Double throat pressure pulsation dampener for oil-free screw compressors

Michael J. Lucas
Principal Engineer
Ingersoll Rand
800A Beaty Street
Davidson NC 28036
USA
Mike_Lucas@irco.com

Abstract

This paper describes a recent invention at Ingersoll-Rand for reducing the pressure pulsations in an oil-free screw compressor. Pressure pulsation refers to the rapid change in pressure with time measured in the downstream piping of the air compressor. The pulsations are due to the rapid opening and closing of the screws as the discharge port ejects air into the piping system. The pulsations are known to produce excessive noise levels and high levels of vibration in the piping system. Reducing these pulsations is critical to achieving a quiet running compressor. This paper will describe the methodology used to analyze the data and show both computational and experimental results achieved while using the pulsation dampener. At Ingersoll-Rand we refer to this silencer as the double-throat pulsation dampener. Ingersoll-Rand has recently received a patent for this design and has begun to implement this design on all of its oil free screw compressors.

Introduction

Pressure pulsations are a well known phenomenon in screw compressors. In fact, the existence of pulsations is found at the discharge of every type and size of turbomachinery (scroll, reciprocating, sliding vane, centrifugal, axial flow, and screw). The severity of these pulsations is dependent on how the gas is compressed. Presented in this paper is an explanation on the noise generated from pressure pulsations. Followed by a description of the silencer used to mitigate these pulsations and attenuate the resulting noise. When designing a discharge silencer for any piece of turbomachinery, it is critical to have some basic understanding on the fluid dynamic and acoustic field near the discharge of the turbomachinery.

Self-Sustained Oscillations

Self-sustained oscillations are characterized by an organized fluid dynamic wave. In this paper, fluid dynamic or aerodynamic will be used to describe the action of the fluid. In oil free screw compressors the term fluid implies the compressed air. Self-sustained oscillations can be decomposed into two components: an acoustic contribution having a wavelength that is described as l_a that has a speed of sound characterized as c and an aerodynamic component described as l_{fluid} that has a corresponding velocity u_{mean}. The mean velocity u_{mean} represents the convective velocity of the vortices. Both the aeroacoustic and the aerodynamic fields share the same frequency. When we measure the sound field with a microphone we are measuring the aeroacoustic sound field that originated from the self-sustained aerodynamic oscillations at the discharge of the compressor. If we were to measure the pressure pulsation inside the pipe near the discharge of the compressor we would measure

$$p(f)_{Total} = p(f)_a + p(f)_{fluid}$$

where $p(f)_{Total}$ is the total pressure field measured by a transducer in the flow. In the near field, a majority of total dynamic pressure is generated by forces acting on the gas ($p_{fluid} >> p_a$). This effect is often referred to in the literature as the added mass effect (Ref. 1). Further downstream from the discharge port of the compressor, the self-sustained aerodynamic component of the pulsation is expected to diminish because of the formulation of small scale eddy structures typically described as turbulence. It is well known in jet-theory that these organized structures end several (3 to 5) aerodynamic (l_{fluid}) wavelengths downstream from the discharge of the jet. The transition from organized vortex structures to small scale turbulent structures denotes the end of the near field. In practice l_{fluid} is the distance measured between two successive vortices. The aerodynamic wavelength is easily determined by the following equation

$$l_{fluid} = \frac{u_{mean}}{f}.$$

The discharge of a screw compressor port is best described as an impulsively generated jet. An impulsively generated jet can be simulated in a laboratory by the experimental set-up shown in Figure 1. In this arrangement a piston is used to generate a vortex pair by raising and lowering the arm illustrated in the diagram. This technique can generate remarkably consistent vortex pairs. Fundamental studies on the generation and interaction of a vortex pair on a solid body provides a basic understanding on how the flow behaves in the near field of the compressor discharge. (Ref. 2)

An analytical description of the vortex pair can be found in Batchelor (Ref. 3). Without going into the details of this formation, it can be shown that a complete description of the flow is a Bessel function of the first order. Figure 2 shows an example of the vortex pair analytical prediction compared to high speed photography.

When the vortex pair is discharged from the compressor port, the vortex pair interacts with any structure that might lie in its path. It is not uncommon for a compressor manufacturer to connect pipe elbows to the compressor discharge without consideration of the problems the elbow might introduce to the flow. Most elbows will cause some abrupt interaction with the vortex pair, which will lead to a self-sustained oscillation. The fundamental characteristic feature of a self-sustained interaction is an unstable disturbance (the vortex pair) impingement on a surface that leads to an upstream interaction with the edge of the discharge port that further amplifies the disturbance. The upstream interaction is an acoustic pressure field that occurs practically instantaneous with the surface impingement because the pressure field traveling upstream is traveling at the speed of sound (see Figure 1). The result is a splintering of the vortex pair into smaller vortices, which are measured acoustically as harmonics of the port passing frequency (Ref. 4). The fluid dynamic – aeroacoustic interaction just described is the basic principal for the pressure pulsation dampener invention (Ref. 5 and 6).

Important Acoustic Terms

Insertion Loss (IL) – The difference between the sound power propagating through a duct *with* and *without* the silencer installed.

Transmission Loss (TL)– The difference between the sound power level measured at the inlet and outlet of the silencer.

Noise Reduction (NR) – The difference of the sound-pressure level measured near the inlet of the muffler and the sound pressure level measured near the discharge.

Insertion loss, transmission loss, and noise reduction are three acoustical terms that are easily confused or misunderstood. They have different physical meanings and uses. Results from noise prediction models are usually presented in the form of transmission loss. Transmission loss characterizes the true performance of a silencer. Transmission loss has the advantage of excluding the confusing effects due to acoustic coupling that occurs between the source, the piping system and the silencer. Insertion loss gives a true measure of a silencer's performance for a given system. However, the measurements of insertion loss require connecting and disconnecting the silencers from the piping system, which is often difficult in real applications. Noise reduction measurements are often the measurement of choice given the practicality of performing a single dynamic pressure measurement at the inlet and discharge of the silencer.

Important Concepts in Nonlinear Acoustics

One of the concerns with pressure pulsation system inside the piping of a screw compressor is the effect of high-amplitude waves. High-amplitude waveforms caused by the non-linear properties of the fluid can lead to a steeping of the acoustic waves where the peaks are travelling faster than the troughs. The distortion in the acoustic wave will lead to additional harmonics which are often described in terms of a Fourier series.

The questioned posed – "Are nonlinear effects important in screw compressors?"

In linear acoustic theory, $P_{Peak} = c^2 r'$ where P is the root mean squared pressure, P_{Peak} is the peak pressure, c is the speed of sound in the media, and r' is the perturbed density of the fluid. In linear acoustic theory it is assumed that the perturbed density of the fluid is much less than the density of the fluid ($|r'| << r_o$). The sound pressure level is described as

$$SPL = 20\log_{10}\left[\frac{P_{Peak}/\sqrt{2}}{P_{ref}}\right]$$

where $P_{ref} = 20mPa$ in air at normal atmospheric conditions. Given that the speed of sound in air is $341 {}^{m}/_{\sec}$ and the density is $1.2 {}^{kg}/_{m^3}$ it can be shown that at a SPL of 140 dB the density ratio (${}^{r'}/_{r_o}$) is equal to 0.002, which is considered the sound pressure level threshold in air where nonlinear acoustic effects becomes important.

Applying similar theory to screw compressors we will assume that the speed of sound c is $420 {}^{m}/_{\sec}$ and the density at 7 bar (100 psig) is $7 {}^{kg}/_{m^3}$. If the peak-to-peak pressure is 137 kPa (20 psi) then the ratio of (${}^{r'}/_{r_o}$) is equal to 0.11, which demonstrates that nonlinear effects are important when describing the acoustics inside a screw compressor piping system.

Description of Invention

The invention is both an aerodynamic and an aeroacoustic device. An example of the device is shown in Figures 3 A & B. The pulsation dampener is made from a single casting to provide the greatest amount of rigidity to prevent structure borne radiation. The internal construction of the device is made of two chambers. The fluid path inside the pulsation dampener is designed to minimize pressure losses by providing smooth transitions between the two chambers, shown in the figures. One of the key features to this device is the jet nozzle formed where the gas enters the first chamber. The existence of the nozzle followed by a sudden expansion formed by the first chamber causes the vortex pair, discharge from the compressor, to rapidly breakdown into smaller vortex structures. When the vortex structures are splintered into smaller structures, the acoustic near field is significantly shortened in length. The shape of the first chamber serves two purposes. First, the shape of the first chamber is designed to minimize the impingement of the vortex structures with walls or solid surfaces. Second, the shape of the first and second chamber is derived from analytical and linear acoustic models to provide the greatest amount of broadband transmission loss. Both chambers are acoustically tuned to minimize the transmission of acoustical waves over a broad range of frequencies from 500 Hz to about 5000 Hz.

Today's silencers used in the compressor industry are either: ¼ wavelength tuned resonators, venturi's, or absorptive. <u>Rarely these silencers consider the aerodynamics of the</u>

flow; their design is purely acoustical and is based on linear acoustic theory. In actuality the acoustic field in the discharge of the compressor is not linear but rather highly nonlinear. Analytical and numerical tools are unavailable to acousticians to design silencers assuming nonlinear acoustics; as a result, most silencers designed today assume linear acoustic theory as a first order approximation.

Figures 4 and 5 show calculated and measured results typical of the pulsation dampener. Figure 4 presents the transmission loss as calculated from commercial acoustical software (SYSNOISE). Figure 5 shows noise reduction loss measured in situ on an actual running compressor over a range of drive speeds. Measured and calculated data is presented in both figures for version 1 and 2 silencers.

Conclusion

Measurement and modelling the fluid flow at the discharge of a screw compressor offers many challenges. It is difficult to know exactly the true character of the flow given the extreme conditions that exist at the compressor discharge. Only through simplified models and basic understanding on the flow mechanics can we begin to estimate the flow character. The double throat pressure pulsation dampener is based on available models, experimental data and experience. It is through the combination of these understandings that Ingersoll-Rand was successful at developing this device for use in our oil free compressors.

References

1. Rockwell, D. and Schachmenmann, A.: Self-generation of organized waves in an impinging turbulent jet at low Mach number. J. Fluid Mech. (1982) pp. 425-439
2. Homa, J., Lucas, M., and Rockwell, D.: Interaction of impulsively generated vortex pairs with bodies. J. Fluid Mech. (1988), vol. 197, pp. 571-594
3. Batchelor, G.K.: An introduction to Fluid Dynamics. Cambridge University Press: 1967
4. Lucas, M. and Rockwell, D.: Self-excited: upstream modulation and multiple frequencies. J Fluid Mech. (1984), vol. 147, pp. 333-352
5. European application was granted February 11, 2009. Patent No. 1715238
6. US patent issued June 23, 2009, as US Patent No. 7,549,509

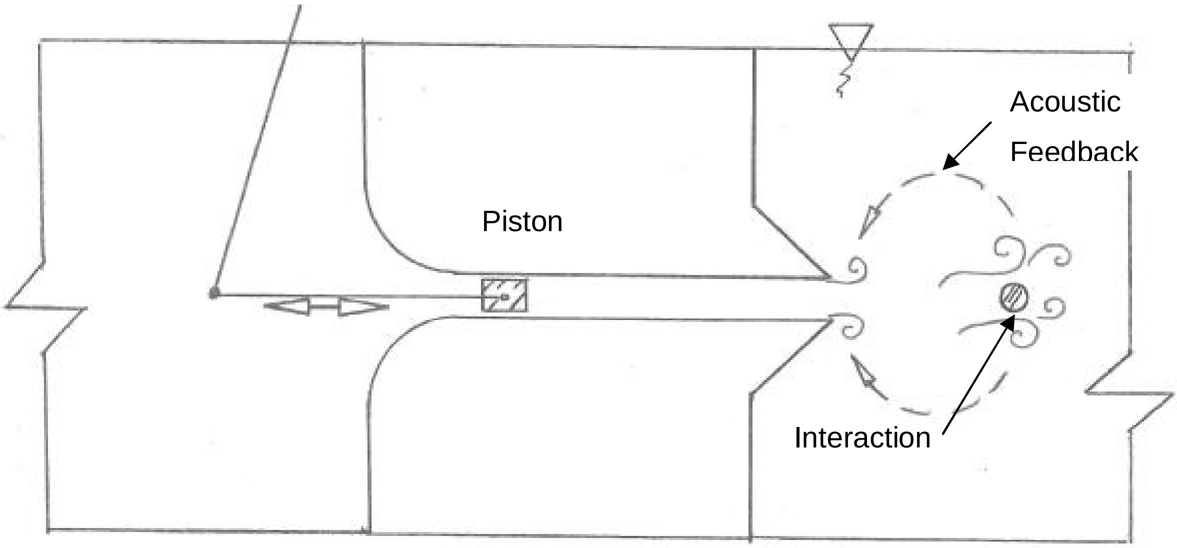


Figure 1. Experimental setup for vortex pair generation and interaction.

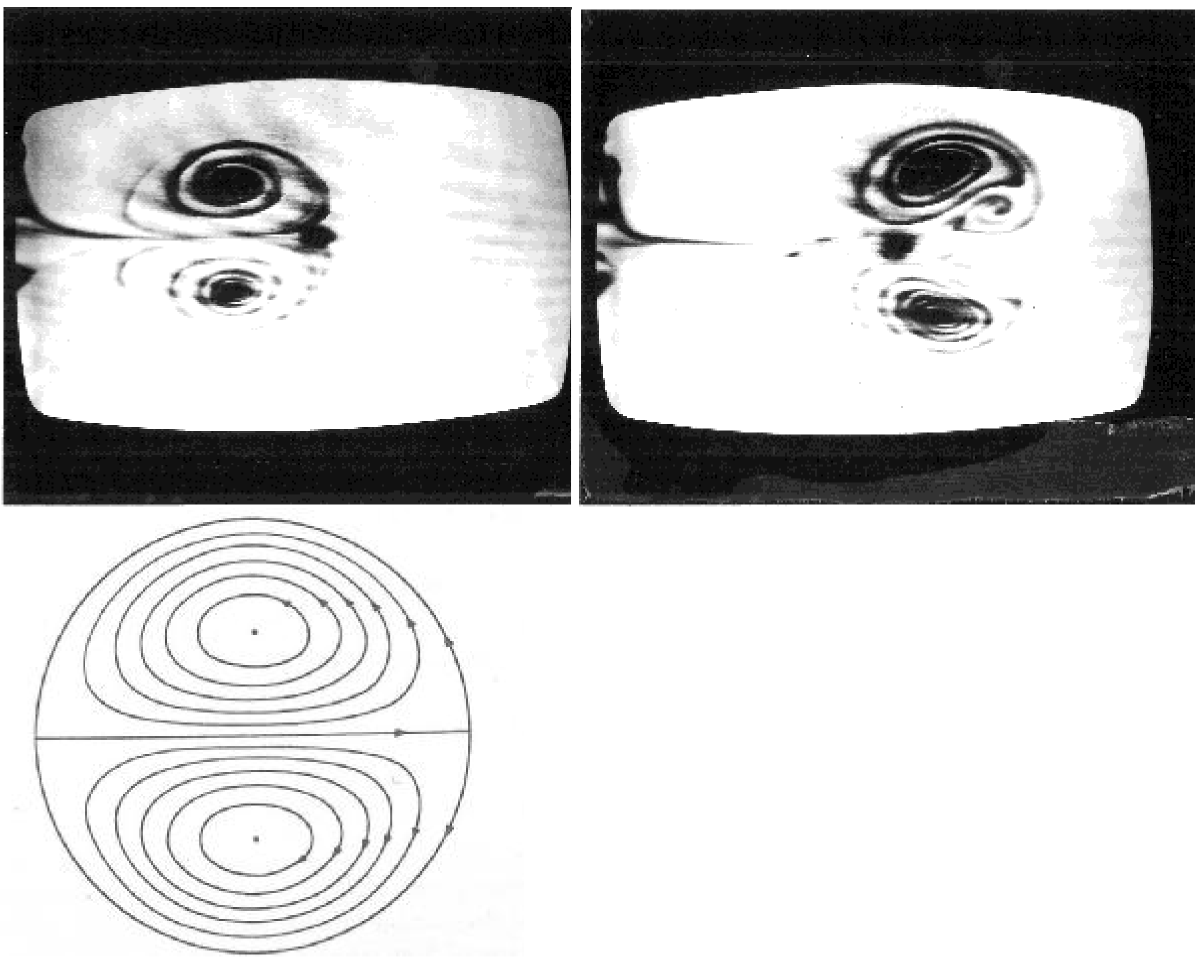


Figure 2. Vortex pair structural interaction visualization compared to analytical prediction.

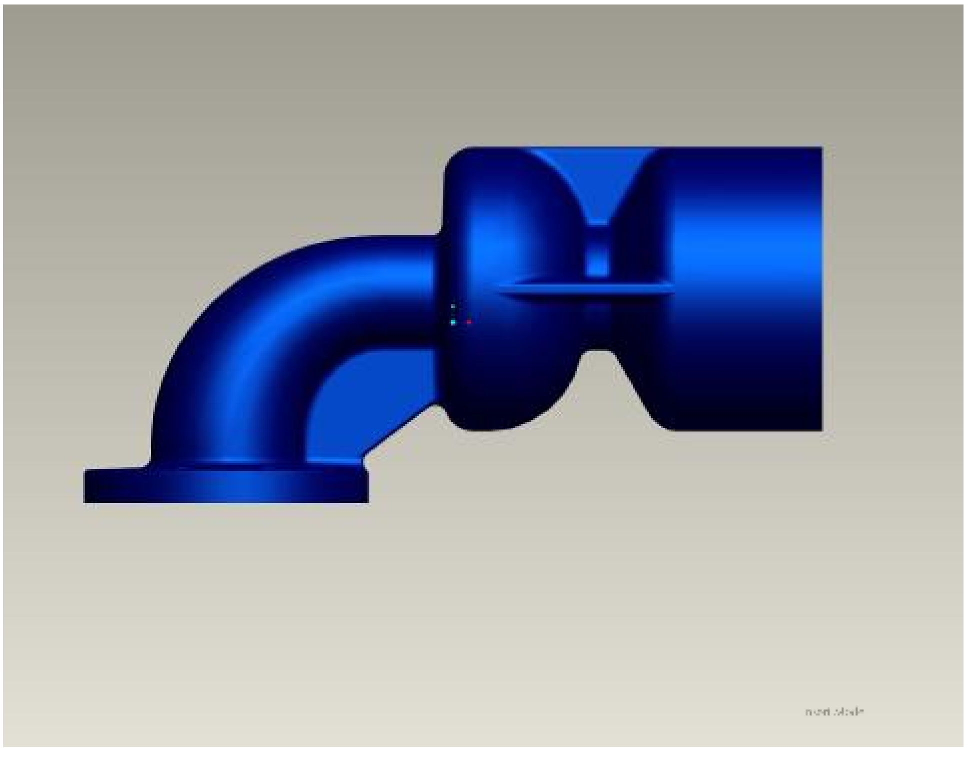


Figure 3. Model of a Double Throat pressure pulsation dampener version 1

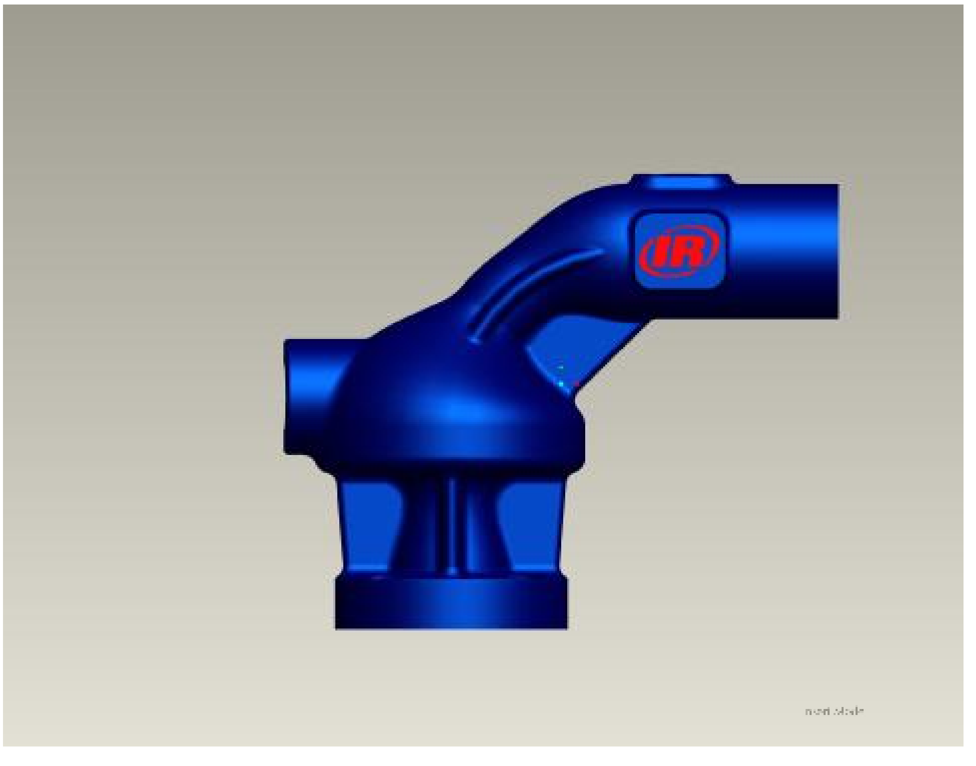


Figure 3 Model of a Double Throat pressure pulsation dampener version 2

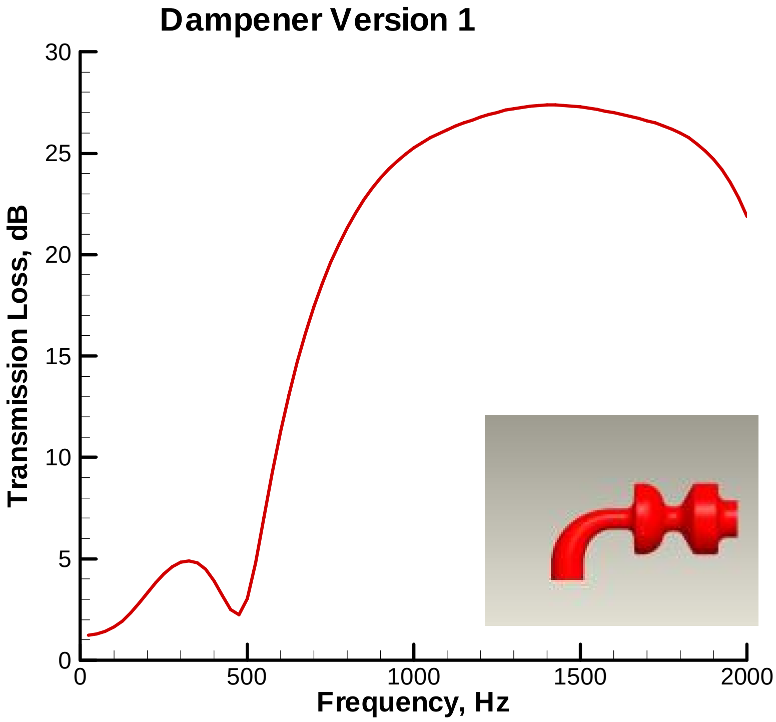


Figure 4A. Transmission loss for the dampener 1.

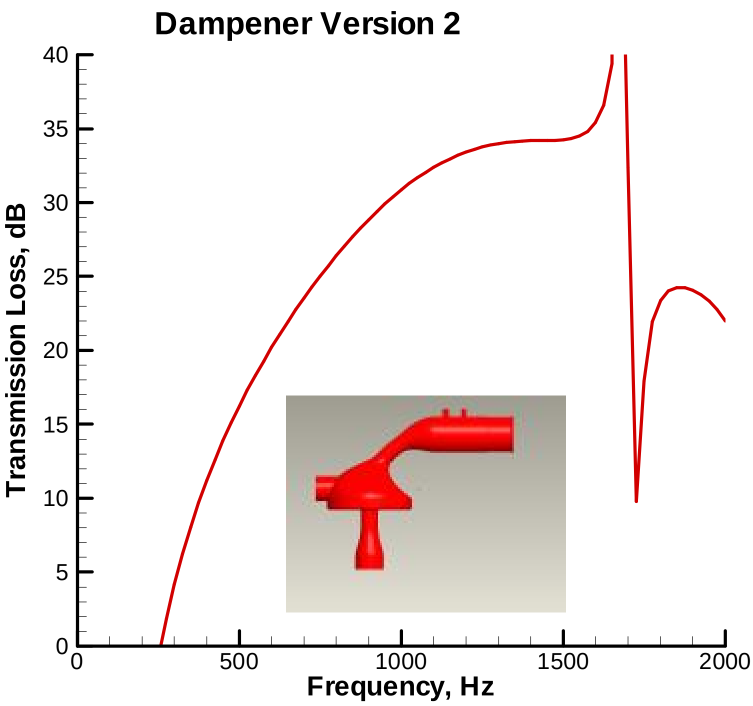
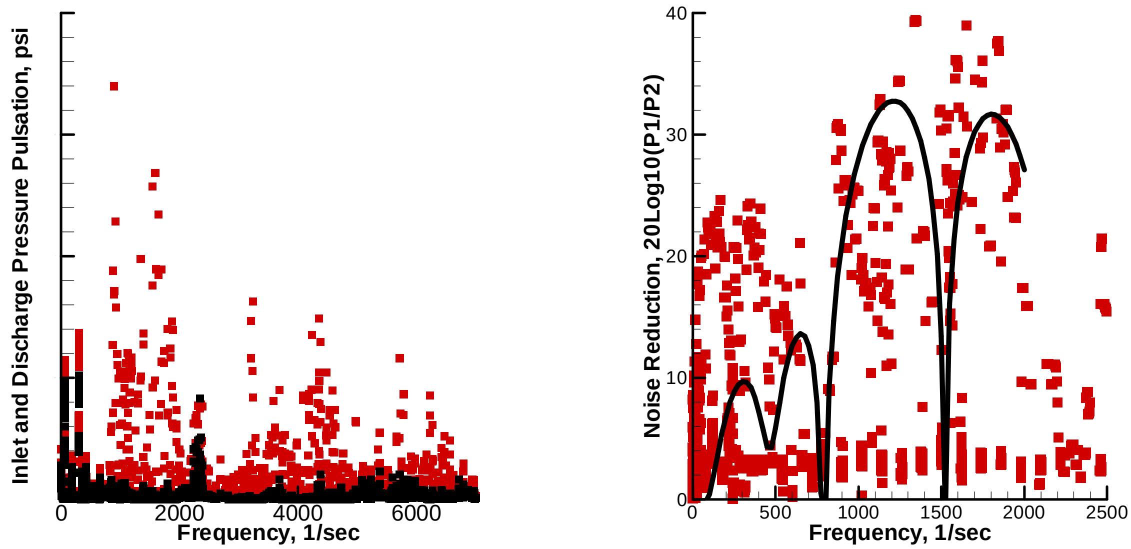


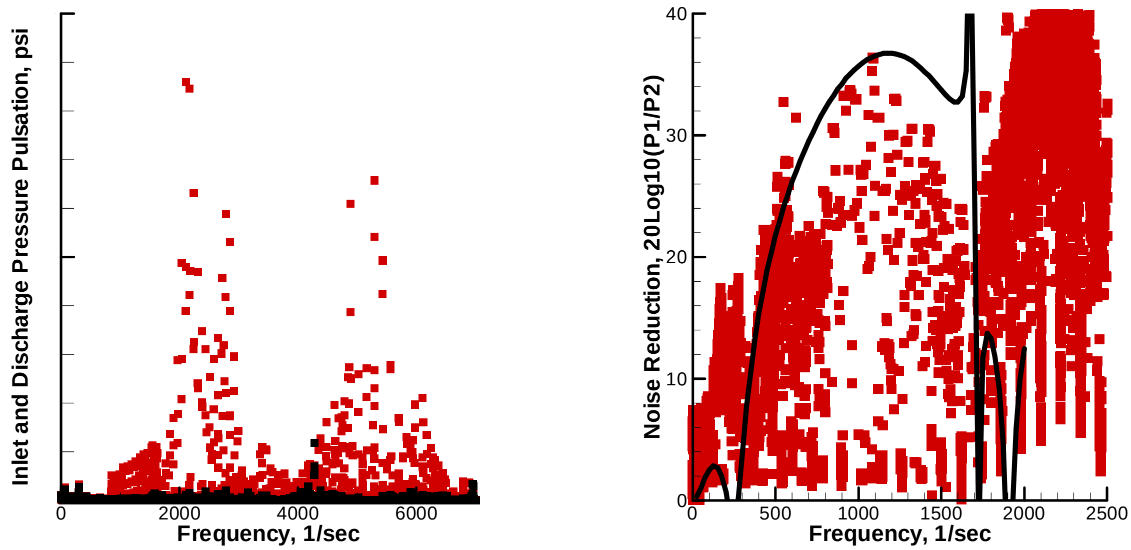
Figure 4B. Transmission loss for the dampener 2.



Inlet pressure pulsation shown as red dots. Outlet pressure pulsation shown as black dots.

Mesaured noise reduction ratio shown as red dots. Calcuated noise reduction ratio shown as black curve.

Figure 5A. Double Throat pressure pulsation dampener version 1.



Inlet pressure pulsation shown as red dots. Outlet pressure pulsation shown as black dots.

Mesaured noise reduction ratio shown as red dots. Calcuated noise reduction ratio shown as black curve.

Figure 5B. Double Throat pressure pulsation dampener version 2