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Simulation of a Refrigeration Compressor Evaluating Accuracy of Results with Variation in 3D Component Discretization

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ABSTRACT

Nowadays, simulation tools are becoming mandatory for refrigeration compressor design, not only to reduce development time improving the responsiveness to the market, but to better understand the complex transient phenomena taking place inside the product. Hybrid models regarding the compressor as a group of components modeled as 1D and 3D are currently the most promising approach for simulation purposes. Flow components with spatial characteristics are normally modeled with CFD, providing excellent accuracy but demanding considerable computational cost due to the high levels of discretization. In this work, three spatial components of a hermetic refrigeration compressor were simulated with low levels of discretization to reduce computational time. Three models were created for each one of these components to consider different levels of discretization, starting from the most basic 1D. The results were compared with experimental data to check the accuracy. For some components, the discretization level didn't seem to make any difference, but others showed better results with higher discretization levels. In general a good agreement was observed between simulation and experiment.

1. INTRODUCTION

The most common approaches used to simulate refrigeration compressors in transient conditions can be classified in three basic groups. The first one considers the components as one dimensional entities, or 1D, where parts with spatial characteristics are simplified to equivalent regular shapes, reducing drastically the number of parameters involved and the time required for data processing as well. The results usually present good accuracy, however, problems might happen when the three dimensional aspects are important for the phenomena in study. Ancel and Ginies (2008) applied a 1D approach to simulate a multi-cylinder reciprocating compressor. As a result, product development time was reduced and many experimental laboratory tests were avoided. Similar work was developed by Damle *et al* (2008) investigating different compressor configurations and calculating PV diagrams for some refrigerants.

The second simulation approach models the system components as three dimensional elements. This is a three dimensional, or 3D, where the solution and the strategy is to discretize the volume, which means to divide it into elements small enough to consider the properties of the fluid constant inside each volume. The equations involved are solved for each element along the whole volume and for each time step along the desired period. This technique is generically called Computational Fluid Dynamics (CFD) and is well known to provide very accurate results for fluids in a three dimensional basis, but on the other hand is very time consuming due to the great amount of calculations involved. Trying to reduce this time, Abidin *et al* (2006) applied a method called Domain Decomposition for the 3D simulation of a piston cylinder system of a hermetic reciprocating compressor which would otherwise take several days with a regular CFD approach. In the same direction, Pereira *et al* (2012) proposed a simplified CFD model reducing the computational cost, making it feasible for optimization purposes. Nakano and Kinjo (2008) listed a variety of CFD applications on reciprocating compressors and described in detail the solution of a pressure pulsation problem in the suction muffler.

Another relevant aspect of this technique is about how to discretize the volumes, for example what kind of shape the elements have to assume and how tiny they have to be. These questions are answered by another sophisticated method called Meshing. Volumes that change with time, like compression chambers, demand a Moving Mesh technique as used by Lv *et al.* (2011) to investigate the effect of the suction line pressure pulsation in the performance of a rotary compressor. Thermal evaluations were conducted by Almbauer *et al.* (2006), comparing the Thermal Network technique applied to a reciprocating compressor with results obtained from a full 3D simulation.
Finally, the third approach is a combination of the 1D and the 3D approaches mentioned above, normally called hybrid, as in the work of Sauls and Novak (2012) developed to estimate compressor pressure pulsations. This is a very interesting way to take advantage of the accuracy and the low computational time simultaneously. In this solution, some components are modeled as 1D and others as 3D according to their characteristics. Tubes, connections and valves, for example, are more likely modeled as 1D elements as the spatial aspects are normally not relevant. Conversely, components like mufflers and chambers are more likely modeled as 3D due to their volume geometric characteristics. Interesting examples of hermetic compressor simulation coupling 0D, 1D and 3D component models were developed by Lang *et al* (2008) and Almbauer *et al* (2010), where the pressure pulsation in the suction line was investigated and validated and the method showed to be an excellent design tool.
The comprehensive work done by Pereira *et al* (2008) compared the accuracy of three different models of a reciprocating compressor; they were carried out using full one, two and three dimensional formulations separately. The full 3D model presented results quite close to the measurements, but is still impractical in terms of processing time. In spite of being less accurate, the 1D and 2D models were not so far from the experiment in most of the results.
The literature reviewed here seems to point to hybrid models as the most interesting approach for refrigeration compressor simulation, combining accuracy with fast response. However, the CFD technique usually demands a high level of discretization for the 3D components, which may not be necessary sometimes. Our contribution with this work is to explore lower levels of discretization for 3D components trying to balance accuracy and low computational time. A hermetic refrigeration compressor was modeled and their 3D components were modeled with different levels of discretization starting from the basic 1D version. Very low levels were considered, especially when compared with the usual CFD, and the results were compared with measurements to check accuracy. The compressor components subject to this coarse 3D discretization were Suction Muffler, Cylinder Head and Discharge Mufflers. Three different levels of discretization were created for each compared to experiment. The other components were modeled as 1D.

2. ANALYTICAL MODEL

The analytical model development was based on a transient regime of a stabilized condition, so the dynamics of the system was observed along the cycle in shaft angle basis after having a stable cycle-to-cycle condition achieved. Some components had their models developed as one dimensional and others as three dimensional. All the development was done in the GT-Suite V7.4 software, where a great amount of templates are available in a library organized in specific groups like Mechanical, Flow, Pneumatic, etc. After choosing the template that best fits each component, some specific aspects about it are provided for characterization purposes. Pipes, in general, have their dimensions and surfaces roughness established, other components, like valves, require some experimental work to have their parameters characterized like Drag Coefficient, Natural Frequency and Damping Factor. For checking purposes, when possible, the parameter was evaluated by more than one method, this is recommended especially when its magnitude is previously unknown.
The software also provides some specific tools to deal with three dimensional components. The tools were employed in this work to create the three dimensional geometry for the parts and to manipulate them along the discretization process and finally generate the file compatible for simulation. Creating the spatial geometry of the component is the first step and there are two possibilities at this point: one is to create it using a tool called "GT-SpaceClaim" and the other is to import it from any other source CAD file provider. Both options were used, the Suction Muffler, for example, was imported from a Parasolid file generated in a UniGraphics platform; the Cylinder Head and Discharge Mufflers were created in the Space Claim. After having the geometry established, it is exported to another tool called GEM-3D, which is able to discretize the volumes manually or automatically. The final result is a file able to run in the simulation platform.
This transformation process was applied to three components of the compressor depicted in Figure 1. The CAD version is shown in the first column on the left, and their levels of discretization are on the three columns on the right. The most basic models are presented in column A, getting gradually more sophisticated in columns B and C. As already mentioned, the discretization can be manually defined using planes to cut the volume into smaller elements, this mode was used in most of the models in columns A and B. Alternatively, the models in column C were generated

automatically after having the element dimensions defined on the three axes' directions. There is a number in brackets beside each model, which is the number of elements the volumes were divided in. The Discharge Mufflers model on column C, for example, had each volume divided in 14 elements totalizing 28, which is the number indicated in brackets.

The wall temperatures along the gas flow path were defined based on experimental measurement and pervious experience, this estimates the heat exchanged between the gas and the components.

2.1 Pipes, bends and orifices

Modeling of these basic flow elements are discussed in Section 3.

2.2 Valves

The valves were modeled considering the spring and mass model, extensively used in refrigeration compressor simulation. The parameters for these components were determined by experimental and theoretical methods. A test rig was developed to determine the drag coefficient curve as a function of the lift evaluated in different valve positions. The natural frequency and spring stiffness were estimated by experimental and theoretical means. A cantilever beam, analog to the valve leaf, was created in the software to evaluate the Natural Frequency and Damping Factor, reproducing the procedure applied on the experimental approach. Resistance of Material theory (Beer and Johnston, 1981) was used to estimate the spring stiffness of the valves, once more trying to reproduce the experimental procedure.

2.3 Suction Muffler

Considering the 3D components, the Suction Muffler is the first one along the refrigerant path. Its geometrical shape has basically two chambers connected by pipes. The most basic model of this component (Figure 1 column A) models the first chamber as one volume, as well as the second chamber. The intermediate model, in column B, considered the first chamber divided into 4 elements or sub volumes. Column C shows the most sophisticated model, where the both chambers were divided in 6 elements each.

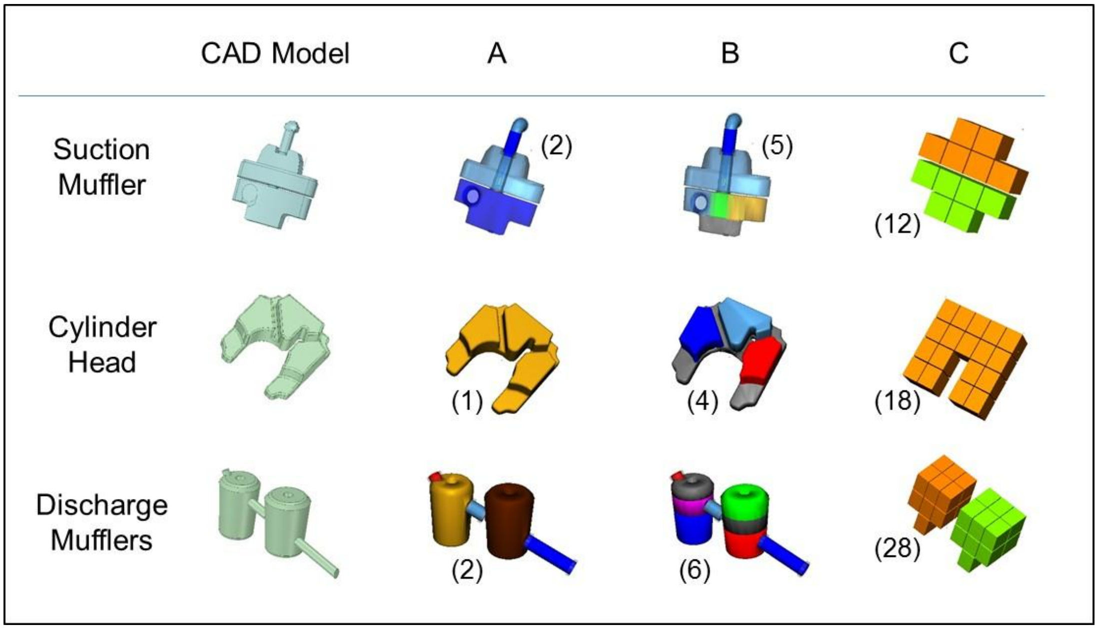

Figure 1: 3D components and their discretization levels.

2.4 Cylinder Head

After the compression chamber the refrigerant goes to the Cylinder Head passing through the discharge valve, the volume is occupied very fast at high pressure. Its spatial geometry was created and the first level of discretization was represented by a single volume, column A in Figure 1. Column B presents the volume divided in 4 elements describing the intermediate level of discretization. Finally the third level, where the Cylinder Head volume was discretized in 18 elements according to column C.

2.5 Discharge Mufflers
Two volumes connected by pipes is basically the geometry of the Discharge Mufflers. They were discretized first as one single volume for each muffler, as in column A Figure 1. The model in column B considered each muffler divided in three elements and the version in column C the mufflers were sub divided in 14 elements each.

2.6 Electrical Motor
The electrical motor torque was assumed to be a function of three variables: voltage applied, winding temperature and motor speed. This model was implemented in GT-Suite by means of tables. The input data was generated in a software package called “Speed”, specific for Electromagnetic Electric Motors simulation. The motor was connected to the mechanical system, also called the pump, and the torque and speed behavior along the cycle could be estimated based on components inertias and a simplified calculation of bearing losses.

2.7 Compressor
This development considered a 60Hz reciprocating compressor with a displacement of 12cc using the R404A gas refrigerant. Dimensions and working condition studied are:

Bore: 27.42mm	Suction Pressure = 437.8kPa
Stroke: 20.32mm	Suction Temperature = 35.4C
Compression Ratio: 75 (Displacement/Clearance)	Discharge Pressure = 2299.4kPa

3. DESCRIPTION OF NUMERICAL METHOD

The simulations performed in this work were done using the GT-Suite simulation software, developed by Gamma Technologies Inc. The software provides a 1D Navier-Stokes solution in most components, with select volumes having a quasi-3D Navier-Stokes solution. The compressor model is built graphically in the tool "GT-ISE", with the appropriate physical volumes (pipes, flow junctions, etc.) placed onto the map and connected together. Relevant geometric inputs are also provided at this time, or are determined by using the "GEM-3D" tool, described later. Such geometry includes measurements like pipe length and diameter, flow junction angles and expansion areas, if applicable, and also numerical parameters like discretization length.

3.1 Basic Flow Solution
Based on the development of the 1D compressor model graphically, all flow volumes are discretized into a staggered grid according to Figure 2, whereby scalar variables (pressure, temperature, density, etc.) are assumed to be uniform throughout each volume. Vector variables (mass flux, velocities, etc.) are calculated for each boundary.

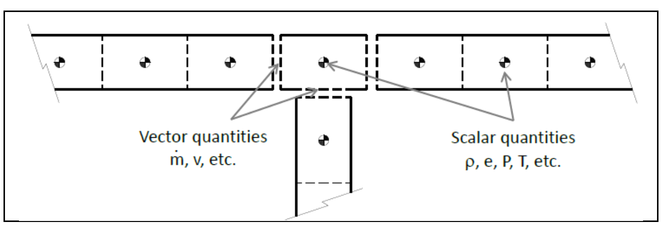

Figure 2: 1D flow volumes discretized into a staggered grid.

The following Navier-Stokes 1-D equations are solved with this grid:

Continuity:
$$\frac{dm}{dt} = \sum_{\text{boundaries}} \dot{m} \tag{1}$$

Energy:
$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_{boundaries} (\dot{m}H) - hA_s (T_{fluid} - T_{wall}) \tag{2}$$

Momentum: $$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{\text{boundaries}} (\dot{m}u) - 4C_f \frac{\rho u|u|}{2} \frac{dxA}{D} - C_p \left(\frac{1}{2}\rho u|u|\right) A}{dx} \tag{3}$$

An explicit forward time marching scheme is used for the solution of these equations. The solution variables are mass flow, density and internal energy. The solution scheme uses each subvolume and its neighbors for numerical integration. The time step taken must obey the Courant condition. A modified version of the Courant condition is used, to account for physics that were not included in Courant's original analysis.

Pipes $$\frac{\Delta t}{\Delta x}\left(|u| + c\right) \leq 0.8 * b \tag{4}$$

'Flow Split' flow junctions $$\Delta t \propto \frac{\rho V}{\rho A_{eff}\, c} \tag{5}$$

Generally this explicit formulation will result in very small time steps, which is desirable for detailed crank-angle based compressor simulations. For these simulations, a high degree of resolution is required to capture the complex flow physics - for example where pressure wave dynamics are important.

3.2 Pipe and Flow Junction Modeling Details:

In 1D piping elements, standard flow losses are calculated in each sub-volume. Such flow losses include effects due to friction (using the explicit Colebrook formulation of the Moody diagram) and geometry like bends or tapers. In between 1D pipe elements, losses due to contractions and expansions (sudden or otherwise) are accounted for via "orifice" models.

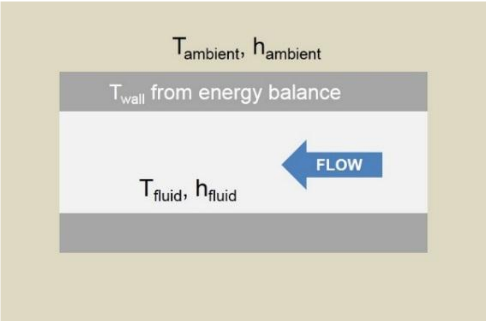

Figure 3: Schematic of Pipe thermal solution method.

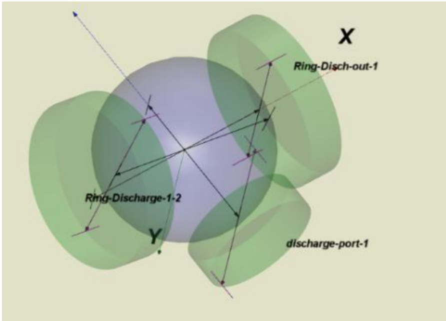

Figure 4: Graphical representation of flow junctions

Thermal effects are also accounted for via a thermal wall solution. This approach allows layers to be modeled and free-convection temperatures to provide heat transfer in or out of the fluid accordingly. Figure 3 illustrates a simple one-layer thermal solution, whereby the internal temperature and heat transfer coefficient (calculated via the Colburn equation) are used in conjunction with an ambient temperature and heat transfer coefficient to calculate the respective heat rates and wall temperature. This concept is expanded upon as more layers are added.

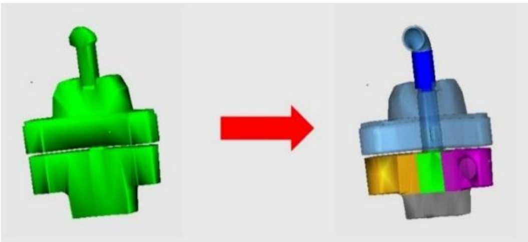

Figure 5: GT-Suite conversion from raw CAD to discretized volumes.

In flow junctions, arbitrary angles in three dimensions can be provided, which will allow what is effectively a 3D momentum solution. This allows improved accuracy over purely 0D flow junctions. Figure 4 is a graphical representation available in the software that illustrates the control volume (blue sphere) and connecting ports (green cylinders) at different angles.

Usually when modeling a 1D refrigerant compressor, CAD data is used for an existing system to provide a starting point. The tools "GEM-3D" and "GT-SpaceClaim" tools were used extensively during this study.

GEM-3D takes the prepared CAD data as input, and then converts to usable numerical models. Angle determination of flow junctions in 3D is also performed to ensure model accuracy.
This process also allows discretization options to be specified upfront, so one can graphically see the resulting numerical model. Figure 5 provides an illustration of the input CAD on the left, and the final discretized volumes on the right that will be used in the software.

4. EXPERIMENTAL PROCEDURE

Pressure transducers were used to measure the pressure in the suction muffler, suction chamber, cylinder and cylinder head pressure and also 1st and 2nd discharge chamber. The sensors were arranged to not reduce the volume in the cylinder head and mufflers or increase the dead volume in the cylinder. The transducers used are high sensitivity with high natural frequencies and are thermal compensated. The sensor range is from 0 to 35 bar and has an accuracy of 0.1% F.S. combining non-linearity and hysteresis. Figure 6 shows a schematic of position of sensors in the compressor.

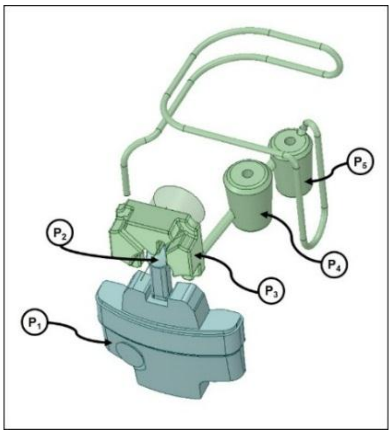

Figure 6: Pressure sensor positions.

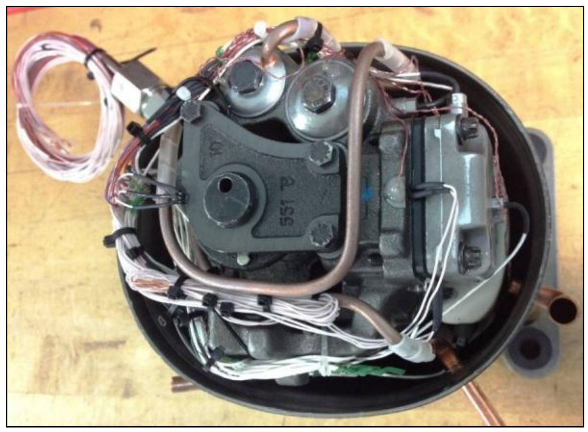
Figure 7: Overview compressor instrumented.

T Thermocouples were prepared and installed to measure the gas temperature inside the suction muffler, cylinder head and 1st and 2nd discharge muffler. The wall temperature was measured in the cylinder, suction and discharge tube and also in the upper and lower housing. The T Thermocouple range is from -200°C to 350°C and has an error of 1°C or 0.75%.
A Hall Effect sensor was used to get the information of crankshaft angular position and motor speed. The top dead center was used as reference (zero) for the crankshaft position.
The valve displacement was measured using strain gages technique. They were calibrated by associating the valve lift with measured strain. They were bonded in the suction valve and discharge valve retainer. The details are illustrated in the Figure 8.
The sensor wires pass through the shell with vacuum pressure feedthroughs. The compressor was assembled in a regular housing and no extra volume was added. In total, the compressors received 10 temperature sensors, 2 displacement sensors, 6 pressure sensors and 1 sensor for speed and angular position. Figure 7 shows an overview of instrumented compressor.
Special software was created using the Labview software to collect and process the sensor's data.

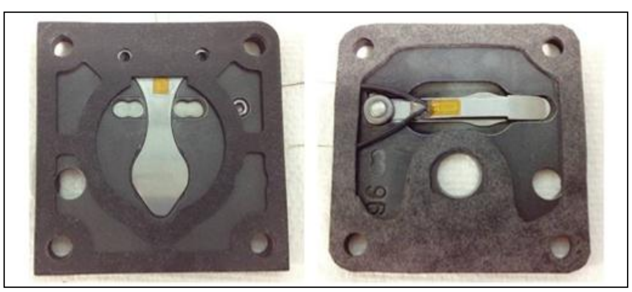
Figure 8: Strain gage bonded in the suction and discharge valve retainer.

5. RESULTS AND DISCUSSION

Three levels of discretization for each one of the 3D components provided different combinations regarding the model of the whole system. 27 combinations are possible, making the analysis process quite complex. The main idea was to choose the combinations that can offer insight regarding the effects of discretization. Varying the discretization level of the same component without changing the others seemed to be a good strategy.

To identify the model configurations of the complete system, a "code" was created based on three letters representing the level of discretization of each 3D component as mentioned in Figure 1, obeying the following the order. The first letter represents the discretization level of the Suction Muffler, the second letter the Cylinder Head and the third letter the Discharge Mufflers. According to this rule the configuration AAB, for example, would represent the compressor model configuration with model level A to the Suction Muffler, level A to the Cylinder Head and level B to the Discharge Mufflers. This rule was followed to codify the compressor model configuration concerning the 3D component levels of discretization.

Figure 6 shows the volume occupied by the refrigerant of the whole system and the points where the static pressures were measured. Figures 9 to 18 show the comparison of measured and simulated pressure results along the shaft angle in a cycle, they are displayed in two columns according to the model configurations. The graphics on the left show the experimental result and different compressor model configurations. The discretization of the components is kept constant except for the component that is being compared to measurement. For example, Figure 9 shows the pressure at the first chamber of the Suction Muffler. Three results are plotted with the three different levels of discretization of the Suction Muffler, represented by the first letter (A, B and C) of each configuration. The other components (Cylinder Head and Discharge Mufflers) were kept constant at their most basic models represented by the second and third letters (all A). Figure 10 shows the most sophisticated model (CCC) considering the level of discretization C for all 3D components. The darker line represents the experimental result.

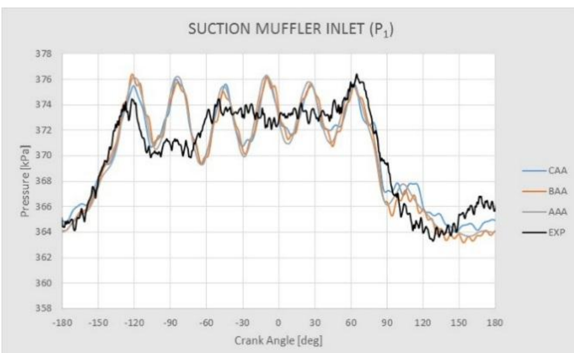

Figure 9: Suction Muffler inlet results.

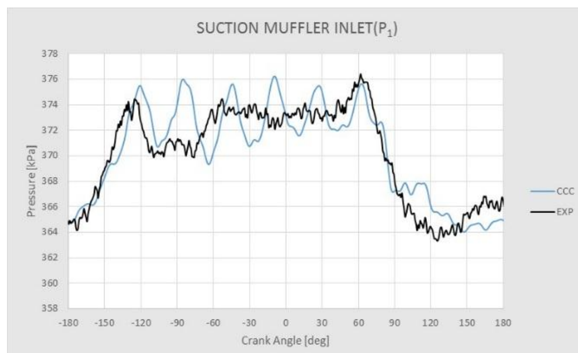

Figure 10: Suction Muffler Inlet results.

Figures 9 and 10 present the pressure in the first chamber of the Suction Muffler. The pressure reduction observed from 60 to -120 degree due to the suction valve opening is very well represented by the model. However the pressure profile from -120 to 60 degree show a slight discrepancy, which seems to be a sensor accuracy problem as the amplitude is quite small. We don’t observe significant differences among the models indicating that the discretization doesn’t make any improvement for the pressure description in this component. The only aspect is about the high frequency pressure pulsation observed in the model BAA, slightly in the model CAA and absent in AAA. The model CCC matches pretty well with the CAA, even being much less sophisticated.

Still in the Suction Muffler, but now at the exit port, Figure 11 shows the pressure profile at this point. Once more the simulated results show very similar results meaning that the discretization is not providing a better description of the phenomena. Comparing the results, there is a smaller amplitude of the measured pressure pulsation from -150 to 60 degree, when compared with the simulation, but the frequency shows good agreement. The valve opening provides a sudden drop in the pressure level, which is quite well described by the model results in all configurations. Once more, CCC model presented a behavior very similar to CAA.

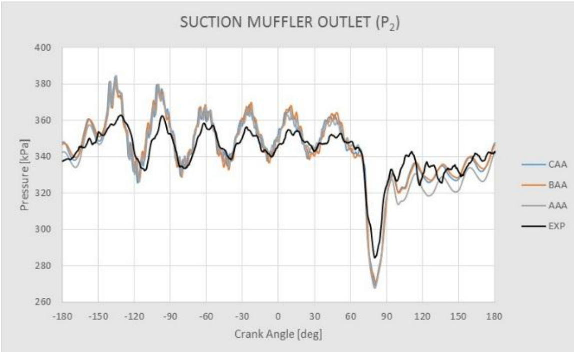

Figure 11: Suction Muffler outlet results.

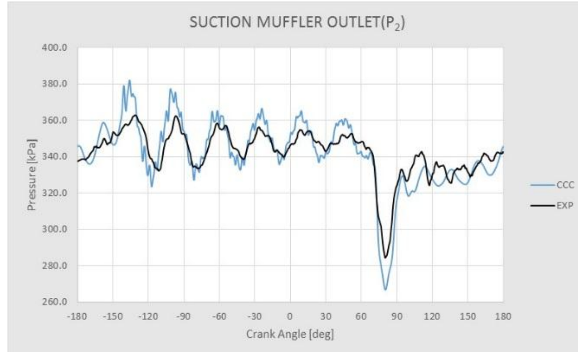

Figure 12: Suction Muffler Outlet results.

Excellent results were obtained for the pressure inside the Cylinder Head for all three models according to Figure 13. Besides the pressure profile, there is a pressure pulsation of high frequency and low amplitude, which shows up in the experimental results and on the models ABA and ACA, but not in AAA. This aspect deserves some attention because if the study is focused in high frequency pressure pulsation then the model AAA is not recommended. Great similarity is observed on the model configuration CCC (Figure 14) and the models ABA and ACA (Figure 13).

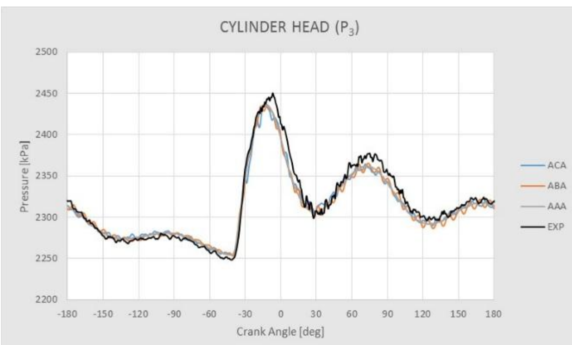

Figure 13: Cylinder Head pressure results.

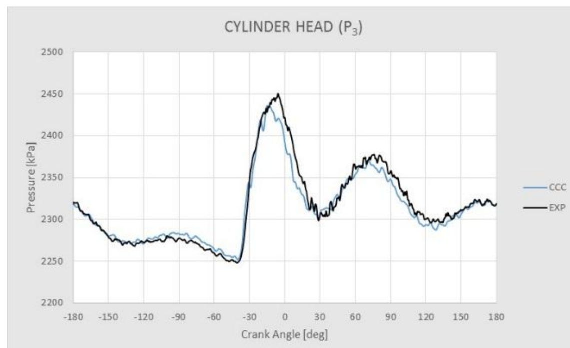

Figure 14: Cylinder Head pressure results.

Results for the First Discharge Muffler are shown in Figures 15 and 16. The amplitude of the pressure profile is slightly smaller compared to the experimental data closer results were provided by model AAC and CCC, although the profiles look similar for all three. The high frequency pressure pulsation shows up again in the measurements and is only slightly represented by the model AAB but better explained by model CCC. These two models are recommended if the idea is to analyze high frequency pressure pulsations on this component. Effects on the simulated pressure behavior in this First Discharge Muffler occurred when the model of the other two components changed. This can be observed comparing the results of the models AAC and CCC, this last one describes the high frequency pressure pulsation better than the other configurations. This is the effect of the Cylinder Head and/or Suction Muffler models, and not of the Discharge Mufflers.

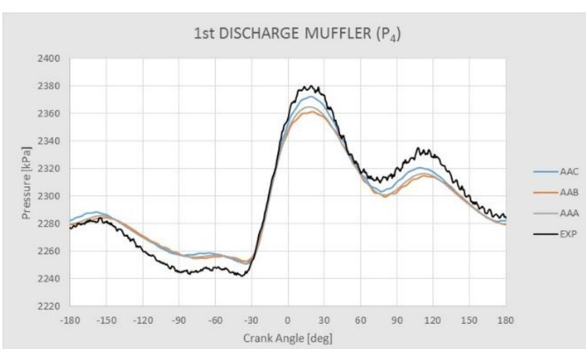

Figure 15: First Discharge Muffler results.

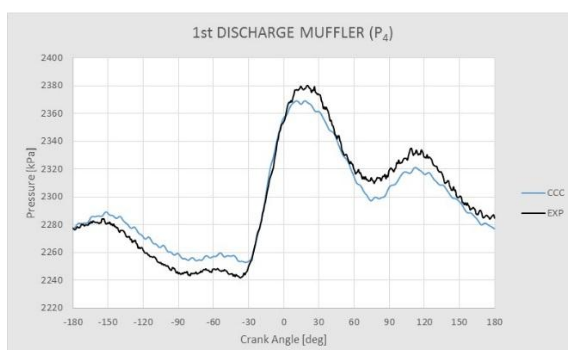

Figure 16: First Discharge Muffler results.

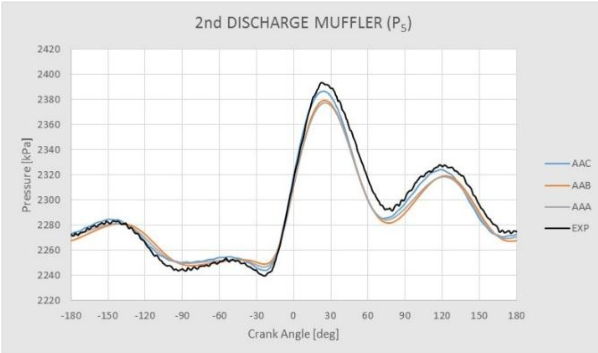

Figure 17: Second Discharge Muffler results

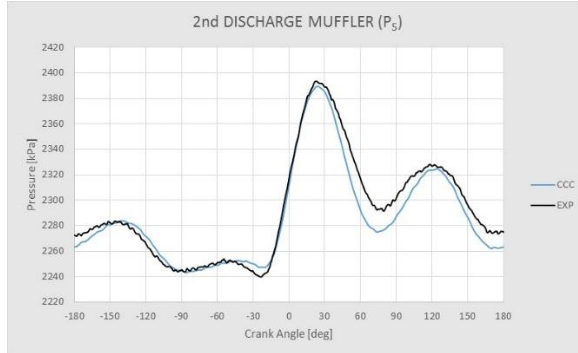

Figure 18: Second Discharge Muffler results.

Similar analysis can be made for the Second Discharge Muffler, in Figures 17 and 18, which presented close results compared to the measurement, but still a slightly smaller amplitude. Model AAC showed the closest result compared to the measurements, very similar to model CCC. Here the high frequency pressure pulsation appears slightly in the measurements and is very slightly represented by the model CCC.
Another metric to examine concerning the configurations simulated in this work is the processing time required to achieve stable state condition. It was considered achieved when a difference smaller than 0.2% between the mass flow, torque and pressure happened in two consecutive cycles. As expected, configuration CCC, the most sophisticated, required the highest calculation time of around 55 minutes, in contrast, the most basic configuration AAA took around 60% of this time. It is important to mention that no optimization work was done to reduce this run time, so there is still room to have this time reduced. However the results we already have make possible any kind of optimization and/or fast evaluation pretty feasible. An Intel Core i7-2.2GHz and 8GB RAM machine was employed in this work.

6. CONCLUSIONS

- The discretization process applied to the Suction Muffler didn't show any significant effect on the results obtained for this component, they are all very similar and showed good agreement with measurements,
- Higher levels of discretization of the Discharge Mufflers provided results closer to the experimental data,
- Simulated results showed that the First Discharge Muffler was affected when different levels of discretization were applied for the others components (Cylinder Head and/or Suction Muffler),
- Higher level of discretization models should be applied when high frequency pressure pulsation are subject of analysis,
- Configurations with components in their lower levels of discretization were not able to predict the high frequency pressure pulsation observed in the calculated results for the Cylinder Head and Discharge Mufflers,
- The most sophisticated model configuration provided the highest processing time to achieve stable state regime, which was around 55 minutes.

7. SUGGESTIONS FOR FUTURE WORKS

Investigate 3D models with higher levels of discretization to better describe high frequency pressure pulsations.

8. NOMENCLATURE

b = time step multiplier specified by the user (less than or equal to 1.0)
m = mass of the volume
V = volume
p = pressure
ρ = density
A = flow area (cross-sectional)
dx = length of mass element in the flow direction (discretization length)
C_p = pressure loss coefficient
D = equivalent diameter
dp = pressure differential acting across dx
Δt = time step [s]
Δx = minimum discretized element length [m]

A_s = heat transfer surface area
h = heat transfer coefficient
T_{fluid} = fluid temperature
T_{wall} = wall temperature
u = velocity at the boundary
$\dot{m}$ = boundary mass flux into volume,

$$\dot{m} = \rho A_{eff} u$$

C_f = skin friction coefficient

u = fluid velocity [m/s]
c = speed of sound [m/s]
V = flow split volume [m^3]
ρ = fluid density [kg/m^3]
A_{eff} = area for flow [m^2]
e = total internal energy (internal energy plus kinetic energy) per unit mass

H = total enthalpy, $H = e + \frac{p}{\rho}$

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