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ABSTRACT

CFD simulations of a hermetic reciprocating compressor became state of the art in the last decade. The numerical procedures become more and more complex and until now almost every part of the compressor can be simulated in a sufficient manner. Most of these methods focus on the determination of boundary conditions or the evaluation of gas guiding components, while assuming the refrigerant as Ideal Gas. In this work, the simulation of a compressor has been chosen to compare the ideal gas assumption on the one hand and a real gas approach on the other. The differences of these two approaches have been shown by comparing pV-diagrams. Furthermore the obtained results have been compared to experimental data, and differences are worked out. All simulations have been done abstracting the valves as flat plates moving parallel to the valve plate.

1. INTRODUCTION

Numerical methods become more and more important in the development process of a hermetic reciprocating compressor. The fast increase of computational power and the progress in modeling of complex physical phenomena, allows engineers to study effects in such a detail, no one expected to be possible a few years ago. The application range for numerical tools in the development process is broad: Beginning with Finite Element Method (FEM) simulations, which can be used to predict the stresses inside the flapper valve, up to boundary element methods, which enable an acoustical characterization of different compressor components.
Another important numerical tool being used excessively in the last few years in compressor development is computational fluid dynamics (CFD). A lot of research work dealing with the use of CFD as a design tool for various compressor components can be found in literature. In some of these works CFD was used quite generally, examples given by Fagotti (2000) or Ottisch (2000). Other authors like Rovaris and Deschamps (2006) or Possamai (2001) have used the method of Large Eddy Simulation (LES) to study flow effects in detail. Possamai (2001) investigated the influence of the valve's inclination on the pressure distribution and furthermore on the forces and the flow field acting on the valve.

The recently published papers deal with the coupling of different simulation methods. Lang (2009) and Shiomi (2009) coupled CFD with FEM, Lang (2008) and Nakano/Kinjo (2008) realized a coupling of CFD with 0D/1D gas dynamic tools. All these works use complex methodologies to model the physics of the flow, but they do not consider the real thermodynamic properties of the gas. Most of these studies consider the refrigerant as an ideal gas, whereas few authors try to implement real gas behaviour by using polynomial or piecewise linear functions. Peskin (1999) studied the effect of different property models on the simulation of a compressor for the refrigerant R134a. He found a strong variance in the gas properties, depending on the equation of state he used. As Peskin (1999) applied a simplified 2D axis symmetric model of the cylinder to simulate the compression and expansion, it was difficult to compare the results directly with experimental data.

The aim of this work now is to investigate the influence of the thermodynamic property model of the refrigerant. This is done by comparing the state of the art approach, which uses the equation of state for the ideal gas, with an equation of state for the real gas. To make simulation results comparable with experimental data, it is tried to keep the 3D simulation domain as close as possible to reality, only the valves are abstracted as flat plates which move parallel to the valve plate. To validate simulation results, the pV-diagram is compared with a measured one.

2. THEORETICAL BACKGROUND

2.1 CFD Equations

For solving the flow behaviour of the refrigerant through the compressor, a finite volume method has been used. The equations for mass, momentum and energy are solved by means of a commercial CFD (computational fluid dynamics) software package (Fluent 12.1). These three equations are shown below (1) – (3).

Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \tag{1}$$

Assuming a Newtonian fluid, the momentum equation can be represented by equation (2).

Conservation of momentum:

$$\rho \frac{\partial u_i}{\partial t} + \rho \frac{\partial u_i u_j}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + S_i \tag{2}$$

Conservation of energy:

$$\frac{\partial (\rho h)}{\partial t} + \frac{\partial (\rho u_i h)}{\partial x_i} = \frac{\partial p}{\partial t} + \frac{\partial}{\partial x_i} \left(\lambda \frac{\partial T}{\partial x_i} \right) + S_h \tag{3}$$

2.2 Equations of State

In addition to the conservative equations for mass, momentum and energy further equations are required to get a closed mathematical description of the regarded thermodynamic system. The required equations, empirically determined, describe the behaviour of the flowing gas, which is R-600a (isobutane) in this case.

The simplest way to describe refrigerant properties is the ideal gas model, based on simple temperature depending relations for thermal and caloric quantities. The thermal equation of state is the well-known and simple expression, shown in equation (4).

$$p \cdot \frac{1}{\rho} = R \cdot T \tag{4}$$

p .... gas pressure in N/m^2
ρ .... gas density in kg/m^3
R...................... gas specific constant in Nm/kgK
T...................... gas temperature in K

The caloric equation of state can be written as:

$$e = c_v \cdot T \qquad (5)$$

e .... internal energy in Nm/kg
c_v .... gas specific heat coefficient under constant volume in Nm/kgK
T..................... gas temperature in K

To consider the dependency of fluid specific parameters on temperature, piecewise linear profiles, shown in Figure 1, Figure 2 and Figure 3, have been used for specific heat, thermal conductivity and viscosity.

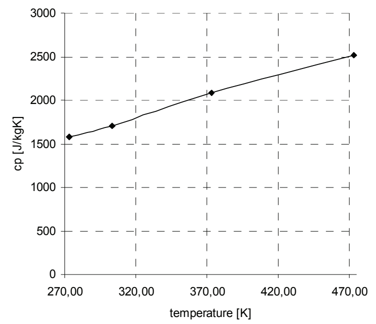

Figure 1: cp profile vs. temperature

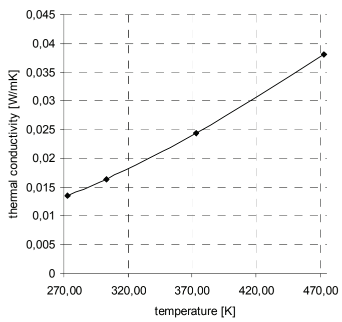

Figure 2: thermal conductivity profile vs. temperature

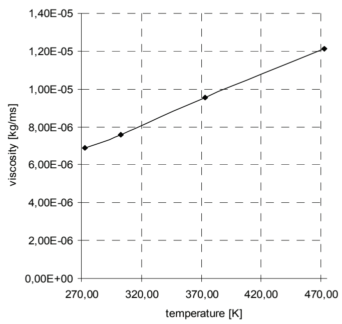

Figure 3: viscosity profile vs. temperature

The ideal gas model represents, according to its notation, an idealized state, except elastic impacts all other interactions are neglected. Although being just an approximation of real physics, the ideal gas model is often used in engineering computations due to its simplicity. The computed results are quite accurate, when pressure level is low and the occurring temperatures are high.
Processes with high densities, like compression in a reciprocating compressor, can lead to drastic deviations of refrigerant's behaviour from the ideal gas model. For obtaining physically accurate results it is indispensable to check if the ideal gas assumption is sufficient enough or if the molecular interactions have to be taken into account.
To describe the thermodynamic behaviour of gases as good as possible, complex experimental measurements have to be done. Based on the measured results, various authors, see Miyamoto and Watanabe (2002), have tried to find simple and even though highly accurate mathematical expressions for picturing the real gas behaviour of R-600a.
In the recently released Fluent version 12, the NIST Refprop version 7 database is implemented, including thermodynamic properties for R-600a. The property model used in Refprop version 7 is a Helmholtz free energy equation of state, proposed by Miyamoto and Watanabe (2002).

3. SIMULATION AND MEASUREMENT SETUP

3.1 SIMULATION SETUP

The compressor which is investigated in this study is quite similar to an HTK 55 ACC production compressor. To obtain results being as close as possible to reality, nearly the whole compressor domain has been modeled. Only two simplifications have been implemented: First of all, the shell has been replaced by a rectangular box having the volume of the original shell. Secondly both valves (suction and discharge) are abstracted as flat plate, moving parallel to the valve plate, following the equation of motion of a one-degree of freedom system, shown in equation (6).

$$\begin{cases} m \times \ddot{x} + d \times \dot{x} + c \times x = \Delta p \times A - F_0 \\ \dot{x}_{(t=n-1)} = -e \times \dot{x}_{(t=n)} \end{cases} \qquad (6)$$

x, $\dot{x}$, $\ddot{x}$ .... valve lift in m, valve velocity in m/s, valve acceleration in m/s²
m .... valve mass in kg
c, d.................. spring rate of the valve in kg/s², damping constant in kg/s
Δp, A............ valve flow pressure difference in N/m², valve cross section in m²
F_0 .... preload force in N
e .... dimensionless restitution coefficient

These simplifications have been done to reduce the numerical effort. For solving the governing equations described in the former section, a pressure based solver has been used. The turbulence behaviour of the fluid has been considered by using the RNG k-ε turbulence model with a standard wall function. The coupling of pressure and velocity has been realized by using the SIMPLEC coupling scheme. The 3-dimensional calculation domain of the compressor consists of suction pipe, suction muffler, shell (not shown in Figure 4), cylinder, suction valve, discharge valve, discharge muffler, the serpentine and the discharge pipe. In Figure 4 the calculation domain, meshed with tetrahedral elements, is shown. The total number of cells is between approximate 400.000 (TDC) and 1.800.000 cells in BDC.

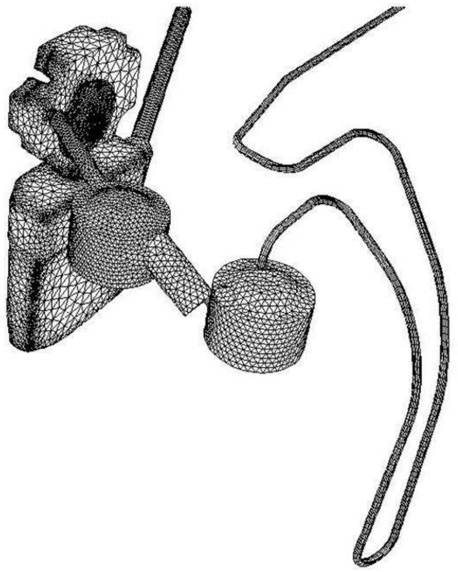

Figure 4: 3 dimensional simulation domain

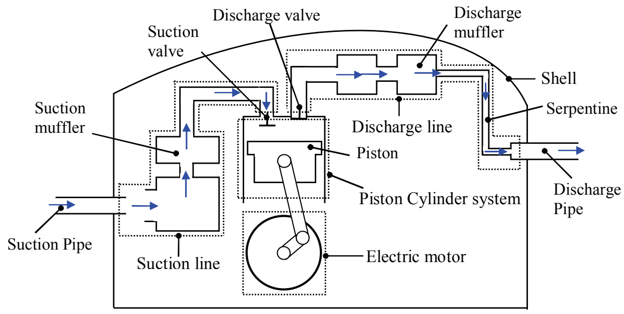

Figure 5: Compressor scheme

In Figure 5 a schematic drawing of the compressor can be seen. At the inlet (inlet suction pipe) a pressure inlet boundary condition has been set with a value of 62700 Pa pressure and a temperature of 305 K. At the end of the discharge line (end discharge pipe) a pressure outlet condition has been applied with a pressure value of 608000 Pa and a backflow temperature of 380 K.

3.2 MEASURING SETUP

Typically, piezoelectric pressure transducers are used to measure the pressure inside the cylinder for thermodynamic investigations. Ideally, the transducers should be flush mounted in the valve plate. Only this application ensures that the compressor working process is not influenced by the measuring setup. Although miniaturized sensors are used, valve plate flush mounting is impossible in this case, because there is hardly space between the two valves, the suction and the discharge system.

Therefore, it was necessary to find another position for the piezoelectric pressure transducer. Currently the piezoelectric pressure transducer is placed in the crankcase and it is connected to the cylinder by a small diameter drill-hole, leading to a slightly increased clearance. It has to be taken into consideration, that this measuring setup influences the compressor working process, especially when comparing the experiment with simulation.

4. RESULTS

Figure 6 shows the pV-diagrams for the simulation with ideal and real gas and the comparison with experimental data, obtained with the strategy described in section 3.2.

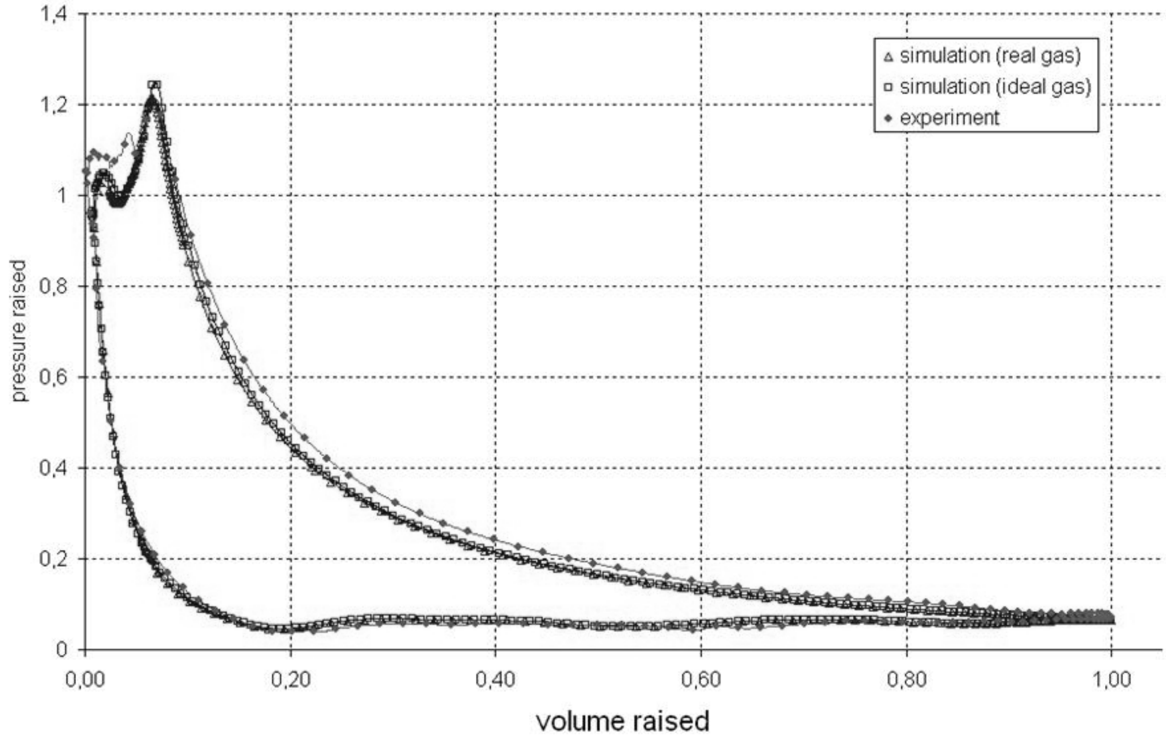

Figure 6: pV-diagram comparison

It can be seen that there are some discrepancies between simulation and experimental results. Compared with the simulation, the experiment shows a bigger amount of heat transferred from the walls of the cylinder to the refrigerant in the cylinder. This is due to the fact that fixed temperature values have been used as boundary conditions for the simulations.
Another obvious discrepancy can be seen in the discharge phase. In this phase small pressure fluctuations occur. These fluctuations are probably spurious fluctuations caused by the measuring setup, as waves are reflected in the small connection bore between cylinder and pressure transducer. To adjust simulations in the way to fit better with the experimental data, it would be necessary to run new simulations with more accurate temperature boundary conditions for the cylinder wall. Moreover, the connection bore should be considered in the simulation domain.
However, the aim of this paper is the comparison of two equations of state and so the correlation between simulation and experiment is sufficient accurate, but the above mentioned details should be taken into account when looking to the results. Having a detailed look to the simulation results obtained with the equation of state for the ideal gas and for the real gas, there is no big discrepancy in the pV-diagrams. In **Table 1** the main performance parameters of the compressor are shown. P_{el} is derived by dividing the closed-loop integral of the pV-diagram by the well known values for the electrical and mechanical efficiency. If comparing the results for the ideal and real gas it can be seen that there is only a difference of about 1.3 percent. In addition the mass flow rate through the compressor which is directly correlated with the cooling capacity (Q_0) differs only of about 0.46 percent.

Table 1**:** Overview of the compressor performance parameters

	Q_0		**P_{el}**		**COP**	
	Watt	**%**	**Watt**	**%**	**W/W**	**%**
Experiment	98.90	100.00	51.70	100.00	1.91	100.00
Ideal gas	97.41	98.50	47.75	92.36	2.04	93.63
Real gas	98.87	99,96	46.87	90.657	2.11	90.56

Figure 7 shows the T-v-diagram for the simulation "ideal-gas" and "real-gas". Especially during expansion and compression, there is a deviation in temperature trend. In the upper and lower dead point the temperature tends to be the same, the reason therefore are again the fixed temperature values for the walls.

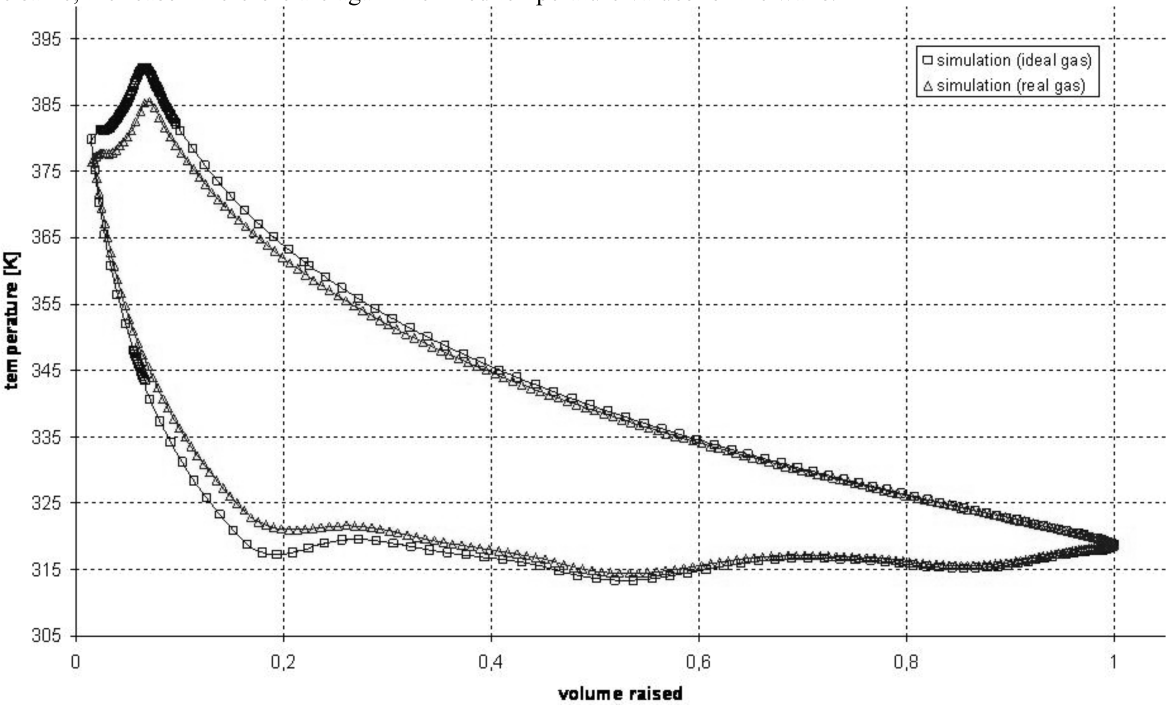

Figure 7: T-v-diagram

All simulations have been done on the same machine, an Intel Xeon X5450@3.0Ghz, with 32GB memory. The biggest difference in using the two different equations of state in a simulation occurs by comparing the simulation times. Whereas a simulation using the ideal gas model is finished in approximately 2 days, it takes 7.5 days until simulation is converged, when the real gas model is used.

5. CONCLUSION

In this work two equations of state are compared to study the influence on the compressor performance and to check if the usage of the ideal gas assumption is capable to describe the behaviour of the refrigerant. The results show that using the equation of state for the ideal gas, the discrepancies to the real gas behaviour are of minor importance. Especially for industrial usage the advantage of significant reduced calculation times is much higher than the slightly improved accuracy. Furthermore the usage of the real gas models tends to destabilize the simulations, leading to non converged simulation results. Finally it can be said that the usage of adequate boundary conditions for the walls of the cylinder and the suction and discharge line is much more important (see Figure 6 & Figure 7) than the usage of an improved gas model.

NOMENCLATURE

u	velocity	(m/s)
f	forces	(N)
μ	viscosity	(kg/(ms))
h	specific enthalpy	(kJ/kg)
Sh	source term of the internal enthalpy equation	(kJ/m^3)
Si	source term of the internal energy equation	(kJ/m^3))
λ	thermal conductivity	(W/(mK))

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