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An experimental and numerical analysis of refrigerant flow inside the suction muffler of hermetic reciprocating compressor

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ABSTRACT
In this study, detailed temperature and pressure measurements were performed at the inlet, outlet and outer boundaries of the suction muffler of a hermetic reciprocating compressor. The measurements were conducted without effecting real phenomena. After experimental studies, detailed computational fluid dynamic analysis of the refrigerant flow (isobutane) in the suction muffler was performed. Experimental pressure and temperature values at the inlet, outlet and outer boundaries of the suction muffler were used as boundary conditions. The effect of suction valve (opening and closing) at the exit of the suction muffler has been considered. 3-dimensional time dependent calculations were completed when statistically steady state convergence was reached for one crank period. Realizable k-ε turbulence model with appropriate parameters, second order discretizations for time and space derivatives and real gas model for isobutane (R600a) were applied for the numerical analysis. Mesh dependency of the analysis and solver algorithms were also investigated.

The results of the numerical analysis has shown that the time integrated average of the numerically calculated mass flow rate is close to average mass flow rate measured with a calorimeter test system. Furthermore thermal mapping inside the suction muffler shows good agreement with experimental results. Time dependent flow analysis results inside the suction muffler help to characterize the flow and acoustic function of the muffler which leads to the new and better muffler designs.

INTRODUCTION
A hermetic reciprocating compressor is the most critical component of a household refrigerator. It consumes approximately 90% energy of overall electrical input power of the refrigerator. Therefore performance improvement studies of the compressor play an important role to reduce overall energy consumption of the refrigerators. The cooling capacity of a compressor is greatly influenced by the volumetric efficiency. The factors affecting the volumetric efficiency are suction gas superheating in the suction path, throttling, clearance volume, valve dynamics, blow back of gases through clearance between piston and cylinder and heat transfer during its transfer into the cylinder.

The design of the suction muffler system greatly influences the volumetric efficiency of reciprocating compressors. In order to increase the COP of the compressor an efficient muffler design which provides minimum temperature increase and pressure loss must be used. Furthermore the suction muffler helps to reduce the noise level produced by pressure pulsations. Inner design of suction muffler, total inner volume, flow path line length, suction and discharge sections' dimensions effect the muffler's function and transmission loss value.
From the viewpoint of COP, the main function of the suction muffler is to reduce pressure drop and heat transfer in the suction system. In order to reduce the noise which originates from the pressure pulsations, it is necessary to know the sources and how the noise is transmitted through the muffler for better designs.

CFD has become an essential development tool in the improvement of performance of reciprocating compressors. Positive efforts have also been made for the complexity of the physical phenomenon of handling compressible flow in oily ambient. A lot of research work dealing with the use of CFD as a design tool for various compressor components can be found in the literature. There is also some remarkable research in the development of suction muffler by using CFD.

V.K. Rao et al [1] applied CFD analysis for suction gas flow through two types of suction mufflers. New suction muffler design which contained a suction separator was verified with experimental studies. A. Nakano et al [2] examined the behavior of the refrigerant gas in a suction muffler by using CFD analysis. Interaction between suction lead valve's time based on opening/closing phases and refrigerant gas flow characteristic was applied by using 4 pole relations of Yoshimura et al. [3]. B-H. Kim et al [4] performed the design of suction muffler with respect to noise reduction and COP. According to the CFD analysis results throat width, tube length, cross-sectional area of the tube were the main design variables to decrease the loss in the suction muffler.

EXPERIMENTAL STUDIES

In order to conduct the thermodynamic analysis and determine the necessary boundary conditions for numerical calculation, a pV set-up was built and calorimeter measurements were performed. Piezo-resistive miniature pressure transducers flush mounted in the valve plate were used to measure the pressure inside the cylinder and suction plenum for thermodynamic investigations. For the determination of the cylinder volume an optical encoder was placed on the shaft, from which the position of the shaft can be determined. From the position of the shaft the piston position was calculated and also the momentary cylinder volume. pV measurements were conducted at ASHRAE test conditions to examine the cooling capacity, power consumption, compression work, pressure characteristics and the thermodynamic losses of the investigated compressor. The results of pV measurements of the reciprocating compressor are shown in Figure 2. Furthermore detailed temperature measurements were also done at various points inside the gas line and the outer surfaces.

![Figure 2: pV-diagram and temperature/pressure measurement locations](image)

NUMERICAL MODELING

The pressure and velocity of the refrigerant gas in the suction muffler change in accordance with the opening and closing of the suction lead valve during the suction and compression processes caused by the back and forth motion of the piston in the cylinder. To understand this complex phenomenon of the refrigerant behaviour through suction muffler Computational Fluid Dynamics was used. Following conditions has been taken into account for carrying out the CFD analysis.

- The gas flow is unsteady inside suction muffler during one crank period
- The gas flow is compressible
- The refrigerant (isobutane) is considered as a real gas and assumed to be pure without oil.
- The refrigerant gas in the suction muffler is a pulsating flow of roughly subsonic velocities
- Experimental temperature measurements and pressure measurements for inlet and outlet boundaries are necessary
- Valve opening and closing times should be considered.

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Computational model

Firstly the moldable two components through which the gas flow path to be analyzed were modeled using NX-IDEAS software (Figure 3). Here the numerical calculation domain was extracted so that only the gas flow paths exists as model. This model was then converted in Parasolid format for mesh preparation.

Discretization

Meshing is a process where the calculation domain is divided into a number of cells. Mesh quality plays significant role in the accuracy and the stability of the numerical solution. The parasolid format of the suction muffler was then imported in Gambit/Fluent(CFD). Hex mesh scheme was applied at inner channel whereas hexcore meshing scheme which provides hex dominant mesh was applied for resonator parts. Figure 4 shows the cross section of the mesh applied. The inner channel was divided with fine mesh elements compared to resonators. The width of the finest mesh is about 0.15 mm at inner channel where we expect to obtain higher velocity gradients. The total number of the cells is nearly 1.7 million which is sufficiently enough for time dependent calculations. Mesh quality was also checked for smoothness, cell shape skewness and aspect ratio.

The external wall of the suction muffler is part of the computational domain in addition to the fluid region in the muffler. Outer boundaries of the suction muffler has been taken zero thickness for numerical calculation.

Boundary conditions

To analyse the 3 dimensional unsteady compressible viscous flow pattern inside suction muffler appropriate boundary conditions must be applied.

The ASHRAE conditions was used as the compressor test conditions (54.4 °C condensing temperature, -23.3 °C evaporating temperature). Table 1 shows the properties and boundary conditions used in the computation.
Table 1: Properties and boundary conditions

<table>
<thead>
<tr>
<th>Properties</th>
<th>Refrigerant</th>
<th>Density (kg/m³)</th>
<th>FEQ Helmholtz equation of state for isobutane of Miyamoto and Watanabe (2001) [6]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermodynamic</td>
<td>NIST Refprop V7 database</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| Boundary conditions (Figure 5) | Muffler Inlet Pressure (Bar) | 0.624          | Muffler Inlet temperature (°C) | 53.5          | Outer wall temperature (°C) | 58.3          | Muffler Exit pressure | Time dependent |

We use experimental results for each of the settings. Figure 5 shows all surfaces where we evaluate results of computation i.e pressure, temperature, mass flow rate, density.

![Figure 5: Defined surfaces for evaluation of the time history of the results](image)

![Figure 6: Boundary condition of the suction muffler exit](image)

Outer boundaries of the suction muffler has no slip wall condition and assigned constant temperature. For the outer boundary of the suction muffler we set a shell temperature of 58.3 °C obtained from experiments.

The boundary conditions for the inlet of suction muffler are set to evaporating pressure (0.624 Bar) and temperature fixed at the experimental value which is 53.5 °C.

As for the boundary conditions of outlet we apply pressure outlet condition when the suction valve is open and apply wall boundary condition when the suction valve plate is closed. The suction valve plate opens when the crank angle reaches 55 degree and closes when it is reached 185 degree. We set the time-dependent variability of pressure in the suction plenum obtained from PV measurements shown in Figure 6. For outlet boundary we also specify backflow total temperature for energy calculations and gave it a fixed wall temperature of 67.5 °C when it is closed.

**Solver properties**

Transient solution should have been run to a point where the transient flow field has become “statistically steady”. At CFD runs, timestep has been taken as one third of one crank angle time. One crank period lasts 20.49 msec obtained from PV measurements.

Second order discretizations for time and space derivatives have been used to get better accuracy. Realizable k-ε turbulence model was used for its robustness and reasonable accuracy similar to wide range of turbulent flows in industrial flows. The turbulence parameters like turbulence intensity and hydraulic diameter were calculated and specified at appropriate zones. Standart wall functions were applied for near wall treatment. SIMPLE algorithm was used for pressure-velocity coupling.

The NIST (National Institute of Standards and Technology) real gas model for vapor phase of isobutane was used. The real gas model allows us to solve accurately for the refrigerant gas flow and heat transfer problems where the working fluid behavior deviates from the ideal-gas assumption. Helmholtz Free Energy (FEQ) equation of state for isobutane of Miyamoto and Watanabe [6] was used for property calculations.

Intel Xeon CPU X7550-2GHz with 32 processor and 48 GB RAM 64 Bit Operating system computer has been used for numerical calculations. One period of analysis lasts around four days.
RESULTS OF COMPUTATION

Here the results of computation in the suction muffler are examined in the 4th cycle of computation. The results of numerical analysis has shown that the time integrated average of the numerically calculated mass flow rate is 10% close to average mass flow rate measured with a calorimeter test system. We were able to confirm that the computational results were in good agreement with the experimental values.

Table 2: Comparison of experimental and numerical mass flow rates

<table>
<thead>
<tr>
<th>Time integrated average of the numerical calculated mass flow rate (gram/sec)</th>
<th>Experimental mass flow rate (gram/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.360</td>
<td>0.323</td>
</tr>
</tbody>
</table>

Figure 7 shows the change of mass flow rate during one crank period of the refrigerant gas in the suction muffler inlet and outlet cross sections. The suction valve opening \(t_1\) and closing \(t_2\) times are indicated with dotted lines. Backflow occurs at muffler exit at 5.58 msec and 8.53 msec after the suction valve is opened. There is cyclic backflow at muffler inlet when the suction valve is closed. Also mass flow rate changes have sharp peaks during one crank period. These results show that the refrigerant gas is sucked less effectively.

Furthermore Figure 9 shows mass flow rate changes at inner channel cross sections when solution results had stabilized. Very small mass flow rate peaks (0.25 g/sec) in cyclic manner occurs at channel section interior and exterior. There are two resonators at the middle of inner channel. At the end of inner channel the refrigerant uses channel exit front because of the main flow direction.
At Figure 11 outlet channel inlet cross section follows the mass flow pattern of muffler exit when the suction valve is open. Too small mass flow rate back and forth patterns occurs when the suction valve is closed which we expect to cause noise.

Pathlines coloured by velocity magnitude at the instant of suction valve opening and closing times are shown at Figure 8 and Figure 12. The highest velocity gradients occur at the instant time of 5.58 msec at the entrance of inner channel. 70 m/s speed is reached at the inner channel which is approximately 0.3 Mach value. (Figure 10) The refrigerant is taken from compressor inner volume 1 msec later than the opening time of the suction valve.

Figure 13 shows the temperature transition of refrigerant at muffler inlet and exit cross sections at one crank period. The rise of temperature in the suction muffler is said to be approximately 6 °C which was confirmed with experimental results. There is also a sharp increase of temperature (3.5 °C) at muffler inlet when the suction valve is closed. This shows hot gas in the muffler returns into inner volume of compressor.

Figure 15 shows the temperature changes at inner channel entrance and exit sections. The gas temperature increases around 2 °C along the path of inner channel for one crank period. The highest temperatures occurs at inner channel outlet. Pathlines coloured by temperature are shown at Figure 14 and Figure 16 at the instant of suction valve opening and closing times. The right part of the outlet resonator has the highest temperature values.
Figure 15: Temperature changes at one crank period

Figure 16: Pathlines at time 10.58 msec while suction valve is just closed (coloured by temperature)

Figure 17 shows the density changes at muffler inlet and exit for one crank period. The density of 1.35 kg/m³ prior to suction valve opening falls to 1.29 kg/m³ when the suction valve is open and the refrigerant gas is sucked into the cylinder.

Figure 17: Density changes at one crank period

When trying to interpret time-sequence data from a transient solution, it is often useful to look at the data’s spectral (frequency) attributes. We computed the spectral distribution of static pressure data recorded at cross sections inside suction muffler. ANSYS FLUENT allows us to analyze time dependent data using the Fast Fourier Transform (FFT) algorithm.

Figures 19 and 20 show the relationship between the pressure data taken from different cross sections and volume natural frequencies of three volumes V₁, V₂ and V₃. (Figure 18) The first volume V₁ can be considered as penetrating chamber with one connecting pipe and V₂ and V₃ as side branch resonators or Helmholtz Resonators.

The side branch resonator-Helmholtz Resonator is an acoustic bandstop filter and works on the principle of dynamic absorber. The fundamental natural frequencies of three volumes are tuned at the approximate frequency bands in order for the attenuation at different cross sections.
Transmission Loss (TL) is one of the design parameters of mufflers. TL is defined as the difference between the sound power level of the incident wave to the muffler system and transmitted sound power. Figure 20 shows the correlation between the TL characteristics of designed muffler and time dependent pressure data using the Fast Fourier Transform (FFT) algorithm muffler inlet and outlet sections. There seems good correlation between these two data as excepted.

CONCLUSIONS

In this paper factors affecting the volumetric efficiency related to the suction muffler are investigated. Suction gas overheating through the muffler and flow patterns inside the suction muffler that cause noise were also studied in detail.

- The kinematics of the suction valve during the suction and compression periods strongly depend on the behaviour of the refrigerant gas in the suction muffler.
- The results of numerical analysis has shown that the time integrated average of the numerically calculated mass flow rate is close to average mass flow rate measured with a calorimeter test system.
- The refrigerant is taken from inner compressor volume 1 msec after the opening time of the suction valve.
- The gas is taken mainly from the inner channel of the muffler.
- The highest velocity gradients occur at the entrance of the inner channel locally.
- After the expansion process is completed, there are also back flows during the exhaust process at muffler inlet.
- The inner channel design is critical to reduce overall pressure loss in the suction muffler.
- The fundamental natural frequencies of three volumes are tuned at the approximate frequency bands in order for the attenuation at different cross sections.
- The correlation between the TL characteristics of designed muffler and time dependent pressure data using the Fast Fourier Transform (FFT) algorithm muffler inlet and outlet sections have good correlation.

REFERENCES

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