

Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2016

Prediction of Compressor Muffler Frequency Response Function using CFD

Tadeu Tonheiro Rodrigues

Embraco, Brazil, tadeu.t.rodrigues@embraco.com

Cristiano Stumpf

Embraco, Brazil, cristiano.stumpf@embraco.com

Ricardo Mikio Doi

Embraco, Brazil, ricardo.m.doi@embraco.com

Follow this and additional works at: <https://docs.lib.purdue.edu/icec>

Rodrigues, Tadeu Tonheiro; Stumpf, Cristiano; and Doi, Ricardo Mikio, "Prediction of Compressor Muffler Frequency Response Function using CFD" (2016). *International Compressor Engineering Conference*. Paper 2412.

<https://docs.lib.purdue.edu/icec/2412>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Prediction of compressor muffler frequency response function using CFD

Tadeu Tonheiro RODRIGUES^{1*}, Cristiano STUMPF², Ricardo Mikio DOI³

Embraco - R&D
Joinville, Santa Catarina, Brazil

¹tadeu.t.rodrigues@embraco.com

²cristiano.stumpf@embraco.com

³ricardo.m.doi@embraco.com

* Corresponding Author

ABSTRACT

The acoustic filters of hermetic reciprocating compressors, also called mufflers, are usually developed through acoustic simulation solving the discretized wave equation to obtain the Frequency Response Function, which translates the acoustic response of the muffler. Nonlinear effects are neglected in this approach, which are attributed to flow patterns, as turbulence phenomena, which occur in the contractions, expansions and changing directions within the geometry. The main aim of this work is to investigate the influence of non-linear effects in the acoustic response of mufflers, solving the flow field by computational fluid dynamics (CFD). A discharge acoustic filter design was simplified for the study purpose and simulated using both CFD and Linear Acoustic techniques; the difference in the two approaches is made by comparing the Frequency Response Function (FRF). The flow effects are analyzed varying the compressor piston displacement and operating conditions. FRF predicted by CFD presents reasonable agreement with acoustics approach for lower frequencies identifying resonances and anti-resonances. It was observed increased disagreement for higher mass flow rates due to the predominance of flow effects over acoustics vibrations modes.

1. INTRODUCTION

Reactive mufflers are widely used in reciprocating compressor design for control and reduction of noise (Soedel, 2006) originated from the intermittent flow through valves, which generates gas pulsation in suction and discharge lines. Engineering approaches to design reactive mufflers are diverse and embrace experimental, analytical and numerical methods.

Lumped formulations were the first known method to evaluate mufflers until the introduction of advanced numerical methods based on the solution of the three-dimensional wave equation to obtain the harmonic response of reactive mufflers. Examples of application of lumped formulation are found in Soedel (2006) and more recently in Park *et al.* (2008), which demonstrated the usefulness of reducing low frequency noise due to discharge pulsation in a reciprocating compressor.

Application of numerical methods is presented in Vendrami *et al.* (2014), which optimizes the suction muffler design by acoustic harmonic analysis, reducing the irradiated noise. Designing reactive mufflers for reciprocating compressors requires multidisciplinary approach, since they imply head losses which jeopardizes the compressor efficiency. The optimization of mufflers aiming improvements in noise and efficiency commonly follows a decoupled approach between thermal and acoustic methodologies as presented in Done *et al.* (2014).

Kim *et al.* (2015) applied CFD simulations to improve both acoustics and flow performance of a suction muffler, for these purposes the flow performance was evaluated in terms of mass flow rate increase while the acoustics performance was verified observing the Transmission Loss curves. The authors validated the muffler improvement

directly in the compressor assembly by standard performance and acoustic tests, with reasonable agreement with numerical predictions.

The previous review show a lack of comparison between the decoupled and coupled approaches, despite the attractiveness of CFD simulations for obtaining reliable data, one can observe that transient simulations are far more costly than acoustic simulation of the wave equation, which is performed in the frequency domain.

The main aim of the present work is to present the results of a coupled evaluation through CFD technique, investigating the influence of non-linear effects related to pump size and operating conditions and comparing the results with linear acoustics simulations. For this purpose, the FRF was chosen as muffler system identity parameters. The discharge muffler was chosen due its importance in the total noise generation of reciprocating compressors, as reported by Bratty *et al.* (2012).

2. NUMERICAL PROCEDURE

2.1 Geometry simplification

The internal geometry of a real discharge system compressor presented in Figure 1.a was simplified to geometry of equivalent dimensions, as illustrated in Figure 1.b; the objective is to obtain a simpler geometry for mesh generation still representative for the purpose of this work.

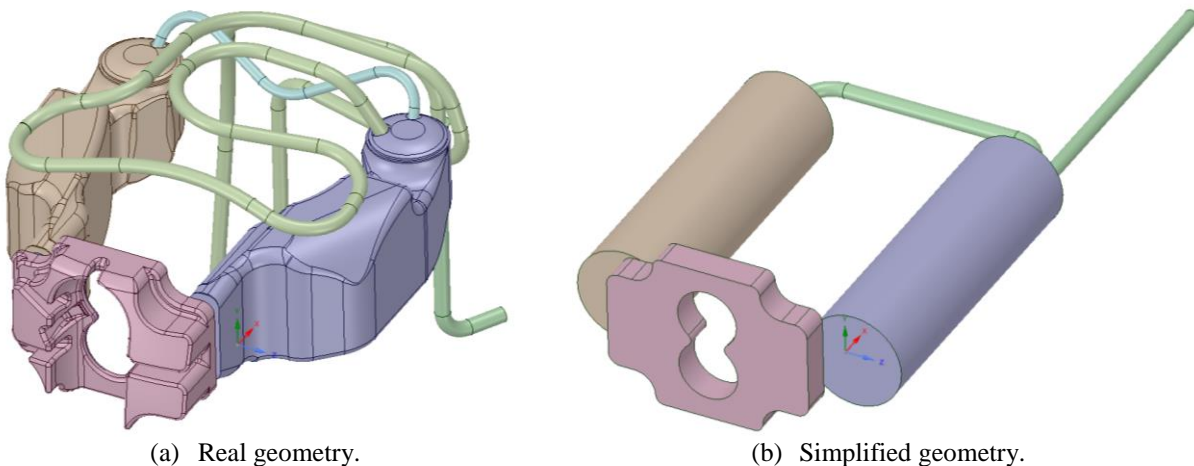


Figure 1: Discharge muffler internal geometry.

2.2 CFD approach

The computational domain was discretized in tetrahedrons, observing skewness quality parameter; at the wall inflation layers were applied for better representation of boundary layer, resulting in computational mesh of 260,000 nodes, presented in Figure 2.

The unsteady, compressible, turbulent fluid flow through the discharge system was solved using the commercial code ANSYS CFX® release 15, based on the element-based finite volume method to discretize the partial differential equations of the mass conservation, momentum, energy and turbulence.

Advection terms were discretized using a High Resolution Scheme and the temporal discretization obtained by a second order backward Euler scheme. The Shear Stress Transport model was set to bridge the turbulent effects to the mean flow, which has been proven as a suitable model for compressor simulation (Rodrigues and Da Silva, 2014). The resultant system of equations is solved through Incomplete LU (ILU) decomposition, algebraic multigrid method and coupled strategy. The pressure evaluation correlation is obtained from the Redlich-Kwong library, available in the numerical tool.

The boundary conditions are illustrated next in Figure 2, with inlet at the discharge ports and outlet at the end of the discharge tube. The boundary conditions are based on performance rating points ARI LBP (Low Back Pressure) and

MBP (Medium Back Pressure) for propane refrigerant. Mass flow rate profile is prescribed at the inlet according to the crank angle (Figure 3), obtained from results of in-house compressor simulation software (Ussyk, 1984) for 60 Hz application, while at the outlet the condensing pressure is applied. At the inlet are set constant temperature, obtained from experimental measurements and turbulence intensity of 5%, which is the recommended value for tubular geometries.

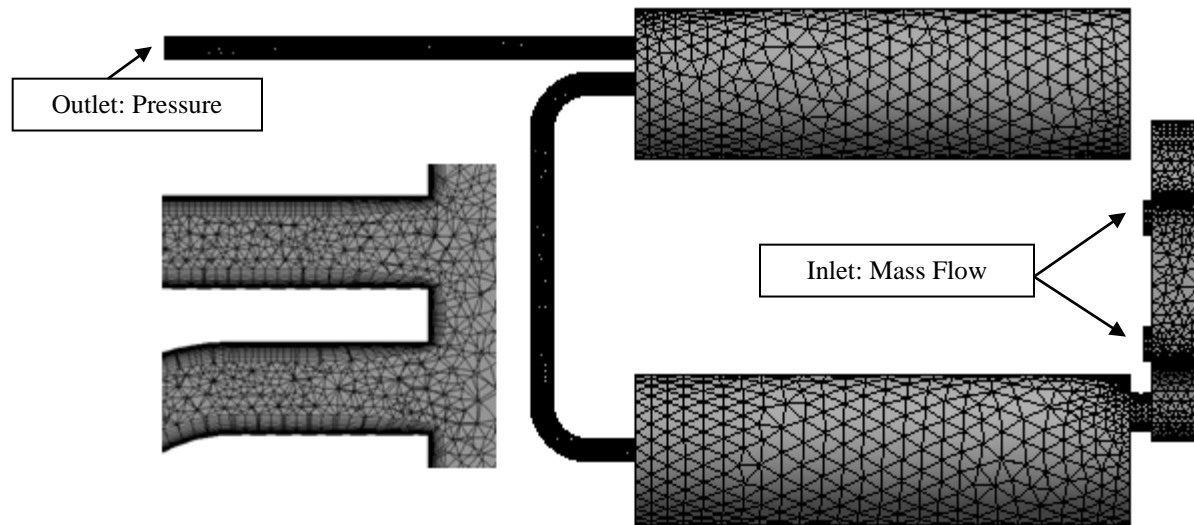


Figure 2: Mesh and boundary conditions for CFD simulation.

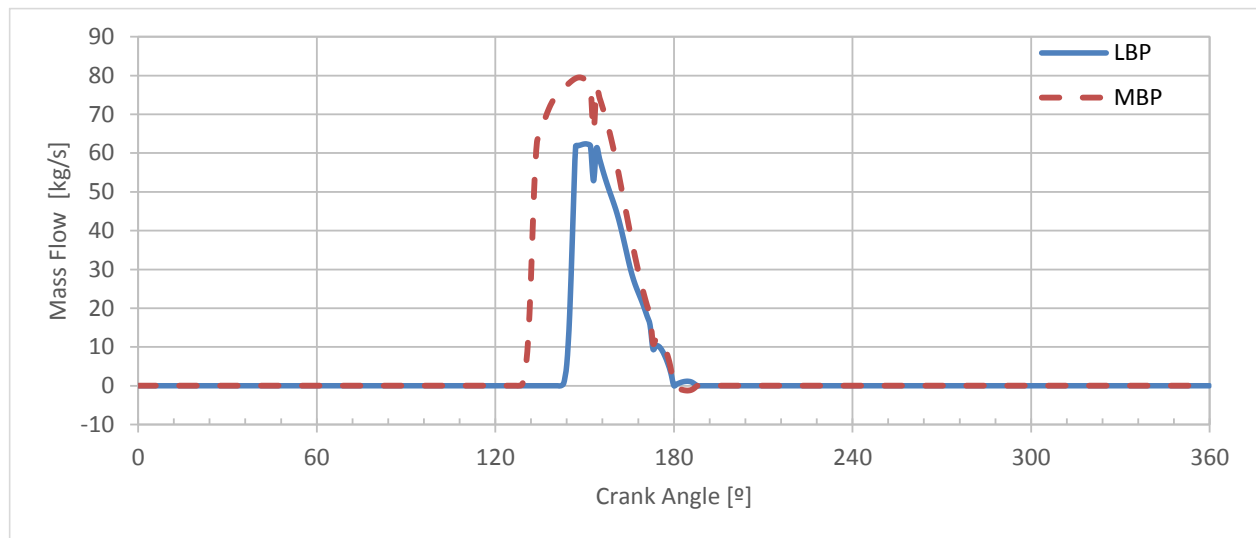


Figure 3: Mass flow rate profiles for inlet boundary condition.

2.3 Acoustics approach

A harmonic response simulation of the acoustic domain was performed in ANSYS Mechanical® release 15 for obtaining the FRF, based on the Finite Element Formulation for solving the governing equation of the physics chosen, in this case, the acoustic wave equation. The physical domain is discretized with fully hexahedral mesh, presented in Figure 4, which is compounded with 9070 second order elements and 41462 nodes. The fluid density and sound speed are set based on experimental measurement of temperature and pressure in the discharge system. At the fluid flow inlet, the system is excited with mass flow, while at the outlet is prescribed an acoustic absorption surface, as illustrated in Figure 4.

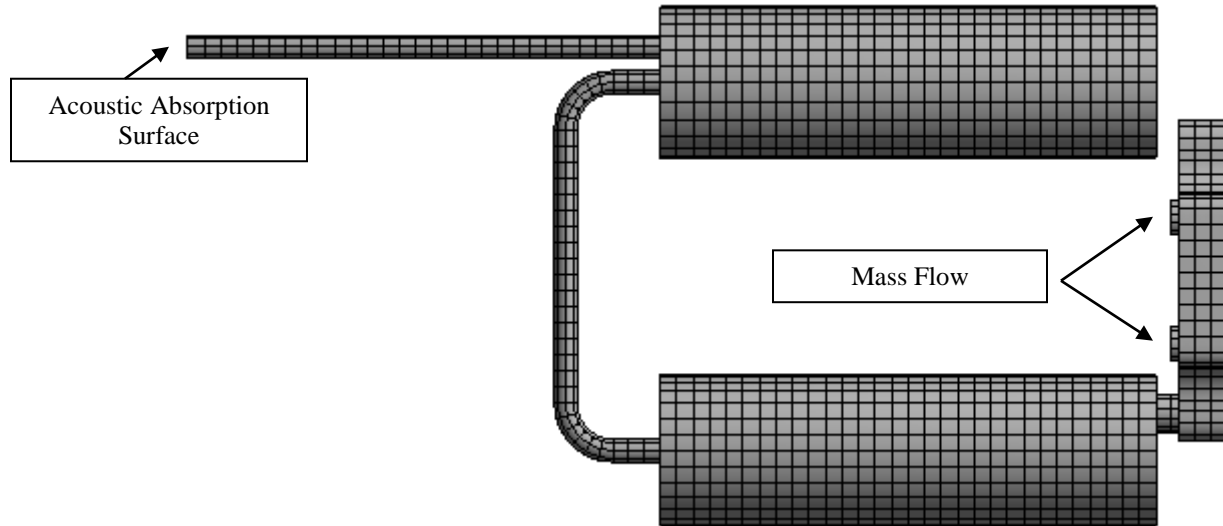


Figure 4: Mesh and boundary conditions for harmonic acoustics simulation.

3. RESULTS

FRF results are obtained by CFD with temporal discretization of 2×10^{-5} s, equivalent to a sampling rate of 50000Hz, the simulations were performed with static flow initial conditions until obtained a statistical stable flow, in this case the sixth cycle. In order to observe nonlinear effects due to the influence of mass flow, the discharge system was simulated for ARI LBP and MBP rating points, each one for three pump sizes requiring correspondent mass flow rate profile exemplified in Figure 3. The results of each pump according to size order are referred as CFD1, 2 and 3. Acoustics results of the FRF were obtained through single simulation for the mentioned rating points, differentiated by the fluid properties set in the acoustics solver.

The FRF calculation through CFD requires temporal sampling of the excitation (inlet) and the resulting (outlet) pressures from the flow field, an example of the sampled pressure signals is illustrated in Figure 5. The sampled pressure signals are treated by Fast Fourier Transform (FFT) method to obtain the pressure magnitude spectrum. The FRF is calculated using the FFT results, based on the estimator provided in Equation 1 for deterministic methods:

$$FRF(f) = \frac{P_{out}(f)}{P_{in}(f)} \quad (1)$$

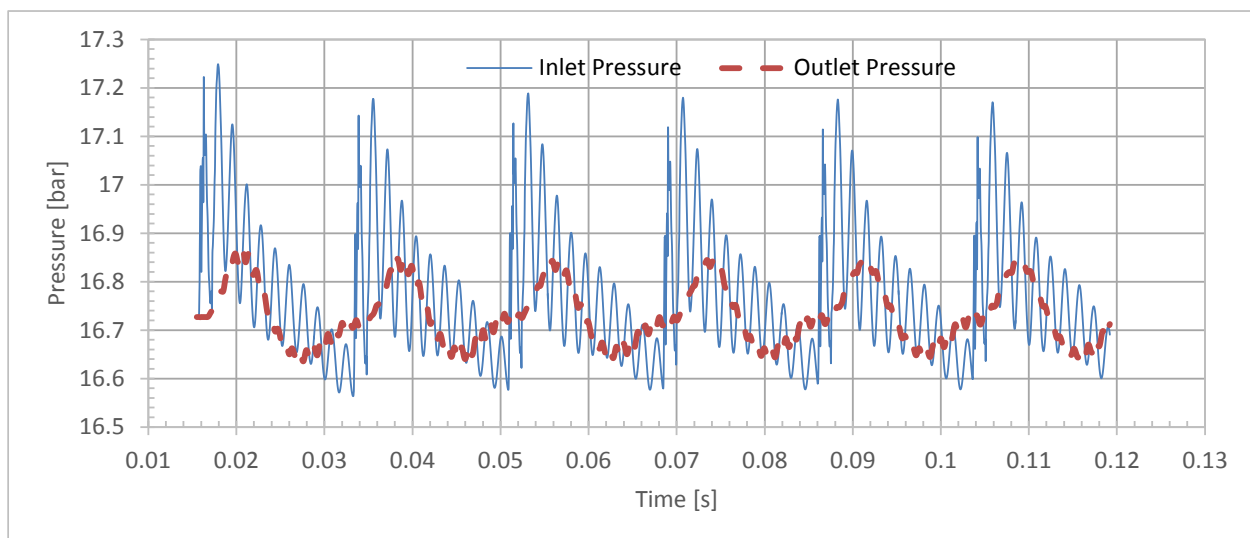


Figure 5: Inlet and outlet pressure temporal evolution.

Pressure spectrum magnitude results are presented in Figures 6 and 7, for inlet and outlet, respectively. One can note good agreement among all conditions simulated up to 1500Hz for inlet and 1400Hz for outlet, especially concerning identification of the main frequencies. Significant difference is found for the outlet pressure spectrum magnitude results for frequencies higher than 1400Hz, mainly related to the operating point. Compared to LBP results, the MBP results presents the main peaks anticipated and an overall increase of the spectrum due to the significant increase of the mass flow.

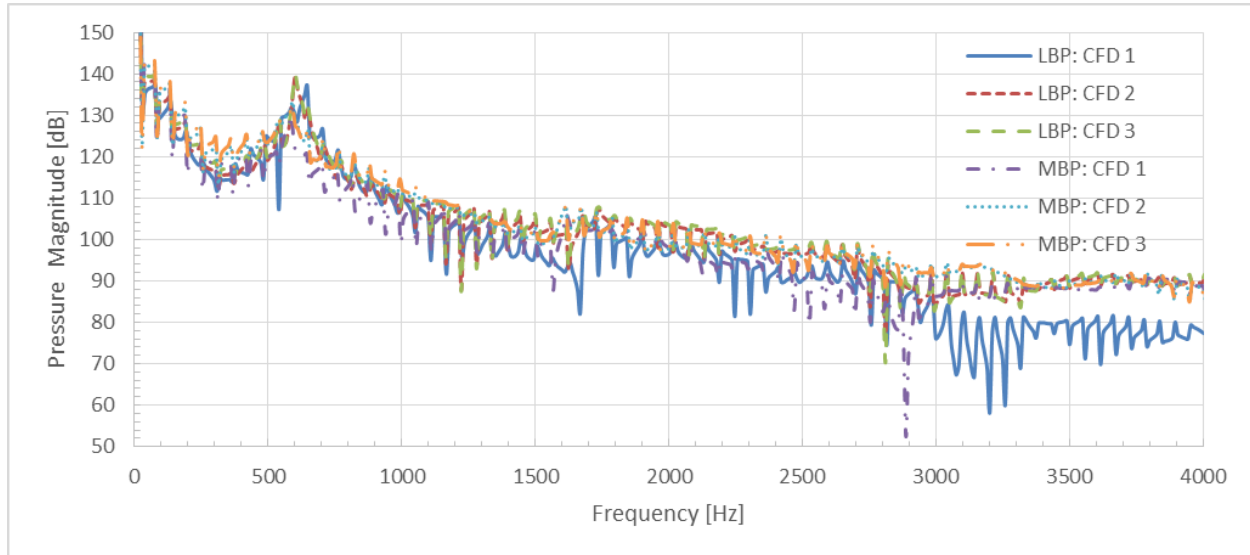


Figure 6: Spectrum magnitude of inlet pressure.

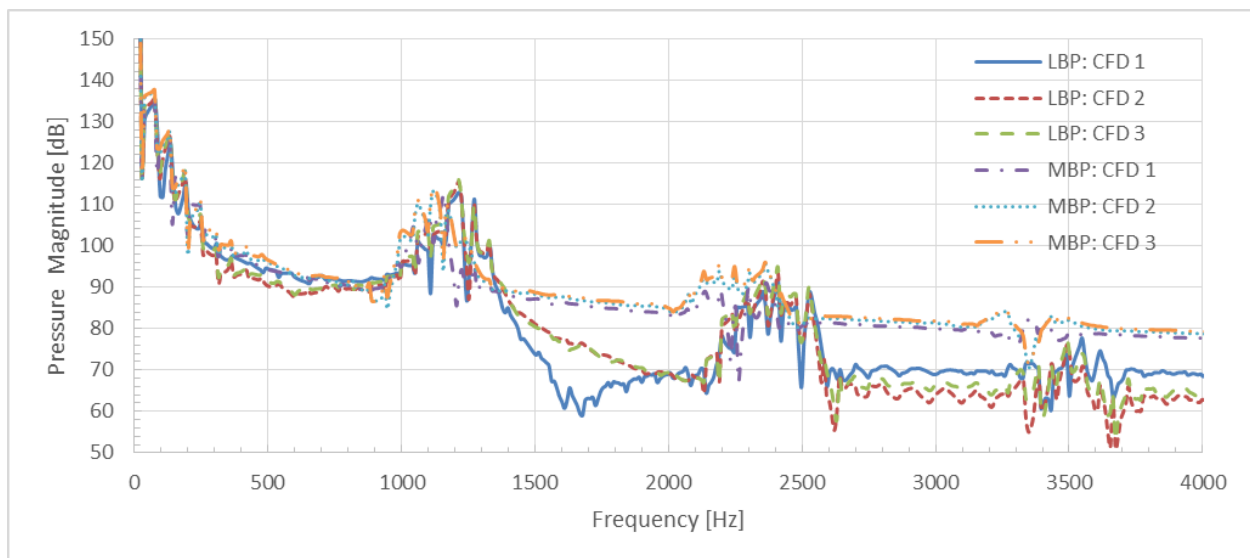


Figure 7: .Spectrum magnitude of outlet pressure.

Results of the FRF's obtained through CFD are compared to acoustics results in Figures 8 and 9, for LBP and MBP rating points, respectively. For LBP, the main profile shape agrees reasonably with the acoustics results until 1500Hz, then the agreement decreases until 2500Hz, from this frequency on the CFD results does not emulate any characteristics observed in the acoustics results.

In the range until 1500Hz, there is no identification of the first two resonances, one possible reason is due their proximity to the average compressor frequency. The first anti-resonance agrees well while the group of resonances

around 1250Hz, the CFD approach identifies only the intermediate. Despite the effects of mass flow rate at LBP conditions, it was not found relevant differences among pump sizes.

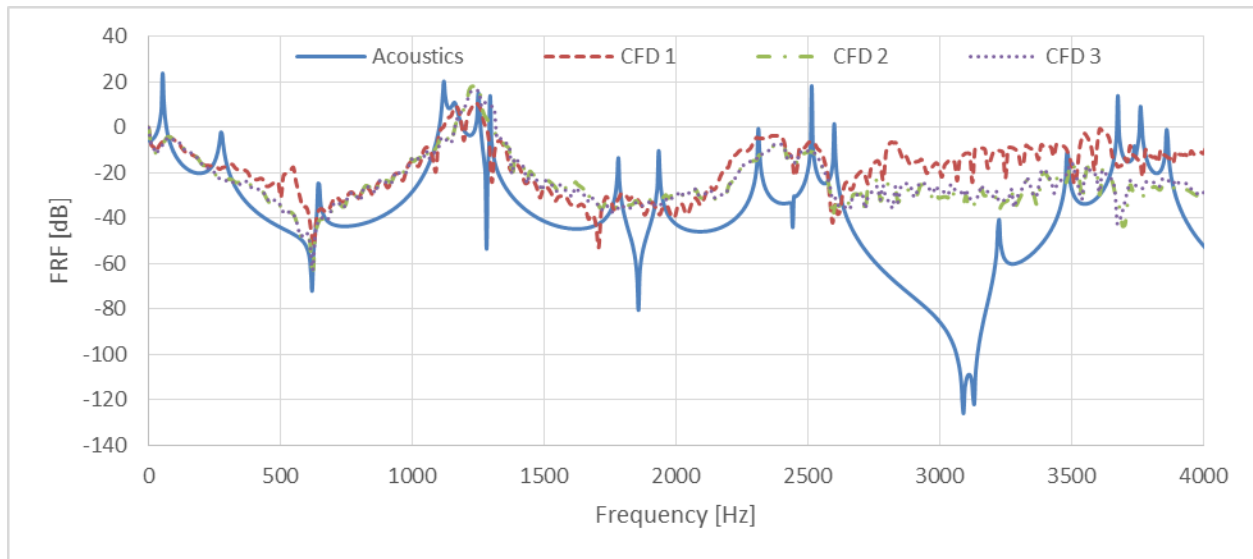


Figure 8: Frequency response function results for ARI LBP.

It is observed higher level of disagreement between CFD and acoustics approaches for MBP condition. In a similar way, it is possible identify the first anti-resonance and the intermediate resonance around resonance group till 1250 Hz, however the values are significantly damped compare to the acoustics results and the ones found for LBP condition.

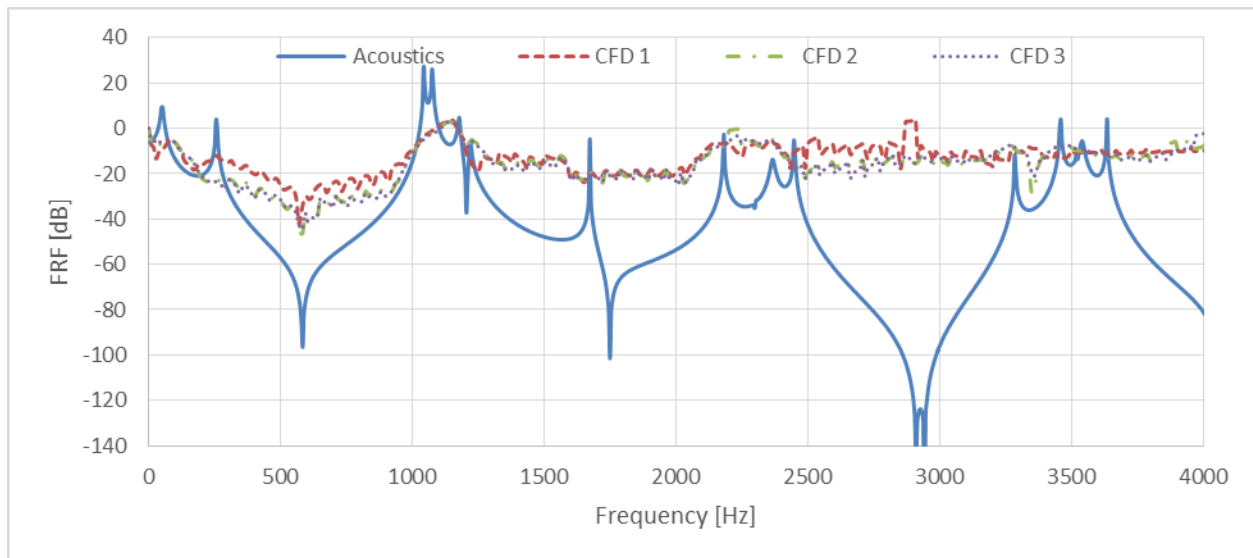


Figure 9: Frequency response function results for ARI MBP.

The interpretation of the results obtained for the rating point conditions indicates relevant influence of mass flow at expressive increase levels, for example, ideally the mass flow rate at MBP condition is 85% higher than LBP. For the pump sizes approached at each rating condition the mass flow rates vary within 36%.

A complementary comparison through visualization of solution fields is performed in order to check the consistency of the FRF results and presented only for LBP. The identification of the first six acoustic modes is presented in Figure 10 as acoustics pressure contours.

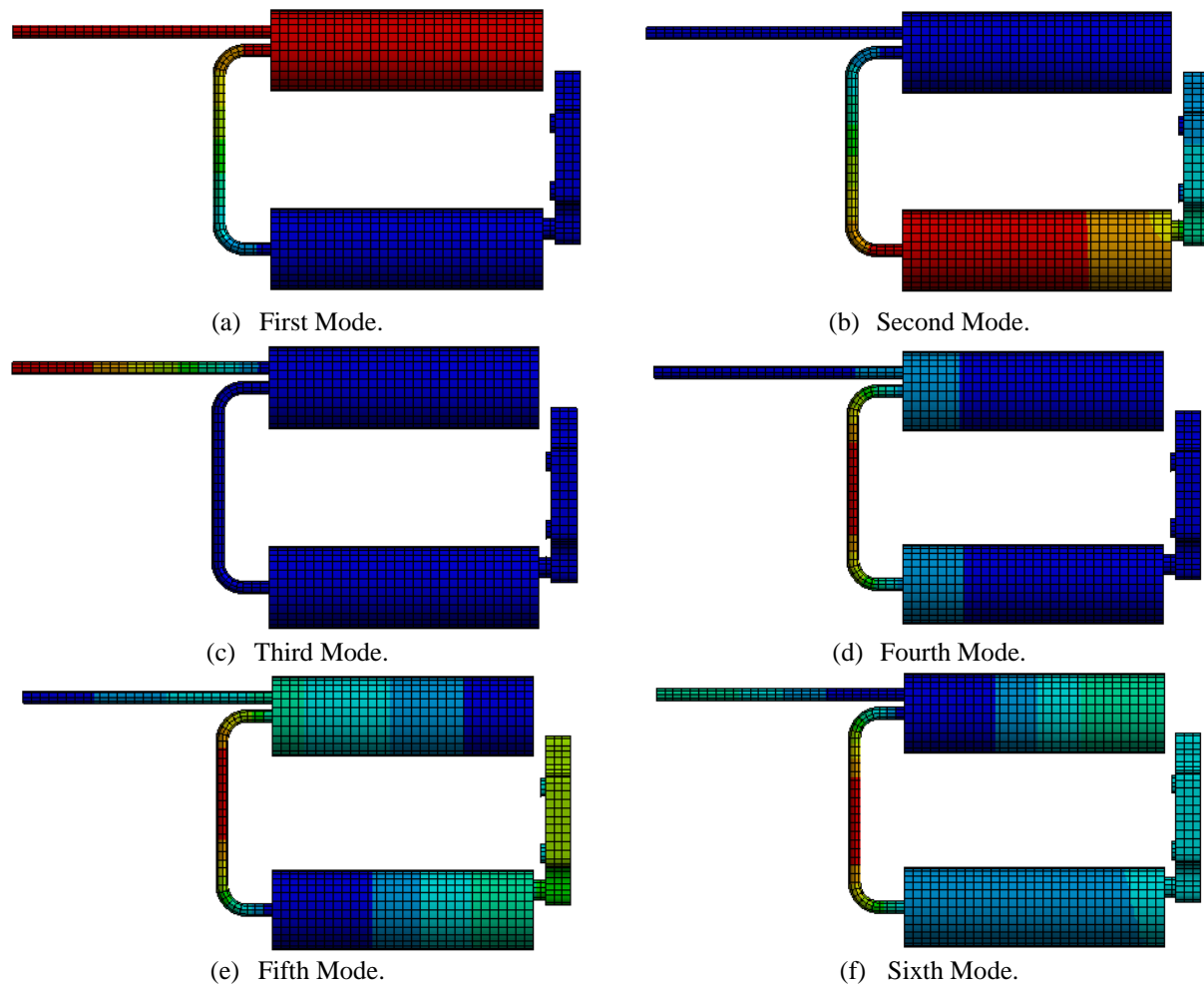


Figure 10: Acoustic mode shapes in pressure contours.

Based on the frequencies and acoustics pressure contours of the acoustic modes the pressure field obtained via CFD is analyzed for identification of similar patterns, depicted in Figures 11 and 12.

Equivalent periodic patterns to the second and third modes are clearly identified in Figure 11, however a deeper insight reveals the equivalent second mode patterns its fully correlated with the mass flow pulse of 60 Hz frequency, which corroborates the absence of the first and second modes identifications in FRF results. The remaining acoustic modes are very similar in patterns, a possible reason why the CFD approach does not distinguish them, due to the mass flow effects, and capture one intermediate mode in FRF, which is presented as time snapshots in Figure 12.

Observing practical applicability the CFD computational cost is significant higher compared to acoustics simulation, it was observed a total calculation time up to 120 times longer. The differences found does not indicate the acoustic method is unsuitable to predicted tendencies, based on this one can conclude is still more effective for large optimization computations.

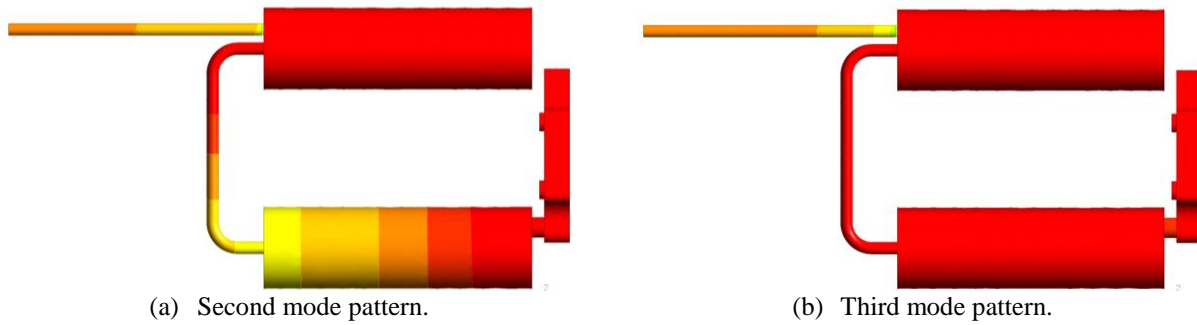


Figure 11: Pressure flow patterns equivalence to acoustic modes.

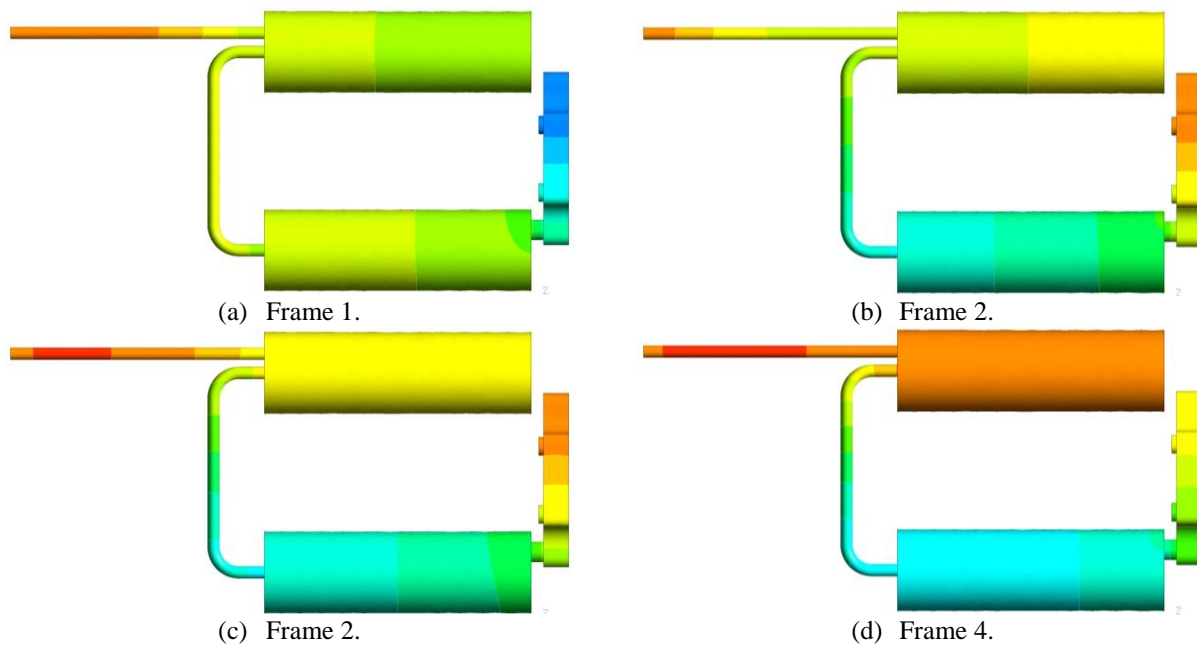


Figure 12: Pressure flow patterns equivalence to fourth-sixth acoustic modes.

4. CONCLUSIONS

In the presented work, an evaluation of CFD approach as coupled technique for analysis of acoustic performance is presented; for this purpose the FRF of a simplified discharge system was obtained through acoustic and CFD numerical methods. It was observed the CFD approach is able to predict FRF with similar profiles to the acoustics method and includes mass flow effects, which was evidenced that for higher mass flow rates the FRF obtained from CFD solution tends to distance from acoustic FRF, as the mass flow rate increases. The FRF results were confirmed observing the pressure patterns in both acoustics and flow solution field. Although the CFD approach is useful to perform acoustics evaluation and it is able to include mass flow effects in the analysis, the computational cost is more expensive compared to acoustics approach, becoming unsuitable for large computation in optimization procedures. On the other hand, due to the wider inclusion of physical phenomena, its applicability is appropriated as complementary evaluation of the acoustics approach in order to understand the effects of the mass flow on the muffler's FRF previously to performing experiments.

NOMENCLATURE

CFD	Computational Fluid Dynamics	(-)
FFT	Fast Fourier Transform	(-)
FRF	Frequency Response Function	(-)
ILU	Incomplete LU factorization	(-)

LBP	Low Back Pressure	(-)
MBP	Medium Back Pressure	(-)
SST	Shear Stress Transport	(-)
P	Pressure spectrum	(Pa)

Subscript

in	inlet
out	outlet

REFERENCES

Bratti, G., Santini Junior, O., Baars, E., Barbieri, R. & Pellegrini, C. (2012). Identification of Energy Path in the Interaction between Compressor and Refrigerator. *Proceedings of the 21st International Compressor Engineering Conference at Purdue* (Paper 2091). West Lafayette, USA: Purdue University.

Done, V., Balide, V., Tamma, B., Soni, K., Dey, S., Angadi, S. & GP, V. (2014). Muffler Design for a Refrigerator Compressor. *Proceedings of the 22nd International Compressor Engineering Conference at Purdue* (Paper 2262). West Lafayette, USA: Purdue University.

Kim, S., Cheong, C., Park, J., Kim, H. & Lee, H. (2015). Numerical and Experimental Investigation into Flow and Acoustic Performances of Muffler for a Hermetic Reciprocating Compressor. *Proceedings of the 22nd International Congress on Sound and Vibration*. Florence, Italy: IIAV.

Park, J., Kang, J., Choi, J., Jang, P., Yoon, M., Kwon, D., & Oh, I. (2008). Reduction of the Low Frequency Noise Due to the Discharge Pressure Pulsation of a Reciprocating Compressor. *Proceedings of the 19th International Compressor Engineering Conference at Purdue* (Paper 1898). West Lafayette, USA: Purdue University.

Rodrigues, T. T., Da Silva, D. L. (2014). Turbulence Modelling Evaluation for Reciprocating Compressor Simulation. *Proceedings of the 22nd International Compressor Engineering Conference at Purdue* (Paper 2297). West Lafayette, USA: Purdue University.

Soedel, W. (2006). *Sound and Vibrations of Positive Displacement Compressors*. Boca Raton, FL: CRC Press.

Ussyk, M.S., "Numerical Simulation of Hermetic Reciprocating Compressor Performance", (in Portuguese), M.Sc. Dissertation, Federal University of Santa Catarina, 1984.

Vendrami, C. E.; Pellegrini, C., Hattori, M. A. & Climaco, D. (2014). Acoustic Improvements for a New Generation of Variable Speed Compressor. *Proceedings of the 22nd International Compressor Engineering Conference at Purdue* (Paper 2308). West Lafayette, USA: Purdue University.

ACKNOWLEDGEMENT

The authors acknowledge the EMBRACO technical and financial support