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ABSTRACT

In this paper, a coupling of 0-, 1- and 3-d tools is applied for the simulation of the suction line of a reciprocating compressor. The 3-d simulation is applied only for the gas inside the suction line while the rest of the domain is modeled in 0- and 1-d way. The 1-d simulation is applied for the gas inside the pipe or any component which can be modeled as pipe. The 0-d simulation is applied for solid components, gas inside the shell and the oil, which leads to the thermal network among them. It eliminates the required thermal boundary condition at the surface of solid components for 3-d simulation. All simulation tools are strongly coupled. In addition, a spring-damping-mass model is used for the modeling of the valves.

1. INTRODUCTION

Increasing the efficiency of a hermetic reciprocating compressor, used in household appliances, is a relevant step for reducing the global energy consumption. To achieve this ambitious target a deeper understanding of flow and thermal effects inside the compressor is required. Many parts of the compressor directly act with these physical effects and influence the compressor efficiency. The point of interest in this paper, which plays an important role concerning the gas flow inside the reciprocating compressor, is the suction muffler. In the suction muffler complex flow phenomena and thermal processes take place. This leads to two different main losses, the pressure loss and the heating of the suction gas in the suction muffler. To reduce these losses a comparison between different muffler designs is indispensable.

A good study of actual muffler designs is prerequisite to understand the influences on the compressor performance. To make this comparison equitable it is necessary to compare the different mufflers on the same compressor. Different mufflers lead to different thermal conditions and different pressure losses inside the compressor. Pressure fluctuations influence the movement of the valve, which directly leads to a different mass flow rate.
Reducing the number of mufflers which have to be tested and increasing the speed of development, the adoption of numerical tools which can predict the pressure loss and the heating of the gas get more and more important. The challenge for those numerical tools is to consider every feedback effect inside the compressor which is relevant.

The most detailed method to predict the pressure loss and the thermal effects (including all feedback effects) in the suction muffler is a 3 dimensional model of the whole compressor. The disadvantage of this method is the extensive numerical effort. Another method is the 1 dimensional modelling of the compressor. A first approach to include most of the feedback effects in a 1 dimensional model is made by Abidin *et al.* (2006).

Filling the gap between the full 3 dimensional approach and the 1 dimensional model is a big challenge for researchers. The idea of zooming into a compressor, by modelling the part of interest in a 3 dimensional way and preparing adequate boundary conditions from a 1 dimensional model, looks like a good method to combine the strengths of both simulation approaches. Several coupling methods have already been developed in the field of I.C.

Engine modelling (for example see Bella *et al.* (2003)). The next step is to transfer these modelling approaches into the field of household compressor design.

In this paper a 0/1 dimensional model with a TNW (Thermal Network) has been combined with the 3 dimensional model of the suction line of a reciprocating compressor, in order to decrease the number of computational cells of a full 3 dimensional simulation. The accuracy of the 1 dimensional model must be high to get an accurate prediction of the suction muffler losses.

2. THE MODEL

The part of interest for this analysis is the suction muffler; therefore the zooming section concentrates on the suction line of the compressor. Figure 1 (a) shows the main parts of the model, figure 1 (b) displays the suction muffler which should be evaluated. The model consists of different subroutines which are implemented into the commercial software package Fluent in the form of a C-program. The subroutines can be classified into four main calculation units, the 0 dimensional calculations, a 1 dimensional part, a routine which contains the thermal connections (TNW) between the masses inside the compressor and a 3-d model of the suction muffler. This modular approach allows studying the different effects which influence the compressor performance.

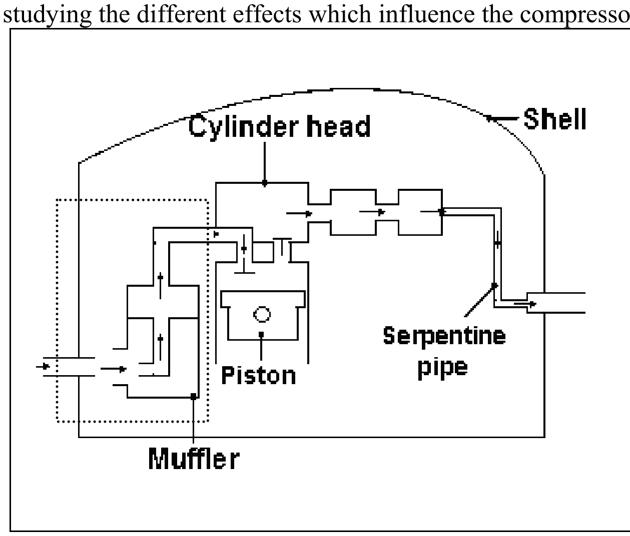

Figure 1 (a): Model of the compressor (the dotted border shows the 3-d Domain of the simulation)

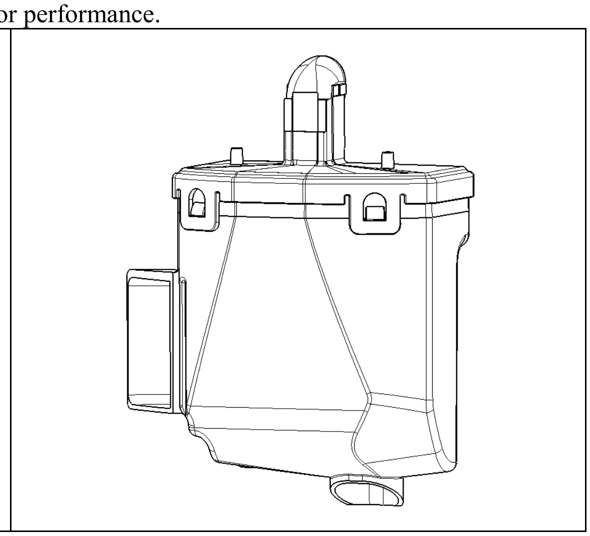

Figure 1 (b): Suction muffler from serial production

2.1 0-dimensional Model

The 0-d routine implements the calculation of the cylinder, the two valves (suction and discharge valve) and a model for the heat transfer. The pressure and the temperature inside the cylinder are calculated by the use of the 1st Law of Thermodynamics (see equation (1)).

$$dW_t + dQ_a + \sum dm_i \times h_i = dU \qquad (1)$$

Relevant literature shows how to create a mathematical model for a valve which can be approximated as a mass-spring system, for example Costagiola (1950), Aigner and Steinrück (2007), Habing and Peters (2006) etc. This former work can be seen as the BVT (basic valve theory) which is used to describe the movement of the valve.

$$\begin{cases} m \times \ddot{x} + d \times \dot{x} + c \times x = \Delta p \times A - F_0 \\ \dot{x}_{(t=n-1)} = -e \times \dot{x}_{(t=n)} \end{cases} \qquad (2)$$

The equations (2) describe the most important factors necessary to model the valve. The integration of the model has been done by the use of a 4th order Runge-Kutta method. There are few other parameters which influence the valve movement but which are not included in the model. For example a stiction force for the suction valve is not implemented in the model, due to the lack of experimental data for this effect. The adjustment of the valve model has been done by experimental data. More detailed information about the BVT can be found in the references above.

Another important point for the simulation of the compressor is the modelling of the heat transfer. The heat transfer inside the cylinder is calculated using the approach described by Adair *et al.* (1972). Adair defines the Reynolds number as a function of the swirl velocity. Furthermore, the swirl velocity has been expressed as a function of the angular velocity of the compressor and the crank angle. A detailed description of the model can be found in Adair *et al.* (1972).

2.2 1-dimensional Model

The high resolution 1 dimensional scheme for the simulation of the flow and the pressure fluctuations inside the discharge line of the compressor can be classified as a 2nd order TVD – Scheme (Total Variation Diminishing) for compressible flows with variable cross sections, based on the solver from Coberán and Gascon (1994). The calculations have been done with characteristic transient boundary conditions for sonic and subsonic flows (Thompson (1990)). The method is based on the solution of the Euler-Equation written in conservative form (see equation (3)).

$$\frac{\partial U}{\partial t}+\frac{\partial F(U)}{\partial x}+S=0 \tag{3}$$

U is the vector which includes the conservative variables (mass, momentum, and energy) in written form: ρ, $\rho \times u$, and E. Whereas F is called the Flux – Vector. The values of the conservative variables for the next time step can be calculated with:

$$U_i^{n+1}=U_i^n-\frac{\Delta t}{\Delta x}\left[F_{i+1/2}^{+}-F_{i-1/2}^{-}\right]+\Delta t\times S_i^n \tag{4}$$

Index i denotes the cell number and index n denotes the time step. In equation (4) the vector S_i is the vector of the source terms of the system, and $F_{i+1/2}^{\pm}$ represents the intercell numerical flux.

2.3 Thermal Network

The temperature connections between the masses inside the compressor are assessed with a thermal network, which is based on the lumped thermal conductance approach (Ooi (2003)). The main idea of the approach is that the compressor is divided into several masses; each of these masses is on a uniform temperature. The calculation of the mass temperature is done by an iterative process solving the 1st Law of Thermodynamics for every mass. Equation (5) shows the 1st Law of Thermodynamics for one exemplary mass.

$$\sum \dot{Q}_{i\,cond.}+\sum \dot{Q}_{i\,conv.}=\frac{d(m_i\times u_i)}{dt} \tag{5}$$

Because of investigating a steady state solution, the time derivative of the internal energy $m_i \times u_i$ of mass i is zero. Due to this fact, and summarizing the convection and conduction parts to one heat flux, equation (5) can be written in the form seen in equation (6):

$$\sum \dot{Q}_i=0 \tag{6}$$

With this equation it is possible to describe every heat flux as seen in equation (7), with HTF_i representing the heat transfer function.

$$\dot{Q}=HTF_i\times\Delta T \tag{7}$$

In this approach the radiation is neglected. Only the conduction and convection parts of the heat transfer are calculated by the use of correlations as worked out in Almbauer *et al* .(2006).

2.4 3-dimensional Model and Coupling

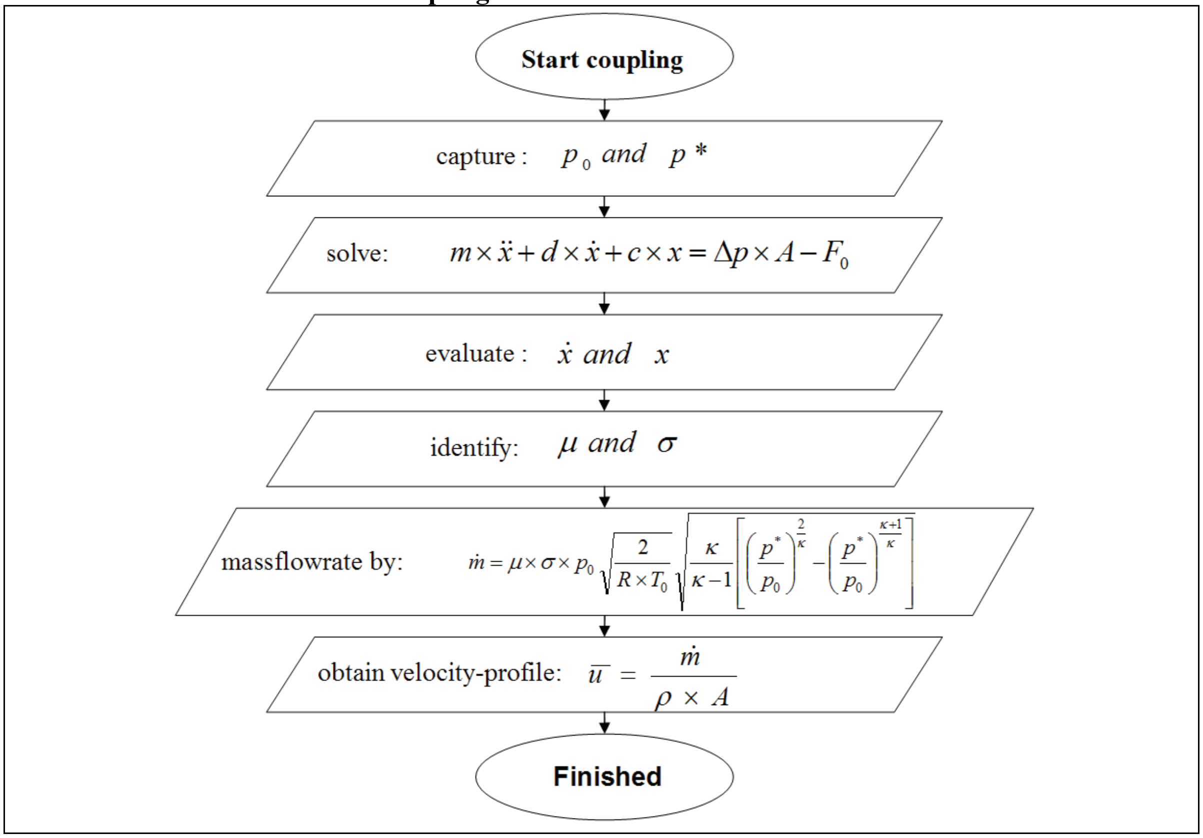

Figure 2: Coupling of the flow variables between 3-d and 0-d

As seen in figure 1 (b), the geometry of the suction muffler is very intricate, which consists of details which are not necessary for the flow or for the thermal behaviour of the muffler. To make modelling easier and to reduce the number of computational cells of the 3 dimensional parts, some details have been removed.
The 3-d model further consists of a shell, which is modelled as a strongly simplified part and the suction pipe. For the pressure-velocity-coupling the PISO scheme has been chosen, recommended for transient flow calculations. At the suction pipe inlet a pressure inlet boundary condition has been adopted, representing the nominal values of a standardized cooling cycle. For the coupling between the 3-d and 0-d domain a velocity boundary condition has been chosen. The coupling procedure for the gas flow works as shown in figure 2.

The first step for the coupling procedure (shown in figure 2) is to read the pressure at the inlet of the cylinder and the pressure inside the cylinder (p_0 and p*). With this pressure difference it is possible to solve the equation of motion for the suction valve by the use of a 4th order Runge-Kutta method. Further 3 dimensional steady state simulations result in values for μ and σ. The parameter μ is called flow coefficient which is a quantity for the flow resistant (losses of friction and flow contraction included). The parameter σ is called "Versperrungsziffer" which regards the change of the area variation; detailed definitions of these values can be seen in Pischinger *et al.* (2002). After these steps the stationary mass flow equation can be evaluated, resulting in the mass flow which runs into the cylinder.
For the thermal coupling it has been decided to use a one way coupling for connecting the 3 dimensional model with the thermal network. This coupling was applied in order to reach a faster convergence and to keep the coupling effort in a controllable range. In the 1 dimensional thermal network the suction muffler is divided into 5 masses. For these masses the temperatures are calculated in the thermal network and the temperatures are sent back to the 3-d simulation as a boundary condition. The detailed 1-d layout for the thermal network can be seen in Abidin *et al.* (2006).

3. RESULTS

For the quality of a suction muffler, two main parameters are relevant. These parameters are pressure loss and temperature gain, both values are measured at five different positions inside the suction muffler. The positions of the measuring points can be seen in figure 3.

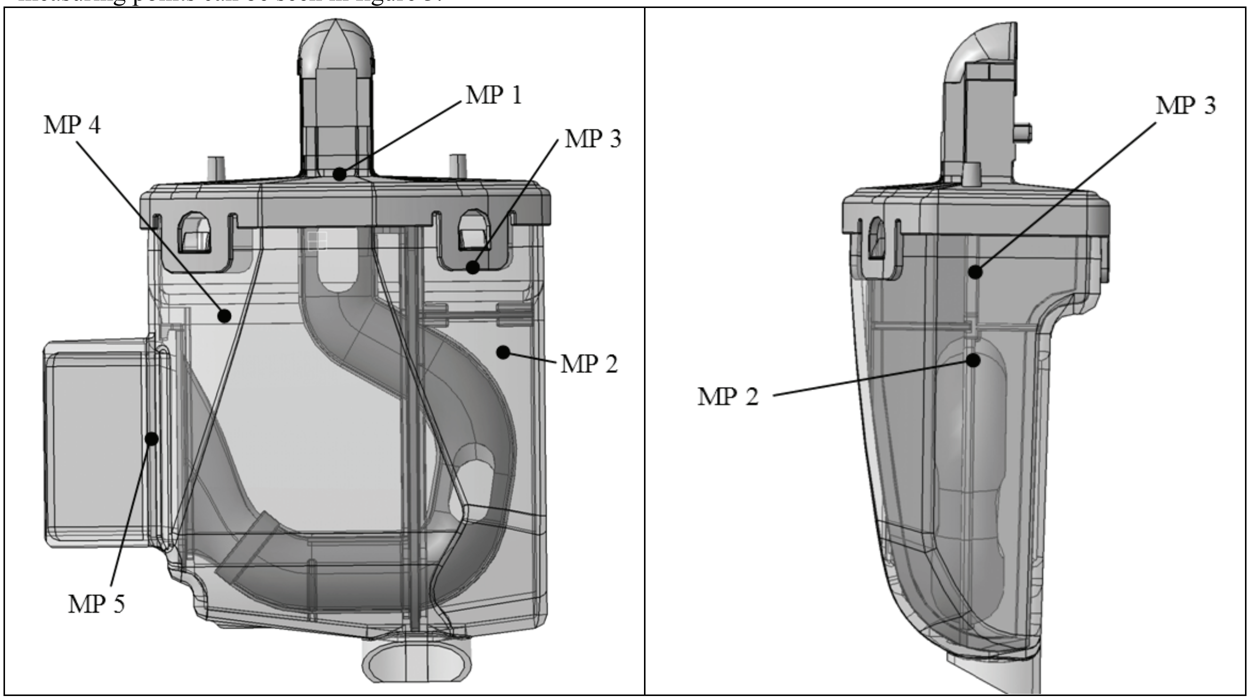

Figure 3 Position of different measuring points

3.1 Pressure Loss

To evaluate the pressure curve of a compressor cycle and to find a comparable unit (regarding different compressors and different suction mufflers) for the pressure loss, a standard method for the evaluation of the pressure loss is defined, first seen in Hanlon (2001) and modified by Lang (2007). This method of evaluation is strongly influenced by the methods used to analyse I.C. Engines, described in equation (8) and equation (9). $W_{suction}$ can be seen as increased work for volume change due to lowered suction pressure, while V_D stands for the displacement of the compressor.

$$\Delta p_{suction} = \frac{W_{suction}}{V_D} \tag{8}$$

$$W_{suction} = \int_{VO}^{VC} (p - p_0)\, dV \tag{9}$$

VC ….. Valve closed
VO ….. Valve opened
p_0 ….. Suction pressure of the ideal cycle

For the suction muffler the important pressure loss is the one which occurs during the opening phase of the suction valve, so the work due to volume change is only evaluated from the moment where the suction valve opens till it closes again. Using above described method applied to the data obtained from measurements and simulations, they can be summarized as seen in figure 4.

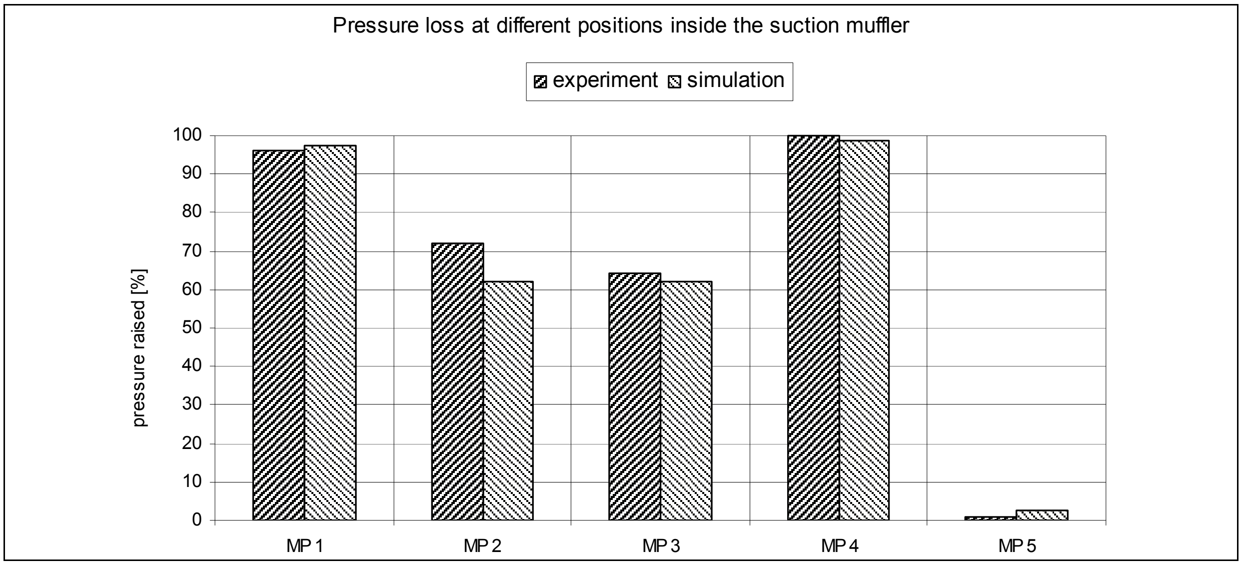

Figure 4: Pressure loss at different positions inside the suction muffler

As shown in figure 4 the integral values of the measured and simulated values fit quite well together. Figure 5 shows the transient pressure curve during the suction valve opening in one specific measuring point. It can be seen that the simulation of the transient pressure in the time domain also matches with the experiment during the valve opening phase. This can be seen as indicator that the valve movement in the model is in a good agreement with the real movement of the valve.

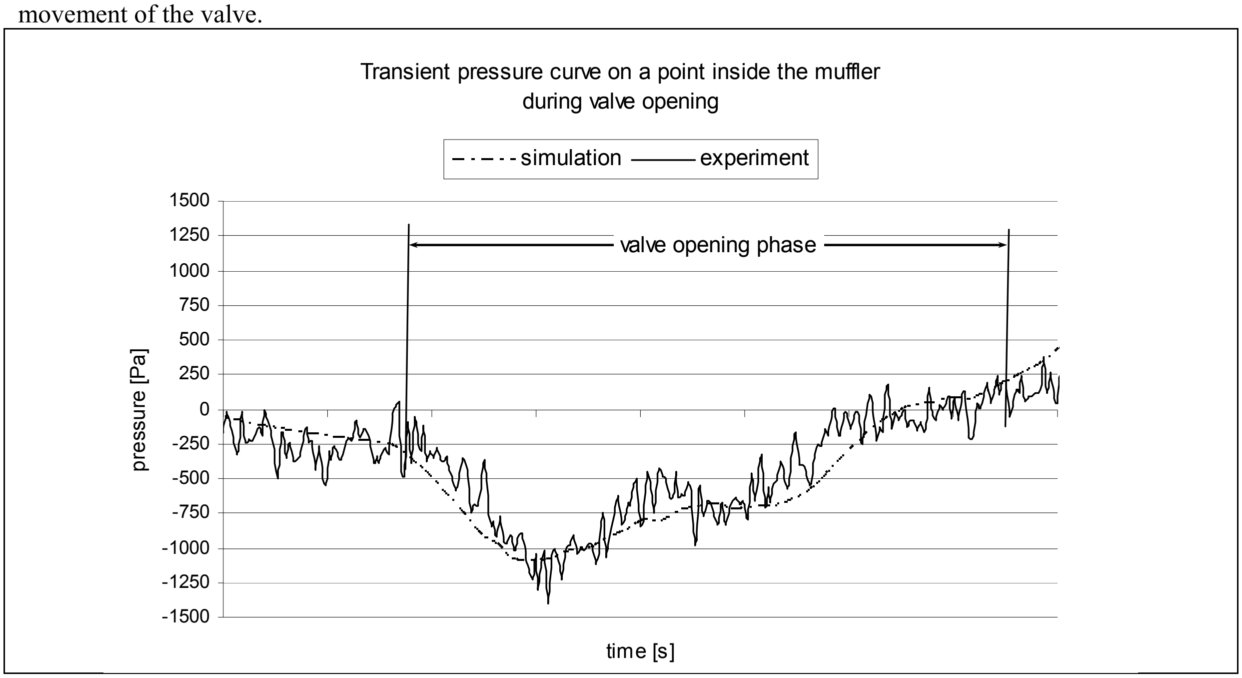

Figure 5: Pressure loss at different positions inside the suction muffler

3.2 Temperature

Measuring a transient gas temperature inside a reciprocating compressor is a difficult task. Due to the inertia of the sensor the fast temperature changes generated by the movement of the gas are not detected. Another effect which influences the quality of the temperature measurement is the connection of the sensor in the compressor. The surface of the wires is exposed to the hot gas inside the compressor shell, resulting in a heat flux through the isolation into the sensor material and influencing the measurement. Because of this, a bigger difference between experiments and simulation is expected.

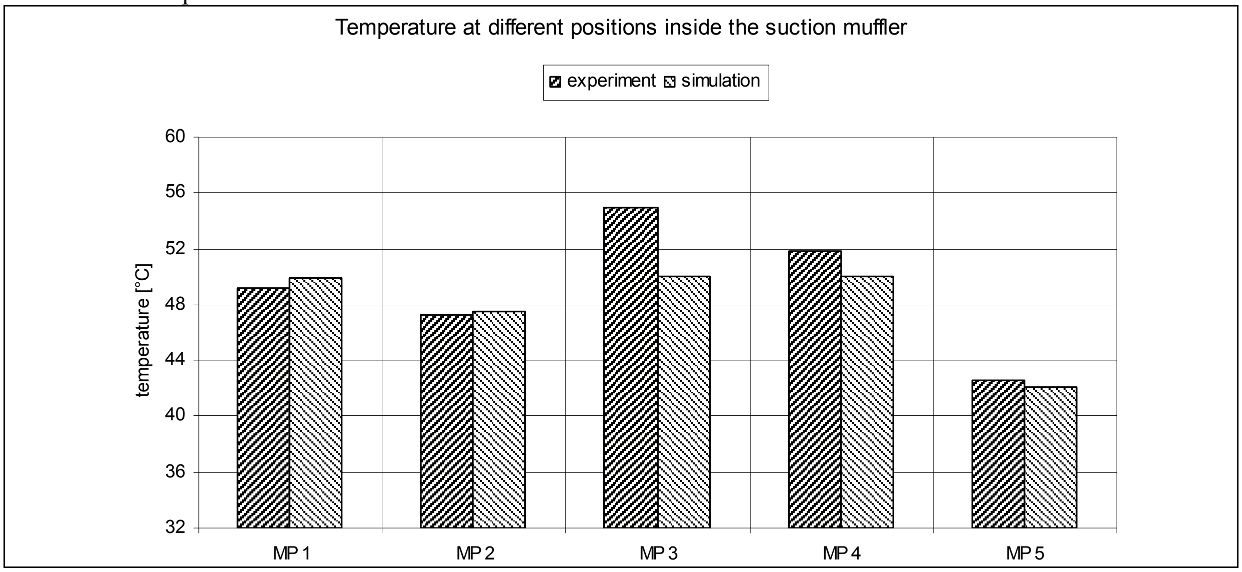

Figure 6: Temperature at different positions inside the suction muffler

Figure 6 shows the temperature distribution inside the suction muffler. With exception of MP 3, experiment and simulation data fit quite well. The bigger difference in measuring point 3 is a hint that the temperature in this position has not converged yet. The time to achieve a converged simulation in measuring point 3 is much longer than the time in the other measuring points, because there is hardly any mass exchange inside this volume. So it takes a plenty of cycles to converge the temperature inside this volume.

4. CONCLUSION

A coupling procedure between a 0-d/1-d model and the 3 dimensional model of a suction muffler of a reciprocating compressor has been successfully developed. The models are strongly coupled; hence the number of unknown boundary conditions has been reduced. The model shows good correlations with the measurements, and thus it can be used to make forecasts of pressure loss, temperature gain, COP (Coefficient of Performance) and cooling capacities of compressors with new muffler designs.

NOMENCLATURE

A	valve cross sectional area	(m²)
F_0	preload force	(N)
Q_a	heat	(J)
$\dot{Q}_i$	heat flux in point i	(W)
U	internal energy	(J)
W_t	work due to volume change	(J)
c	spring rate of the valve	(kg/s²)
d	damping constant	(kg/s)
e	restitution coefficient	(-)
h_i	specific enthalpy	(J/kg)
m	effective valve mass	(kg)
m_i	mass of cell i	(kg)

p_0	upstream stagnation pressure	(Pa)
$p*$	downstream static pressure	(Pa)
Δp	valve flow pressure difference	(Pa)
Δt	time step size	(s)
u	gas velocity	(m/s)
$\overline{u}$	averaged gas velocity	(m/s)
u_i	internal energy	(J/kg)
x	valve movement	(m)
$\dot{x}$	valve velocity	(m/s)
$\ddot{x}$	valve acceleration	(m/s²)
Δx	cell size	(m)
ρ	gas density	(kg/m³)

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