Experimental validation of mathematical model for small air compressor

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Abstract. Development process of reciprocating compressors can be simplified by using simulation tools. Modelling of a compressor requires a trade-off between computational effort and accuracy of desired results. This paper presents experimental validation of the simulation tool, which can be used to predict compressor behaviour under different working conditions. The mathematical model provides fast results with very good accuracy, however the model must be calibrated for a certain type of compressor. Small air compressor was used to validate an in-house simulation tool, which is based on mass and energy conservation in a control volume. The simulation tool calculates pressure and temperature history inside the cylinder, valve characteristics, mass flow and heat losses during the cycle of the compressor. A test bench for the compressor consisted of pressure sensors on both discharge and suction side, temperature sensor on discharge side and flow meter with calorimetric principle sensor.

1 Introduction

Reciprocating compressors are used in wide range of application and in every application they are used in broad spectrum of operating conditions. Refrigeration compressors are produced in big series, however they have to fulfil various requirements of cooling demand. Large scale industrial compressors are mostly different piece by piece as the application changes from chemical processing, gas transportation and storage to compressed air technologies. Keeping the efficiency of a compressor at highest level requires using sophisticated technologies and smart regulation, which must be designed and tested during development process. Simulation tools play a significant role in this process.

There are several approaches how to simulate the cycle of a compressor. First category of models is based on polytropic or isentropic principle inside the cylinder. In the work of Li [1] and Posch [2] there is a deep review of these models and their experimental validation. The models are used to predict mass flow rate, electrical power and discharge temperature, however Posch [2] concluded that no model became apparent to fulfil accuracy demands.

Another models for compressors are based on the energy balance over a control volume. Numerical solution is not complicated, although the results are correct and the models are easy to use. They are often used in combination with different tools, e.g. one dimensional model for suction/discharge line [3], thermal model [4] or electrical model of motor [5]. One of the problem is that these models must be experimentally calibrated for specific type of machine, which reduces the universality for all compressors.

Most advanced tools use complex numerical simulations to predict fluid flow inside the cylinder of a reciprocating compressor. Posch [6] analysed the whole compressor using CFD. Aigner [7] developed his own three dimensional numerical code for compression chamber and suction and discharge line. These models offer very detailed results, however their demand for computational power and time cost is much higher compared to previous models.

2 Simulation model

The model used in this work is based on the principle of energy (1) and mass conservation (2).

$$dW + dQ + \sum_i dm_i h_i = dU \tag{1}$$

$$\frac{dm_{cyl}}{dt} + \sum_i m_i = 0 \tag{2}$$

Q stands for heat transfer from/to the cylinder, $dm_i h_i$ represents inflow or outflow through the valves, dU is inner energy, dW is the work of the piston and the change of inner energy is dU. Movement of valves is described by a single degree of freedom equation (3) [8],

$$m\ddot{x} + d\dot{x} + kx = \sum_i F_i \tag{3}$$

where m is valve mass, d is damping constant and k is spring stiffness. F is the force acting on the valve and x is its position. Flow through the valves is considered as steady, one dimensional and isentropic, solved by following equation.

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$$\dot{m} = \phi_{eff} \cdot \rho_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}} \sqrt{\frac{2\kappa}{\kappa-1}\frac{p_1}{\rho_1}\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}\right)} \tag{4}$$

Effective flow area φ_{eff} is determined using results from numerical simulations of particular valves. Heat transfer inside the cylinder is calculated by improved Annand approach [9], eq. (5)

$$Nu = 0.7\,\mathrm{Re}^{0.7}\,\mathrm{Pr}^{0.7} \tag{5}$$

Thermodynamic properties of the working fluid are gathered from CoolProp library [10]. More information on the simulation model could be found in [11].

3 Experimental setup

The compressor used in this work is one-cylinder air compressor. Parameters are shown in Table 1.

Table 1. Compressor parameters.

	Value	**Unit**
Theoretical volumetric flow	7.4	m^3/hr
Actual volumetric flow	4.3	m^3/hr
Maximum discharge pressure	10	bar
Weight	20	kg
Dimensions	460x300x380	mm

Compressor was mounted with rubber silent blocks to a test bench in order to reduce vibration and noise level, however it still produced a lot of vibrations. Therefore the suction and discharge line were connected to sensors through rubber hoses. Figure 1 illustrates a schematic of a test bench composed of pressure sensors, mass flow meter, temperature sensors and wattmeter.

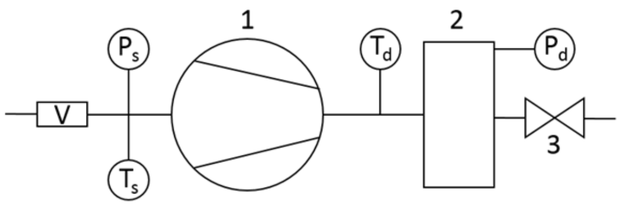

Fig. 1. Schematic of test bench.

Table 2. Test bench parts.

1	Air compressor	T_s	Suction temperature
2	Pressure vessel	T_d	Discharge temperature
3	Control valve	P	Power consumption
m	Flow meter	P_d	Discharge pressure
P_s	Suction pressure		

The compressor is sucking fresh air from surroundings through mass flow meter. Temperature sensor is part of the mass flow meter and it measures the inlet temperature, yet it does not change significantly from room temperature (22°C). Absolute pressure is measured directly in front of suction port of the compressor. Inside the cylinder, the gas is compressed and discharged through the discharge valve into the pipeline, where temperature sensor PT100 is positioned. Following safety valve is used to protect compressor from higher pressure loads than 10 bar. As the reciprocating compressor is not pumping compressed air continuously, but rather in specific intervals, it is necessary to reduce pressure waves before the discharge pressure is measured. Therefore a 20 litres pressure vessel was inserted in the test rig and the pressure is sampled there.

3.1 Methodology

The measurements were performed in two different ways: steady state measurements and continuous measurements. First option means, that the discharge pressure was set to a specific value and measurement lasted long enough to stabilize the discharge pressure and temperature. Stabilization took around 20-30 minutes, especially due to the temperature sensor, which has large thermal inertia and therefore needs more time to reach actual discharge temperature. Second option means that the compressor was started from “no-load” condition and it was run until the discharge pressure reached 10 bars, when the safety valve opened. Temperature was not evaluated in this case.

All measurements started with control valve fully opened. This settings are considered as “no-load” case. In steady state measurements the control valve was manually moved towards closed position to increase the pressure in the vessel. Several points were measured between opened and closed position. Both types of measurements were performed three times, however it was not possible to reach the same discharge pressures in steady state measurements, as the control valve was operated manually. Nevertheless the trends of all measured variables were almost identical.

Suction pressure and discharge pressure obtained from the experiments were used as a boundary conditions for the simulation tool. Validation of the model was based on comparing the mass flow, discharge temperature and power consumption (efficiency) of the compressor.

4 Results and discussion

The air compressor is supplying compressed gas at various pressure levels, depending on particular application. Therefore it is essential to examine the compressor behaviour at different discharge pressures.

Mathematical model of the compressor provides results of cylinder pressure and temperature through the cycle of crankshaft. Ideal validation would be a pressure indication, however the experimental equipment is much more complicated and rather expensive. Comparing “global” parameters of the compressor could still give an answer to model accuracy.

4.1 Volumetric flow

Main parameters that are influencing mass flow are suction and discharge pressure, suction temperature and valve properties. Suction pressure and suction temperature were kept constant during the experiments. Discharge pressure was the parameter that was changed. Valve properties were evaluated by series of CFD simulations and valve spring stiffness measurements.

The flowmeter is measuring volumetric flow under normalized conditions, meaning pressure equal to 1013 hPa and temperature equal to 273.15 K. Density of air at this condition is 1.293 kg/m^3. Mass flow obtained from mathematical model was recalculated for normalized conditions as well and the results are displayed in Figure 2.

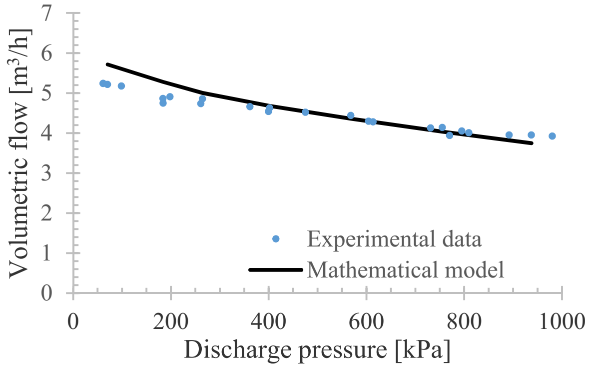

Fig. 2 Volumetric flow through the compressor.

The results from mathematical model show good agreement with experimental data, especially for higher discharge pressures. The difference is below two percent for discharge pressures above 300 kPa. Lower discharge pressures show bigger difference, reaching up to twelve percent for 70.4 kPa discharge pressure. Decreasing volumetric flow is also causing decrease in volumetric efficiency of the compressor. Theoretical volumetric flow of the compressor is 7.4 m^3/hr. This value is dependant just on dimensions of the compressor and its speed. It does not include any losses caused by valves, death volume or superheating of working gas. Following Figure 3 shows decrease in efficiency caused by increasing discharge pressure.

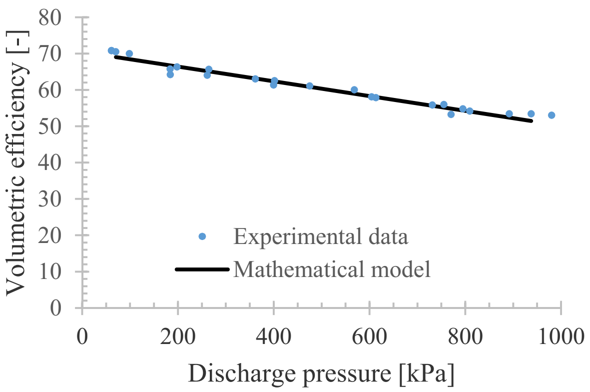

Fig. 3 Volumetric efficiency of the compressor.

4.2 Compressor efficiency

The performance of the compressor is formed by three main factors: electrical efficiency of motor, mechanical efficiency of drive shaft and thermodynamic efficiency of the processes inside the cylinder. However, the character of measurement did not allow us to evaluate every efficiency individually. Overall compressor efficiency was evaluated, comparing power consumption and work done by the compressor, eq. (6).

$$\eta = \frac{W_c}{W_{in}} \tag{6}$$

Actual work of the compressor W_c is calculated using 1st Law of Thermodynamics, equation (7).

$$W = Q - m \cdot \Delta h \tag{7}$$

W is compressor work, *Q* is heat transferred between the cylinder wall and working gas, *m* is the mass inside the cylinder and *h* represents enthalpy.

Results in Figure 4 show experimentally measured power consumption multiplied by estimated efficiency and the work done by the compressor. The efficiency was estimated from series of measurement and was set to 50%. As it can be seen in the Figure 4, the estimated efficiency seems to fit quite adequately with results from mathematical model above 300 kPa discharge pressure. In the region under 300 kPa discharge pressure the work done by compressor decreases significantly, but the power consumption stays under the same trend through the whole measured interval. This means that the overall efficiency is lower when the discharge pressure is decreasing, Figure 5.

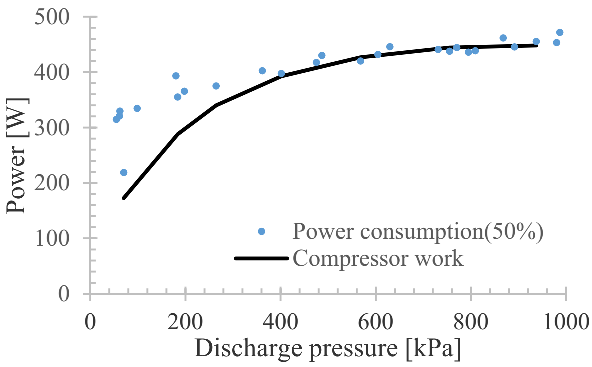

Fig. 4 Power consumption and compressor work.

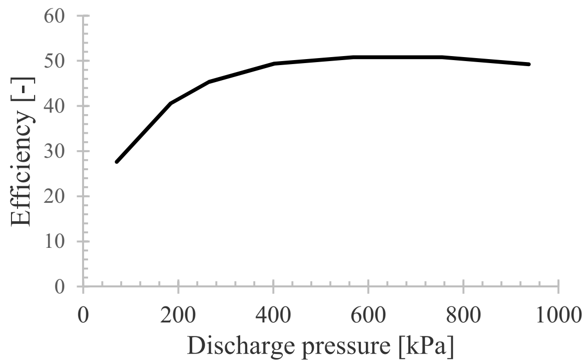

Fig. 5. Overall efficiency of the compressor.

The reason for that could be the electrical efficiency of the motor, which is decreasing as the load on the motor is dropping down [12]. Burt [12] stated, that the efficiency of the motor changes by approximately 10%, when the relative load decreases from 60% to 20%. This is also shown in Figure 5, where the decreasing efficiency of actual compressor is shown with a respect to the discharge pressure. The electric motor should be examined more deeply to prove this decrease in overall efficiency of the compressor. Heat transfer inside the compressor might also cause inaccuracy in efficiency calculation. Lower load on motor and higher mass flow through the compressor cause lower thermal load on walls, which are surrounding working gas. The effect of gas superheating is lower and the work done by compressor will be affected. The same reason could be behind the discrepancies in volumetric flow rate at lower pressures, see Figure 2. The mathematical model uses constant suction temperature, but during the experiments, the gas in the suction line is heated up by the surrounding wall, e.g. valve plate or cylinder head. Measuring the temperature right in front of suction valve could help to predict volumetric flow and work done by compressor more accurately.

4.3 Discharge temperature

The highest gas temperature occurs when the discharge valve is opened. Mathematical model calculates temperature in the cylinder, however this temperature is difficult to measure as it changes very fast. Therefore the probe was positioned in the discharge line. Omitting the effect of re-expansion of compressed gas in discharge chamber and heat transfer through the walls of discharge line caused rather big discrepancies between mathematical and experimental results in Figure 6. A CFD simulation of fluid flow in discharge line proved that the temperature in measuring point will not significantly overcome 100 °C, even for the highest discharge pressure. As there is a significant temperature drop in discharge line a thermal model must be developed and included in the presented mathematical model. Also the experimental setting has to be changed and the thermal probe must be positioned right behind the discharge valve.

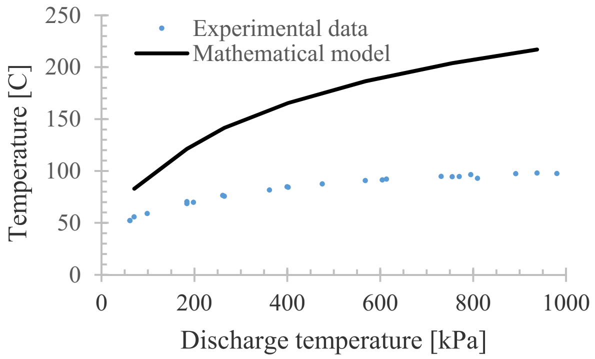

Fig. 6 Discharge temperature.

Conclusion

In this work a validation of developed mathematical model for reciprocating compressor is presented. Comparison of calculated and measured volumetric flow shows good agreement in whole range of working conditions, which is indicating that the mathematical model is giving reasonable results. In case of power-work comparison or discharge temperature the results were not completely satisfying. The efficiency of the compressor was estimated, but it was found out that it is not a constant value and at lower loads it decreases. Therefore it is necessary to perform more experiments focused on this problem. Another issue was found out with discharge temperature. The mathematical model calculates the temperature inside the cylinder, however the in the discharge line of the compressor. The influence of heat transfer in discharge line and gas expansion must be examined more deeply in order to calculate temperature distribution in discharge line accurately.

Another thing which is influencing the calculation is the wall temperature of the cylinder. In the mathematical model the temperature was set to 80 °C, which may not be the actual temperature. Moreover, it may change with varying discharge temperature. Due to this fact, the wall temperature will be measured exactly with thermocouples in future work.

More experimental data are necessary, especially the temperature profile of the compressor in different points. Performing experiments and development of new models for reciprocating compressor will be the goal of future works.

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