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Compressor models under pressure: modeling impact on refrigerant performance in vapor-injection heat pumps

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Abstract

In cold climate regions with ambient temperatures down to −20 °C, air-to-water heat pumps (AWHP) using single-stage standard cycles face technical limitations. Low source temperatures reduce evaporation pressure and suction gas density, decreasing refrigerant mass flow and heating capacity. Simultaneously, high pressure ratios lead to low isentropic efficiencies and elevated discharge temperatures, restricting compressor operation. Both effects reduce system efficiency, expressed by the coefficient of performance (COP), and increase reliance on auxiliary heating, raising operational costs. To address these limitations, quasi-two-stage vapor injection is identified in literature as a promising enhancement. However, the effectiveness of such cycle modifications depends on accurate compressor modeling. This study investigates how the choice of compressor model affects refrigerant performance predictions in vapor-injection cycles. Three compressor models are compared in a quasi-steady-state simulation framework with thermodynamic properties from REFPROP. One assumes constant isentropic and volumetric efficiencies, while furthermore an empirical and semi-physical model is used. The vapor injection cycle is benchmarked against a simple cycle across source temperatures from −20 °C to 0 °C and a sink temperature of 75 °C. Key performance indicators are COP, heating capacity, and compressor discharge temperature. Refrigerants considered are R290, R1270, and R1243zf. Results show that refrigerant ranking changes with the compressor model. For example, R290 outperforms R1243zf in the semi-empirical model but not in the constant-efficiency case. Different compressor models lead to deviations within the vapor-injection cycle up to 15.9 % for COP, 5.4 % for heating capacity and 11.4 K in discharge temperatures. The findings show that refrigerant selection depends on compressor modeling assumptions. Since performance rankings of refrigerants shift depending on the used model, a more detailed refrigerant-dependent model is required. Experimental validation on a dedicated vapor injection test rig will follow to quantify these interactions under real conditions.

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Keywords: Cold Climate; COP; Heating Capacity ; Natural Refrigerants; Efficiency

1. Introduction

Decarbonizing the heating sector is essential for achieving climate targets. Heat pumps enable this transition when operated with renewable electricity [1]. They extract heat from a low-temperature source, such as ambient air, and deliver it at a higher temperature level to the heating system by compressing the refrigerant. As a result, they can supply several times the amount of heat relative to the electrical energy required for compression. This efficiency is one of the reasons heat pumps are already common in new residential buildings in Germany [2].

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New buildings represent a small share of the overall building stock. Most existing buildings in Germany are still heated by condensing boilers. A central barrier to heat pump deployment in these buildings is the requirement for higher supply temperatures. Radiators provide less heat transfer surface than, for example, underfloor heating systems, which necessitates higher supply temperatures to meet the required heating load. Consequently, a heat pump for such applications must achieve a large temperature lift between the ambient source temperature and supply temperature, potentially up to 75 °C [3]. This work investigates concepts for implementing heat pumps capable of operating at high temperature lifts.

In the literature, the simple cycle is the basic reference process from a thermodynamic perspective. It comprises four main components of a heat pump: the compressor and the expansion valve, to establish the pressure levels, and the condenser and evaporator, to transfer heat from the ambient and to the building's heating system. Several studies have identified the compressor as the key component for evaluating heat pump performance [4], [5]. Simulations of heat pump processes demonstrate a high sensitivity of the results to the choice of compressor model.

A common approach in previous studies to represent the compressor in simulations is the assumption of constant compressor efficiencies [6]. Subsequent studies have addressed the lack of operating-point dependence under the constant assumption. For instance, Neumaier et al. observed changes in the efficiency ranking of refrigerants in brine-source residential heat pumps (low temperature lifts) when varying operating conditions with a semi-physical fluid-dependent compressor model [4]. Other studies derived empirical correlations, often in polynomial form, to express compressor efficiency as a function of operating point. For example, Xu et al. and Moreno-Rodríguez et al. formulate efficiencies as functions of the pressure ratio [2], [7]. The mathematical correlation is matched with experimental data, thus, including the behavior of specific refrigerants. A limitation of these empirical correlations is their poor extrapolation beyond the validity range and for different refrigerants. Most previous studies focused on moderate temperature lifts, limiting the applicability of these correlations to high temperature lift scenarios [4], [5], [7]. Similarly, Höges reported that considerable effort was required to compile and adjust suitable correlations in order to achieve reasonable agreement for his operating conditions, further illustrating the lack of robust models for broader applicability [8]. This work therefore evaluates the transferability of existing correlations to heat pumps operating under high temperature lift conditions.

In addition to models assuming constant compressor efficiencies and purely empirical correlations, semi-physical and fluid-dependent models have been developed in the literature. These models fit parameters that relate to physical behavior to empirical data. This approach allows within certain limits predictions beyond the range of the available data, making semi-physical models superior to purely empirical models when parameter fitting is accurate [9]. Semi-physical models, such as those proposed by Molinaroli et al. for rolling piston compressors [10], have primarily been developed for applications with moderate temperature lifts. Their applicability to high temperature lift scenarios is not discussed in the literature.

Neumaier et al. compared refrigerants such as R290, R1270 and R1243zf in simple cycle heat pumps for operating point B5W35 using two compressor models for a conventional piston compressor [4]. The first model assumes constant compressor efficiencies, while the second is a semi-physical reciprocating compressor model from Roskosch et al. [11]. The efficiency ranking of refrigerants differs between the two models [4]. Therefore, this work investigates whether similar behaviour occurs at high temperature lifts using three compressor models considering rolling piston compressors and investigating further refrigerant cycle configurations compared to the standard cycle.

In cold climate regions with ambient temperatures down to −20 °C, air-to-water heat pumps (AWHP) using single-stage standard cycles face technical limitations [3]. Low source temperatures reduce evaporation pressure and suction gas density, decreasing refrigerant mass flow and heating capacity ($\dot{Q}_{\mathrm{con}}$). Simultaneously, high pressure ratios lead to low isentropic efficiencies and elevated discharge temperatures ($T_{\mathrm{discharge}}$), restricting compressor operation. Both effects reduce system efficiency, expressed by the coefficient of performance (COP), and increase reliance on auxiliary heating, raising operational costs [3]. To mitigate losses in efficiency (COP), heating capacity ($\dot{Q}_{\mathrm{con}}$), and operational limits ($T_{\mathrm{discharge}}$) caused by decreasing ambient temperatures, modifications to the refrigerant cycle (also referred to as cycle configuration) have been discussed in the literature [12]. For applications involving high temperature lifts in cold climates, vapor injection cycles represent a promising solution. Ostlender et al. and Höcker et al. have already investigated this application case using R290 and evaluating different modifications to the cycle configuration. However, both studies relied on constant compressor efficiencies and a mass-specific assessment approach, limiting the transferability of their findings to more detailed component-level analyses [13], [14].

In heat pump processes with vapor injection, an intermediate pressure level is used to inject gaseous refrigerant into the compressor. Depending on the operating point, the injection can increase condenser capacity and the COP. The injection also provides internal cooling of the compression process, resulting in lower compressor discharge temperatures. For these reasons, vapor injection cycles are considered highly suitable for applications with high temperature lifts. Particularly the vapor injection flash tank cycle is considered with high potential in the literature [12]. Previous studies assessing vapor injection cycles under cold-climate conditions, e.g. Ostlender et al. and Höcker et al., relied on constant compressor efficiencies. Since compressor efficiencies vary significantly near the limits of the operating envelope, this assumption limits the ability to accurately quantify the performance potential of vapor injection cycles [13], [14].

In summary, the literature provides several approaches to improve heat pump operation under cold-climate conditions and achieve high temperature lifts, including modified cycle configurations and different compressor modelling strategies. However, most studies either rely on constant compressor efficiencies or on empirical correlations validated only for moderate temperature lifts, leaving their applicability to high-lift scenarios uncertain. This gap motivates the present work, which investigates how different compressor models influence the predicted performance of high-lift heat pumps and whether conclusions from moderate-lift studies remain valid. Section 2 introduces the compressor models, the considered refrigerants and cycle configurations, and the operating conditions of the application scenario. Section 3 presents and discusses the results, first comparing cycle configurations for R290 and then extending the analysis to multiple refrigerants. The main findings are summarized in the conclusion.

2. Method

The preceding literature review shows that compressor modelling has a significant impact on the refrigerant efficiency ranking for moderate temperature lifts. To quantify this impact for high temperature lifts, this work compares three different compressor modelling approaches. The comparison is based on the work conducted by Höges et al. [8]. The first modelling approach assumes constant isentropic efficiency (η_{is}) and volumetric efficiency (η_{vol}), with values as commonly used in the literature [8]:

$$\eta_{\mathrm{is},1} = 0.7 \tag{1}$$

$$\eta_{\mathrm{vol},1} = 0.9 \tag{2}$$

The second compressor modelling approach introduces a dependency of η_{is} and η_{vol} on the pressure ratio Π. For this purpose, a correlation for η_{is} proposed by Xu et al. and a correlation for η_{vol} proposed by Moreno-Rodríguez et al. are applied [2], [7]. Both correlations are derived from empirical data. The correlation from Xu et al. stems from a CO_2 cold climate heat pump application with an operation point of A-20W65 [2]. Moreno-Rodríguez et al. derived the correlation from a R134a heat pump supplying domestic hot water with evaporating temperatures between -10 °C and 20 °C and condensation temperatures of up to 55 °C [7].

$$\eta_{\mathrm{is},2} = \frac{35.23 \cdot \Pi^3 - 108.5 \cdot \Pi^2 + 141.2 \cdot \Pi - 70.85}{\Pi^4 + 54.71 \cdot \Pi^3 - 189.9 \cdot \Pi^2 + 263.7 \cdot \Pi - 132.5} \tag{3}$$

$$\eta_{\mathrm{vol},2} = 0.9 - 0.035 \cdot \Pi \tag{4}$$

The third compressor modelling approach is a semi-physical model based on the formulation by Molinaroli et al. [10]. This model simulates the compressor behaviour through a thermodynamic equivalent path composed of sequential sub-processes. The refrigerant undergoes isobaric heating at the suction side (modelled via a fictitious isothermal wall), followed by isobaric mixing with internal leakage and re-expansion flows. These leakage flows are modelled physically as isentropic flows through convergent nozzles. Subsequently, the model calculates the isentropic compression and accounts for pressure losses at the discharge valve before the final heat transfer at the discharge side. Unlike simple efficiency maps, this method predicts the mass flow rate and power consumption using physically parameters identified from reference data [8], [10].

The fitted Parameters in this work result from a R290 rolling piston compressor with an operating envelope ranging for the evaporation temperature between -25 °C and 15 °C and condensation temperatures between

30°C and 60 °C [10]. Based on the calculated mass flow and power consumption, the resulting η_{is} and η_{vol} were derived:

$$\eta_{\mathrm{is,3}} = \frac{P_{\mathrm{com,is}}}{P_{\mathrm{com}}} = \frac{\dot{W}_{\mathrm{in}} - \dot{W}_{\mathrm{loss}}}{\dot{W}_{\mathrm{in}}} \tag{5}$$

$$\eta_{\mathrm{vol,3}} = \frac{\dot{m}}{V_{\mathrm{h}} \cdot n \cdot \rho_1} \tag{6}$$

For visualisation purposes the model outputs η_{is} and η_{vol} for R1243zf are plotted over Π and the evaporation pressure p_{eva} in Figure 1.

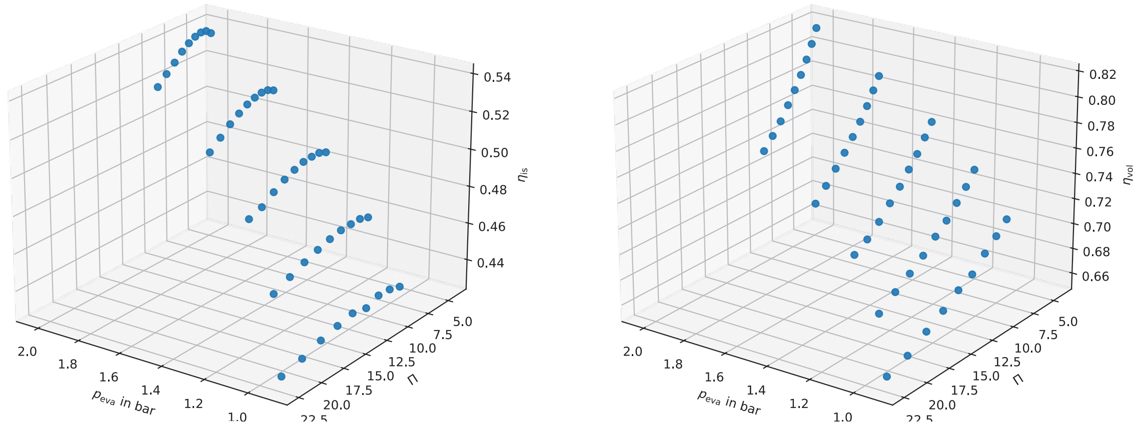

Figure 1: Model outputs η_{is} and η_{vol} from the semi-physical model of Molinaroli et al. over the pressure ratio and the evaporation pressure.

The literature review highlighted that evaluating compressor models requires considering the operating point conditions and refrigerants [8]. For the model comparison in this work, R1270, R290, and R1243zf are analysed. R290 is already widely used in the building sector and, as a natural refrigerant, is considered compliant with current and future regulatory restrictions [5]. R1270 exhibits thermodynamic properties similar to R290. The main difference lies in the isentropic exponent, which is higher for R1270 than for R290, leading to higher discharge temperatures for R1270. R1243zf is considered as a synthetic hydrofluoroolefin refrigerant containing three fluorine atoms per molecule. Currently, no regulatory restrictions apply to R1243zf. However, uncertainties remain regarding PFAS-related discussions, such as those addressed in the REACH regulation [8]. This refrigerant is also included in the investigations by Neumaier et al., allowing a comparison of results [4]. Since vapor injection cycles are considered suitable for cold climate applications, the VI-FT is examined in this work. To evaluate the performance of the VI-FT it is compared to the simple cycle.

The choice of refrigerant, in combination with the selected design point and design capacity, plays a decisive role in determining the heat pump dimensioning. In this work, a 10 kW AWHP heat is examined at the design point A-10W75. The design is based on a refrigerant-specific cycle optimization using the pinch-point method. Particular attention is given to the compressor displacement volume (V_{h}), which, in positive displacement compressors, has a major influence on the heat pump capacity. Consequently, the choice of compressor model affects the heat pump design, resulting in different geometric dimensions for the system. Table 1 shows the dependence of V_{h} on the compressor model and the refrigerant for the simple cycle and the VI-FT. The differences in V_{h} between the models illustrate the influence of η_{vol} in the design point. Following the system design, the operation of the heat pump can be simulated. In this work, the VcLibPy library is used for the operational simulation [15].

Table 1: Influence of the compressor model on the dimensioning of the displacement volume.

V_{h} in cm^3	Simple Cycle			Vapor Injection Flash Tank		
	R1270	**R290**	**R1243zf**	**R1270**	**R290**	**R1243zf**
Constant	37.20	46.62	79.30	28.45	34.14	59.97
Empirical	49.55	63.46	128.00	31.35	37.53	55.50
Semi-physical	40.60	49.41	82.59	27.51	31.62	48.38

The VcLibPy follows a map-based approach. At the system level, information on heat pump states and the heat pump itself is provided to the calculation algorithm. The algorithm yields the corresponding steady-state cycle configuration states. Different cycle configurations are available at the system topology level. Component-level models aggregate the cycle configurations. This approach enables a comparison of different refrigerants, cycle types, and compressor models. In summary, in this work two cycle configurations and four refrigerants are examined with three different compressor models. For the calculation of fluid properties, the REFPROP database from NIST is used [16].

Recent literature proposes advanced semi-empirical and physics-inspired formulations to improve predictive accuracy for vapor-injected compressors, such as the dimensionless mapping approaches by Lumpkin et al. [17] or the characterization methodologies by Khan and Bradshaw [18]. However, these data-driven frameworks require specific calibration coefficients, which are currently not available in the literature for the considered natural refrigerants at high-temperature lifts. Therefore, this study utilizes the established constant, empirical, and semi-physical approaches to fundamentally quantify the model-induced uncertainty in heat pump dimensioning before applying such advanced calibration methods.

For the application case in this study, AWHP are considered in cold climate regions. Among heat pumps AWHP are the most practical solution for retrofitting existing buildings [1]. The focus is on existing buildings that require high supply temperatures of up to 75 °C. To ensure comparable condensing pressures across all models, the sink outlet temperature was fixed at 75 °C with a constant mass flow. Consequently, the sink inlet temperature serves as the dependent variable, adjusting to the resulting heating capacity. In this study, cold climate refers specifically to source temperatures between −20 °C and 0 °C. This application case is similar to previous studies from Ostlender et al. and Höcker et al. [13], [14].

3. Results & Discussion

The analysis compares the cycle configurations and compressor models for the single refrigerant R290. The comparison focuses on $\dot{Q}_{\mathrm{con}}$, the COP, and $T_{\mathrm{discharge}}$. Figure 2 shows $\dot{Q}_{\mathrm{con}}$ as a function of the inlet source temperature $T_{\mathrm{source,in}}$ at a constant sink outlet temperature $T_{\mathrm{sink,out}}$ of 75 °C. The three compressor models are shown side by using different marker saturations to distinguish their respective results.

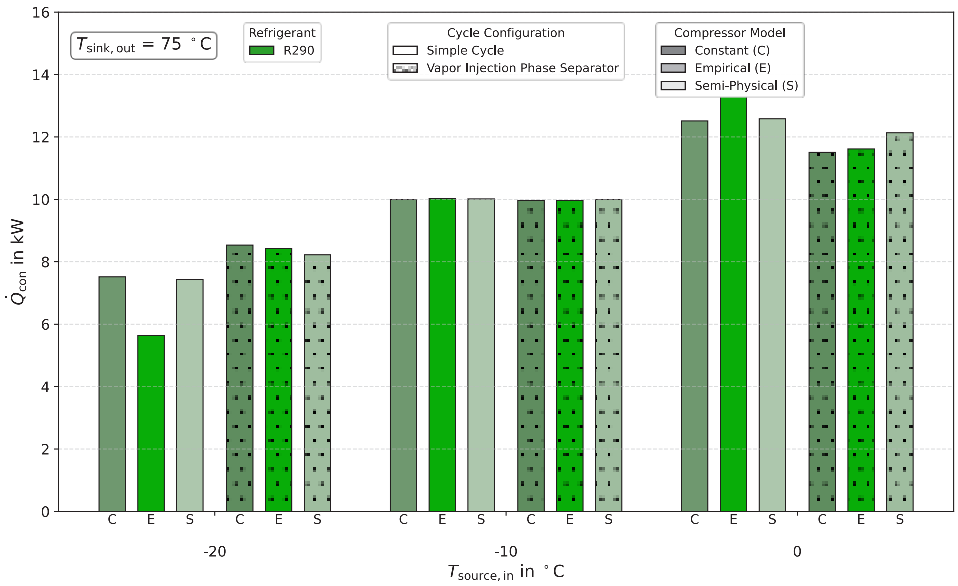

Figure 2: Heating capacity over source inlet temperature depending on cycle configuration and compressor model for R290.

The results show that all systems are designed to provide 10 kW at the design point A−10W75. The simple cycle exhibits stronger deviations from the design capacity when operating under off-design conditions. This behavior is caused by operating-point-dependent density variations at the compressor inlet, which affect the simple cycle more strongly than the vapor-injection cycle. In the VI-FT configuration, part of the mass flow enters the compressor at an elevated pressure level, reducing sensitivity to these density fluctuations.

At low source temperatures, the VI-FT cycle provides higher heating capacities than the simple cycle for all compressor models. This observation is consistent with findings in the literature, which identify vapor-injection cycles as better suited for cold-climate operation than single-stage standard cycles [1], [3], [12], [19]. Regarding the influence of the compressor models, the results show that the volumetric efficiency exhibits a particularly strong impact on system performance. The empirical correlation by Moreno-Rodríguez et al. predicts the lowest η_{vol} across all operating points. For the simple cycle, the deviation of the empirical model from the other models increases (at A-20W75 25 % to the constant and 24.1 % to the semi-physical model) as source temperature decreases, leading to a higher reduction in $\dot{Q}_{\mathrm{con}}$ at $T_{\mathrm{source,in}} = -20\ °C$.

Because all systems are sized to 10 kW at A−10W75, the higher deviation at low ambient temperatures is a consequence of the reduced volumetric efficiency. Conversely, at higher $T_{\mathrm{source,in}} = 0\ °C$, the differences in η_{vol} between the empirical model and the constant model approach each other. As a result, the larger displacement volume, selected during design and depending on the refrigerant choice and cycle configuration, becomes more dominant for the empirical model at these operating points, allowing a higher refrigerant mass flow and, thus, increased $\dot{Q}_{\mathrm{con}}$ compared with the other compressor modelling approaches.

While the heating capacity primarily reflects how the compressor model and the cycle topology influence the mass flow rate at off-design conditions, the overall system performance must additionally be assessed in terms of efficiency. Therefore, the COP is evaluated regarding the compressor models and cycle configurations. Analogous to figure 2, figure 3 shows the simulation results of the COP for R290.

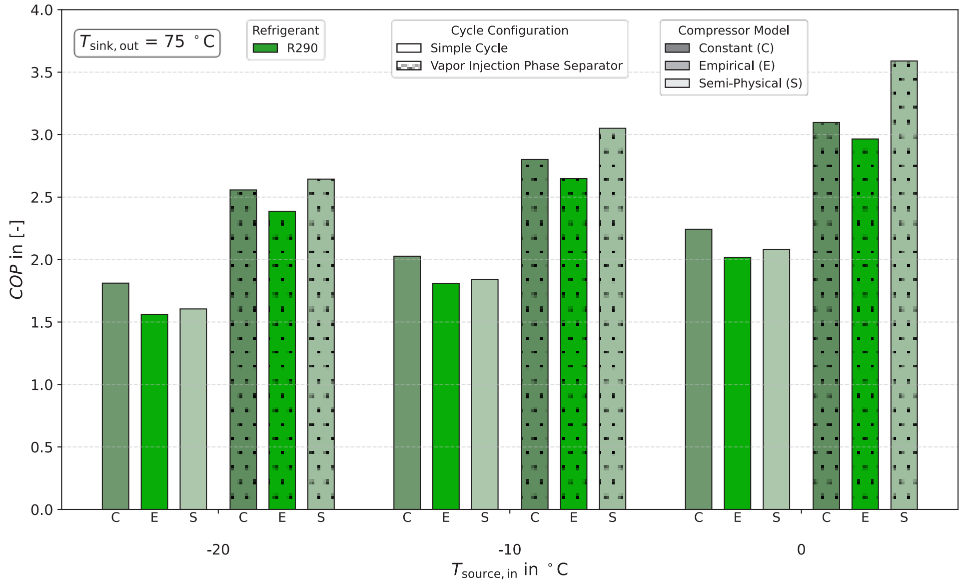

Figure 3: COP over source inlet temperature depending on cycle configuration and compressor model for R290.

All cycle configurations exhibit decreasing COPs with decreasing source temperatures, which can be explained by the behaviour of the corresponding Carnot cycle. The simple cycle shows particularly low COPs below $T_{\mathrm{source,in}} = 0\ °C$, ranging between 1.5 and 2.3 depending on the compressor model. The VI-FT configuration achieves higher efficiencies at all operating points, with COPs ranging from 2.4 to 3.6, again depending on the chosen compressor model. The impact of compressor models on COP is primarily determined by η_{is}. The empirical model predicts the lowest η_{is} in the considered operating range, resulting in the lowest COP. For the simple cycle, assuming a constant η_{is} of 0.7 is not reasonable at low source temperatures, since the deviation from the other two models increases.

The semi-physical model shows differences depending on the cycle configuration. For the simple cycle, the COP tends to be lower, like the empirical model, whereas for the VI-FT configuration, it can exceed the constant-efficiency compressor model. At higher source temperatures, COP increases more strongly with the semi-physical model than with the other models. Notably, assuming η_{is} as 0.7 does not consistently lead to higher COP estimates for the VI-FT cycle, whereas it does for the simple cycle. These findings indicate that comparing cycle configurations based on a constant compressor efficiency can lead to misleading conclusions. Therefore, the literature should provide more research towards choice of refrigerants and cycle configurations especially at high temperature lifts.

In addition to $\dot{Q}_{\text{con}}$ and the COP, $T_{\text{discharge}}$ is a critical performance parameter. $T_{\text{discharge}}$ directly affects the operational limits of the compressor and the suitability of the heat pump for high temperature lift applications. The following analysis examines how the different cycle configurations and compressor models influence $T_{\text{discharge}}$ across the considered source temperatures. Figure 4 shows the simulation results of $T_{\text{discharge}}$ for R290, comparing the cycle configurations and compressor models over $T_{\text{source,in}}$ at constant $T_{\text{sink,out}}$.

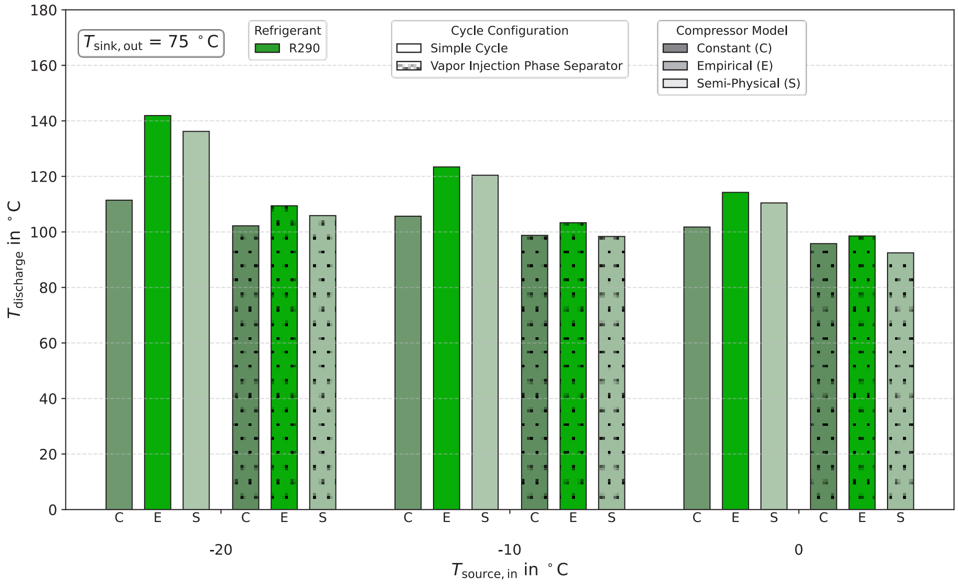

Figure 4: Discharge temperature over source inlet temperature depending on cycle configuration and compressor model for R290.

The VI-FT cycle reaches lower $T_{\text{discharge}}$ than the simple cycle. This improvement is attributed to the intermediate cooling provided by the injection process. These findings are consistent with the literature [12].

The intermediate cooling contributes to a lower $T_{\text{discharge}}$. This way, the compression process operates closer to the isentropic state change of the entire compression process. For the simple cycle, assuming a constant isentropic efficiency of 0.7 leads to significant deviations from the other models, as reflected in $T_{\text{discharge}}$. The difference in $T_{\text{discharge}}$ is up to 30.5 K between the constant model and the empirical model for the simple cycle at A-20W75. For the empirical model, it is again apparent that the isentropic efficiency is estimated lowest, which corresponds to higher discharge temperatures of R290.

To prove the findings also for other refrigerants, the refrigerants R1270 and R1243zf are investigated for both cycle configurations and all compressor models. This approach allows assessing how the choice of compressor model influences the predicted performance of each refrigerant. It also addresses the influence of the refrigerant ranking regarding different compressor models.

$\dot{Q}_{con}$ is shown in Figure 5 depending on $T_{source,in}$ for the three refrigerants, both cycle configurations and the three compressor models.

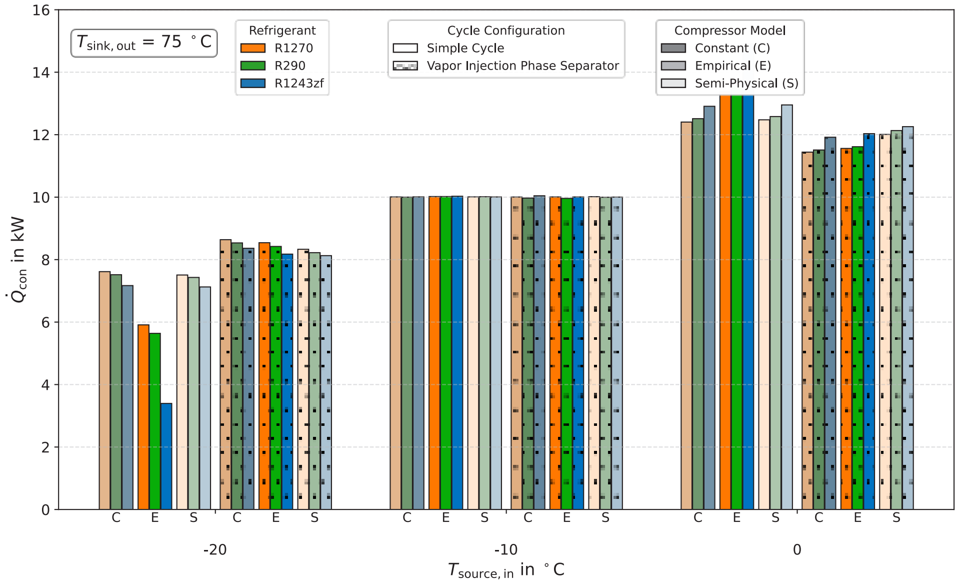

Figure 5: Heating capacity over source inlet temperature depending on cycle configuration, compressor model and refrigerant.

Regardless of the compressor model, the refrigerants exhibit different levels of variation in $\dot{Q}_{con}$ under off-design conditions. R1270 shows the smallest deviation from the design capacity of 10 kW, followed by R290. R1243zf exhibits the largest deviations in $\dot{Q}_{con}$ from the design point in the simple cycle. Within the VI-FT cycle, the largest deviation between compressor models is at A0W75 for R290 between the constant and semi-physical model with 5.4 %. Within the simple cycle the largest deviation between compressor models is at A-20W75 for R1243zf between the constant and empirical model with 52.6 %.

Using the empirical compressor model, it becomes evident that variations in $\dot{Q}_{con}$ are higher for the simple cycle than for the VI-FT configuration. This is reflected in the compressor displacement volumes (V_h) shown in Table 1. The sizing of V_h is strongly influenced by η_{vol} at the design point conditions.

In the empirical model, which depends on the pressure ratio, volumetric efficiencies vary more strongly than in the other two models. The pressure ratio for R1243zf at the design point A-10W75 is higher than for R290 and R1270, resulting in a lower volumetric efficiency for R1243zf. Consequently, a smaller fraction of V_h contributes effectively to the mass flow rate. To achieve the required mass flow and thus provide 10 kW of $\dot{Q}_{con}$ at the design point, V_h is dimensioned larger for R1243zf. Due to the low volumetric efficiency, the compressor for R1243zf is more than twice the size of the compressors for R290 and R1270 (see table 1).

For the considered source temperatures, the pressure ratio of R1243zf changes more strongly than for R290 and R1270. E.g. from A-10W75 to A-20W75 the pressure ratio of R1243zf changes for the constant model from 12.8 to 19, while for R290 it changes from 9.7 to 13.8 and for R1270 from 9.2 to 13.1. In the model comparison the pressure ratio varies no further than 0.8 for all fluids for the considered operation points. Consequently, due to the higher differences in the pressure ratio, η_{vol} also varies more for R1243zf, which alters the effectively utilized compressor displacement volume and the resulting mass flow rate. Therefore, deviations in $\dot{Q}_{con}$ are observed for R1243zf in the empirical model compared with R290 and R1270. In this regard, the applicability of the empirical model for the present application is not recommended. The differing predictions depending on the refrigerant and compressor model introduce a significant degree of uncertainty in the results and, thus, in the decision process.

Despite the smaller V_h, the VI-FT exhibits less degradation in $\dot{Q}_{con}$ at source temperatures below the design point. This indicates that a compressor may be sized smaller while maintaining robustness against capacity reduction at lower source temperatures (neglecting effects on the *COP*). One reason for this behaviour is the split mass flow, which is recombined within the compressor. The suction gas at low pressure and low density is supplemented by the injected mass flow at higher pressure and higher density. As a result, a share of the mass flow is drawn at the low density corresponding to the low evaporation pressure (lower expansion valve losses). Consequently, a smaller V_h may be required to achieve the same mass flow compared with single-stage cycle configurations. The magnitude of the described effects is observed by numerical models and are subject to the deviations between the different compressor models. The robustness of these results should be validated against experimental data.

To consider the efficiency, Figure 6 depicts the COP for the refrigerant comparison.

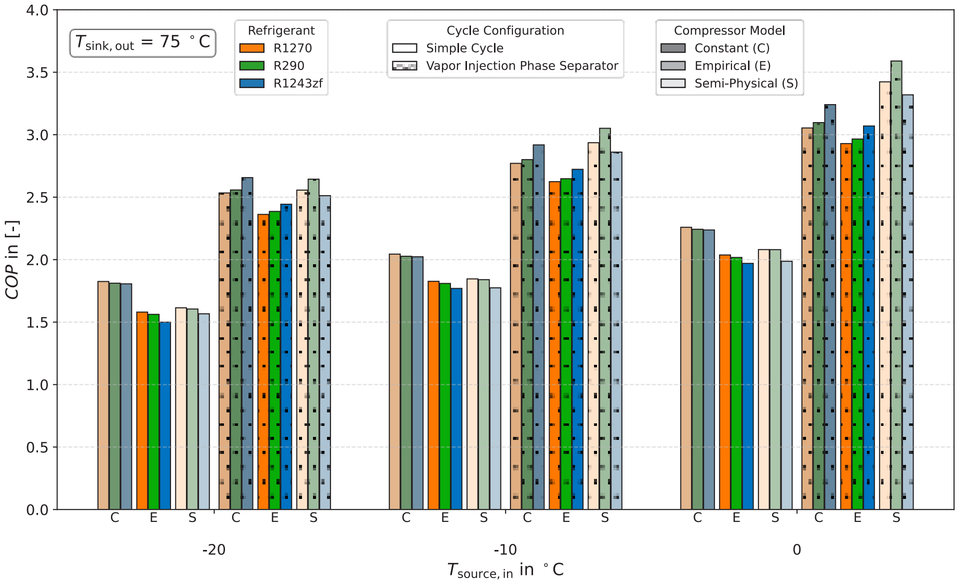

Figure 6: COP over source inlet temperature depending on cycle configuration, compressor model and refrigerant.

The largest efficiency differences are observed between the cycle configurations, followed by smaller differences between refrigerants. In the extreme case at the operating point A-20W75 for R290, the COP difference between cycle configurations is 41.2 %, when the constant model is used. Using the empirical model the difference between the cycle configurations is 52.8 % and 64.7 % with the semi-physical model (all relative to the simple cycle).

Within the VI-FT the largest deviation between compressor models is at A0W75 for R290 between the empirical and semi-physical model with 21.8 %. Within the simple cycle the largest deviation between compressor models is at A-20W75 for R1243zf between the constant and empirical model with 17 %. Furthermore, it is evident that the influence of the compressor model on COP for the VI-FT increases at higher source temperatures. This is illustrated by the relative deviation in COP between the empirical and semi-physical models for R290, which rises from 10.8 % at the operating point A-20W75 to 21.8 % at A0W75, both values expressed relative to the empirical model. The relative difference between the constant and empirical models remains approximately constant across the considered source temperatures. For the simple cycle, no significant changes in the COP ratios are observed with respect to the compressor model.

For the simple cycle, R1243zf exhibits the lowest COP regardless of the compressor model. In the VI-FT configuration, however, Figure 6 shows a change in the refrigerant ranking depending on the compressor model, consistent with observations by Neumaier et al. [4]. While the constant and empirical models predict the highest COPs for R1243zf, the semi-physical model estimates the lowest COPs for the considered operating points. This demonstrates when comparing refrigerants and cycle configurations, the choice of compressor model can significantly influence the refrigerant ranking and, thus, the choice of refrigerant.

Furthermore, in the VI-FT configuration, the COP differences between R290 and R1270 are larger when using the semi-physical model. At the operating point A0W75, the relative differences are 4.6 % for the semi-physical model, 1.4 % for the constant model, and 1.2 % for the empirical correlation, all expressed relative to the value for R290.

Figure 7 depicts how the compressor models and refrigerants affect $T_{\mathrm{discharge}}$ across the different cycle configurations.

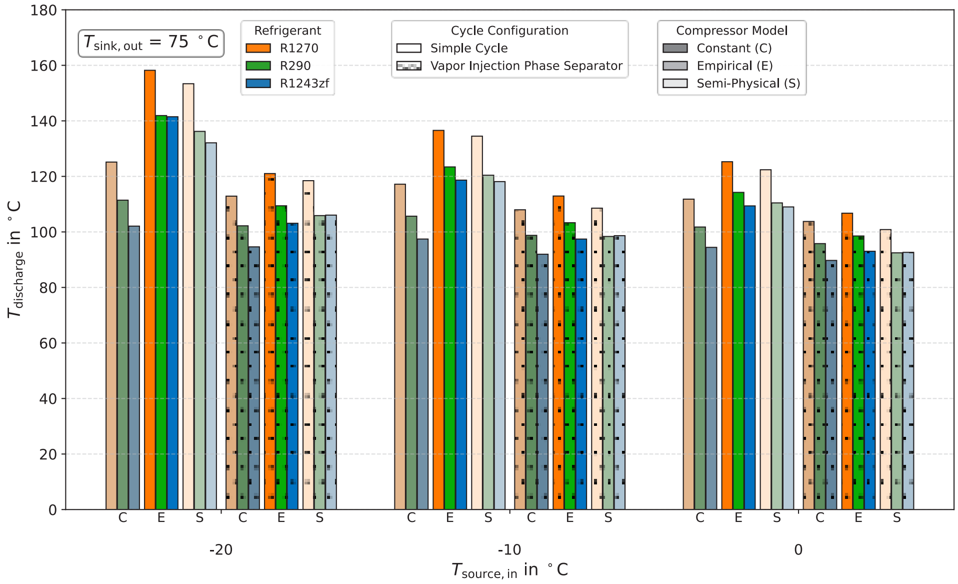

Figure 7: Discharge temperature over source inlet temperature depending on cycle configuration, compressor model and refrigerant.

It is evident that R1270 exhibits the highest $T_{\mathrm{discharge}}$ among the refrigerants, which is consistent with literature reports [8]. For all refrigerants and cycle configurations, $T_{\mathrm{discharge}}$ increases as the source temperature decreases. This behavior is consistent with literature and can be attributed to the isentropic efficiency. A lower isentropic efficiency causes the compression process to terminate at higher entropies (due to the definition of isentropic efficiency) and, thus, at higher temperatures, for the same discharge pressure. The rate at which $T_{\mathrm{discharge}}$ rises with decreasing source temperature depends on the compressor model, as different models predict different isentropic efficiencies. This highlights the uncertainties in the results that arise from the choice of compressor model regardless the refrigerant.

Analogous to the COP, higher $T_{\mathrm{discharge}}$ values are observed for the simple cycle when using the empirical and semi-physical models. This again indicates that assuming a constant isentropic efficiency of 0.7 for single-stage cycle configurations may be inaccurate for this application. For all compressor models, the VI-FT exhibits a reduction in $T_{\mathrm{discharge}}$ compared with the simple cycle, consistent with literature [12]. This reduction is attributed to the intermediate cooling of the compression process provided by the injection.

The evaluation of the $T_{\mathrm{discharge}}$ reduction is subject to considerable uncertainty, as substantial quantitative differences occur depending on the compressor model. At the operating point A-20W75 for R290, the difference in $T_{\mathrm{discharge}}$ between the simple cycle and the VI-FT cycle is 12.3 K for the constant model, 37.2 K for the empirical model, and 31 K for the semi-physical model. Within the VI-FT the largest deviation between compressor models is at A-20W75 for R1243zf between the constant and semi-physical model with 11.4 K. Within the simple cycle the largest deviation compressor models is observed at A-20W75 for R1243zf between the constant and the empirical model with 39.4 K. Qualitative differences are also apparent between results obtained with different compressor models. In the VI-FT, using the constant and empirical models, R1243zf exhibits the lowest $T_{\mathrm{discharge}}$. When the semi-physical model is used, however, R290 shows the lowest $T_{\mathrm{discharge}}$.

The qualitative and substantial quantitative differences indicate that standard assumptions are insufficient for designing high-lift heat pumps with natural refrigerants. For the considered heat pump application of high temperature lifts, no compressor models with confirmed validity are known to the authors. The combination of the correlations from Xu et al. and Moreno-Rodríguez et al. may also lead to additional deviations that remain unknown. The combination was taken from Höges and done to avoid physically unplausible results for the compressor efficiencies due to the smaller temperature lifts in the previous studies [8]. Therefore, for all studies addressing high temperature lifts, it is essential to clarify which model has been used. While previously mentioned studies mainly focused on low-temperature applications, high temperature-lift heat pumps are subject to the same effects. Variations in suction density due to changes in the evaporation pressure must also be accounted for, and appropriate compressor modelling is equally critical.

Accurate comparisons of cycle configurations in terms of efficiency, capacity, and maximum process temperature require valid computational models. Model validation, in turn, relies on experimental data. These data must subsequently be incorporated into prediction models to enable the development of improved models. Even if the absolute values of efficiency or capacity are uncertain, maintaining a consistent ranking of cycle configurations or refrigerants across operating conditions would already provide valuable guidance for system design and selection. While advanced characterization methodologies like those proposed by Khan and Bradshaw [18] or dimensionless mappings by Lumpkin et al. [17] offer robust frameworks to minimize errors, they rely on calibrated coefficients that are currently unavailable for R290 at these extreme operating conditions.

Consequently, the identified discrepancies underscore the necessity of the experimental investigation to generate the reference data required to parameterize such high-fidelity models. The Institute for Energy Efficient Buildings and Indoor Climate is constructing a test rig that allows a direct comparison of different cycle configurations under experimental conditions. Future experimental data will support the assessment of compressor model validity. If existing models are found to be unsuitable, the new test rig data can serve as a basis for the further development of these models. Such validation is important not only for low-temperature applications but also for high-temperature heat pumps, as the same physical effects (such as variations in suction density and efficiency at large temperature lifts) impact compressor performance in both cases.

4. Conclusion

In summary, the present study shows that compressor models predict markedly different values for both isentropic and volumetric efficiencies when applied to high temperature-lift heat pump applications. These differences directly affect the predicted heat pump performance indicators, including efficiency, heating capacity, and maximum process temperature for three refrigerants (R290, R1270 and R1243zf) and two cycle configurations (simple and vapor injection with flash tank). This finding reinforces the relevance of the compressor (model) as the key component governing overall heat pump behavior, as highlighted in previous work [4], [5].

Different compressor models (constant, empirical and semi-physical) lead to deviations within the VI-FT and refrigerant up to 15.9 % for COP, 5.4 % for heating capacity and 11.4 K in discharge temperatures. The quantitative and qualitative deviations between the models demonstrate that current assessments of high temperature-lift processes remain uncertain and even might yield different decisions when it comes to realization of cycle configurations.

Under these conditions, the limitations of extrapolating empirical correlations become evident, as they can lead to inconsistent or unreliable predictions. While advanced semi-empirical frameworks theoretically offer the potential to minimize errors, they rely on calibrated coefficients which are currently unavailable for natural refrigerants in high-lift applications. Consequently, experimental data are required to generate the necessary reference data to parameterize such high-fidelity models. For this purpose, a dedicated test rig is currently being developed at the Institute for Energy Efficient Buildings and Indoor Climate. Future experimental results will support the evaluation of compressor model validity. If existing models are found to be unsuitable, the new data will provide a basis for improving current modelling approaches. This validation is crucial for developing reliable predictive tools not only for the high temperature-lift applications investigated but also for the expanding field of high-temperature heat pumps. Both areas are subject to the same underlying physical effects (such as variations in suction density and efficiency drops at large pressure ratios) making validated compressor models equally critical for their successful design and implementation.

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