International Conference on Applied Energy 2020
Dec. 1 - Dec. 10, 2020, Bangkok / Virtual
Paper ID: 0169

Validation of a Semi-Empiric Model for Twin-Screw Compressors with an Adaptable Internal Volume Ratio

Florian Kaufmann[1*], Christoph Wieland[1,3], Hartmut Spliethoff[1,2]

1 Institute for Energy Systems, Technical University of Munich, Boltzmannstraße 15, 85748 Garching, Germany

2 Bavarian Center for Applied Energy Research, Walther-Meißner-Str. 6, 85748 Garching, Germany

3 Munich School of Engineering, Lichtenbergstr. 4a, 85748 Garching, Germany

ABSTRACT

Twin-screw compressors with an adaptable built-in volume ratio can significantly increase the overall process efficiency for applications with fluctuating operating conditions. Detailed component models, necessary to assess their full potential, are rare in literature. Therefore, this work develops and validates a novel semi-empiric compressor model taking into account the mechanical adaptation of the built-in volume ratio. The model shows good agreement with operating data provided by the manufacturer.

Keywords: twin-screw compressor, semi-empiric model, variable built in volume ratio, model validation

NOMENCLATURE

Abbreviations	
BVR	Built-in volume ratio
Symbols	
A	Area, m²
a_{tl}	Torque loss coefficient, -
a, b, c	Polynomial coefficients, -
b_{hl}	Ambient heat transfer factor, $W{\cdot}K^{-5/4}$
c_p	Isobaric heat capacity, J/(kg·K)
h	Specific enthalpy, J/kg
$\dot{m}$	Mass flow, kg/s
n	Compressor speed, rpm
N	Number of data points, -
P	Pressure, Pa
P	Power, W
$\dot{Q}$	Heat flow, W
s	Specific entropy, J/(kg·K)
T	Temperature, K
UA	Overall heat transfer factor, W/K
v	Specific volume, m³/kg
V_s	Swept volume, m³
μ	Dynamic viscosity, Pa·s

1. INTRODUCTION

In a world with an ever-increasing energy demand, the development of more efficient processes and machines has become a major focus of industry and academia. This especially applies to refrigeration processes, which will become more and more important as global warming continues. Manufacturers of compressors for heat pump and refrigeration cycles have reacted to this necessity by developing more flexible compressor types that show high efficiencies over a wide range of operating conditions. Among the volumetric compression machines, which are commonly used in heat pump cycles [1], twin-screw compressors are often chosen for their robustness and wide range of operating conditions [2]. The efficiency of volumetric compression machines strongly depends on the applied pressure ratio and its match to the compressor's built-in volume ratio (BVR). Therefore, multiple manufacturers have developed twin-screw compressors, which can vary the BVR by means of a sliding piston that changes the geometry of the compression chamber. Figure 1 compares the catalog efficiency of a compressor with a variable BVR (Bitzer CSW10593) to the catalog efficiency of a compressor with a fixed BVR (Bitzer CSW95103). The compressor with fixed BVR is smaller and shows a generally lower efficiency. However, the relevant distinction of the compressor with variable BVR is its ability to reach high efficiencies over a significantly wider range of applied pressure ratios compared to the fixed-BVR compressor.

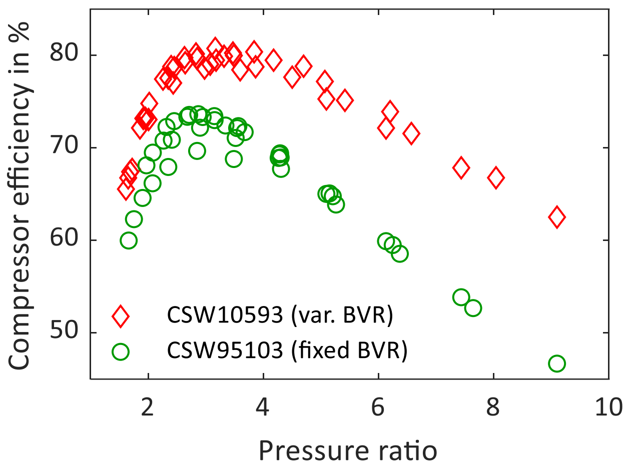

Figure 1: Comparison of twin-screw compressors with (CSW10593) and without (CSW95103) variable BVR

The assessment of processes utilizing variable-BVR machines requires reliable models, which are also valid in the part-load region. However, existing modeling approaches for twin-screw machines are unable to reproduce the effects of a varying BVR or require extensive knowledge of the machine geometry [3], which is often not available. This work extends an existing semi-empiric twin-screw compressor model from Giuffrida [4] in order to account for a varying BVR. The model is validated for a specific compressor using publicly available operational data provided by the manufacturer.

2. TWIN-SCREW COMPRESSOR MODEL

2.1 Base model from the literature

The new model is based on a semi-empiric model for twin-screw compressors from Giuffrida [4] (subsequently referred to as "base model"), which was developed from a previous model for Scroll-compressors by Winandy et al. [5]. The base model is depicted in Figure 2 and takes into account the most relevant characteristics of a twin-screw compressor. This includes mechanical losses in the compressor as well as internal leakage, heat losses to the environment and heat transfer between the working fluid and the compressor housing. In order to account for the effect of over- and under-compression related to the BVR, the base model applies an isentropic compression (3⟶4) followed by an isochoric compression (4⟶5). In a first step the leakage mass flow $\dot{m}_{leak}$ is determined by assuming a single leakage path connecting exhaust and inlet stream.

The leakage path is modeled as an isentropic flow through a convergent nozzle with the cross-sectional area A_{leak}.

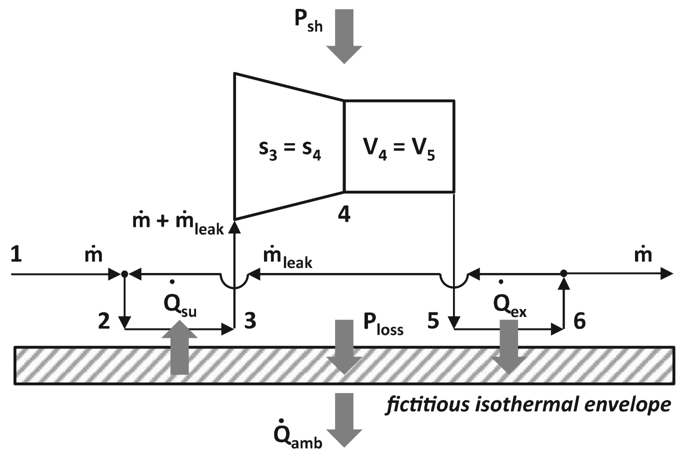

Figure 2: Base model developed by Giuffrida [4]

The resulting mass flow can be determined by

$$\dot{m}_{leak} = \frac{A_{leak}}{v_{leak}} \cdot \sqrt{2 \cdot (h_6 - h_{leak})}\ . \quad (1)$$

For more detailed information about the calculation of the specific enthalpy h_{leak} and specific volume v_{leak} in the nozzle please refer to Giuffrida [4]. Equation (1) allows the determination of the swept mass flow $\dot{m}_s$ by combining the suction mass flow $\dot{m}$ and the leakage mass flow:

$$\dot{m}_s = \dot{m}_{leak} + \dot{m} \quad (2)$$

The adiabatic mixing of both flows is followed by heat exchange with the compressor housing, which is modeled as a fictitious isothermal envelope of the temperature T_w. The heat flow to the fluid $\dot{Q}_{su}$ (with the index su referring to the suction side) can be described in dependence of the fluid's isobaric heat capacity c_p by the equation

$$\dot{Q}_{su} = \dot{m}_s \cdot c_{p,2} \cdot (T_w - T_2) \cdot \left[1 - e^{\frac{-UA_{su}}{\dot{m}_s \cdot c_{p,2}}}\right]. \quad (3)$$

The flow-regime dependency of the lumped heat transfer factor UA_{su} is accounted for by assessing the relation between $\dot{m}_s$ and a nominal mass flow $\dot{m}_{nom}$ for which the model parameter $UA_{su,nom}$ is fitted:

$$UA_{su} = UA_{su,nom} \cdot \left(\frac{\dot{m}_s}{\dot{m}_{nom}}\right)^{0.8} \quad (4)$$

Knowing the fluid state after suction warming (i.e the specific volume v_3), the compressor speed n can be calculated with the swept compressor volume V_s:

$$n = \frac{v_3 \cdot \dot{m}_s}{V_s} \tag{5}$$

The total internal power P_{int} required for both continuous compression steps is described by

$$P_{int} = \dot{m}_s \cdot \left((h_4 - h_3) + v_4 \cdot (p_5 - p_4)\right) \ . \tag{6}$$

The state variables after the isentropic compression (i.e. the specific enthalpy h_4 and the pressure p_4) are calculated from the specific inlet entropy s_3 and the specific volume v_4, which is in turn calculated from the inlet state by the equation

$$v_4 = \frac{v_3}{BVR} \ . \tag{7}$$

The first contributor to the mechanical losses $P_{loss,1}$ is modeled as a constant fraction of the internal power using the dimensionless model parameter $a_{tl,1}$:

$$P_{loss,1} = a_{tl,1} \cdot P_{int} \tag{8}$$

A second mechanical loss $P_{loss,2}$, which is depending on the compressor speed n, can be calculated by the equation

$$P_{loss,2} = a_{tl,2} \cdot V_s \cdot \mu \cdot \left(\frac{\pi \cdot n}{30}\right)^2 , \tag{9}$$

including the dynamic viscosity of the lubrication oil μ as well as the dimensionless model parameter $a_{tl,2}$. Using the equations (6), (8) and (9) allows the calculation of the total required shaft power P_{sh} as

$$P_{sh} = P_{int} + P_{loss,1} + P_{loss,2} \ . \tag{10}$$

The heat transfer $\dot{Q}_{ex}$ (with the index ex referring to the exhaust side) between fluid and compressor housing after the compression steps is modeled equivalent to the suction warming in equation (3). However, the model parameter $UA_{ex,nom}$ is used in equation (4) to determine the corresponding heat transfer coefficient. In addition, the temperature difference in equation (3) is (T_5-T_w) assuming heat flow from the fluid to the compressor housing.

In order to calculate a heat balance around the compressor housing, also the heat loss to the surrounding $\dot{Q}_{amb}$ has to be considered:

$$\dot{Q}_{amb} = b_{hl} \cdot (T_w - T_{amb})^{\frac{5}{4}} \tag{11}$$

It depends on the ambient temperature T_{amb} as well as the heat transfer factor to the surrounding b_{hl}. The exponent in equation (11) is due to its derivation from natural convection phenomena.

For more information please refer to Giuffrida [4]. For the model to be solved, the energy balances around the compressor housing and around the whole compressor have to be satisfied.

The first is defined as

$$P_{loss,1} + P_{loss,2} - \dot{Q}_{amb} + \dot{Q}_{ex} - \dot{Q}_{su} = 0 \ , \tag{12}$$

while the latter can be written as

$$P_{sh} - \dot{Q}_{amb} + \dot{m} \cdot (h_6 - h_1) = 0 \ . \tag{13}$$

It is worth mentioning, that the mechanical losses are completely transferred to the compressor housing as friction heat. Since the thermal envelope temperature and the outlet state are not known a priori, they have to be determined iteratively by numerical minimization of the residuals from equations (12) and (13).

According to the previous section, the base model is fully defined by the eight parameters A_{leak}, b_{hl}, $UA_{su,nom}$, $UA_{ex,nom}$, $a_{tl,1}$, $a_{tl,2}$, V_s and BVR. For information about each parameter's unit, please refer to the nomenclature section. The swept volume V_s and the built-in volume ratio BVR are constant parameters defined by construction and hence do not need to be identified in the fitting procedure. For further details on the base model please refer to Giuffrida [4].

2.2 Model adaption for variable BVR

The base model described in section 2.1 is extended by the subsequently described equations in order to account for the variable BVR. Since they are not influenced by the mechanical variation of the BVR, the calculation of the heat flows $\dot{Q}_{su}$, $\dot{Q}_{ex}$ and $\dot{Q}_{amb}$ is performed according to the base model. Also the swept volume V_s stays constant during BVR variation. The speed dependent torque losses $P_{loss,2}$ contribute to the total losses very little and are therefore regarded as independent from the BVR in order to reduce the model complexity. The remaining model parameters change together with the BVR. The BVR can be adapted in the allowed range between BVR_{min} and BVR_{max} whose values depend on the compressor type. In a first step, the model estimates the ideal BVR (BVR_{id}, which is used as the input for further calculations) depending on the applied pressure ratio.

$$BVR_{id} = \frac{v_1}{v_{id}} \tag{14}$$

In this equation, v_1 and v_{id} are the specific volumes of the working fluid at the compressor inlet and after an isentropic compression to the specified outlet pressure.

If the value for the BVR_{id} exceeds the allowed range it is set to the nearest permitted value. After the BVR_{id} is identified, the depending parameters are calculated by polynomial correlations of the form:

$$A_{leak} = a_A \cdot BVR_{id}^{\,b_A} + c_A \tag{15}$$

$$a_{tl,1} = a_a \cdot BVR_{id}^{\,b_a} + c_a \tag{16}$$

The parameters a, b and c are fitted to the compressor data. For pressure ratios resulting in BVR_{id} values outside the allowed range, the values for A_{leak} and $a_{tl,1}$ are calculated using BVR_{min} and BVR_{max} respectively. The polynomial correlation of the given form is chosen since preliminary comparisons to other correlation types have proven it delivers the best results.
In addition to the above model parameters, the temperature, pressure and mass flow of the inlet stream as well as the outlet pressure are used as input values. The model output consists of the mechanical shaft power, the compressor speed, the ambient heat losses and the outlet stream temperature. While similar models use the compressor speed as an input instead of the mass flow, the approach in this work is more convenient for the subsequent use in process simulations.

3. METHODOLOGY AND DATA SELECTION

The compressor model is validated using catalog data for the semi-hermetic twin-screw compressor CSW10593 provided by the manufacturer Bitzer [6]. The compressor control system automatically adapts the BVR in a range from 1.7 to 3.5 to achieve the maximum compressor efficiency at a constant swept volume of 11.495 L. The working fluid used in the compressor is the refrigerant R134a.
In order to provide a good database for the fitting and validation procedure, eighty data points are generated with a software tool, which is provided by the manufacturer based on experimental data. For given evaporation and condensation temperatures (corresponding to the inlet and outlet pressure) and a specified compressor speed, the software calculates the necessary shaft power as well as the gas outlet temperature and the mass flow through the compressor. Forty data points are created for compressor speeds of 2900 rpm and 3500 rpm each, with an inlet superheating of 10 °C. Figure 3 shows the corresponding temperatures used in the data set. As displayed in Figure 3, the data points are distributed over the recommended range of operation indicated by the black pentagon.

The data points used for validation (red diamonds) are selected in a way to cover a wide range of operating conditions. During the fitting procedure, the error function given in equation (14) is minimized numerically.

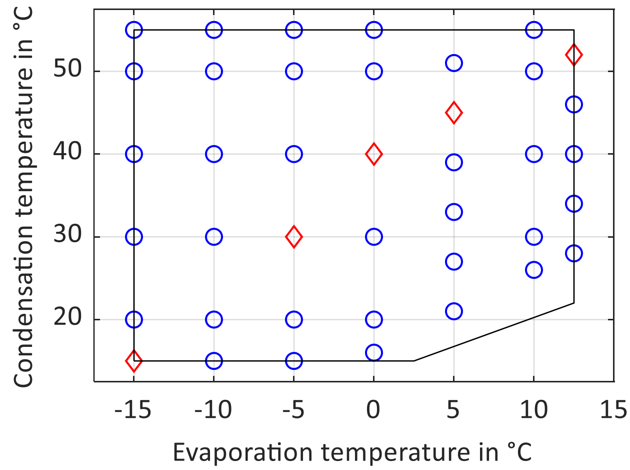

Figure 3: Catalog data used for fitting (blue circles) and validation (red diamonds)

In this function n, P and T_6 are the compressor speed, shaft power and outlet temperature respectively while the subscripts sim and cat indicate the simulated and catalog values. N is the total number of data points used in the fitting procedure.

$$err = \frac{1}{3} \cdot \left(\sqrt{\frac{1}{N} \sum \left(\frac{n_{sim} - n_{cat}}{n_{cat}} \right)^2} + \sqrt{\frac{1}{N} \sum \left(\frac{P_{sim} - P_{cat}}{P_{cat}} \right)^2} + \sqrt{\frac{1}{N} \sum \left(\frac{T_{6,sim} - T_{6,cat}}{T_{6,cat}} \right)^2} \right) \tag{14}$$

The model implementation and solving is conducted in *Matlab 2018b* [7] with the fluid properties being calculated from *Refprop 10* [8].

4. VALIDATION AND DISCUSSION

After fitting the model to the data set described in section 3, the model parameters given in Table 1 are obtained. For the corresponding unit of each parameter, please refer to the nomenclature section.

Table 1: Model parameters of a Bitzer CSW10593 compressor

b_{hl}	8.908	a_A	4.390-10	a_a	11.137
$UA_{su,nom}$	41.995	b_A	14.158	b_a	-7.299
$UA_{ex,nom}$	39.602	c_A	0.6287	c_a	0.224
$a_{tl,2}$	122.80				

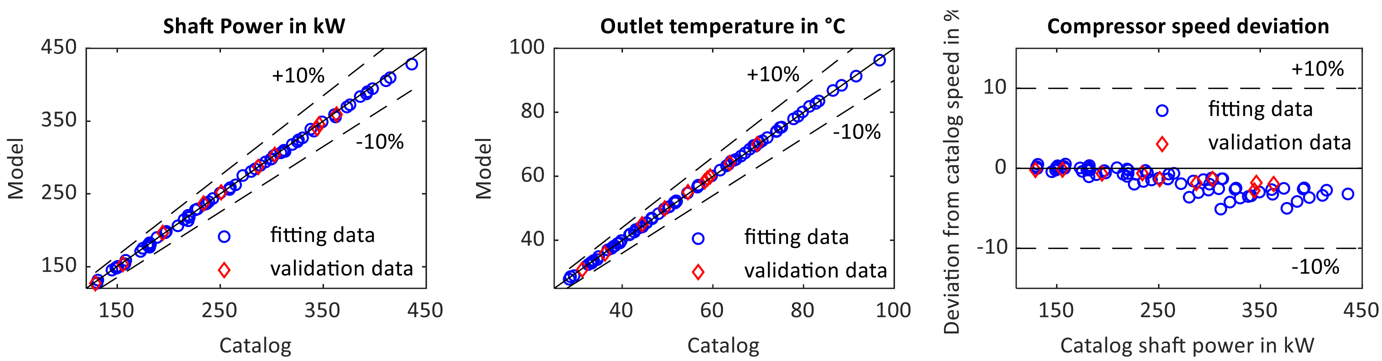

Figure 4: Model validation with catalog data provided by the manufacturer

The results of the compressor fitting and validation are displayed in Figure 4, which shows parity plots for the shaft power and the compressor outlet temperature as well as the deviation of the compressor speed for different catalog shaft powers. Since only two values for the compressor speed are used for the fitting procedure, the compressor speed deviation cannot be reasonably displayed in a parity plot. Hence, the catalog shaft power is used instead of the catalog speed to display the compressor speed deviation. An ideal model would have all data points lying on the continuous black line indicating zero deviation from the catalog data. The model generally shows very good agreement with the catalog data with the mean model deviation from the catalog data being 0.83 %, 0.52 % and 1.50 % for the shaft power, the outlet temperature and the compressor speed respectively. This can also be concluded from the dashed lines in Figure 3 indicating a model deviation of ±10 % from the catalog data. The validation data points in Figure 4 lead to the conclusion that the new model also predicts operating points not included in the fitting procedure with very high accuracy.

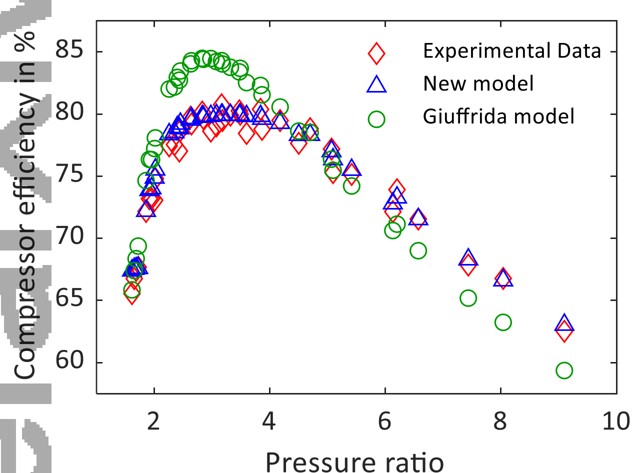

Figure 5: Model comparison with catalog data for a Bitzer CSW10593 compressor

When the model proposed by Giuffrida and the new model are fitted to catalog data for a compressor model with variable BVR, the superiority of the novel model becomes apparent. Figure 5 compares the catalog compressor efficiency to the efficiencies calculated by the new model and the Giuffrida model. Both model versions were fitted to the catalog data described in section 3 before. To avoid biased results, the BVR was considered a fitting parameter for the Giuffrida model, resulting in a BVR of 2.81. It can be observed, that the new model aligns with the catalog data very well since it can account for the variable BVR. Using a fixed BVR, the Giuffrida model naturally exhibits a slimmer peak for the efficiency since the effects of over- and under-expansion manifest for smaller deviations from the BVR. As displayed in Figure 5, this leads to flawed predictions when the model is fitted to experimental data of a compressor with variable BVR.

5. CONCLUSION AND OUTLOOK

This work validates an improved semi-empiric twin-screw compressor model for compressors with a variable built-in volume ratio using catalog operational data provided by a manufacturer. The new model shows significantly higher accuracy than existing model of a similar complexity. In the future, the proposed modeling approach could also be applied to twin-screw expanders. Moreover, the model will be integrated in process simulations of heat pumps in order to investigate the potential benefits on process efficiency and flexibility.

ACKNOWLEDGEMENT

Funding from the Bavarian State Ministry of Education, Science and the Arts in the framework of the project Geothermal-Alliance Bavaria is gratefully acknowledged.

REFERENCE

[1] Arpagaus, C., Bless, F., Uhlmann, M., Schiffmann, J., and Bertsch, S. S. 2018. High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy, 152, 985-1010. *Energy* 152, 985–1010.

[2] Liu, J., Li, Q., Wang, F., and Zhou, L. 2012. A new model of screw compressor for refrigeration system simulation. *International Journal of Refrigeration* 35, 4, 861–870.

[3] Bianchi, G., Marchionni, M., Miller, J., and Tassou, S. A. 2020. Modelling and off-design performance optimisation of a trilateral flash cycle system using two-phase twin-screw expanders with variable built-in volume ratio. *Applied Thermal Engineering* 179, 115671.

[4] Giuffrida, A. 2016. A semi-empirical method for assessing the performance of an open-drive screw refrigeration compressor. *Applied Thermal Engineering* 93, 813–823.

[5] Winandy, E., O, C. S., and Lebrun, J. 2002. Experimental analysis and simplified modelling of a hermetic scroll refrigeration compressor. *Applied Thermal Engineering* 22, 2, 107–120.

[6] https://www.bitzer.de/websoftware/, accessed 10. September 2020

[7] MATLAB 2018b, The MathWorks Inc., Natick, Massachusetts, United States.

[8] Lemmon, E.W., Bell, I.H., Huber, M.L., McLinden, M.O. , NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, 2018.