ICSV14
Cairns • Australia
9-12 July, 2007

14th International Congress on Sound & Vibration

NOISE REDUCTION OF A RECIPROCATING COMPRESSOR BY ADDING A RESONATOR IN SUCTION PATH OF REFRIGERANT

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Abstract

The hermetically sealed reciprocating compressor used for air conditioning applications predominantly possesses four noise sources viz. motor, impact of reed (due to suction gas pulsations), mounting vibrations and discharge gas pulsations. The interaction of air-borne noise (by the flow of refrigerant inside compressor) and structure-borne noise (by the components of reciprocating mechanism) becomes cause of concern if a resonance condition is reached. This paper deals with noise reduction of a compressor by suppressing the air-borne noise. A resonator structure is added in the suction path of compressor. The resonator adds desired values of lumped inertance and capacitance to the whole acoustic domain of compressor. The frequency at which this resonator will generate maximum transmission loss is dependent on its geometrical features. The sound spectrum of the compressor in 1/3rd Octave band is studied and on this basis the geometrical features of resonator are decided. The location of resonator is decided based on manufacturing considerations. The negative volume of resonator (the cavity inside resonator) is solved in Sysnoise. The transmission loss through the resonator is calculated over a range of frequency. A bench test of resonator prototype is conducted to establish a relation between the theoretical calculations and physical testing. The whole compressor domain (negative volume of refrigerant passage) is solved in Sysnoise. The transmission loss properties of the compressor with resonator and without resonator are compared. It is observed that the compressor with resonator gives higher transmission loss at the frequency of interest. The pressure drop of the refrigerant across the suction path is calculated in Fluent. The difference of pressure drop is insignificant in both the cases which indicate the compressor's performance will not be affected. The compressor is built and tested for sound power level. 3 dBA of absolute SPL reduction is seen after testing four samples.

1. INTRODUCTION

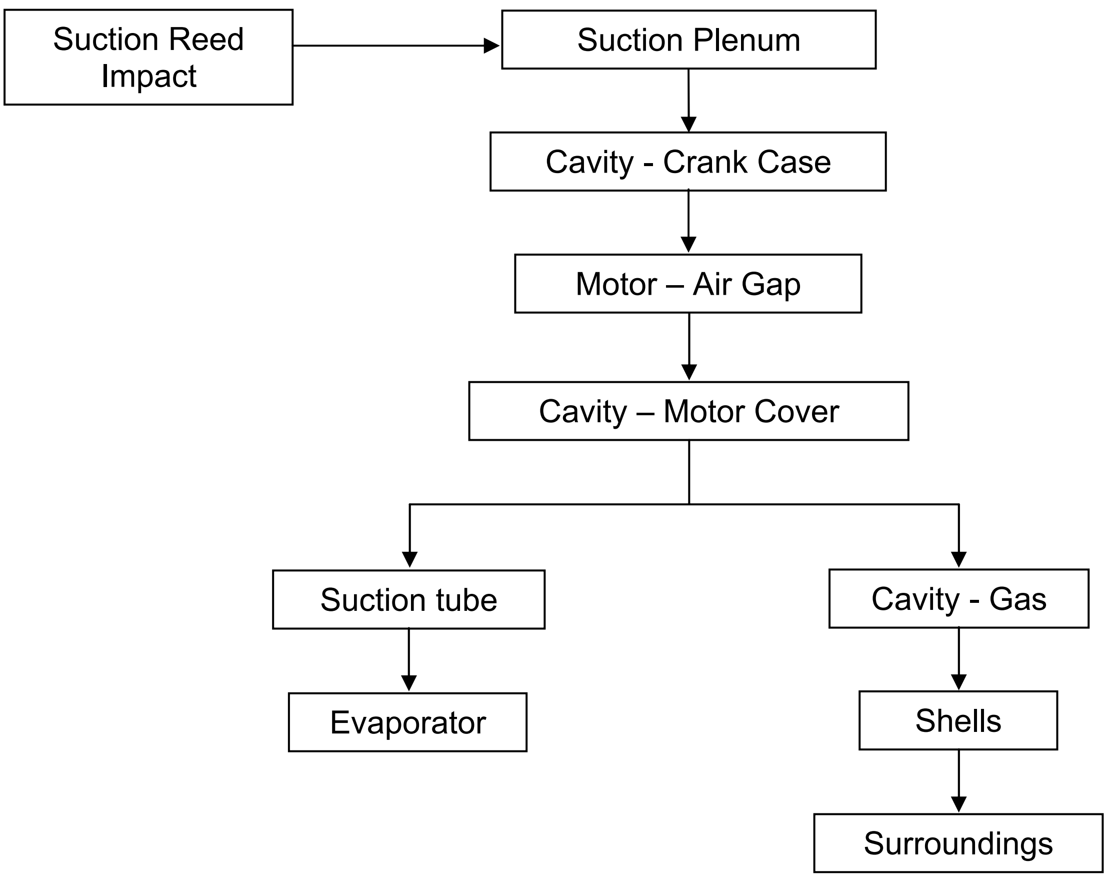
The home appliances like refrigerator or air conditioner emit sound waves that are unpleasant to human ear. Especially, the low frequency sound waves (20 ~ 1000 Hz) coming out from the appliance can cause irritation to the human being and have been a cause of concern for the manufacturers. Essentially, there are several noise sources in an appliance – compressor, fan, condenser, evaporator etc. Out of these, the compressor is a major contributor of noise due to the compression mechanism. A hermetically sealed reciprocating compressor used for air conditioning application is taken as a case in this paper. This case is done to improve noise level of the compressor. Five compressors were picked form a batch of 150 and its Sound Pressure Levels were measured as per IS 3745 in a semi-anechoic Sound Test Lab. The SPL was plotted in 1/3rd Octave Band and the results were studied. The jury tests were conducted to test the Quality of Sound.

The results were not very encouraging. The Sound Power Level of the compressor was higher than competitor's by 2 dBA and compressor sounded very harsh to jury with presence of chattering noise.

The Sound waves spectrum in 1/3rd Octave band delineated that the low frequency Sound Pressure values were significantly high with 250 and 315 Hz being the major culprits. Hence, the synthesis of the compressor noise path was done to analyze the noise sources and their effect on Sound Pressure Level.

2. NOISE PATH SYNTHESIS

The suction side noise path of the compressor can be broadly charted as follows –



The sound waves generated are predominantly air-borne. This can be attributed to the flow restrictions of refrigerant due to suction and discharge valves. This results in a higher than desired pressure at discharge side of compressor. High pressures lead to a disheveled Pressure – Time diagram (non – sinusoidal) and leads to higher pumping harmonics, which if resonate with natural frequency of a structure like tubes or shells can cause severe oscillations.

It is obvious that for suction side the sound waves travel in a direction opposite to that of refrigerant. The primary noise sources as heard by jury, the chattering noise, can be accredited to the impact of reeds (here, spring type valve reeds are used). The natural frequency of these reeds in hinged condition is close to 120 Hz. This is a cause of concern as the compressor operating frequency is 60 Hz. These waves travel through the suction muffler and then to gas cavity. Some of the waves will get transmitted to the surroundings through outer shell and others will go to evaporator. In this case, the suction muffler is an enclosed cavity cast in crank case of the compressor. It acts like a series of dissipative attenuators by the virtue of presence of obstacles like motor windings, air gap and connector duct between muffling cavity and suction plenum. This configuration converts the acoustic energy into heat energy and attenuation of sound is achieved. But as sound waves spectrum suggests, the low frequency zone is hardly affected by this configuration.

3. PRINCIPLE & THEORY

Dampening of low frequency sound waves thus becomes very critical. The low frequency sound means the wavelength of sound waves is high. Introduction of impedance discontinuities in the suction path can help in attenuating frequencies in the low frequency zone. A reactive muffler based on the Helmholtz's principle is a simple expansion chamber with a neck and cavity being its essential components. The resonator adds the lumped inertance and lumped compliance to the system.

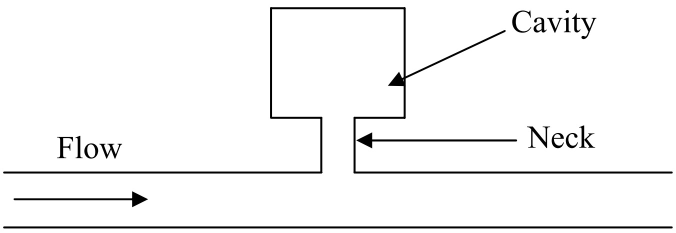


Fig. 1 – Helmholtz Resonator

The neck portion represents the lumped inertance and acts as a restriction to flow of waves and the cavity represents the lumped compliance which acts as a reservoir of acoustic energy. The sound pressure at any point inside Helmholtz resonator, for a given source distribution q is given by,

$$\nabla^2 p(x,y,z) + k^2 p(x,y,z) = -j\rho\omega q(x,y,z) \qquad (1)$$

Equivalent integral formulation for any weighing function $\tilde{p}$, of equation 1 can be done by Galerkin's Weighted Residual Method,

$$\int_V \tilde{p}\left(\nabla^2 p + k^2 p + j\rho\omega q\right) dV = 0 \qquad (2)$$

Term	Description
k	Acoustic Wave Number
ρ	Ambient Fluid Mass Density

Equation 2 can be solved by discretizing the volume and then formulating mass, stiffness and acoustic damping matrices.

The impedance of the Helmholtz's resonator can be written as,

$$Z_{neck} = i\omega \frac{l}{A}$$
$$Z_{cavity} = \frac{1}{i\omega\left(\frac{V}{c^2}\right)}$$
$$CorrectionFactor = \frac{ck^2r^2}{A}$$

$$Z_{resonator} = i\omega \frac{l}{A} + \frac{1}{i\omega\left(\frac{V}{c^2}\right)} + (CorrectionFactor)$$
$$Z_{resonator} = i\left[\omega \frac{l}{A} - \frac{c^2}{\omega V}\right] + \frac{\omega^2}{\pi c} \qquad (3)$$

The radiation impedance, from equation 3 equals Zero when –

$$\omega = c\sqrt{\left(\frac{A}{lV}\right)} \qquad (4)$$

Term	Description
l	Length of Neck
A	C/S Area of Neck
V	Volume of Cavity
c	Speed of Sound

Equation 4 represents the frequency at which radiation impedance is Zero, which means it is the resonant frequency. At this frequency, the resonator will have a high value of transmission loss.

Targeting the frequencies 250 and 315 Hz, the geometrical features of resonator structure were decided. While deriving the Helmholtz's equation for resonant frequency, the shape of resonator is not considered. The volume of cavity is the only parameter that determines the performance of the resonator and not its shape. So, until all dimensions of cavity and neck opening are considerably smaller than wavelength, the resonator will abide to the frequency equation stated above. The positioning of resonator was a critical aspect. From manufacturing and assembly point of view it was selected close to the suction plenum. This also benefited in less pressure drop since the gas path through connector hole increased by 40%, otherwise remaining constant.

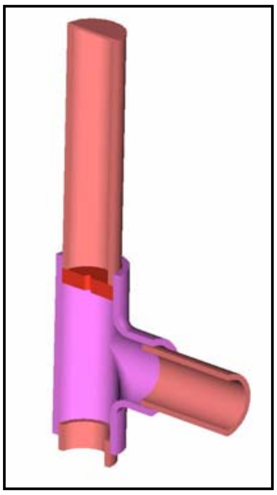


Fig. 2 – C/S of Resonator

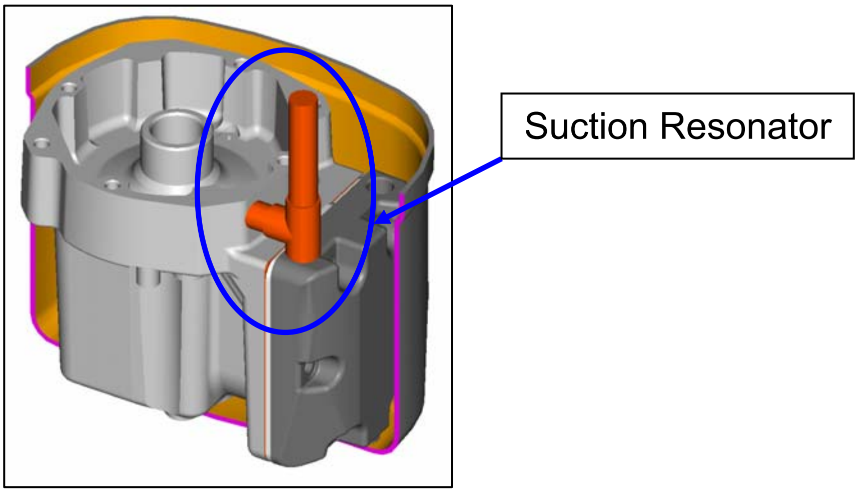


Fig. 3 – Resonator Assembly

4. ANALYSIS

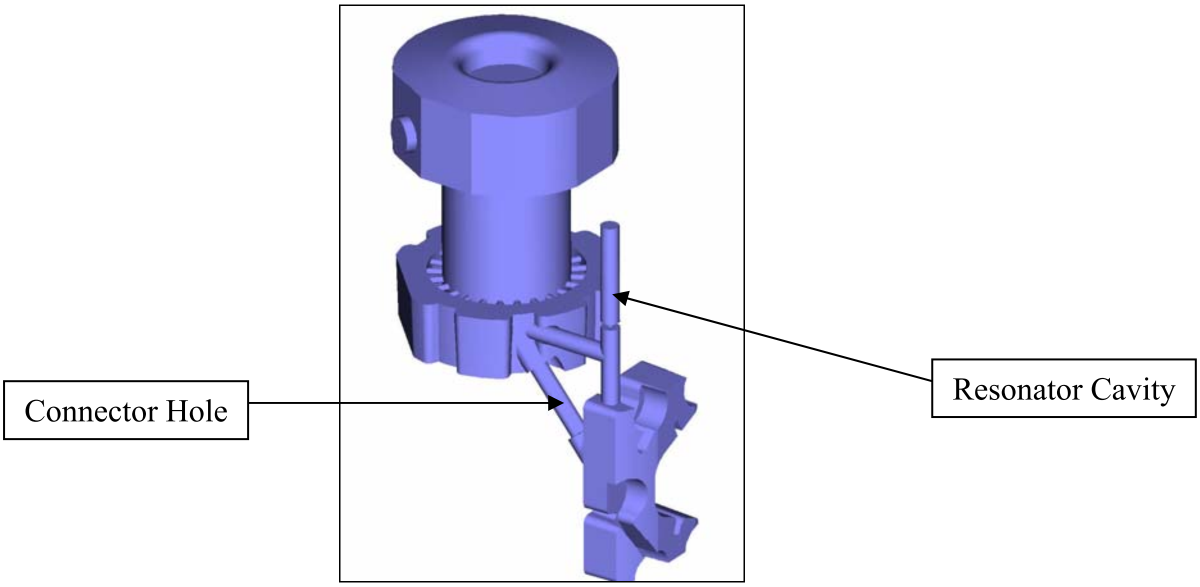


Fig. 4 – Compressor Suction Domain

The characterization of resonator can be done by transmission loss, insertion loss and level difference (or, noise reduction). As resonator outlet is assumed to have anechoic termination, transmission loss was taken as parameter to characterize performance of resonator. The transmission loss of the resonator was calculated in Sysnoise. The calculation of Transmission Loss is based on the finite element method as mentioned earlier. The target was to achieve high magnitude of transmission loss at 250 and 315 Hz or a wide attenuation curve peaking at 300 Hz. Different versions of resonator with variables such as, length of cavity, diameter of cavity, length and diameter of neck portion were analyzed using Taguchi's D.O.E. method and optimized configuration was selected.

Some assumptions are required to be made while proceeding with the analysis such as homogeneous refrigerant gas, uniform sound velocity, perfectly non-reflecting resonator outlet etc. The particle velocity boundary condition was assumed to be unity at the inlet of resonator. With these assumptions the analytical results vary from the actual results. To correlate the two results, bench test of resonator is conducted. The set up for bench test includes a sound source to the (noise) inlet of resonator and FFT analyzing equipment at the outlet. The results of bench test and resonator show fair agreement as the gas path is simple. But as it is very difficult to prototype only suction domain of compressor, it was decided to test the whole compressor and establish the relation between the Transmission Loss and Sound Spectrum of compressor.

The pressure drop is a prime parameter in compressors as it can reduce the cooling capacity substantially. The addition of resonator in existing suction path meant lower mass flow through the resonator. Hence, the desired results were not seen. In order to overcome this, diameter of connector hole between crank case and suction plenum was reduced so as to get maximum mass flow through resonator. This was analyzed in FLUENT. The aforesaid resonator structure was built in plastic and fitted into the cast iron crank case by means of 'O' rings. The compressor was tested in the semi-anechoic chamber as per IS-3745.

5. RESULTS

1. As seen from figure 6, the resonator gives effective gain in 'Transmission Loss (TL)' at low frequency range of 200 – 350 Hz. The pressure contours plotted for 250 Hz also suggest the same. Further analysis of the whole compressor suction domain was propelled by these results.
2. When the compressor domain with resonator was solved in Sysnoise and its TL curve was compared with original compressor, it was apparent that there was significant gain in the range of 200 – 500 Hz. However, the higher side frequencies like 800 – 1250 Hz have shown low side drift in TL. The sound pressure plot across suction domain of compressor at a frequency, say 250 Hz shows the effect of presence of resonator in the suction path.
3. Physical compressors were assembled with resonator and were tested in the sound test lab. The results were confirming CAE analysis. Gain of 10 – 12 dBA was seen in the low frequency zone of 100 – 500 Hz.
4. Overall reduction of 3 dBA was observed after testing a batch of five compressors.

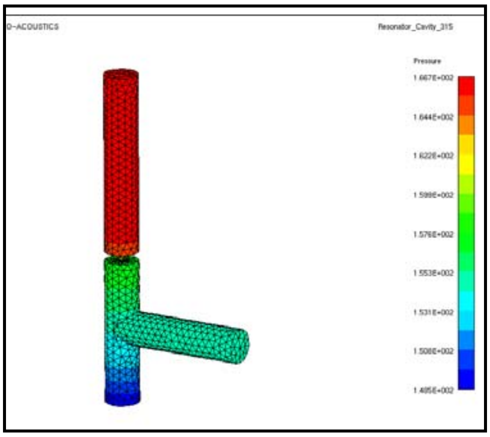


Fig. 5 –Resonator Sound Pressure Plot

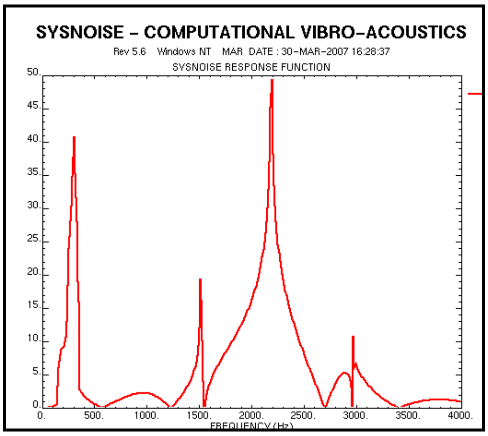


Fig. 6 –Resonator TL Curve Plot

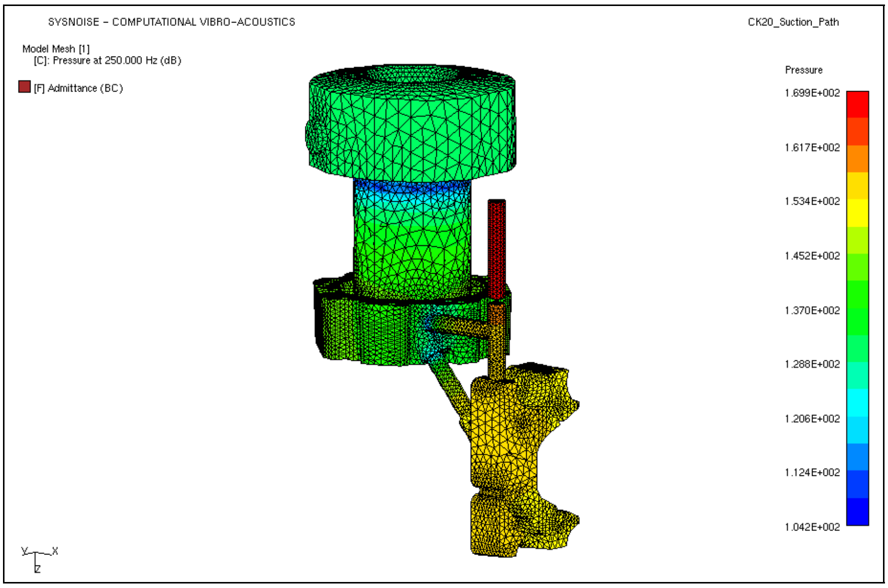


Fig. 7 – Compressor Suction Domain Pressure Plot

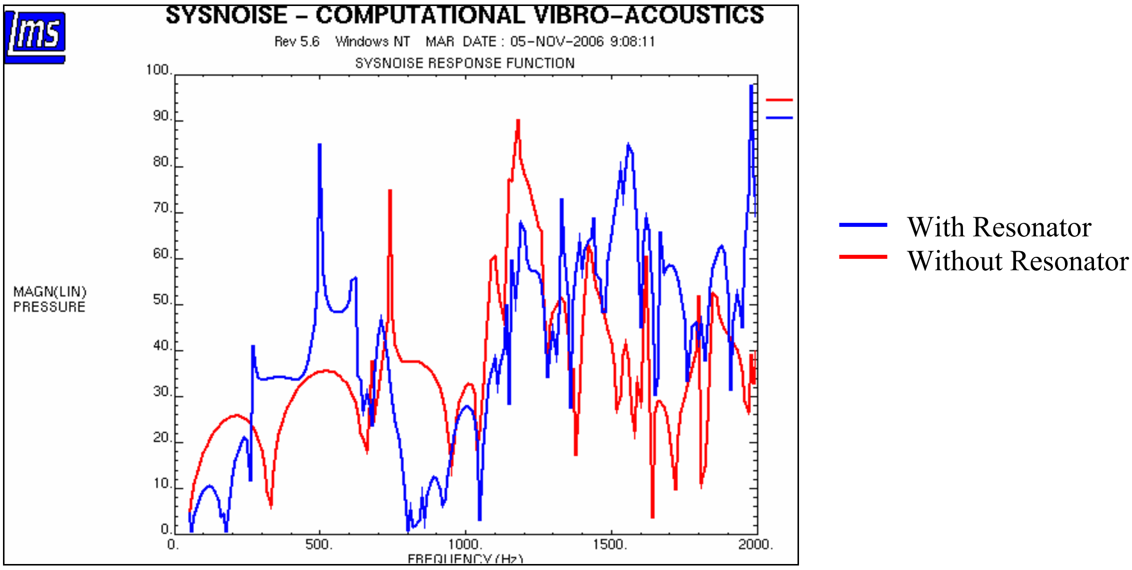


Fig. 8 – TL Curve Comparison of compressors

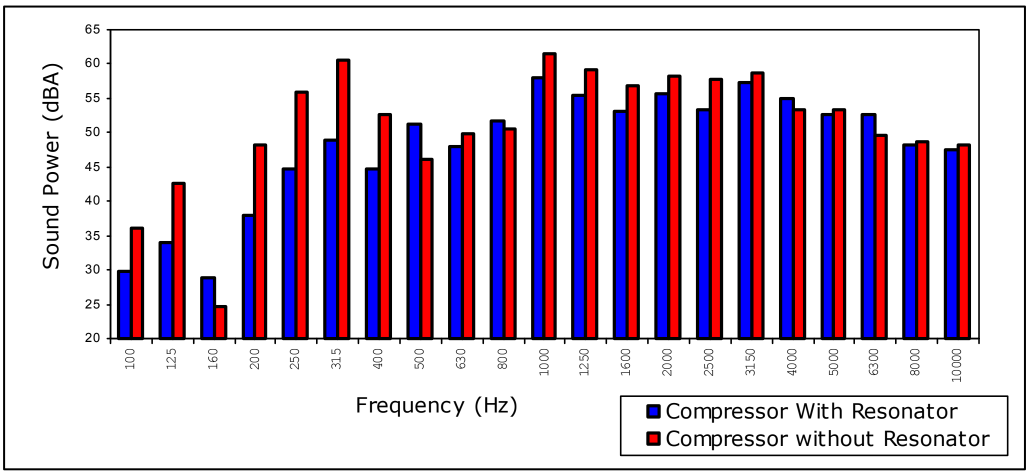


Fig. 9 – Sound Pressure Spectrum of Compressors (Lab Tested)

6. CONCLUSIONS

The resonator structure gives effective noise reduction. It has a high transmission loss values for narrow band frequency ranges and it is very effective in attenuating pure tone frequency pulsations. Another simpler structure like the one in automobiles (simple expansion muffler with chambers and acoustic dampening material inside) can also give similar results. But these types of mufflers involve huge pressure drops across suction side of compressor which is undesirable considering the cooling capacity requirements. The Helmholtz's resonator being totally isolated structure form the gas path achieves the noise attenuation from a very marginal magnitude of pressure drop (6% in addition to original pressure drop). Furthermore, no separate tooling is necessary to fit in the resonator in compressor domain. So, if the suction reed modifications was targeted the cost and performance of compressor would have been affected. Hence, the resonator is an acceptable solution for attenuation of low frequency noise.

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