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Semi-empirical Scroll Compressor Model with Optional Vapor-injection

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ABSTRACT

The development and evaluation of heat pump system designs and control strategies is often based on simulative experiments on a virtual test rig. Such system simulations require component models of sufficient prediction precision while not being over-complex. One key component of every vapor compression heat pump is the compressor.
This study presents a semi-empirical model of a scroll compressor with optional vapor-injection. The model predicts the electric power input and refrigerant mass flow as well as the hot gas temperature at the compressor's discharge port. It is based on thermodynamic subprocesses covering refrigerant transitions. Important refrigerant mass flows, losses and heat transfers within the compressor are considered as well as its interaction with the surroundings. A set of parameters is used to adapt the generic model to a specific compressor. All parameters are derived based on data which is typically provided by compressor manufacturers in their catalogues or online tools. The parameters are determined via optimization. An important advantage of the proposed model is that the parameters are not purely empirical, but they have a physical significance. The optimization is monitored so that a global optimum is found. The model is then validated with multiple test cases.
The combination of a physically consistent generic model with empirically determined parameters of physical meaning has multiple advantages: A model for one specific compressor can be derived from information typically provided by manufacturers. Other than polynomial models, this model can be used for extrapolation, i.e., to predict compressor performance outside of the envelope of the data it was derived from. It can also be applied to test the use of new refrigerants which are not the design refrigerant.

1. INTRODUCTION

Using simulation tools for the larger part of heat pump development is expected to decrease engineering cost and shorten time to market. The idea is to integrate multiple capabilities in order to model and test components, systems and controls in one simulation framework: A virtual test rig (VTR). Such a VTR is being developed for subcritical vapor compression heat pumps. The following critera are required:

- Accurate representation of the components and the system topology
- Flexibility in refrigerant choice
- Capability to test design decisions on the system topology level
- Capability to test design decisions on the component level
- Capability to test control strategies
- Capability of wellfounded extrapolation

To model and test realistic heat pump behavior, especially concerning the controls, (quasi-) dynamic simulation is necessary. The VTR functionality shall be demonstrated at economizer heat pump systems with one or multiple scroll compressors using hydrocarbon refrigerants.

The compressor or compressor unit is a key subsystem of the heat pump from a cost, performance and constrols perspective. This necessitates a good understanding and model quality of it. Three classes of scroll compressor models exist: Analytical models, empirical models and semi-empirical models (Byrne et al., 2014).
Analytical models are also called geometrical or "mechanistic" (Bell & Team, 2020) because they consider the geometry and mechanics of compressor parts, e.g., the scrolls. They are used for compressor optimization and design. Much information needed for this type of model is only available to compressor manufacturers and not to system designers. They are also complex and computation-intensive. (Byrne et al., 2014; Winandy et al., 2002)
The standard approach to incorporate compressor behavior into a system simulation of a heat pump is the use of empirical data. Following EN 12900 (Refrigeration Technology Standards Committee, 2013), manufacturers of positive displacement refrigerant compressors can provide this data either in graphical or tabular form or as polynomials. These polynomial equations are specified in the before mentioned norm and in AHRI 540 (AHRI, 2020). Due to the merely mathematical character of these polynomials, they are only valid within the envelope of the data they are based on. Polynomials are valid for one refrigerant and compressor speed. As stated in EN 12900, extrapolation is not allowed. Since extrapolation and refrigerant switch capabilities are on the criteria list for the VTR, an empirical modeling approach of the compressor is discarded.
Based on a literature review by Byrne et al. (2014), semi-empirical models are often used for performance assessment of refrigeration technology such as heat pumps. Typical model outputs are compressor power input, mass flow rate and discharge temperature. They combine the two aforementioned approaches. Generalizable equation sets are derived from physical principles of the described compressor type with parameters to adapt the model to a specific machine. These parameters are derived from reference data and thereby determine the character of the model. Only, if all parameters have physical significance, e.g. describe geometric properties or ratios, the model character is physics-based. And only physics-based models can be trusted to reproduce the behavior of a real machine when design changes are applied or when extrapolating.
Thus, the lack of detailed compressor information available to the system designer combined with the required criteria for the VTR necessitate that the compessor model is semi-empirical and physics-based. A scroll compressor model with optional vapor injection of that type is presented and analysed in this study.

The model is particularly inspired by two approaches from literature:
Winandy et al. (2002) and Winandy and Lebrun (2002) present a simple model with a two-step adiabatic compression and a thermal mass. This thermal mass exchanges heat with the suction and discharge gas as well as with the environment. All thermal power due to electro-mechanical loss is directly added to it. The two-step adiabatic compression is separated into an isentropic and an isochoric part. A simple injection model is presented in (Winandy & Lebrun, 2002).
Tello-Oquendo et al. (2019) propose a more complex model based on the same adiabatic compression submodel combined with multiple loss factors and heat transfer paths. The model is extended for vapor injection and heat exchange with the refrigerant takes into account the Nusselt correlation for turbulent tube flow from Dittus and Boelter (1985).

The semi-empirical compressor model presented in this work can be used for scroll compressors with or without vapor injection. It predicts the electric power input, the refrigerant mass flow and the discharge gas temperature. Further, the most influential subprocesses, e.g. motor and mechanical losses as well as internal heat transfer and heat transfer to the surroundings, are considered. There is a strong focus on physics-based equations so that the model produces wellfounded extrapolation predictions. The refrigerant is an integral part of the model and can be altered. And the motor cooling concept, namely suction- or discharge-gas cooling, is implemented so that an estimation for a change of design in this aspect is possible.
For validation of the presented model, publicly available data of a scroll compressor with optional vapor injection for use with refrigerants R410A and R32 is used.

2. SCROLL COMPRESSOR MODEL

This model is quasi-stationary. The compression process is divided into subprocesses. These are described by simplified physics-based principles, derived from thermodynamics. Fluid properties are implemented using CoolProp (Bell et al., 2014; Bell & Team, 2020). A refrigerant is chosen by selecting one of the available fluids in the library. The overall topology of the model concept is depicted in (Figure 1).

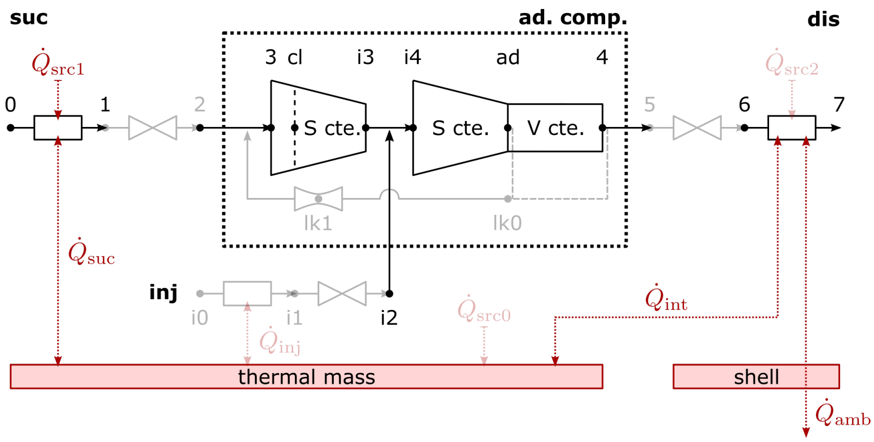


Figure 1: Compressor model (Greyed out parts are neglected in this paper.)

During compression, the refrigerant undergoes the following changes of state:

(0) Suction port condition

- **(0 - 1)** Isobaric heat transfer, suction plenum
- **(1 - 2)** Isenthalpic pressure loss at the scroll intake

(i0) Injection port condition

- **(i0 - i1)** Isobaric heat transfer, injection channel
- **(i1 - i2)** Isenthalpic pressure loss at the injection port

(2) Beginning of adiabatic compression

- **(2 - 3)** Isobaric mixing with the leakage
- **(3 - cl)** Supercharging effect before the closing of the scrolls
- **(3 - i3)** Isentropic compression to the injection pressure
- **(i3 - i4)** Isobaric mixing with the injected fluid
- **(i4 - ad)** Isentropic compression to the adapted pressure
- **(ad - 4)** Isochoric compression with over-/ undercompression

(5) End of adiabatic compression

- **(5 - 6)** Isenthalpic pressure loss at the scroll outlet
- **(6 - 7)** Isobaric heat transfer, discharge plenum

(7) Discharge port condition

(s) Isentropic lossless reference process

- **(0 - i3s)** Isentropic compression to the lossless injection pressure
- **(i3s - i4s)** Isobaric mixing with the lossless injected fluid
- **(i4s - 7s)** Isentropic compression

The influence of lubrication oil is neglected. In this paper leakage, pressure losses and heat transfer to the injection channel are also neglected: $\dot{m}_{lk} = 0$, $\Delta p_{12} = 0$, $\Delta p_{i0i1} = 0$, $\Delta p_{56} = 0$, $\dot{Q}_{inj} = 0$.

The scrolls have a fixed geometry with a built-in volume ratio.

$$\varepsilon_v = \frac{V_{cl}}{V_{ad}} \tag{1}$$

Nieter (1988) has identified a "supercharging effect" before the sealing of the scrolls. This is taken into account with the ratio between the geometric displacement with closed scrolls V_{cl} and the effective displacement V_3 at the beginning of the compression.

$$\varepsilon_{eff} = \frac{V_3}{V_{cl}} = \frac{\rho_{cl}}{\rho_3} \tag{2}$$

Hermetic scroll compressors are powered by an electric motor. According to Häberle et al. (2003), the rotational speed of the axle in relation to the synchronous speed $n_s = f/p$ of an electric motor is:

$$n = n_s(1 - s) \tag{3}$$

The motor has p pole pairs, the frequency after the inverter is f and s is the slip.
Electro-mechanical motor loss is modeled as a linear function of the electric power consumption P_{el} with the idle loss L_{mo0} and the constant motor efficiency η_{mo}. This approach is equivalent to the electro-mechanical loss model according to ASHRAE Toolkit (Bourdouxhe et al., 1994). Here, it is extended by the friction loss of the bearings and the scrolls with a constant friction moment M_{fr}.

$$L_{mo} = L_{mo0} + (1 - \eta_{mo})P_{el} \tag{4}$$

$$L_{fr} = 2\pi\, n\, M_{fr} \tag{5}$$

Mass flows can change between operating points and are constant at one. The sum of all mass flows suffices a steady-state balance without storage terms.
The mass flow at the scroll intake is the same at the effective beginning of the compression (3) and when the scrolls seal off (cl). It combines the suction mass flow $\dot{m}_{suc}$ and the leakage mass flow $\dot{m}_{lk}$. Scroll compressors are positive displacement machines. Thus, mass flow is a function of rotational speed of the axle n and the combination of matching displacement V and density ρ.

$$\dot{m}_3 = nV_3\rho_3 = nV_{cl}\rho_{cl} = \dot{m}_{cl} = \dot{m}_{suc} + \dot{m}_{lk} \tag{6}$$

When the injection port is unused, no injection takes place. When vapor injection is activated, the injection mass flow ratio $\dot{m}_{inj}/\dot{m}_{suc}$ is linearly related to the injection pressure ratio p_{i0}/p_0 (Tello-Oquendo et al., 2017).

$$\dot{m}_{inj} = \begin{cases} 0, & \text{no inj.} \\ \dot{m}_{suc}\left(A + B\,\frac{p_{i0}}{p_0}\right), & \text{vapor inj.} \end{cases} \tag{7}$$

At the adapted point, the compressor conveys the discharge mass flow $\dot{m}_{dis}$ and the leakage $\dot{m}_{lk}$. Both, the suction mass flow and the injected mass flow are combined at the discharge port.

$$\dot{m}_{ad} = \dot{m}_{dis} + \dot{m}_{lk} = (\dot{m}_{suc} + \dot{m}_{inj}) + \dot{m}_{lk} \tag{8}$$

Similar to the mass flows, a steady-state energy balance is always valid for the sum of electric power input P_{el}, adiabatic compression power P_{adc} and electro-mechanic losses L_{fr}, L_{mo}. Like for all positive displacement machines with a fixed geometry of the compression organ, a deviation from the design pressure ratio leads to a performance decrease. This is accounted for with a term for over- or undercompression as presented by Winandy and Lebrun (2002).

$$P_{el} - P_{adc} - L_{fr} - L_{mo} = 0 \tag{9}$$

$$P_{adc} = (\dot{m}_{suc} + \dot{m}_{lk})(h_{i3} - h_3) + (\dot{m}_{dis} + \dot{m}_{lk})(h_{ad} - h_{i4}) + P_{ou} \tag{10}$$

$$P_{ou} = nV_{ad}(p_4 - p_{ad}) \tag{11}$$

All electro-mechanical loss is converted into heat. In this model, the heat is fully added to the refrigerant in the suction plenum.

$$\dot{Q}_{src1} = L_{mo} + L_{fr} \tag{12}$$

This represents suction gas cooling. For discharge gas cooling it shall be added to the discharge plenum ($\dot{Q}_{src2}$).
Heat is exchanged between the discharge and the suction plenum via a thermal mass of uniform temperature T_{tm}. The heat exchange between the thermal mass and the refrigerant in the plena is modeled with a power-type Nusselt correlation (pt) derived from the boundary layer equations for single-phase flow (Baehr & Stephan, 2019). The flow is assumed turbulent. All fluid properties are calculated with the mean bulk temperature $T_m = (T_{fluid,in} + T_{fluid,out})/2$.

$$\dot{Q}_{suc} = A_{ht}h_{ht,suc}(T_{tm} - T_{m01}) = C_{ht,int}k_{m01}\left(\frac{\dot{m}_{suc}}{\mu_{m01}}\right)^{0.8} Pr_{m01}^{1/3}(T_{tm} - T_{m01}) \tag{13}$$

$$\dot{Q}_{int} = A_{ht} h_{ht,int} (T_{tm} - T_{m67}) = C_{ht,int} k_{m67} \left(\frac{\dot{m}_{dis}}{\mu_{m67}} \right)^{0.8} Pr_{m67}^{1/3} (T_{tm} - T_{m67}) \tag{14}$$

$$h_{ht} = \frac{k}{L_{ht}} Nu \tag{15}$$

$$Nu_{pt} = c_{Nu} Re^{0.8} Pr^{1/3} \tag{16}$$

The heat transfer area A_{ht} and characteristic length L_{ht} are estimated to be the same on the suction and discharge side. The average heat transfer coefficient h_{ht} is determined separately for both sides with the respective fluid properties and Nusselt correlation. Pr is the Prandtl number, k is the fluid's thermal conductivity, μ is the fluid's dynamic viscosity and Nu is the Nusselt number. The heat transfer coefficient $C_{ht,int}$ is a design parameter which summs up the Nusselt coefficient c_{Nu} and the geometric properties.

Goossens et al. (2017) find that "80 % of heat losses in scroll compressors occur from the lateral zones that are at the level of compression chamber, discharge plenum, and the top horizontal plate" at the examined operating point ($t_{cond} = 40\,°\mathrm{C}, t_{evap} = 0\,°\mathrm{C}, t_{amb} = 10\,°\mathrm{C}$). Thus, it is assumed here that all heat loss to the ambience (amb) occurs through heat exchange of the top third of the compressor shell with the ambient air. On the outside the shell is approximated as a cylinder (cyl) with a circular top plate (tp). It is of uniform temperature T_{sh}.

$$\dot{Q}_{amb} = \dot{Q}_{cyl} + \dot{Q}_{tp} \tag{17}$$

$$\dot{Q}_{cyl} = \pi D_{cyl} k_{film} Nu_{CC} (T_{air} - T_{sh}) \tag{18}$$

$$Nu_{CC} = \left(0.825 + \frac{0.387\, Ra^{1/6}}{\left[1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right]^{8/27}} \right)^2 \tag{19}$$

$$\dot{Q}_{tp} = \pi D_{cyl} k_{film} Nu_{McA} (T_{air} - T_{sh}) \tag{20}$$

$$Nu_{McA} = \begin{cases} 0.54\, Ra^{1/4}, & Ra < 2 * 10^7 \text{(laminar)} \\ 0.14\, Ra^{1/2}, & \text{else (turbulent)} \end{cases} \tag{21}$$

According to Baehr and Stephan (2019) the heat transfer through free convection from a vertical cylinder to the ambient air is calculated with the Nusselt correlation from Churchill and Chu (1975) (CC) for the vertical wall when $D_{cyl}/H_{cly} \geq 35 Gr^{-1/4}$. Gr is the Grashof number. The heat transfer through free convection from the top plate to the ambient air is determined with the Nusselt correlation for the horizontal plate according to McAdams (McA) (Incropera & DeWitt, 2002) with the characteristic length $L_{tp} = D_{cyl}/4$. D_{cyl} is the cylinder outer diameter and $Ra = Gr\,Pr$ is the Rayleigh number. All fluid properties are obtained at the film temperature $T_{film} = (T_{sh} + T_{air})/2$. T_{air} is the temperature of the ambient air. For the isobaric expansion coefficient, air is assumed as an ideal gas: $\alpha_p = 1/T_{air}$. On the refrigerant side, the heat transfer is calculated similar to equations (14 - 16).

$$\dot{Q}_{amb} = C_{ht,amb} k_{m67} \left(\frac{\dot{m}_{dis}}{\mu_{m67}} \right)^{0.8} Pr_{m67}^{1/3} (T_{sh} - T_{m67}) \tag{22}$$

3. EFFICIENCIES AND DESIGN PARAMETER DETERMINATION

Parameter determination in semi-empirical models is usually realized with optimization routines.The presented model and optimizer is implemented in python because python provides many libraries for data science and optimization. The optimization algorithms are from SciPy (The SciPy development team, 2021). The dual annealing method from SciPy.optimize (Tsallis, 1988; Tsallis & Stariolo, 1996; Xiang et al., 1997; Xiang & Gong, 2000) is found suited. This method is a multidimensional bound constrained global optimization routine. It combines a statistic simplistic annealing approach with a local optimizer. Here, the local optimizer is the bounded gradient based quasi-newton algorithm for nonlinear problems L-BFGS-B (Byrd et al., 1995; Zhu et al., 1997).

For the optimizer, comparing efficiencies instead of absolute values has two advantages: They are dimensionless and in the same size range. The volumetric efficiency η_v of the compressor is the suction mass flow normalized with the theoretic ideal suction mass flow $\dot{m}_{suc,id} = n_s V_{cl} \rho_0$. The overall compressor efficiency η_c is the ratio of the isentropic power consumption for compression and the electric power input. The proposed discharge temperature efficiency η_{T7} is a dimensionless temperature. It is defined as the isentropic discharge temperature T_{7s} divided by the discharge temperature T_7.

$$\eta_v = \frac{\dot{m}_{suc}}{\dot{m}_{suc,id}} = \frac{\dot{m}_{suc}}{n_s V_{cl} \rho_0} \tag{23}$$

$$\eta_c = \frac{\dot{m}_{suc}(h_{i3s} - h_0) + \dot{m}_{dis}(h_{7s} - h_{i4s})}{P_{el}} \tag{24}$$

$$\eta_{T7} = \frac{T_{7s}}{T_7} \tag{25}$$

In the model without vapor injection the unknown design parameter set is: $\{\varepsilon_{eff}, \varepsilon_v, M_{fr}, L_{mo0}, \eta_{mo}, C_{ht,int}, C_{ht,amb}\}$. The optimizer finds these by minimizing a cost function. Here, it is a combination of the differences between model and reference data in the three efficiencies presented in equations (23 - 25).

$$cost = \frac{1}{N} \sum_{i=1}^{N} \left(\Delta\eta_{v,i}^2 + \Delta\eta_{c,i}^2 + \Delta\eta_{T7,i}^2 \right) \tag{26}$$

The cost function is a mean squared error over all N data points. The cost at each data point i is the Euclidean norm of the three efficiency deltas.

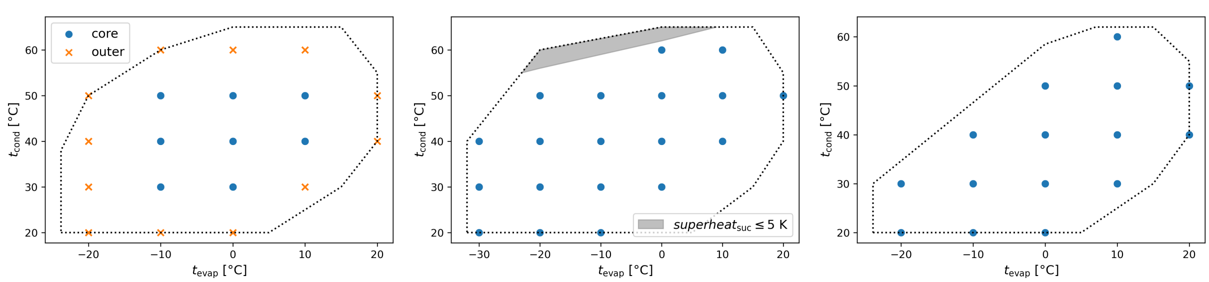
In order to preserve the deterministic physics-based character of the model, it is of utmost importance to find the parameter set of the unique global minimum of the cost function. In addition to the applied global optimization routine, this is ensured by multiple parallel and independent executions of the optimizer with randomized initiation. Only, if the same parameter set with the minimal cost is found by several executions, it is assumed the global optimum.

With vapor injection, $\{A, B\}$ from eq. (7) are additional design parameters. These are determined previous to the optimization routine via linear regression as proposed by Tello-Oquendo et al. (2017).

4. VALIDATION

This model is assessed with data of a hermetic scroll compressor with optional vapor injection, provided by the manufacturer via a software tool. Operating points are selected by superimposing an equidistant 10 K step grid over the compressor map. All points, where data is available in the manufacturer's tool, are taken into account. They are marked in the three test cases, which are presented (Figure 2):

(a) No injection with refrigerant R410A, 20 operating points (8 in core + 12 in outer)
(b) Vapor injection with refrigerant R410A, 19 operating points
(c) No injection with refrigerant R32, 15 operating points



(a) R410A, no inj. **(b)** R410A, vapor inj. **(c)** R32, no inj.

Figure 2: Compressor map and operating points of the scroll compressor

Four synchronous speeds are taken into account: $n_s = \{40, 50, 60, 70\}$ Hz. Figure (3) shows that the linear relation between $(\dot{m}_{inj}/\dot{m}_{suc})$ and (p_{i0}/p_0) proposed by Tello-Oquendo et al. (2017) is only justified at constant synchronous speed. Thus, for vapor injection only $n_s = 50$ Hz is considered.

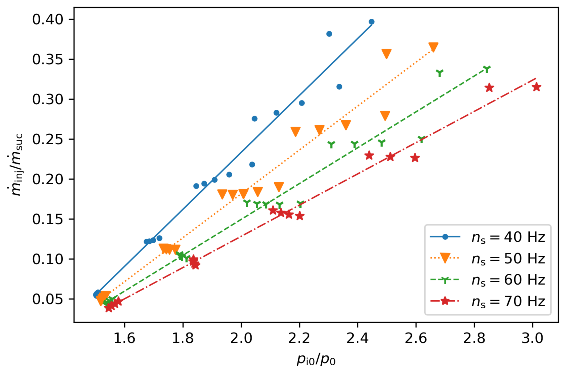


Figure 3: Injection mass flow ratio over injection pressure ratio with regression lines

The used scroll compressor supports variable speed operation and has a nominal synchronous speed $n_{s,nom} = 50$ Hz. It uses an induction motor with an estimated constant motor slip $s = 1/30$. Following AHRI 540 (AHRI, 2020), this compressor has a cooling capacity of approx. 69 kW (235 kBtu/h). The shell has an upper diameter $D_{cyl} = 0.239$ m and a total height $H_{sh} = 0.564$ m. Since the geometric displacement with closed scrolls V_{cl} is not provided, it is estimated from the volume flow at nominal synchronous speed. This has no influence on the model performance because the estimate $\tilde{V}_{cl}$ is used for nomalizing in eq. (23), both for the model and the reference data. Here, with $\dot{V}_{ref}(50\,\text{Hz}) = 37.6\,\text{m}^3/\text{h}$ equation (27) results in $\tilde{V}_{cl} = 216.09\,\text{cm}^3$. The superheating at the suction and at the injection port is 10 K and the ambient air temperature is 35 °C.

$$\tilde{V}_{cl} = \frac{\dot{V}_{ref}(n_{s,nom})}{n_{nom}} \tag{27}$$

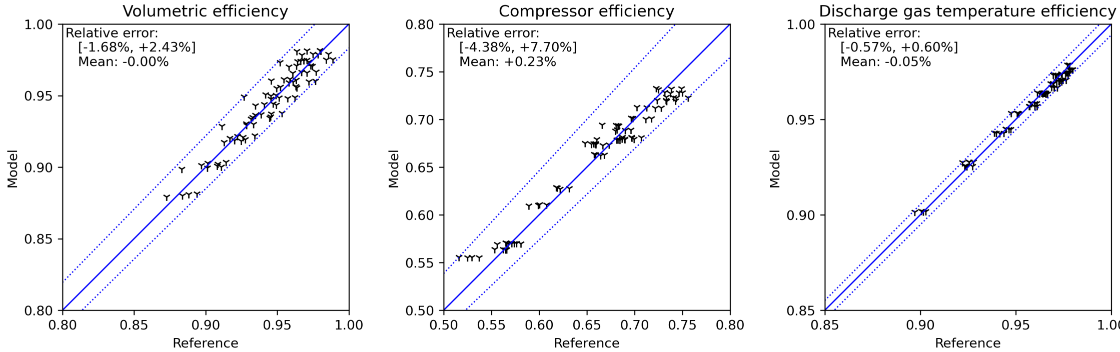
The two design parameter sets in (Table 1) are determined from test case (a). One only from the operation points marked as "core", the other from all. They are used for the validation tests in (Fig. 4). The core set is only applied to test the extrapolation capability of the model. For all other validation tests, the set derived from all operation points in (Fig. 2(a)) is used. With linear regression the design parameters for vapor injection are determined at $n_s = 50$ Hz with a correlation coefficient $R^2 = 98.57\,\%$ as $A = 0.368$ and $B = 0.274$.

All parameters in (Table 1) are in a physically reasonable size range. Only $C_{ht,amb}$ of the core set limits out close to the upper set bound. This is not a major issue because it will only limit the amount of heat loss to the environment. However, it is an indicator that the core set contains too few operating points to properly train the model. This issue is resolved when training the model with all data from test case (a).

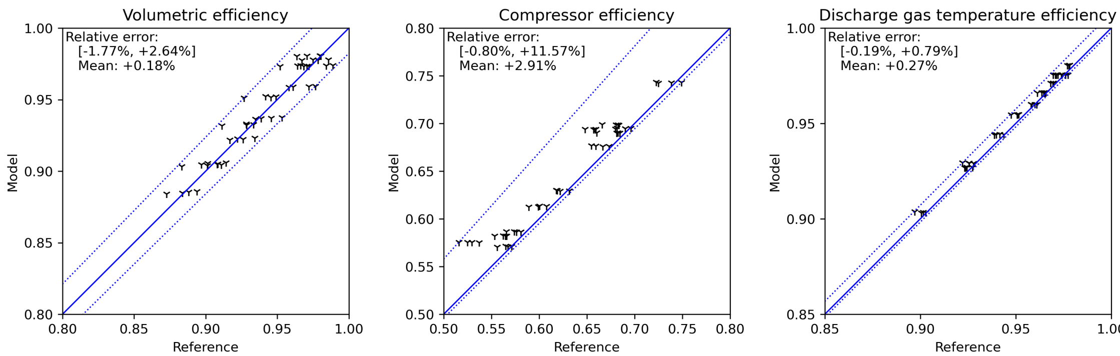
The quality of the model is assessed with the relative deviation of the model from reference data in the three efficiencies from eqs. (23 - 25). The results are presented in form of correlation plots in (Fig. 4).
(Fig. 4a) shows that the model reproduces its training data with a max. mean deviation of 0.23 %. Only few operating points with low overall compressor efficiency are notably overestimated.
(Fig. 4b) indicates that extrapolating from training data from the core of the compressor envelope to the outer field of it leads to overestimating the compressor and discharge temperature efficiencies. This is not surprising because losses

Table 1: Design parameter sets

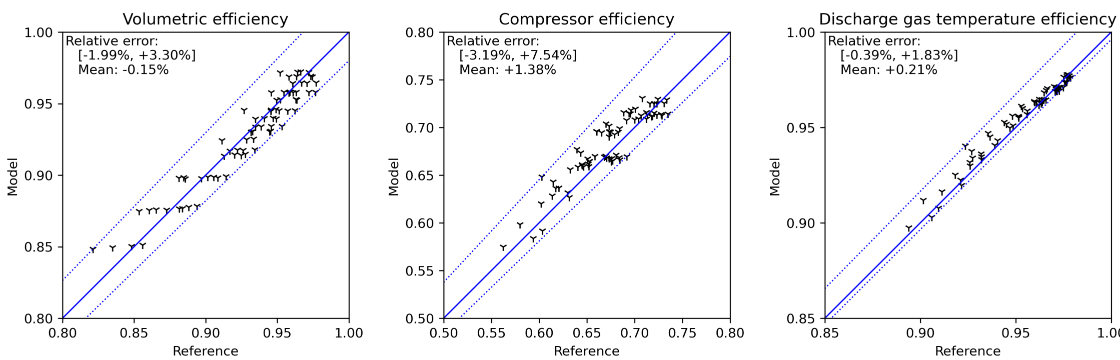
data	ε_{eff} [-]	ε_v [-]	M_{fr} [Nm]	L_{mo0} [W]	η_{mo} [-]	$C_{ht,int}$ [$m^{0.2}$]	$C_{ht,amb}$ [$m^{0.2}$]
R410A, no inj., core	1.049	2.871	1.770	0.000	0.827	0.296	4.814
R410A, no inj., all	1.053	2.956	1.548	0.000	0.816	0.402	0.021



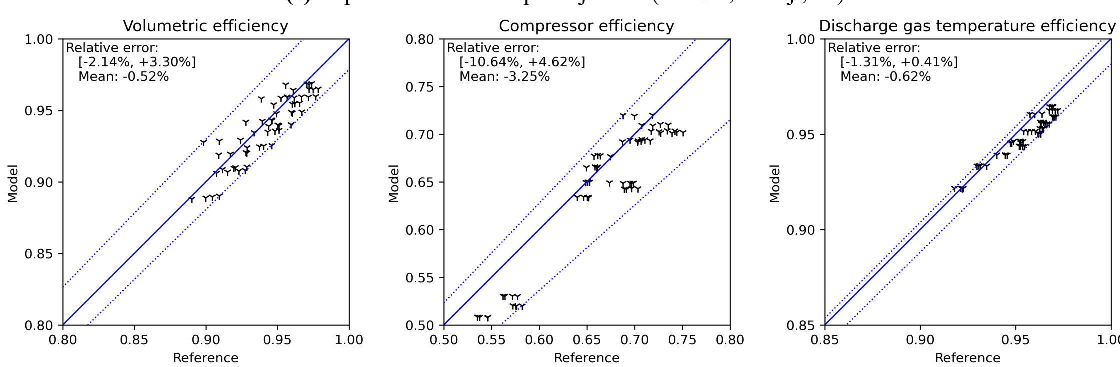
(a) Evaluation of the model fit (R410A, no inj., all)



(b) Extrapolation of core model to outer data (R410A, no inj., core)



(c) Implementation of vapor injection (R410A, no inj., all)



(d) Refrigerant switch from R410A to R32 (R410A, no inj., all)

Figure 4: Comparison between model and reference data (Underlying design parameter set in parentheses.)

are expected to be less dominant near the design point of the compressor. The mean deviation is still below 3 % for all three efficiencies. It remains an open question whether a denser distribution of test points in the core region of the compressor map leads to a better training result.
Implementing vapor injection (Fig. 4c) has a limited effect on model quality. The most distinct impact is an increased overestimation of the discharge temperature efficiency. Mean relative errors of all three efficiencies are smaller than 1.5 % magnitude.
(Fig. 4d) shows that a replacement of the design refrigerant is possible in this model. With a switch from R410A to R32, the model has a tendency to underestimate the compressor and discharge temperature efficiencies by up to −10.64 % and −1.31 %, respectively. Still, the mean deviation of all three efficiencies remains smaller than or equal to 3.25 %.

5. CONCLUSIONS

A quasi-stationary semi-empirical scroll compressor model is proposed and evaluated. It is physics-based and data enabled. Further, an optimization routine to determine the design parameter set of the model is presented.

- The proposed model shows small deviation when reproducing the data it is trained on.
- A capability to extrapolate from the training operation range is verified.
- The model is vapor injection ready.
- Refrigerant switch from R410A to R32 is possible.

In a future version, the model can be extended by the influences of leakage, pressure losses and heat transfer to the injection channel. A relation between injection mass flow ratio and injection pressure ratio considering the compressor speed would be an improvement. Further, it is of interest to evaluate the minimum amount and distribution of operating points, necessary for a proper training result.

NOMENCLATURE

Subscript

cond	condensation
dis	discharge
evap	evaporation
inj	injection
int	internal
lk	leakage
ref	reference
sh	shell
src	source
suc	suction
tm	thermal mass

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