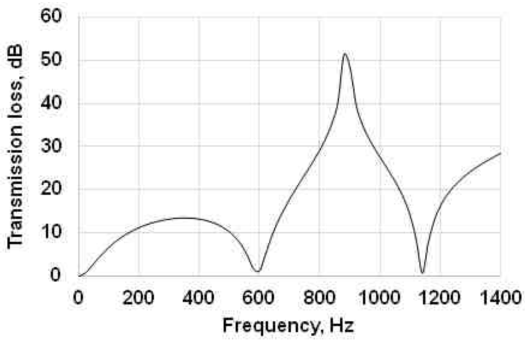

Figure 5 TL Prediction – Numerical analysis

The transmission loss behaviour of the numerical analysis of the base geometry displays a narrow band behaviour, which is inadequate for the installation variations. Subsequently a design optimisation was carried out by the conversion of the in-plane inlet into a side inlet to take advantage of the higher order modes to enhance the bandwidth of the transmission loss behaviour. The optimized geometry is shown in figure 6.

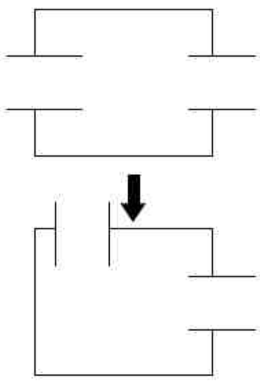

Figure 6 Optimized geometry

The undamped transmission loss was predicted using numerical analysis for the optimized geometry of the resonator. The predicted transmission loss of the optimized geometry is shown in figure 7 together with the TL of the base geometry for comparison. It can be observed that the bandwidth of the TL in the vicinity of the frequency of interest, 950Hz is increased from 200Hz to 400Hz. Even with the increased bandwidth, a minimum TL of 30dB is still available, which is adequate for the expected noise reduction.

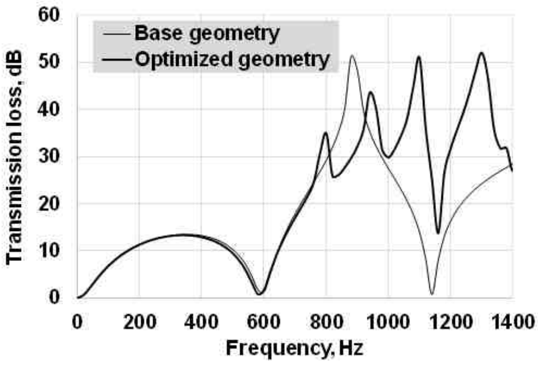

Figure 7 TL of optimized geometry

The exploited transverse modal response behaviour of the optimized resonator is illustrated in figure 8. With the geometrical optimisation, the higher order modes are so envisaged that the transmission loss behaviour is fairly constant in the frequency range from 800 to 1200Hz.

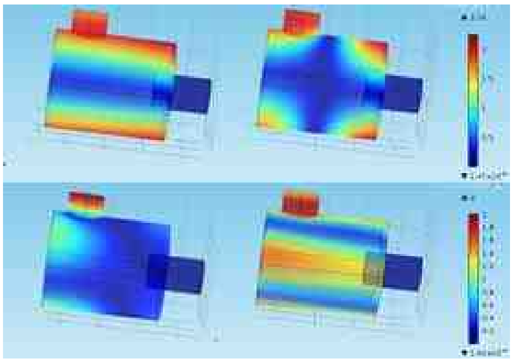

Figure 8 Higher order modal behaviour

4.0 EXPERIMENTS

The predicted transmission loss of the optimized resonator is experimentally verified in the acoustic impedance test facility as shown in figure 9. The impedance tube test facility is of 100mm diameter and is made of cast nylon and houses a loud speaker source, power amplifier and a random noise generator. This is designed to analyse the normal incidence sound absorption coefficient of acoustic materials and transmission loss of resonators and mufflers till 2000Hz. The transmission loss is evaluated using the two load method [3]. It should also be noted that since the temperature at the installation varies depending on the operating condition, whereas the TL evaluation at the impedance tube test facility is done at room temperature. Hence it would be more appropriate to use normalised wave number for the transmission loss characteristics instead of frequencies.

Figure 9 Acoustic Impedance test facility

The optimized resonator was manufactured and experimentally evaluated for the transmission loss behaviour, as shown in figure 10. This part level validation of the optimized design allows for the tuning of the intended frequency to the required accuracy before implementation in the complete machine due to practical limitations. However, in this case, the need for acoustic fine tuning did not arise as the correlation of the results is adequate in the first iteration itself.

Figure 10 TL measurements in acoustic Impedance test facility

5.0 RESULTS

The measured transmission loss from the impedance tube test facility is plotted in figure 11 along with the numerical prediction results. It can be observed that the numerical predictions are very close to the measured transmission loss behaviour, except at the anti-resonant frequencies of the acoustic domain. This is expected, since the numerical model did not incorporate the damping effects in the acoustic model. The flow induced damping in the compressor in operation will further dampen the peaks and troughs of the characteristics to a constant TL behaviour in the frequency range of interest.

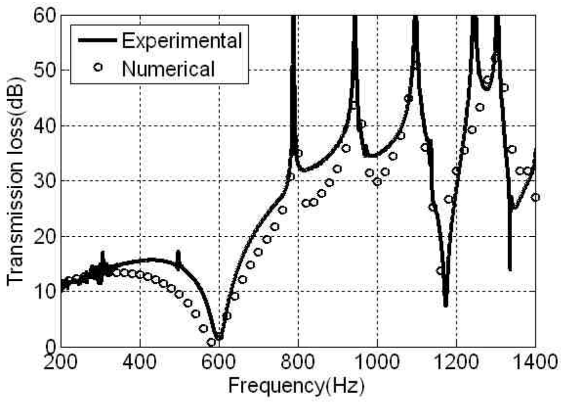

Figure 11 Comparison of Transmission Loss

The validated extended bandwidth resonator is then installed in the first stage outlet of the oil free compressor package as shown in figure 12.

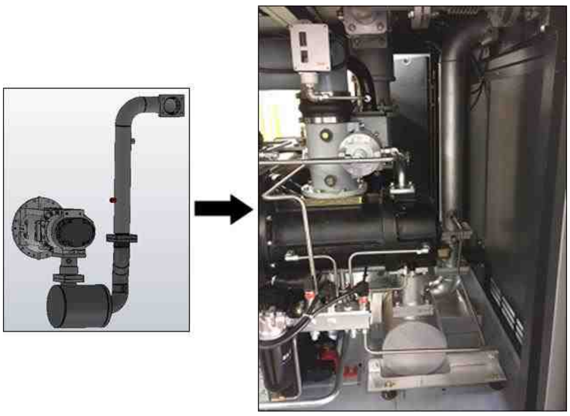

Figure 12 Installation of the tuned resonator in the compressor package

The near field acoustical measurements using condenser microphones are made and the results are shown in figure 13. The results indicate a significant reduction of radiated noise in the frequency range of 900Hz to 1000Hz.

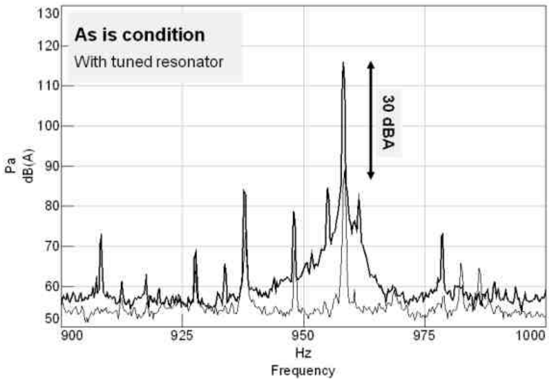

Figure 13 Comparison of Near field acoustic measurements

A 30dB reduction of the sound pressure amplitude at the tonal frequency is observed. The annoying tonality behaviour of the noise pattern is completely eliminated with substantial enhancement of sound quality. In addition, conventional sound pressure measurements in parallelepiped measurement grid as per ISO 3744 standard at a distance of 1m from the compressor shows a sound pressure reduction of 9dBA.

6.0 CONCLUSION

The tonal noise of a screw compressor was eliminated by the design and implementation of a tuned resonator. The bandwidth of the resonator is to be extended for standardisation as well as to accommodate the variations of the tonal frequency in the installation. The analytical design was optimized by numerical analysis for extending the bandwidth of the resonator by exploiting higher order acoustic dynamic behaviour. This was then validated at a part level in the acoustic impedance test facility. Subsequent implementation in the compressor resulted in a significant reduction of the tonality and the overall noise levels of the compressor. Hence a significant sound quality enhancement was achieved through tonal noise reduction of an oil free screw air compressor with an extended bandwidth tuned resonator.

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NOMENCLATURE

p	Acoustic pressure
x, t	Space and time variables
c_0	Speed of sound
f, g	Arbitrary continuous functions
k	Wave number
ω	Angular frequency.
A, B	Complex wave amplitudes
Q	Volume velocity
Y	Acoustic impedance

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