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EVALUATING REFRIGERANTS USING A DYNAMIC RECIPROCATING COMPRESSOR MODEL

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ABSTRACT

This paper presents an approach to investigate the refrigerant's influence on a reciprocating compressor. Highlight of this model is the one-dimensional semi-physical approach, which is general fluid independent. Compared to compressor models from literature, this model handles with real gas behaviour. The applied fluid property models have been specifically designed to be fast and accurate. This is given by the implementation of a hybrid approach based on the simplified Helmholtz Equation of State (EoS) and polynomial functions. Considering rotational speeds between 30 s^{-1} and 70 s^{-1} and pressure ratios from 5 up to 13, results show a maximum isentropic efficiency of 67 %, 63 %, 66 %, 53 % for the refrigerants R410A, R134a, R744, R32, respectively.

Keywords: Heat Pump, Refrigerant-Retrofitting, Modelica, Fluid Model, Hybrid approach

1. INTRODUCTION

The European F-Gas Regulation accelerates the exchange of conventional fluids in refrigeration cycles (European Union, 2014). Typically, conventional refrigerants in heat pumps are highly harmful to the climate. Thus, it is necessary to find alternative environmental friendly refrigerants. Considering a heat pump in detail, the compressor performance is crucial for the overall efficiency. Various approaches have been made in the literature to compare different refrigerants in compressors (Cimmino and Wetter, 2017; Li, 2013; Navarro-Peris et al., 2013). These approaches are often stationary and well applicable to conventional heat pumps with constant speed. In recent years, however, the demands on heat pumps have continued to grow. For example, speed-controlled heat pumps are used to provide domestic hot water, space heating and, by reversing the process, also for cooling. This leads to high operating dynamics and justifies the analysis of transient states. In addition, publicly, available data on efficiencies is very limited to compare different refrigerants reasonably. Supporting the screening and evaluation process, open-source dynamic simulation models are a promising method to have a reasonable decision tool in the assessment process.

This paper presents an approach to investigate the refrigerant's influence on a reciprocating compressor. For this purpose, we develop a dynamic simulation model in Modelica based on a semi-physical compressor model from the literature (Roskosch et al., 2017). The highlight of this model is the one-dimensional white-box approach, which is fluid independent. In comparison to compressor models from the literature, this model handles with real gas behaviour and, therefore, it is able to predict volumetric and isentropic efficiencies depending on variable operating points. The used fluid property models were specially developed to be fast and accurate (Sangi et al., 2014; Vering et al., 2020). The refrigerants R410A, R134a, R744 and R32 are evaluated under dynamic conditions in this work. R140A and R134a currently provide most data and models for validation, whereas R32 and R744 are good candidates for the new generation of refrigerants for both, heating and cooling applications. Considering rotational speeds between 30 s^{-1} and 70 s^{-1} and pressure ratios from 5 up to 13, the simulation model calculates isentropic and volumetric efficiency of the process. Hence, we are able to compare different refrigerants under operating conditions. To further increase simulation speed at almost the same accuracy, we deduce correlations that can be used in e.g. annual simulations.

2. PHYSICAL COMPRESSOR MODEL

The presented compressor model is implemented in Modelica and based on a semi-physical compressor model from the literature (Roskosch et al., 2017). Therefore, only the most important fundamentals are explained in this chapter. For a more detailed description of the compressor model, see (Roskosch et al., 2017). Figure 1 shows the general structure of the implemented model in Modelica. The great advantage of such a polymorphic model structure is the possibility to realize a universally applicable compressor model. With a corresponding adaptation, further compressor geometries and even compressor types can be modelled and calculated in a relatively short time. Lines marked in blue are fluid flows, red lines are heat flows and black lines are geometric quantities. The geometry model describes the volume of the piston. For this purpose, it uses the crank angle of the motor shaft ϕ as input variable. The sub-model "Compression process" describes the compression of the refrigerant by calculating the states of the gas and the mass flows determining energy and mass balances. In addition, the model

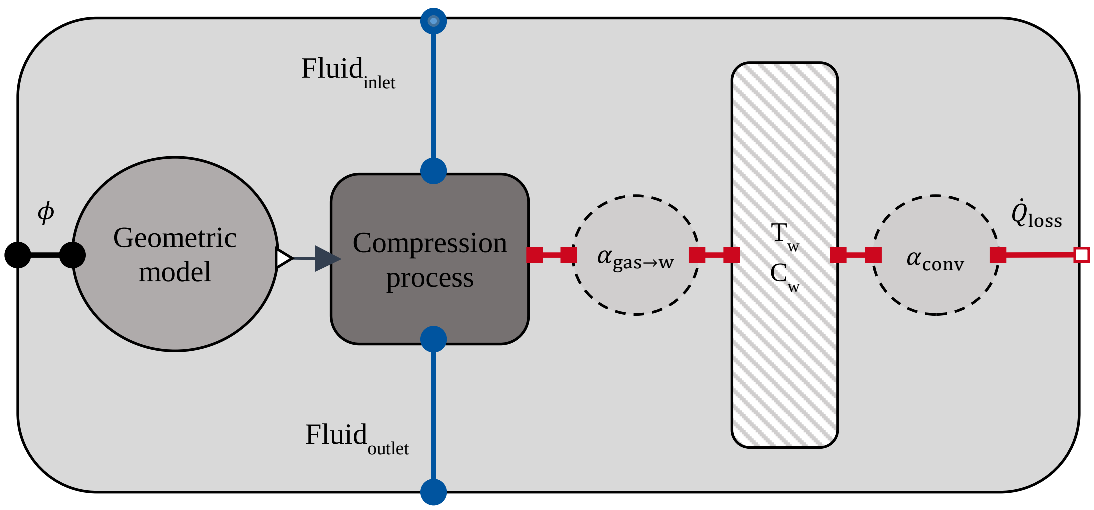

Figure 1: Modular structure of the compressor model in Modelica

has sub-models for the heat transfer between gaseous refrigerant and piston wall $\alpha_{gas \to w}$, and between piston wall and environment α_{conv}. The model's modular design enables the modification of the geometry or the heat transfer models without much effort.

The submodel "Compression process" is independent of the compressor's geometry and the used refrigerant. It calculates the current thermodynamic state by solving the energy balance (Eq.: 1) and the mass balance (Eq.5). The control volume of all following balances refers to the inside of the piston. The first law of thermodynamics leads to

$$\frac{d}{dt}\left(m_{gas} * u_{gas}\right) + \dot{m}_{outlet} * h_{gas} - \dot{m}_{inlet} * h_{inlet} - \frac{dQ_{gas \to w}}{dt} - \frac{dW_{rev}}{dt} - \frac{dW_{irr}}{dt} = 0 \qquad \text{Eq. (1)}$$

where m_{gas} describes the actual mass of the gas in the piston and u_{gas} the actual specific energy of the gas. The actual specific enthalpy of the gas and the specific enthalpy of the gas at the inlet is given by h_{gas} and h_{inlet}. The mass flow at the outlet and inlet are described by $\dot{m}_{outlet}$ and $\dot{m}_{inlet}$. The reversible and irreversible work is represented by W_{rev} and W_{irr} respectively. The reversible work W_{rev} is obtained by assuming quasi-static changes of state according to the following equation using the actual pressure inside the piston p_{gas}:

$$\frac{dW_{rev}}{dt} = -p_{gas}\frac{dV_{gas}}{dt} \qquad \text{Eq.: (2)}$$

The irreversible work considers different processes but is mainly based on the sliding friction between the piston rings and the cylinder. For this reason, the friction pressure $p_{friction}$ is introduced to describe the dissipated work

$$\frac{dW_{\text{irr}}}{\text{dt}} = -p_{\text{friction}} \frac{dV_{\text{gas}}}{\text{dt}} \qquad \text{Eq. (3)}$$

The heat transfer between gas and cylinder $\frac{dQ_{\text{gas}\to\text{c}}}{dt}$ is a dynamic quantity, since the gas temperature has a high-temperature change especially during the compression. Equation 4 describes $\frac{dQ_{\text{gas}\to\text{c}}}{dt}$ according to Newton's cooling law:

$$\frac{dQ_{\text{gas}\to\text{c}}}{dt} = \alpha_{\text{gas}\to c} A_{\text{cg}} \left(T_{\text{c}} - T_{\text{gas}}\right) \qquad \text{Eq. (4)}$$

whereas A_{cg} represents the area of the cylinder that is in contact with the refrigerant. The heat transfer coefficient $\alpha_{\text{gas}\to c}$ is calculated using the Woschni correlation (Woschni et al., 1998). This correlation originates from combustion technology and is used with neglect of combustion but also predict convection coefficients within a compressor (Tuhovcák et al., 2016).

To determine the current refrigerant mass, the mass balance around the control volume is considered:

$$\frac{dm_{\text{gas}}}{dt} - \dot{m}_{\text{inlet}} + \dot{m}_{\text{outlet}} = 0 \qquad \text{Eq. (5)}$$

The loss-free Bernoulli equation provides the mass flows $\dot{m}_{\text{inlet}}$ and $\dot{m}_{\text{outlet}}$. In addition, effective flow areas are used, which are refrigerant and flow-dependent (Roskosch et al., 2017).

The thermodynamic state is determined by the two independent state variables density and enthalpy using our fluid models in Modelica. Both, the used fluid models and the compressor model is open-source available on GitHub: https://github.com/RWTH-EBC/AixLib/issues/739.

3. RESULTS

The compressor performance is analysed for the fluids R134a, R410A, R32 and R744. The reciprocating compressor is simulated for 20 seconds at each operating point.

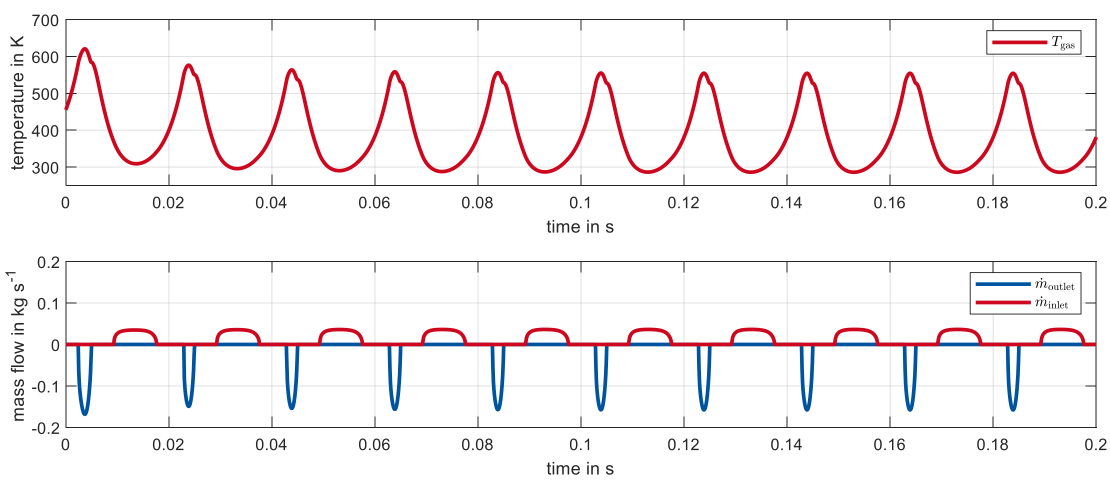

Figure 2: Simulated start-up of the reciprocating compressor of refrigerant R744 at an inlet pressure of 10 bar, an inlet temperature of 293.15 K, an outlet pressure of 100 bar and a rotational speed of 50 1/s.

This simulation time has proven to be sufficiently long to have stationary conditions since the transient process is already completed after a few cycles (see Figure 3). The inlet temperature is constant 290.15 K for all considered fluids.

Compared to the inlet pressure for the refrigerants R32 and R134a (saturated vapour pressure at 273.15 K) the inlet pressure for the fluids R744 and R410A was set lower to allow a larger variation range of the pressure ratio Π. Typical measures like the isentropic and the volumetric efficiency are used to analyse the compressor performance. Initially, the general operation of the compressor model is considered. To demonstrate the functionality of the compressor model, the temperature and mass flow curves are shown in Figure 2 as an example for the refrigerant R744 for $\Pi = 10$. Since R744 cycles operate at a very high pressure and temperature level (supercritical) this figure underlines that the model works in the supercritical range. It can be seen, that the model provides realistic curves for all variables and allows detailed insight into significant thermodynamic values during the compression process. It is also noticeable that the gas temperature assumes values over 500 K. This can be explained by the high pressures that occur in an R744 refrigeration circuit, which rise above 120 bar. By the known time-dependent variables temperature, pressure and mass flow, the influence of the pressure ratio can be analysed by calculating the isentropic and volumetric efficiency.

3.1. Influence of the pressure ratio

In the following, the isentropic, as well as the volumetric efficiency, are used to evaluate the overall compressor efficiency. In order to systematically investigate different operating points and the resulting influence on compressor efficiency, the pressure ratio at constant compressor speed will first be analysed in detail. For constant compressor inlet conditions (pressure and temperature) the pressure ratio Π is varied by changing the outlet pressure. The isentropic efficiency is calculated according to the well-known equation Eq. (7) as a function of the discharge enthalpy, the inlet state and the isentropic work $(h_{\mathrm{outlet}}^{\mathrm{is}} - h_{\mathrm{inlet}})$.

$$\eta_{\mathrm{C}}^{\mathrm{is}} = \frac{\int \dot{m}_{\mathrm{outlet}}\,\mathrm{d}t\left(h_{\mathrm{outlet}}^{\mathrm{is}} - h_{\mathrm{inlet}}\right)}{\int \dot{m}_{\mathrm{outlet}}\,h_{\mathrm{outlet}}\,\mathrm{d}t - h_{\mathrm{inlet}}\int \dot{m}_{\mathrm{inlet}}\,\mathrm{d}t} \quad \text{Eq. (7)}$$

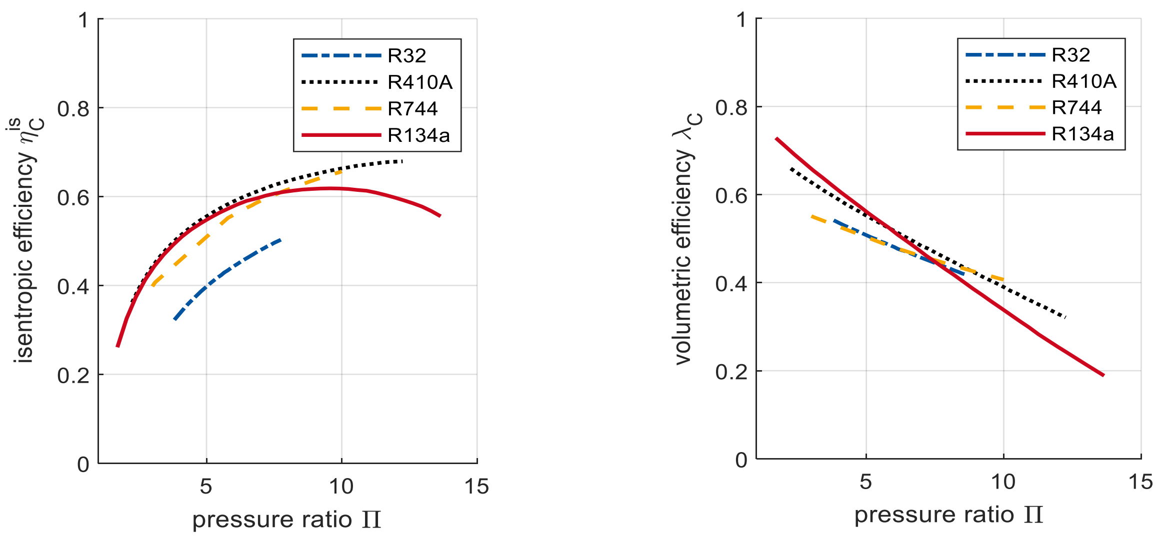

Figure 3: Simulated isentropic efficiency (left) and volumetric efficiency (right) of the refrigerants R32, R410A, R134a and R744; $T_{\mathrm{inlet}} = 290.15$ K, $p_{\mathrm{inlet,R32}} = 8.13$ bar, $p_{\mathrm{inlet,R410}} = 4$ bar, $p_{\mathrm{inlet,R744}} = 10$ bar, $p_{\mathrm{inlet,R134a}} = 2.93$ bar

Since the compressor outlet enthalpy is not constant during the discharge process, it must be integrated along with the discharge time duration. The same procedure is used for volumetric efficiency:

$$\lambda_{\mathrm{C}} = \frac{\int \dot{m}_{\mathrm{inlet}}\,dt}{V_0 \rho_{\mathrm{inlet}}} \quad \text{Eq. (8)}$$

In Eq. (8), V_0 ist the volume of the piston and $\int \dot{m}_{\mathrm{inlet}}\,dt$ describes the time integral of the mass flow during the suction process. Therefore, the results of Eq. (7) and Eq. (8) are analysed in the following section.

Figure 3 (left) shows the isentropic and Figure 3 (right) the volumetric compressor efficiency of the refrigerants as a function of the pressure ratio. The maximum pressure ratio varies depending on the fluid due to the different definition ranges of the fluid models. Regardless of the fluid, the isentropic efficiency first reaches a maximum with increasing pressure ratio and then decreases again. The highest efficiency is observed at the lowest pressure ratio and decreases linearly with increasing pressure ratio. Both, the curves for the isentropic efficiencies, and the curves of the volumetric efficiencies show typical characteristic curves (Sánchez et al., 2017).

Our model predicts differences for the isentropic efficiencies depending on the fluid. This applies to the position of the maxima in relation to the pressure ratio, the curvature of the parabolic curve before and after the maximum, as well as the values of the absolute maxima. Of the considered refrigerants, R410A reaches the highest value in absolute terms. The efficiency characteristics of R744 are similar to those of R410A. The isentropic efficiency of R134a reaches its maximum at a pressure ratio of $\Pi = 10$ with a value of 62%. For R32 the model predicts significantly lower efficiencies than for the other considered refrigerants. In addition, isentropic efficiency can no longer be calculated for pressure conditions above $\Pi = 8$, because the substance data are not sufficiently available in RefProp. In general, the difference between the values of R32 in absolute terms and in comparison to R134a is confirmed in the literature. (Antunes and Bandarra Filho, 2016).

In approximation, the volumetric efficiencies dependent on the pressure ratio of all considered refrigerants (Figure 3, right) show a linear behaviour. Furthermore, at a pressure ratio of approximately $\Pi = 8$, all straight lines show a volumetric efficiency of approximately 45% and only differ in their different gradients.

3.2. Influence of the compressor rotation speed

Various simulation studies in the literature do not take into account, that the isentropic efficiency is strongly influenced by the compressor rotation speed. Therefore, in addition to the simple variation of the pressure ratio, the influence of the compressor rotation speed on the compressor efficiency is examined in this paper. Our preliminary investigations have shown that the compressor rotational speed has no significant effect on the characteristic course of the isentropic efficiency in dependence on the pressure ratio. For this reason, the fluid-dependent pressure ratio Π at the point with the maximum observed isentropic efficiency was chosen according to Figure 3. For this constant pressure ratio, the compressor rotation speed is varied**.** Figure 4 shows the isentropic (left) and volumetric (right) compressor efficiency as a function of the compressor rotation speed for the considered refrigerants. It can be observed, that the isentropic efficiency shows an almost linear behaviour for all considered fluids (see Figure 3). With increasing rotational speed the isentropic efficiency of all considered fluids decreases almost linear. This behaviour can be generally observed and confirmed in an experimental investigation in the literature (Mendoza-Miranda et al., 2016; Venzik et al., 2017; Venzik and Atakan, 2017). These effects can be explained as follows. Due to increasing compressor speed, the piston speed also increases and thus both the irreversibilities caused by the mass acceleration of the piston and the stronger frictional influences between piston and cylinder wall (viscous friction) increase. In addition, higher refrigerant mass flows also lead to greater flow losses during the suction and discharge process (Roskosch et al., 2017). All these effects lead to a significant reduction of the isentropic efficiency and thus to the reduction of the overall efficiency when the compressor speed rises. However, the increasing compressor rotation speed also influences the volumetric efficiency directly.

The volumetric efficiency in Figure 4 (right) also shows a linear decrease for increasing rotation speed. This behaviour also applies to all considered fluids and can be explained by higher valve flow resistance. The valve resistance at the outlet depends on the mass flow density. This is higher since the discharge process takes shorter time due to the higher rotational speed. For this reason, more residual gas remains within the cylinder after the discharge process. As a consequence less fresh gas get into the cylinder during the suction process, whereby the volumetric efficiency decreases. This leads to a decrease of the isentropic efficiency of 0.03 and of 0.037 for the volumetric efficiency at a speed increase of 10 Hz (e.g. for R410A).

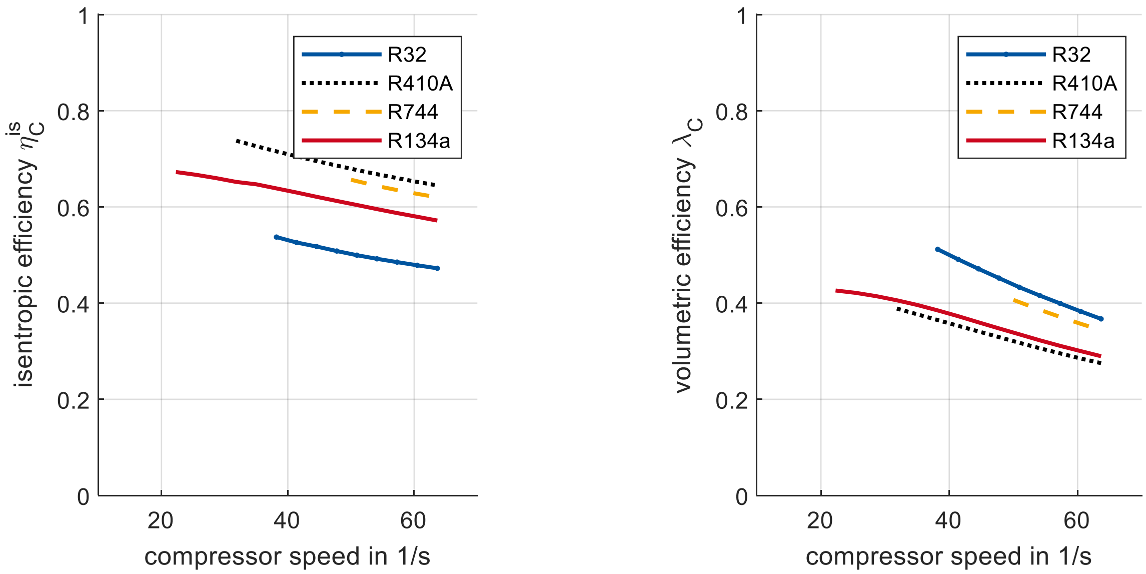

Figure 4: Simulated isentropic (left) and volumetric (right) efficiency of the refrigerants R32, R410A, R744 and R134a as a function of the compressor rotational speed

3.3. Reduction of model's complexity

The presented detailed model allows refrigerant-dependent predictions about its operating behaviour. From this, simplified correlations for the isentropic and volumetric efficiency are developed for a constant compressor rotational speed of 50 s^{-1}. These correlations allow reducing the complexity of the model. This increases the simulation speed, which enables new simulation studies, such as annual simulations.

A polynomial function of the form

$$\eta_{\mathrm{C}}^{\mathrm{is}} = \eta_0 + a(\Pi - \Pi_0)^2 \qquad \text{Eq. (8)}$$

is used to describe the isentropic efficiency. From the polynomial vertex shape in Eq. (8), the maximum achievable isentropic efficiency η_0 and its position Π_0 is directly apparent. The quantity a is a fluid-dependent regression coefficient. The Root Mean Square Error (RMSE) is below 0.02 for all investigated refrigerants. Table 1 shows the accuracy and all coefficients of the correlations for the isentropic efficiency.

Table 1: Accuracy and coefficients of correlations for isentropic efficiency

Fluid	RMSE	η_0	Π_0	a
R32	0.0012	0.5284	10.0286	-0.0052
R410A	0.0100	0.6725	10.8725	-0.0037
R744	0.0025	0,6570	11,1987	-0,0038
R134a	0.0170	0.6317	9.4101	-0.0051

The correlation for the volumetric efficiency is based on a linear approach and is shown in Eq. (9):

$$\lambda_C = b_{\mathrm{fluid}} + m * M_{\mathrm{fluid}}\, \Pi \qquad \text{Eq.(9)}$$

In equation 9, b describes a fluid-dependent and m a fluid-independent regression coefficient. In the literature, the integration of the molar mass is proposed for the consideration of fluid-specific quantities (Mendoza-Miranda et al., 2016). For this reason, the linear approach includes the molar mass M_{ref}. Thus, the correlations for the volumetric efficiency only have one fluid-dependent regression coefficient. The RMSE for the volumetric efficiency for all refrigerants is less than 0.006.

Table 2: Accuracy and coefficients of correlation for volumetric efficiency

Fluid	RMSE	b_{fluid}	M_{fluid}	m
R32	0.0034	0.6187	52.02 g mol^{-1}	-4.40E-4
R410A	0.0058	0.7124	72.60 g mol^{-1}	-4.40E-4
R744	0.0027	0.5977	44.01 g mol^{-1}	-4.40E-4
R134a	0.0058	0.7898	102.04 g mol^{-1}	-4.40E-4

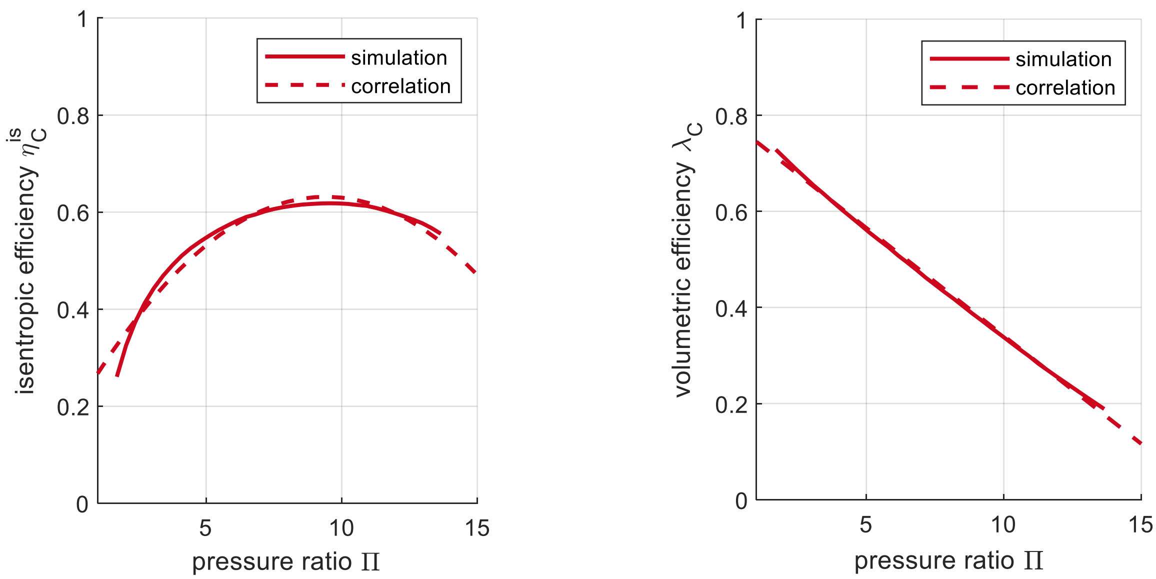

Figure 5: Simulated isentropic (left) and volumetric (right) efficiency as a function of the pressure ratio and fluid-specific correlations for R134a

4. CONCLUSION AND OUTLOOK

The presented compressor model is based on a semi-physical approach, which is fluid-independent. The thermodynamic state of the refrigerant is determined by energy and mass balances around the cylinder. The modularity of the model allows the application of further geometries and other compressor types. To be able to analyse different fluids in a short time, it is necessary to connect the compressor model with a fast and high accurate fluid model. Especially for this issue, fluid models for the considered fluids R744, R32, R134a and R410A based on EoS were developed.

Using the compressor model in combination with the fluid model, it is possible to show the influence of the fluid, pressure ratio and compressor rotational speed on the compressor efficiency. The compressor model provides physically plausible values concerning isentropic and volumetric efficiencies. In comparison to common simulation models from the literature, this compressor model considers also the compressor rotational speed and their influence on the isentropic and volumetric efficiency. For nominal rotational speed (50 s^{-1}), the highest isentropic efficiency was observed for R410A at a pressure ratio of about 12. In contrast to the lowest rotational speed (30 s^{-1}), the isentropic and volumetric efficiencies of R410A decrease with increasing rotational speed from 0.74 to 0.64 and from 0.39 to 0.28 respectively (const. $\Pi = 12.5$).

In future work, this compressor model will be evaluated by a compressor test bench. To use the simulation results in a fast way for further simulation studies, such as annual simulations, correlations for isentropic and volumetric efficiency were developed. These correlations provide accurate data and can be easily implemented in other models.

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REFERENCES

Antunes, A.H.P., Bandarra Filho, E.P., 2016. Experimental investigation on the performance and global environmental impact of a refrigeration system retrofitted with alternative refrigerants. International Journal of Refrigeration 70, 119–127.

Cimmino, M., Wetter, M., 2017. Modelling of Heat Pumps with Calibrated Parameters Based on Manufacturer Data. Presented at the The 12th International Modelica Conference, Prague, Czech Republic, May 15-17, 2017, pp. 219–226.

European Union, 2014. REGULATION (EU) No 517/2014 OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 16 April 2014 on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006.

Li, W., 2013. Simplified steady-state modeling for variable speed compressor. Applied Thermal Engineering 50, 318–326.

Mendoza-Miranda, J.M., Mota-Babiloni, A., Ramírez-Minguela, J.J., Muñoz-Carpio, V.D., Carrera-Rodríguez, M., Navarro-Esbrí, J., Salazar-Hernández, C., 2016. Comparative evaluation of R1234yf, R1234ze(E) and R450A as alternatives to R134a in a variable speed reciprocating compressor. Energy 114, 753–766.

Navarro-Peris, E., Corberán, J.M., Falco, L., Martínez-Galván, I.O., 2013. New non-dimensional performance parameters for the characterization of refrigeration compressors. International Journal of Refrigeration 36, 1951–1964.

Roskosch, D., Venzik, V., Atakan, B., 2017. Thermodynamic model for reciprocating compressors with the focus on fluid dependent efficiencies. International Journal of Refrigeration 84, 104–116.

Sánchez, D., Cabello, R., Llopis, R., Arauzo, I., Catalán-Gil, J., Torrella, E., 2017. Energy performance evaluation of R1234yf, R1234ze(E), R600a, R290 and R152a as low-GWP R134a alternatives. International Journal of Refrigeration 74, 269–282.

Sangi, R., Jahangiri, P., Klasing, F., Streblow, R., Müller, D., 2014. A Medium Model for the Refrigerant Propane for Fast and Accurate Dynamic Simulations. Presented at the the 10th International Modelica Conference, March 10-12, 2014, Lund, Sweden, pp. 1271–1275.

Tuhovcák, J., Hejcik, J., Jicha, M., 2016. Heat Transfer Analysis in the Cylinder of Reciprocating Compressor. Presented at the International Compressor Engineering Conference, Indiana, USA.

Venzik, V., Atakan, B., 2017. Temperature glide matching in a zeotropic mixture heat pump. Presented at the Proceedings of Ecos 2017, San Diego, CA, USA.

Venzik, V., Roskosch, D., Atakan, B., 2017. Propene/isobutane mixtures in heat pumps: An experimental investigation. International Journal of Refrigeration 76, 84–96.

Vering, C., Engelpracht, M., Nürenberg, M., Müller, D., 2020. An open-source Modelica library for design and control: Modular and scalable heat pump models using natural refrigerants. Presented at the 14th Gustav Lorentzen Conference, Kyoto, Japan.

Woschni, G., Klaus, B., Zeilinger, K., 1998. Untersuchung des Wärmetransportes zwischen Kolben, Kolbenringen und Zylinderbüchse. MTZ Motortech Z 59, 556–563.