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ABSTRACT

During development of refrigeration cycle simulations, the modelling of the compressor requires a trade-off between computational speed and level of detail. In that case, the use of semi-empirical models makes sense instead of complex 1-dimensional models which are common in the development phase of a compressor. Also the very low measuring effort for the adaption of these models favours their use.

The present work contains a comparison of different semi-empirical models found in the literature which simulate the dynamic performance of reciprocating compressors. These models calculate the refrigerant mass flow rate and the compressor power based on polytropic compression and the formulation of a volumetric efficiency. The model parameters are determined by fitting calorimeter data of two compressors. To validate the transient prediction-capability of compressors by these models, start-up and cycling measurements were carried out and compared with the computed data. Due to the common requirements in cycle simulation tools, a model for the discharge temperature was developed and validated additionally.

The goal of the present study is to find a combination of accurate models of mass flow rate, compressor power and discharge temperature which can be applied to cycle simulations of the whole refrigeration system.

1. INTRODUCTION

Despite a reasonably low energy consumption of a single refrigerator, the percentage of the worldwide energy consumption of domestic refrigeration applications is significantly high because of the high number of worldwide refrigerators. To improve the efficiency of household refrigerators, the development process of these devices has changed. In addition to the calculation of the component characteristics and experimental research the simulation of the refrigeration cycle provides a better understanding of the thermal behaviour of the system. Several models have been developed to describe the refrigeration cycle in transient working conditions (Hermes *et al.,* 2008; Philipp, 2002; Jakobsen, 1995). The modelling of a refrigeration system requires sub-models which describe the physical behaviour of the individual components like heat exchangers, expansion devices or the reciprocating compressor. The development of a compressor model for the usage in a cycle simulation requires a trade-off between computational speed and level of detail. Several studies dealt with the topic of modelling reciprocating compressors for household refrigeration appliances but not all concepts are qualified to be used in cycle simulations.

A possibility to model hermetic compressors is a polynomial fit (ASHRAE, 1993). Based on data measured in a calorimeter the correlations are found by fitting high order polynomials. Due to the fact that there are no physical models included and the extrapolation to operational ranges outside the measuring data base does not provide reliable results such models should not be used.

Another category of methods are complex 1-dimensional models. These models use the fundamental equations (conservation of mass, momentum and energy), detailed information of the geometry of the compressor components and they are often combined with 3d-CFD simulations additionally (Lang, 2010). Due to the high computational effort and the need of detailed input-data which can only be available from manufacturers, these methods are avoided in refrigeration cycle simulations. It is more suitable to use them in the development phase of a compressor.

A third approach are so called semi-empirical methods. Such models are combinations of polynomial fits and simple thermodynamic correlations. They also use experimental data from calorimeter measurements but compared to simple polynomial fits the implement physical meaning allows extrapolations so they can be used also in transient applications like dynamic cycle simulations. Semi-empirical methods found in the literature contain models to calculate the refrigerant mass flow rate and the electrical power of reciprocating compressors (Jähnig *et al.*, 2000; Negrao *et al.*, 2011; Li, 2012).

The current work presents a validation of published semi-empirical methods for transient operation conditions. The usage of compressor models in cycle simulations also requires a calculation of the discharge temperature which is presented in this work.

2. MATHEMATICAL MODEL

The bases for the use of semi-empirical methods for reciprocating compressors are separate thermodynamic models for each individual property. The following equations from open literature show the modelling of the mass flow rate and the compressor power. To fulfil the input requirements for refrigeration cycle simulations, a model for the discharge temperature is presented additionally.

2.1 Mass Flow Rate
The highly unsteady compression process and the related unsteady volume flow rate of a reciprocating compressor compared to the time constant of refrigeration systems enables a quasi-steady treatment of the mass flow rate. To model deviations of the constant volume flow a volumetric efficiency is defined.

$$\dot{m}_{calc} = \eta_v \cdot \frac{\dot{V}}{v_{suct}} \tag{1}$$

The definition of the volumetric efficiency is not clearly defined in literature. Jähnig *et al.* (2000) used the clearance volume ratio C and a pressure drop term δp which is a constant percentage of the evaporating pressure. Instead of formulating a polytropic compression process they used the isentropic coefficient κ, which is the ratio of the constant pressure to constant volume specific heats related to the compressor suction conditions. The unknown parameters which have to be fitted are C and δp.

$$\eta_v = 1 - C \left[\left(\frac{p_{cond}}{p_{evap}(1 - \delta p)} \right)^{1/\kappa} - 1 \right] \tag{2}$$

Negrao *et al.* (2011) proposed a simple linear correlation between the mass flow rate and the discharge and suction pressure ratio without any pressure loss using two fitting parameters a and b.

$$\eta_v = a + b \frac{p_{suct}}{p_{dis}} \tag{3}$$

In dependence on the formulation of the compressor mass flow of Jähnig *et al.* (2000), Li (2012) also took the suction pressure drop and the isentropic coefficient related to suction conditions. The unknown parameters in this formulation are b_1, b_2 and δp.

$$\eta_v = b_1 + b_2 \left[\left(\frac{p_{cond}}{p_{evap}(1 - \delta p)} \right)^{1/\kappa} \right] \tag{4}$$

2.2 Compressor Power

The modelling of the electrical compressor power has to account for the sum of loss mechanisms between the electrical supply and the ideal isentropic compressor work. Typical losses are thermodynamic losses due to heat release of the electric motor in the shell or of mechanical friction. A practical approach is to split the compressor power in a constant power loss and in an efficiency corrected isentropic power. The constant power loss contains the power consumption of an unloaded compressor and other influences. The overall efficiency η_{all} contains the efficiency of the motor, the mechanical efficiency and the isentropic efficiency.

$$P = \frac{P_{is}}{\eta_{all}} + P_{loss} \tag{5}$$

The isentropic compressor power can be calculated applying the following correlation:

$$P_{is} = \dot{m} \cdot \frac{\kappa}{\kappa - 1} \cdot p_{suct} \cdot v_{suct} \left[\left(\frac{p_{dis}}{p_{suct}} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right] \tag{6}$$

Jähnig *et al.* (2000) neglected the constant power loss term and introduced a combined efficiency which contains also the constant part. They found an exponential function with three unknown parameters (d, e, f) which provided the best agreement to their experimental data.

$$P = \frac{P_{is}}{\eta_{comb}} \tag{7}$$

$$\eta_{comb} = d + e \cdot exp(f \cdot p_{evap}) \tag{8}$$

In addition to the constant power loss Negrao *et al.* (2011) assumed the overall efficiency as constant so they got two unknown parameters P_{loss} and η_{all}.

Li (2012) proposed a calculation of the overall efficiency which contains suction and discharge pressure. Finally, he got four parameters to be fitted, these are a_1, a_2, a_3 and P_{loss}.

$$\frac{1}{\eta_{all}} = a_1 + \frac{a_2}{p_{dis}} + \frac{a_3}{p_{suct}} \tag{9}$$

Additionally to the models found in literature an in-house model is also used for comparison. The model contains a constant part and a variable part which depends on the Coefficient of Performance (COP) of the ideal process and the cooling capacity Q_0. This model needs two parameters, P_{fix} and η.

$$P = P_{fix} + \frac{Q_0}{COP_{carnot} \cdot \eta} \tag{10}$$

2.3 Discharge Temperature

The use of semi-empirical compressor models in refrigeration cycle simulations requires the estimation of the discharge temperature to get reliable boundary conditions for the simulation of the condenser. The main difficulty in formulating a correlation for the compressor discharge temperature is to find parameters which show similar time behaviour.

Li (2012) presented a formulation of the discharge temperature by modelling the shell temperature. The calculation of the compressor discharge temperature results from the formulation of the energy balance of the entire compressor.

$$P = \dot{m}(h_{dis} - h_{suct}) + \dot{Q}_{amb} \tag{11}$$

The heat transfer rate from the compressor to the ambient can be calculated as:

$$\dot{Q}_{amb} = \alpha \cdot A_{shell}(T_{shell} - T_{amb}) \tag{12}$$

For the formulation of the compressor shell temperature he used a term including the ratio of discharge and suction pressure and an individual term including the discharge pressure. The discharge temperature model by Li (2012) results in three unknown fit parameters, c_1, c_2 and c_3.

$$T_{shell} = c_1 + c_2 \frac{p_{dis}}{p_{suct}} + \frac{c_3}{p_{dis}} \tag{13}$$

The disadvantage of this formulation is the influence of the heat transfer coefficient α between the compressor shell and the ambient. It depends on several parameters like the shape of the compressor or ventilation. In this work the shell temperature is already known so a direct correlation between the discharge temperature and the compressor shell temperature in addition with the isentropic temperature calculation as a function of the pressure ratio has been developed.

$$T_{dis} = T_{shell} \left(f_1 + f_2 \cdot \left(\frac{p_{dis}}{p_{suct}} \right)^{\frac{\kappa-1}{\kappa}} \right) \tag{14}$$

Table 1: Summary of the required fit parameters for the listed models.

model	mass flow rate	power	discharge temperature
Jähnig	2	3	n.a.
Negrao	2	2	n.a.
Li	3	4	n.a.
Authors	n.a.	2	2

2.4 Curve Fit

The unknown parameters in the formulations listed above can be estimated by using a least square curve fit. To solve the least square problem the Levenberg Marquardt method has been used. The objective function for the curve fit is defined as:

$$dX = min \sqrt{\sum \left(\frac{X_{exp} - X_{calc}}{X_{exp}} \right)^2} \tag{15}$$

Where X is replaced by the modelled properties, mass flow rate, compressor power or discharge temperature.
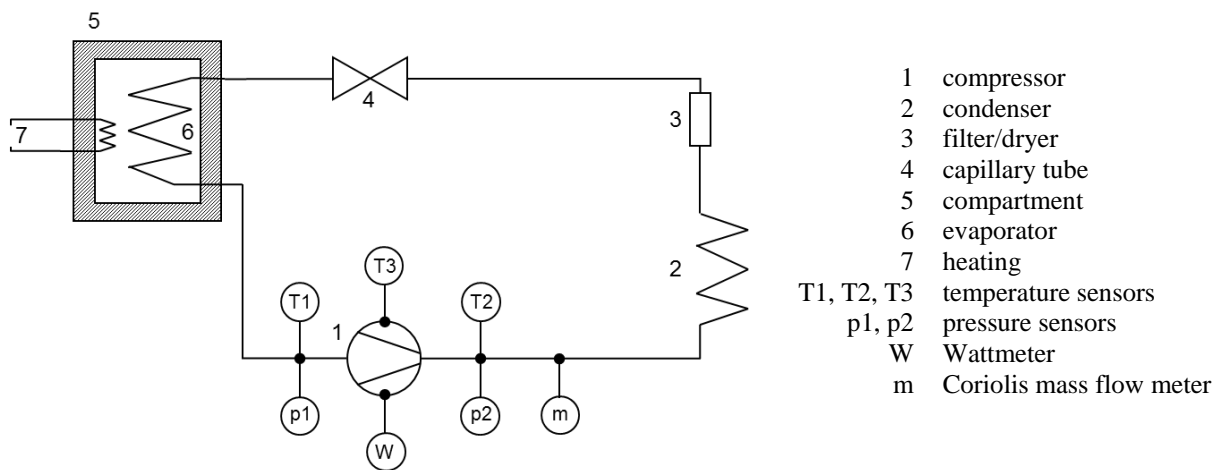
3. EXPERIMENTAL WORK

The validation of the semi-empirical compressor models presented in the previous chapter is done by a comparison of the prediction behaviour for start-up and cycling transients. Two different reciprocating compressors manufactured by Secop Austria GmbH are used (C1 and C2). The main difference between the compressors is the design of the suction muffler. Compressor C1 sucks the refrigerant mainly from the compressor shell, compressor C2 has a direct suction unit. The refrigerant which is used in this study is R600a (isobutane).

The main idea of using semi-empirical methods is the combination of basic physical models and measured data. In this study the measured data is obtained by calorimeter tests. Based on the variation of unknown parameters of different models the number of calorimeter test points is set to six which is sufficient for each model. The calorimeter points are a combination of two condensing temperatures (45°C, 55°C) and three evaporating temperatures (-10°C, -23.3°C and -30°C). The modelled properties mass flow rate, power and discharge temperature has to be measured for the validation. Additionally the input parameters for the described formulations, evaporating and condensing pressures (or temperatures), shell temperature, ambient temperature, suction line temperature (to determine the specific volume and the isentropic coefficient) have to be measured. The shell temperature is measured at the top of the compressor. The mass flow rate is calculated via energy balance using the inlet and outlet conditions of the calorimeter vessel and the vessel power. Table 2 shows the measured values at a specified calorimeter test point.

Table 2: Calorimeter test data. Condensing temperature = 45°C, evaporating temperature = -10°C, ambient temperature = 32°C.

| physical quantity	unit	C1	C2
mass flow rate	kg/h	2.04	2.01
power	W	74.9	71.6
shell temperature	°C	45	39.8
discharge line temperature	°C	54.3	64.5
suction line temperature	°C	32.3	32.3

The determination of the start-up and cycle simulation data for the validation of the investigated models is carried out on a refrigerator test rig. The test rig consists of the following refrigerator parts: a natural convection condenser, a capillary tube (without heat exchanger), an evaporator, a compartment with defined heat input and a compressor. The test rig is equipped with temperature and pressure sensors at the suction and discharge line of the compressor. The mass flow is measured with a Coriolis mass flow meter which is positioned between the compressor discharge line and the condenser. Similar to the calorimeter tests a temperature sensor at the top of the compressor measures the shell temperature.

1	compressor
2	condenser
3	filter/dryer
4	capillary tube
5	compartment
6	evaporator
7	heating
T1, T2, T3	temperature sensors
p1, p2	pressure sensors
W	Wattmeter
m	Coriolis mass flow meter

**Figure 1:** Test rig design.

4. VALIDATION

This chapter contains the comparison of the measured data of the two different compressors and the calculated data. The calculation was carried out with the semi-empirical models fitted to six calorimeter test data points. The quality of the prediction behaviour of the models should be tested for a compressor start-up and cyclic behaviour. The central issue of the calculation is the definition of the suction conditions (specific volume, isentropic coefficient) because it is not possible to measure the temperature of the gas at the intake valve in a direct way. The suction temperature is essential for the simulation because the suction pressure is defined so the suction density is only addicted to the temperature. Three different suction temperatures are investigated: shell temperature, suction line temperature and constant suction temperature at 60°C (and the polytropic coefficient of 1.05) which is usually used for the determination of calorimeter data with the authors' model. The mean absolute deviation of each model to the measured data was evaluated for the start-up and for each cycle (three cycles for C1 and two cycles for C2). The discharge temperature model is validated with each suction condition because a comparison with an existing model cannot be carried out. The measured mass flow rate and the compressor power for both compressors for a start-up and cycle operation conditions can be seen in Figure 2. The rapid decrease of the mass flow rate and the compressor power for compressor C2 during the start-up after 25 minutes is caused by a change of the flow characteristics of the capillary tube.

compressor C1

(a)

compressor C2

(b)
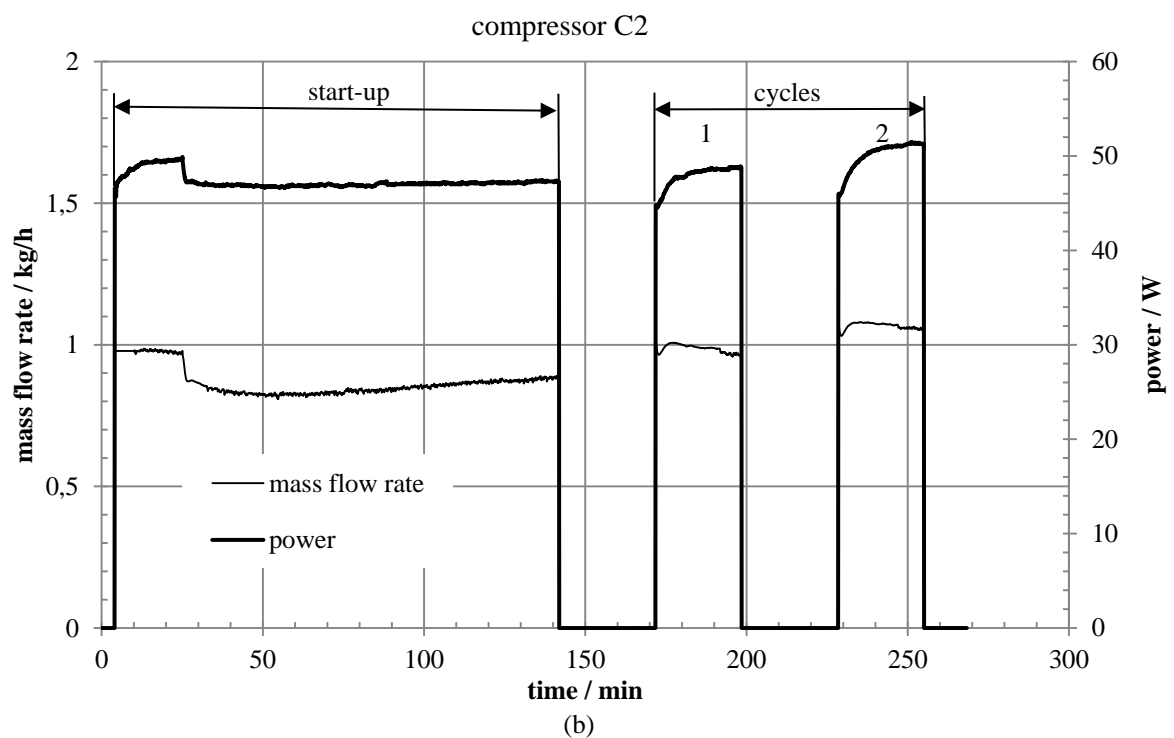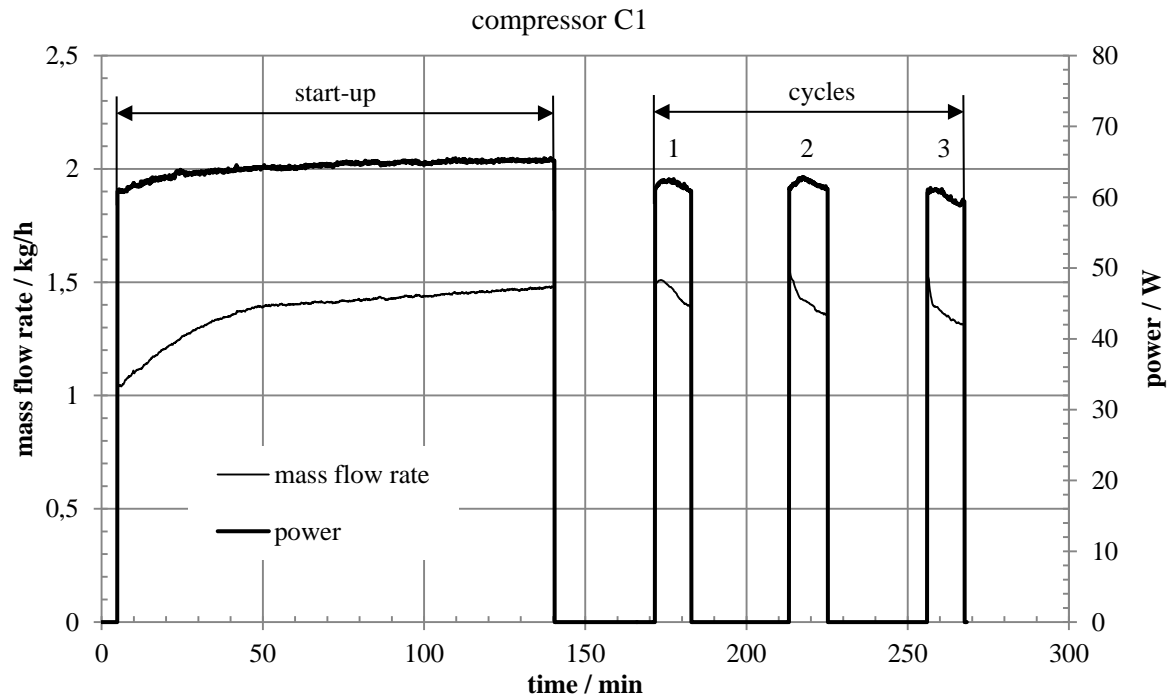
Figure 2: Measured mass flow rate and compressor power of compressor C1 (a) and C2 (b).

Figure 3 shows the measured discharge temperature for compressor C1.
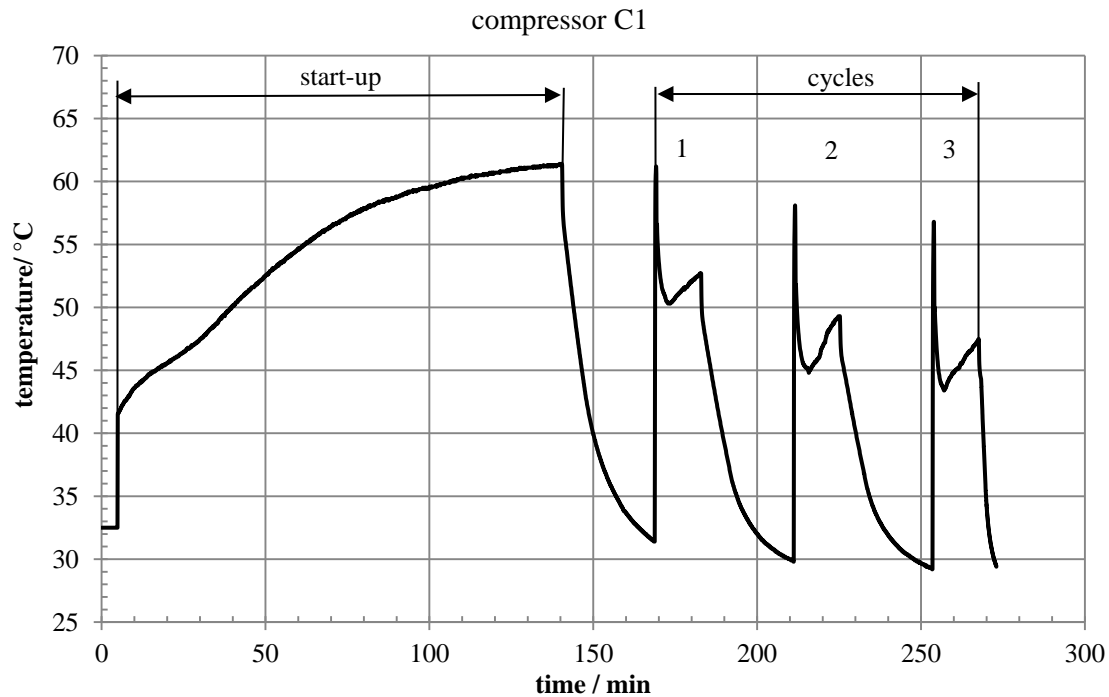
compressor C1

Figure 3: Measured discharge temperature of compressor C1.

4.1 Mass Flow Rate

The analysis of the calculated mass flow rate shows the sensitivity of the different models to the suction conditions and the type of compressor. The simulation of the start-up of compressor C1 shows the best results with Li (2012) using suction line temperature. The prediction of the behaviour of the compressor for cycle tests is well represented with Jähnig *et al.* (2000) using the suction line temperature. The combination of the models with constant temperature and shell temperature do not provide accurate results for compressor C1.

The calculation of the mass flow rate of compressor C2 shows that the use of only one constant temperature for that kind of compressor is not meaningful. The best prediction is achieved with Negrao *et al.* (2011) and 60°C and a polytropic coefficient of 1.05 for the start-up and Jähnig *et al.* (2000) with suction line temperature for cycle mode. Jähnig *et al.* (2000) do not provide reliable results for a constant temperature of 60°C so the results are not considered in Table 3.

A general result of the application of the models to calculate the mass flow rate for cycle simulation is the decrease of accuracy between start-up and cycle modes.

Table 3: Mean absolute deviation between measured and calculated mass flow rate in percent for start-up and cycle mode.

		C1				C2		
		start-up	1.cycle	2.cycle	3.cycle	start-up	1.cycle	2.cycle
Jähnig et al. (2000)	shell	10,36	17,71	15,79	15,60	3,18	11,27	9,82
	suction	3,06	9,76	8,47	8,83	7,36	3,94	3,30
Negrao et al. (2011)	shell	10,15	17,67	15,80	15,57	2,38	10,46	9,48
	suction	3,09	11,237	9,89	10,03	4,08	13,00	12,03
	60/1,05	8,35	17,307	16,20	16,43	1,69	10,17	9,51
Li (2012)	shell	10,33	17,82	15,92	15,68	2,46	10,62	9,62
	suction	2,96	11,06	9,71	9,84	7,81	3,97	3,44
	60/1,05	8,40	17,37	16,26	16,50	1,70	10,23	9,57

4.2 Compressor Power

The simulation of the compressor power shows the influence of the isentropic and the polytropic coefficient in the models respectively. The use of a constant suction temperature of 60°C and a polytropic coefficient of 1.05 results in smaller deviations between the models compared to non-constant values.

Li (2012) applied to compressor C1 gives best results for the calculation of the start-up and cycle mode applying suction line temperature. Similar to the results for compressor C1 Li (2012) in combination with suction line temperature gives the most accurate results for start-up and cycle mode. Jähnig *et al.* (2000) in combination with a constant suction temperature of 60°C and Li (2012) in combination with shell temperature are not considered in Table 4 because they do not provide reliable results.

Despite the low number of fit parameters compared to the models from literature the authors' model does not show significant disadvantages of the prediction-capability.

Table 4: Mean absolute deviation between measured and calculated compressor power in percent for start-up and cycle mode.

		C1				C2		
		start-up	1.cycle	2.cycle	3.cycle	start-up	1.cycle	2.cycle
Jähnig et al. (2000)	shell	1,89	0,72	1,10	0,59	6,86	7,60	6,66
	suction	2,71	3,81	2,58	4,17	12,27	12,51	11,09
Negrao et al. (2011)	shell	2,44	2,16	3,63	2,64	1,77	1,49	1,15
	suction	1,69	1,61	3,14	2,18	1,87	1,60	1,28
	60/1,05	2,38	2,08	3,60	2,64	1,21	0,84	0,65
Li (2012)	suction	1,41	1,45	3,02	2,08	0,50	0,59	0,71
	60/1,05	2,26	2,16	3,72	2,81	0,93	0,44	0,40

4.3 Discharge Temperature

The different design of the investigated compressors results in a considerably influence in the simulation of the discharge temperature. Using the shell temperature gives the best results for both compressors for start-up and cycle mode. The longer discharge line of compressor C1 leads to a decrease of the difference between the discharge and shell temperature and a higher accuracy of the simulation. The use of the suction line temperature or constant 60°C does not lead to desired results in combination with the presented model.

Table 5: Mean absolute deviation of the measured and calculated discharge temperature in Kelvin for start-up and cycle mode.

	C1				C2		
	start-up	1.cycle	2.cycle	3.cycle	start-up	1.cycle	2.cycle
shell	1,37	0,13	1,46	0,57	5,81	6,03	6,68
suction	3,85	1,88	0,51	1,08	8,65	8,38	8,96
60/1,05	1,61	0,16	1,50	0,72	6,45	6,56	7,05

5. CONCLUSION

In this study a comparison of the transient prediction-capability of different semi-empirical models for the simulation of mass flow rate, compressor power and discharge temperature are presented. Two compressors with different design were used to validate the models. The fit parameters of the models were determined with calorimeter test data. Three different reference temperatures were used to define the suction conditions because this temperature is essential for the estimation of the suction density which has a big influence on the modelled properties.

Based on the investigation of the prediction-capability of the transient behaviour, no model has become apparent to fulfil accuracy demands for both compressors. The development of a universal compressor model which could be used in dynamic cycle simulations of refrigeration systems with respect to variable speed compressors is going to be the main goal of future works.

NOMENCLATURE

$\dot{m}$	mass flow rate	(kg/s)	**Subscripts**		
$\dot{V}$	volume flow rate	(m³/s)	calc	calculated	
v	specific volume	(m³/kg)	v	volumetric	
C	clearance volume ratio	(-)	suct	suction condition	
δp	percentage pressure loss	(-)	cond	condenser	
p	pressure	(Pa)	evap	evaporator	
a, b, b_1, b_2	fitting parameters for mass flow rate	(-)	is	isentropic	
P	power	(W)	comb	combined	
COP	coefficient of performance		amb	ambient	
Q_0	cooling capacity	(-)	dis	discharge	
T	temperature	(W)	shell	compressor shell	
$\dot{Q}$	heat flux	(K)			
A	area	(W)			
d, e, f	fitting parameters for compressor power	(m²)	**Greek symbols**		
a_1, a_2, a_3	fitting parameters for compressor power	(-)	η	efficiency	(-)
c_1, c_2, c_3	fitting parameters for shell temperature	(-)	κ	isentropic coefficient	(-)
f_1, f_2	fitting parameters for discharge temperature	(-)	α	heat transfer coefficient	(W/m²K)
X	modelled property	(-)			

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