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A Compressor Model to Determine Cooling Cycle Efficiencies and Performances Without or With Internal Heat Exchanger for Specific Refrigerant

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Keywords: cooling system compressor model, refrigeration cycle EER, internal heat exchanger

ABSTRACT

Today, the importance of refrigeration has increased due to the shortage of food, food storage, health, and industrial cooling. Global warming is the most important environmental problem that the world has faced recently. Global Warming Potential (GWP) values of HFC type refrigerants are high. In order to contribute to this environmental problem, HFC type refrigerants are replaced with alternatives. In some cases, especially when flammability is critical, alternative refrigerants are rarely preferred yet. Therefore, it is very important to use appropriate charge values. Cooling systems are modeled in order to use them under necessary and sufficient minimum conditions. While cooling system modeling, the most important problem is to choose the compressor model according to the appropriate refrigerant. In this study, a cooling system modeling code was developed and integrated with CoolProp software in MATLAB environment. The purpose of this investigation is to develop a method comp simulation software for determining which compressor model to use for a selected fluid. The modeling of refrigeration cycle with and without Internal Heat Exchanger (IHX) cases has been investigated. Analyses were made on different compressor correlations and the results were compared with the experimental data from literature. As the second purpose of the present study a compressor model has been proposed in the literature. The proposed model results are good agreement with EN12900 and experimental results.

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INTRODUCTION

Regarding the environmental consequences of HFC-refrigerants, as well as the usage of flammable refrigerants, have necessitated a reduction in the refrigerant charge in refrigeration and heat pump systems. At the component level, it can be seen that the primary refrigerant content is generally found in the heat exchangers. The charge may be decreased to exceedingly low levels by using compact designs Palm (2007) and Palm (2008). Optimal charging values can be calculated with modeling and each parameter can be checked beforehand with modeling. Cooling systems have been investigated both experimentally and numerically using various modeling techniques in recent years. Based on the rapidly developing computer technology the number of numerical studies has surpassed that of experimental investigations. Different methods have been developed to calculate compressor performance. In a modeling study by Sun et al. (2016), thermodynamic performances of R23/404A and R41/404A refrigerant couples in a cascade cooling system were compared using MATLAB software. The compression processes were considered adiabatic, and the target functions of optimization work were selected as Coefficient of Performance (COP), exergy destruction, and exergetic efficiency. It has been shown that R41 has better thermodynamics performance and thus can be used in low-temperature cycles in cascade cooling systems Sun (2016). Another numerical study was conducted in Organic Rankine Cycle by Imran et al. in 2020 who listed different models in their study. By modeling, they were able to examine the behavior of the system in terms of safe and efficient design and operation of the loop for heat sources Imran et al. (2020). In 2017 a modeling work by Molinaroli et al., introduces a semi-empirical model of a rolling piston compressor and validates it by evaluating four different compressors. Over 96 percent of the computed refrigerant mass flow rates and more than 97 percent of the calculated compressor electrical powers were found to match within 5% Molinaroli (2017). Another work studied a compressor modeling comparing work in 2020. Modeling the compares compressor modeling platforms to aid in the selection of an appropriate modeling platform for a broad research project. User-generated models for the reciprocating compressor are created in MATLAB and Modelica. The identical compressor is modeled using PDSim and GT-Suite, two modern compressor modeling tools. It has been emphasized that each program has its own advantages and disadvantages Tanveer (2020). Wang studied the properties of the RC01 refrigerant mixture, which consists of 55% R1234ze(E) and 45% R290 by mass, were extracted using REFPROP software and compared with other refrigerants. They concluded that RC01 could be used instead of R22 Wang et al. (2021). Roskosch and friends studied that a differential compressor model in 2017 for reciprocating compressors is developed, which quickly predicts volumetric and isentropic efficiency and can be easily fitted with observed data at only one operation point of an existing compressor. The volumetric efficiency is found a mean forecast error of 3.0 percent, and the isentropic efficiency is found a mean prediction error of 2.3 percent Roskosch et al. (2017). In 2018 another modeling study carried out by Roy and Mandal investigations were made using R161 instead of R404A in the high-temperature cycle based on the COP, exergy destruction, and second law efficiency of the system over different evaporator temperatures. The authors concluded that R161 which has higher performance can be an alternative to R404A which is flammable. Compressor power was calculated using enthalpy throughout this study Roy and Mandal, (2018). Similarly, in 2019 and 2020, enthalpy differences were used for the compressor in the experimental and modeling studies of García del Valle et al. (2020); García del Valle et al. (2019). Dutra purposed a new approach indeed to simulate hermetic reciprocating compressors with a specific model for the electrical motor. The estimation of motor slip, and thus the compressor speed, is an important component of the suggested model. Under a high-pressure ratio scenario (Te = −35 °C; Tc = 70 °C) the discrepancies between the predicted and measured values for the volumetric and isentropic efficiencies were 10.2 percent and 7.8 percent for the standard model and 5.3 percent and 5.6 percent for the suggested model, respectively Dutra et al. (2015). In 2013 C. K. Lee and friends studied on, three models for compressor models were analyzed for 4 different fluids. It was stated which compressor model should be rejected Lee et al. (2013). Likewise in 2011 Elgendy work on compressor model is proposed in the study on cooling system modeling Elgendy et al. (2011). Similar to our study of Encabo Cáceres et al. (2017) performed an optimization study for the Organic Rankine Cycle using RefProp table data and modeling in MATLAB environment. The study, 39 heat transfer fluids were examined. The aim of his study is to determine the best performing fluid base on the energetic and exergetic performance assessment. In 2016 Müller investigated the effect of IHX effect on zeotropic refrigerants. His calculations made for R22 and R407C show that IHX is always improving the EER for the cycle with the refrigerant mixture, while there is no improvement in EER in the cycle using R22 with the low evaporation temperature. Compressor work was calculated as isentropic under the ideal gas assumption in Müller’s study Müllern (2016). Altunacar used EN 12900 as the theoretical compressor model in his experimental and theoretical study Altunacar (2020); DIN (2013). Also, there are numerous compressor models developed by various researchers such as Ramaraj et al. (2014), Bell et al. (2011) and Nelson James et al. James et al. (2016). In this present study, different compressor models used in cooling system modeling and these models effect on performance calculation are investigated. Despite the reduced charge values by using IHX, it has been reported that not choosing the appropriate compressor model will change the cooling performance values significantly. For this reason, a developed compressor model is suggesting to literature. The suggested model's outcomes show good agreement with those of EN12900 and experiments.

PROBLEM DEFINITION

Compressor models plays an important role that strongly affects of true calculation of EER and COP values in a refrigeration system. The work developed cooling system solver on MATLAB. Seven different compressor models were investigated in developed solver. Compressor models are given in Table 1 which are EN 12900 DIN (2013), Ramaraj et al. model Ramaraj et al. (2014), thermodynamic table from CoolProp software Bell et al. (2014), Elgendy et al. (2011), Lee et al. (2013), James et al. (2016) and isentropic approach under the ideal gas assumption. Developed code of cooling system on MATLAB can be solve with and without IHX. Pressure drops and heat losses from the system were neglected. Investigated refrigeration cycle with internal heat exchanger is mirrored in Fig 1.

The basic equations of the developed codes are the following,

$$COP = \frac{Q_{eva}}{W_{comp}} \quad (1)$$

Since super-heating and sub-cooling are done with an IHX, super-heating and the increase in compressor work are added to the definition of EER, it shown in Fig 2.

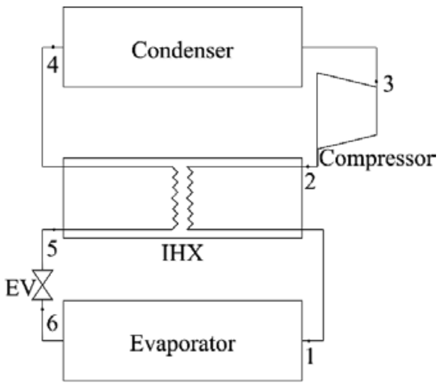

Fig 1. Schematic representation of the investigated refrigeration cycle with internal heat exchanger.

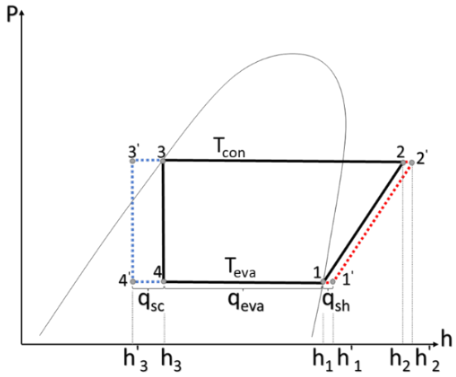

Fig. 2. The P-h diagram showing the regions of latent heat for sub-cooling and sensible heat for super-heating.

For the calculation of performance, energy efficiency ratio (EER) is defined as in Eq. 7.

$$h_{fg} = h_{vap} - h_{liq} \quad (2)$$

$$q_{sh} = \mathrm{h}_{1'} - h_1 \quad (3)$$

$$q_{eva} = h_{fg} - (h_4 - h_{liq}) = h_{fg} - c_p(T_{con} - T_{eva}) \quad (4)$$

$$q_{sc} = \mathrm{h}_3' - h_3 \quad (5)$$

$$w_{comp} = h_{2'} - h_{1'} \quad (6)$$

EER is the ratio of a unit's cooling output relative to its input power adding superheat and subcooling effects.

$$EER = \frac{h_1 - \mathrm{h}_{3'}}{\mathrm{h}_{2'} - \mathrm{h}_{1'}} \quad (7)$$

Thermal Modeling of Cooling System

The cooling system was initially modeled in MATLAB toward being able to apply the compressor models. CoolProp software has been integrated with the MATLAB environment. Flowchart of the code procedure is shown in Fig 3. Seven different models and approaches have been investigated, including the EN 12900 compressor standard, as shown in Table 1.

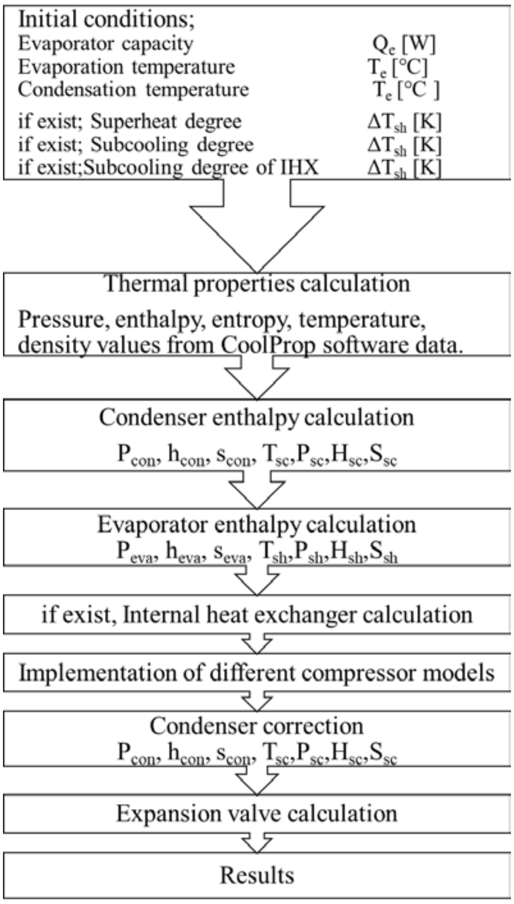

Fig. 3. Flowchart depicting the code established in MATLAB to simulate the cooling cycle.

Table 1. Investigated models and approaches.

Model	**Explanation**
EN 12900	EN 12900 compressor model standard
Ramaraj model	Model of Ramaraj et al. in 2014
Nelson J. model	Model of Nelson James et al. in 2016
CK Lee approach	The approach CK LEE et al. studied in 2013
Ideal gas approach	Ideal gas compressor work calculation, Eq. (5)
Real gas approach	CoolProp thermodynamics table value
Elgendy model	Model of Elgendy et al. in 2011

The first model described in EN12900 standard defines the compressor performance in a polynomial form where the variables are calculated using the evaporation temperature at the suction point and condensation temperature at the dew point. It is used by giving polynomial coefficients based on the experimental data provided by the compressor manufacturers with the EN12900 standard. Therefore, the most appropriate values can be obtained for real results. The only problem is that the compressor model will change in the modeling system depending on the cooling capacity, evaporator and condenser temperatures, and the polynomial coefficients will constantly change. The results of this standard are used by software of compressor producers which are FrasCold and Bitzer. Where, X cooling capacity (W) or flow rate (kg/h) with according to C coefficients from standart, S is evaporator temperature (℃), D is condenser temperature (℃). With the EN12900 standard, compressor manufacturers present the performance data of the compressor they produce as a polynomial. However, the biggest disadvantage of this model is that it is product-based. The polynomial coefficients presented for a particular product are not fixed. It takes different values for different compressors.

$$X = C_1 + C_2(S) + C_3(D) + C_4(S^2) + C_5(S\,D) + C_6(D^2) + C_7(S^3) + C_8(D\,S^2) + C_9(S\,D^2) + C_{10}(D^3) \tag{8}$$

Second model developed by Ramaraj et al. (2014) encompasses a compressor performance calculation method in a semi-empirical way. The model uses experimental data for the creation of a performance map. Where, $\dot{W}$ compressor power [W], n polytropic exponent,$\dot{m}$ mass flow rate [kg/s], P absolute pressure [kPa], X_{oil} mass fraction of oil, subscripts m is mixture of refrigerant, and ref is refrigerant.

$$\dot{W} = \frac{n}{n-1}\left(\frac{\dot{m}}{\rho_{1m}}\right)(C_{17} + C_{18}X_{oil} + C_{19}X_{oil}^2)(C_{20}P_1 + C_{21})\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \tag{9}$$

$$n = \frac{\ln\left(\frac{P_2}{P_1}\right)}{\ln\left(\frac{\rho_{2m}}{\rho_{1m}}\right)} \tag{10}$$

$$\rho_m = \frac{\rho_{oil}\rho_{ref}}{(1-x_{oil})\rho_{oil}+x_{oil}\,\rho_{ref}} \tag{11}$$

In the Ramaraj model lubrication is also taken into account as a mass fraction of oil is 0.1. Third model published by Nelson James et al. James et al. (2016). The frictional losses within the machine are considered to be precisely proportional to the rotational speed (ω) at which the compressor runs, according to the research. Also, it was emphasized that the compression process has two steps. The first involves isentropic compression of a fluid mixture along the machine's internal volume ratio. A comparison is done after the isentropic compression process to check if the pressure after compression is lower than the pressure in the discharge plenum. If the fluid is under compressed, further work must be done to bring it up to the discharge pressure.

$$\dot{W}_{friction} = 2\pi\tau_{loss}\omega \tag{12}$$

$$\dot{W}_s = \dot{m}(h_2 - h_1) \tag{13}$$

$$\dot{W}_v = v_{int}(P_2 - P_{int}) \tag{14}$$

$$\dot{W} = \dot{W}_s + \dot{W}_v + \dot{W}_{friction} \tag{15}$$

Where $\dot{W}$ is total power and compression along internal volume ratio $\dot{W}_s$, compression of discharges process $\dot{W}_v$, [W], τ_{loss} frictional torque [N.m], ω angular velocity [Hz]. Fourth model studied by Lee et al. (2013). Three different models were compared in his study. One of the models used under the condition of a constant specific heat capacity.

$$w = h_2 - h_1 - \frac{(s_2-s_1)(T_2-T_1)}{\ln\left(\frac{T_2}{T_1}\right)} \tag{16}$$

Fifth approach ideal gas, the gas enthalpy at evaporation pressure can be taken as the compressor inlet enthalpy. Under the isentropic compression assumption, the enthalpy value at the evaporator pressure is calculated so that the input entropy equals the output entropy. Specific work of compressor can be described by an equation of an isentropic work of the heat capacity for ideal cooling cycle and cooling cycle with IHX.

$$w = \frac{n}{n-1}P_1 v_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right] \tag{17}$$

Sixth approach used for the calculation of compressor performance in this study utilizes enthalpy tables. CoolProp software is integrated into MATLAB program in order to automatize the use of thermodynamic data. Compressor performance can be calculated from the inlet and outlet enthalpy differences. Thermodynamic table values were obtained using CoolProp software which was created and published by Bell et al. (2014). Seventh model is published by Elgendy et al. (2011). In the study has a correlation of gas engine energy consumption as a function of compressor power, engine speed. It has been established based on experimental data. Isentropic efficiency, $\eta_{pseduo-is}$ is included, Φ and γ represent the coefficients of the linear law of the pseudo-isentropic efficiency and Neng is the engine speed of the compressor. Coefficient a, b, c, d, e are from the article.

$$\eta_{pseduo-is} = \phi\left(\frac{P_{int}}{P_{suc}}\right) + \gamma \tag{18}$$

$$Y = \exp\left(aN_{eng} + bp_{suc} + cp_{dis} + dT_{suc} + e\right) \tag{19}$$

$$\dot{W} = \left(\frac{\dot{m}(h_3-h_2)}{\eta_{pseduo-is}}\right) + \dot{m}v_3(p_{dis} - p_{int}) \tag{20}$$

Suggested Compressor model

Compressor manufacturers provide EN12900 standard data for each compressor. These polynomial values obtained with experimental data are product-based. For this reason, it limits the ability of parametric programming. Nelson-James compressor model and Ramaraj model are the closest models to EN-12900 results among the models investigated. In this study, the approaches of Ramaraj and Nelson-James were examined in the proposed model. In the proposed developed compressor model, frictional losses and the effect of lubrication are taken into account. The model was developed on the basis of compressor inlet and outlet pressures, enthalpy difference, mass fraction of oil and shaft torque.

The mean density expression is used in equation 21 as suggested by Ramaraj et. al Ramaraj idr., (2014).

$$\rho_{1m} = \frac{\rho_{oil}\rho_{ref,1}}{(1-X_{oil})\rho_{oil}+X_{oil}\rho_{ref,1}} \tag{21}$$

Where ρm from Ramaraj average density [kg/m^3]. $\rho_{ref,1}$ inlet refrigerant density and $\rho_{ref,2}$ outlet refrigerant density. ρ_{oil} is oil density, constant 981 kg/m^3. n isentropic exponent.

$$n = \frac{\ln\left(\frac{P_2}{P_1}\right)}{\ln\left(\frac{\rho_{2m}}{\rho_{1m}}\right)} \tag{22}$$

$$w = \frac{v_{int}(P_2-P_{int})}{2} + \frac{n}{\rho_{1m}(n-1)}(42-8X_{oil}+23X_{oil}^2)(14P_1+1676)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right]+0.5(h_2-h_1)+\frac{\pi\tau_{loss}\omega}{\dot{m}1000} \tag{23}$$

All terms in the suggested compressor model are kJ/kg since the product of specific volume and pressure is kJ/kg. In the second term, the enthalpies are taken directly as kJ/kg. In the third term, since the product of torque and frequency will give Watt, it is used by dividing it by the mass flow rate. In the equation 23, indices 1 are compressor input, and indices 2 are compressor output indices. The pressure and enthalpy values at the inlet and outlet of the compressor and the density and specific volume value at the compressor inlet were calculated. Shaft torque can be obtained from compressor manufacturers and is available in the literature for many capacities and evaporation temperatures. It is the product of the shaft torque of the compressor and the friction factor. The shaft torque can be obtained from the manufacturer's data. But this value could be taken as $\tau = 0.65$ Nm. The frictional torque τloss value can be calculated by multiplying the shaft torque by the friction coefficient. The friction coefficient is around μ~0.008 and it can be seen in 2014 study of Hwang et al. (2002). Mass fraction of oil in the equation 0.1 was taken. Inlet refrigerant density X mass fraction. τ shaft torque. ω is rotational speed. As an example of calculation R1234yf refrigerant, for a 2.7-kW cooling capacity, the work consumed by the compressor for the evaporator temperature of 10 °C and the condenser temperature of 50 °C was examined. $\omega = 50hz, \tau_{\text{loss}} = 0.65\ \text{Nm}, \text{v}_1 = 0.041\ m^3, P_2 = 1302.32\ kPa, P_1 = 437.53\ kPa,\ X_{oil} = 0.1, h_2 = 388.94\frac{kJ}{kg},\ h_1 = 369.70\frac{kJ}{kg}$. [w=33.1 kJ/kg]

RESULTS AND DISCUSSIONS

The developed compressor model has been investigated in the modeled cooling system on MATLAB for cases with and without IHX. Investigations were made for R134a, R404A, R407C and R1234yf refrigerants. The EN12900 standard is the presentation of the polynomial function based on the evaporator and condenser temperatures of the compressor's experimental data produced by the compressor manufacturers DIN, (2013). Various compressor models are available in the literature. Six of these models were compared to EN12900, as presented in Fig. 4. and Fig. 5.

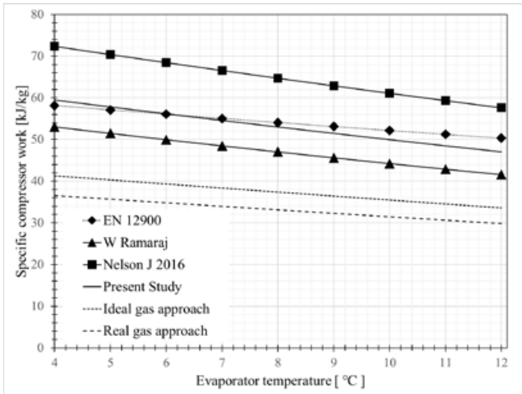

Fig. 4. Specific compressor work of present developed model without IHX R407C, Qe = 10kW, calculated; Tc=57.6 °C.

Models are investigated in a cooling system with R407C refrigerant, evaporator temperature Te = 6.770 °C, compressor temperature Tc = 57.609 °C with internal heat exchange. Fig 5 shows that the models in the literature give results that are far from the EN12900 standard data. Among the models examined, the closest to EN12900 is the Ramaraj and Nelson-James model. The results are examined, the theoretical compressor models and the experimental based compressor models are clustered among themselves. Therefore, while making the investigation, the models were also compared with different studies in the literature. Theoretical and experimental studies were selected from the literature. for example, compressor models were compared with the theoretical work of Arora et al. (2008). COP values was evaluated with -50°C to 0°C different evaporator temperatures in Fig. 6. for condenser temperature Tc = 55°C and R404A refrigerant. Theoretical approaches and experimental-based models formed two separate groups. The theoretical calculation made by Arora et al. (2008) is good agreement with the ideal gas approach.

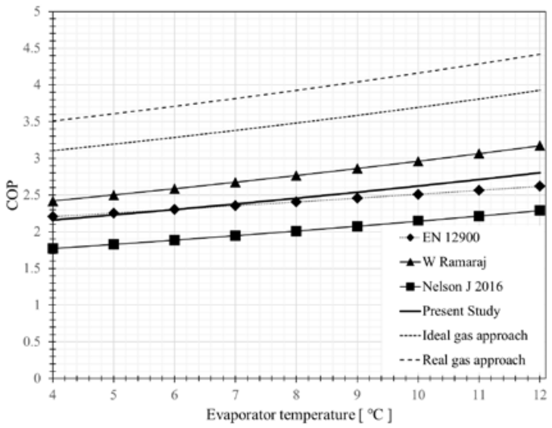

Fig. 5. COP changing with evaporator temperature of present developed model without IHX, R407C, Qe = 10 kW, Te=6.7°C, Tc=57.6 °C.

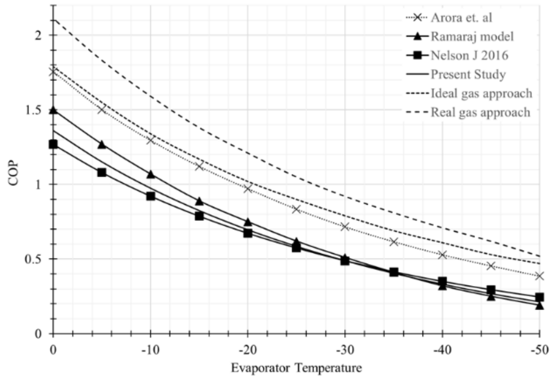

Fig. 6. Comparison of the change of COP according to evaporator temperatures with the theoretical study of Arora et. al, refrigerant R404A without IHX, Tc=55°C.

The effect of friction and lubrication should also be included in the compressor calculation to be compatible with the experimental calculations. Experimental results differ from theoretical results. There are many reasons for this. Irreversibility, the effect of lubrication are examples. It is necessary to compare experimental-based models with experimental data. Oruç et al. (2020) published experiments for different evaporator and condenser temperatures and evaporator capacities. The experimental results and the Nelson-James model, the Ramaraj model, and the results of the compressor model proposed in this study were compared. As shown in Fig. 7. the evaporator temperature is -5°C and the condenser temperature is 30°C, the compressor work is 940W, while the result of the purposed model in present study is 971W. The deviation is of the order of 3 %. Similarly, for the experiment with the evaporator temperature of 0 °C and the condenser temperature of 30 °C, the deviation is in the order of 0.5%. it can be seen that the model proposed in this study has a good agreement with the experimental results.

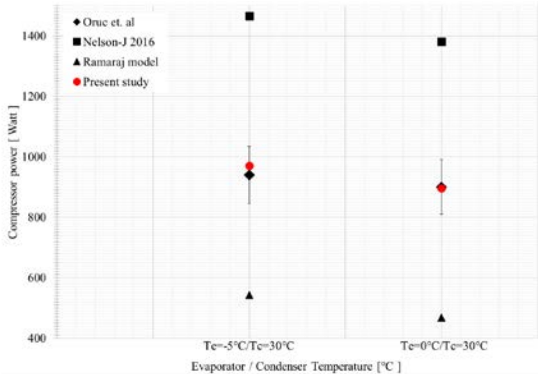

Fig. 7. Comparison of the experimental results of Oruc et. al and the model proposed in this study, refrigerant R404A without IHX.

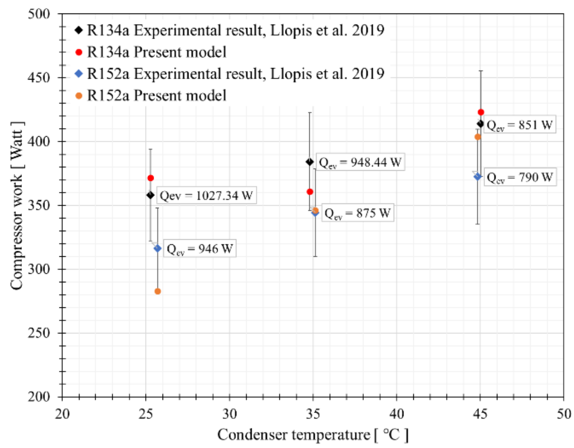

Fig. 8. Comparison of the Experimental results Llopis et al. 2019 and the proposed model, Te=0°C.

The developed compressor model is compared work of Llopis et al. (2019). In the results presented in Fig. 8. 6 different datapoints were examined for two different refrigerants with R152a 2 K subcooling degree and 4K subcooling degree for R134a refrigerants. As a result of the examination, it was seen that the developed model was below ten percent. The developed model and the experimental results are in good agreement. In the study of Devecioglu (2018), R1234ze(E), R1234yf, R134a refrigerants were examined for 5 K subcooling degree. When the results were examined, the error in the examination of R1234ze(E) and R134a fluids was below ten percent as shown Fig. 9. Compared to R1234yf refrigerant, the error rate increased to over ten percent as the condenser temperature increased. is at the level of 11 percent.

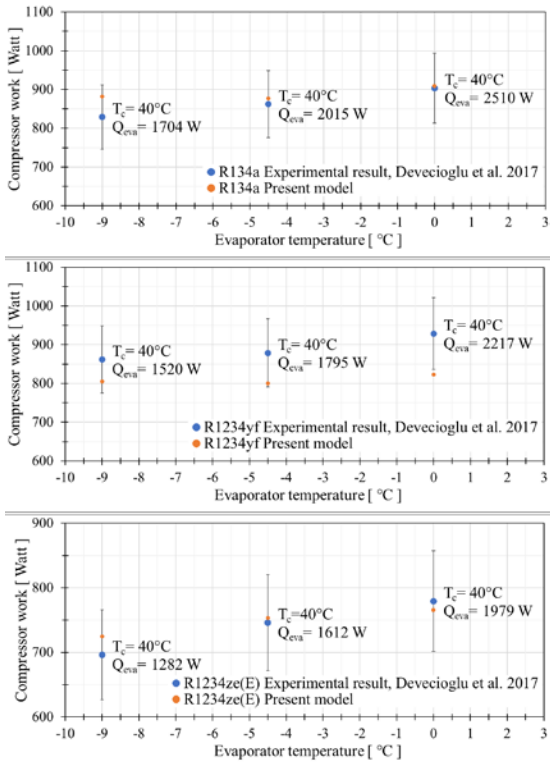

Fig. 9. Comparison of the Experimental results Devecioglu et al. 2017 and the proposed model.

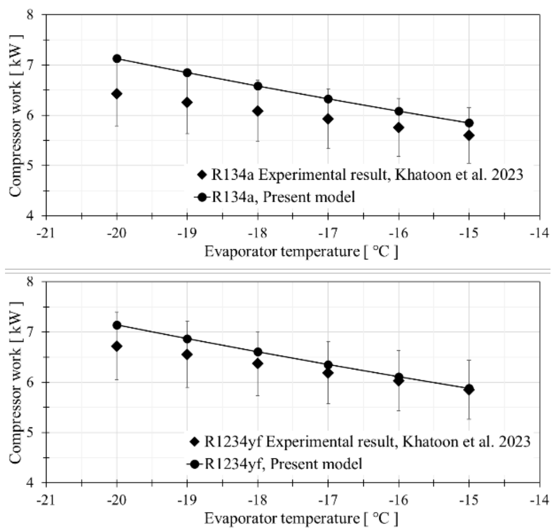

Fig. 9. Comparison of the Experimental results Khaaton et al. 2023 and the proposed model, Tcon=35°C, Qeva=10 kW.

In the study of Khatoon et al. (2023), R134a, R1234yf refrigerants were examined. It was observed that the error increased in the proposed model as the difference between the evaporator condenser temperatures increased as shown Fig. 10. All inspections as a result of the inspections, the error rate was found to be in the order of ten percent and below. Experimental data provided to us by the laboratories of Bitzer company have been compared with the purposed model for R134a and R404A as shown Figure 11.

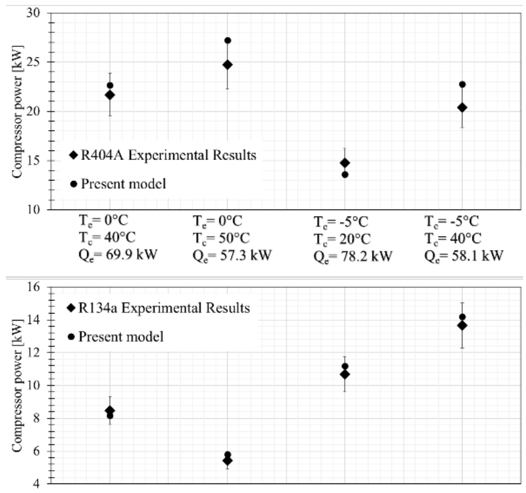

Fig. 10. Comparison of the Experimental results and the proposed model.

It was observed that the error rate increased as the evaporator condenser temperature difference increased. The investigation reveals that the proposed model had a deviation rate of less than 10%. The developed model and the experimental data correlate well.

CONCLUSION

CoolProp software has been integrated MATLAB Software which was used as the modeling environment in the current study. The compressor models were also applied to MATLAB code.

- Compressor models were compared. The compressor model selection results for Cooling System simulation are at the following for Fig 4-7. The specific compressor work and COP values of EN 12900, Nelson J. and Ramaraj models are close to each other. Also, the Elgendy model has a minor difference. However, there is a great deviation of the values of ideal and real gas approach from those of the EN12900 model. EN12900, Nelson J. and Ramaraj models differs from other theoretical models. Other theoretical models are also compatible with each other. This is because of the difference between ideal and real working conditions.
- When the case with an internal heat exchanger is compared with the case without an internal heat exchanger, the results are similar.
- With the compressor model developed within the scope of this study, the closest results to the EN12900 standard data presented by compressor manufacturers were obtained, compared to other models.

- The compressor model proposed within the scope of this study, the results obtained for IHX-containing and without IHX cases are in accordance with EN12900 results.
- EN12900 is product bases, so it is not suitable for programming and parametric analysis. The developed compressor model offers the advantage of parametric programming, because of this model mainly dependent on inlet and outlet conditions. So, this model is not product bases except torque. But frictional torque loss could be taken consant values as given section 2.2.
- The agreement with the experimental results with developed compressor model for the 30 K and 35 K temperature difference between the evaporator and condenser temperatures is at the level of 5%, Fig 7-11.
- For the three refrigerants investigated in this study, the proposed compressor model was good agreement with previous empirically based models. It has been found that it is appropriate to use the recommended compressor model. Finally, best model must be searched for the specific refrigerant.

REFERENCES

ALTUNACAR, T. *Amonyaklı Soğutma Sistemi Tasarımı*. İstanbul Technical University. (2020.).

Arora, A. in Kaushik, S. C. Theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A. International Journal of Refrigeration, 31(6), 998–1005. https://doi.org/10.1016/j.ijrefrig.2007.12.015 (2008.).

Bell, I. H., Groll, E. A. in Braun, J. E. Performance of vapor compression systems with compressor oil flooding and regeneration. International Journal of Refrigeration, 34(1), 225–233. https://doi.org/10.1016/j.ijrefrig.2010.09.004 (2011.).

Bell, I. H., Wronski, J., Quoilin, S. in Lemort, V. Pure and pseudo-pure fluid thermophysical property evaluation and the open-source thermophysical property library coolprop. Industrial and Engineering Chemistry Research, 53(6), 2498–2508. https://doi.org/10.1021/ie4033999 (2014.).

Devecioğlu, A. G. in Oruç, V. Improvement on the energy performance of a refrigeration system adapting a plate-type heat exchanger and low-GWP refrigerants as alternatives to R134a. Energy, 155, 105–116. https://doi.org/10.1016/j.energy.2018.05.032 (2018.).

DIN, N. K. (FINKä) im. Kältemittelverdichter-Nennbedingungen, Toleranzen und Darstellung von Leistungsdaten des Herstellers; Deutsche Fassung EN 12900:2013. ICS 23.140; 27.200. https://doi.org/10.3139/9783446461208.007 (2013.).

Dutra, T. in Deschamps, C. J. A simulation approach for hermetic reciprocating compressors including electrical motor modeling. International Journal of Refrigeration, 59, 168–181. https://doi.org/10.1016/j.ijrefrig.2015.07.023 (2015.).

Elgendy, E., Schmidt, J., Khalil, A. in Fatouh, M. Modelling and validation of a gas engine heat pump working with R410A for cooling applications. Applied Energy, 88(12), 4980–4988. https://doi.org/10.1016/j.apenergy.2011.06.046 (2011.).

Encabo Cáceres, I., Agromayor, R. in Nord, L. O. Thermodynamic Optimization Of An Organic Rankine Cycle For Power Generation From A Low Temperature Geothermal Heat Source. Proceedings of the 58th Conference on Simulation and Modelling (SIMS 58) Reykjavik, Iceland, September 25th – 27th, 2017, 138, 251–262. https://doi.org/10.3384/ecp17138251 (2017.).

García del Valle, J., Quinatoa Quinatoa, K. J. in Castro Ruiz, F. Experimental study of the static and dynamic behavior of a novel heat driven electronic controlled expansion valve. Applied Thermal Engineering, 168(July 2019), 114718. https://doi.org/10.1016/j.applthermaleng.2019.114718 (2020.).

García del Valle, J., Sierra-Pallares, J., Rodríguez Vega, A. S. in Castro Ruiz, F. Influence of the degree of superheating on the performance of a R134a condenser by means of experimental and numerical studies. International Journal of Refrigeration, 98, 25–34. https://doi.org/10.1016/j.ijrefrig.2018.10.026 (2019.).

Hwang, J. J., Hwang, G. J., Yeh, R. H. in Chao, C. H. Measurement of interstitial convective heat transfer and frictional drag for flow across metal foams. Journal of Heat Transfer, 124(1), 120–129. https://doi.org/10.1115/1.1416690 (2002.).

Imran, M., Pili, R., Usman, M. in Haglind, F. Dynamic modeling and control strategies of organic Rankine cycle systems: Methods and challenges. Applied Energy, 276(July), 115537. https://doi.org/10.1016/j.apenergy.2020.115537 (2020.).

James, N. A., Braun, J. E., Groll, E. A. in Horton, W. T. Semi-empirical modeling and analysis of oil flooded R410A scroll compressors with liquid injection for use in vapor compression systems. International Journal of Refrigeration, 66(2016), 50–63. https://doi.org/10.1016/j.ijrefrig.2015.12.011 (2016.).

Khatoon, S. in Karimi, M. N. Thermodynamic analysis of two evaporator vapor compression refrigeration system with low GWP refrigerants in automobiles. International Journal of Air-Conditioning and Refrigeration, 31(1), 2. https://doi.org/10.1007/s44189-022-00017-1 (2023.).

Lee, C. K. in Lam, H. N. A comparison of different generalised modelling approaches for a scroll refrigerant compressor. International Journal of Refrigeration, 36(4), 1369–1375. https://doi.org/10.1016/j.ijrefrig.2013.01.002 (2013.).

Llopis, R., Calleja-Anta, D., Sánchez, D., Nebot-Andrés, L., Catalán-Gil, J. in Cabello, R. R-454C, R-459B, R-457A and R-455A as low-GWP replacements of R-404A: Experimental evaluation and optimization. International Journal of Refrigeration, 106, 133–143. https://doi.org/10.1016/j.ijrefrig.2019.06.013 (2019.).

Molinaroli, L., Joppolo, C. M. in De Antonellis, S. Modèle semi-empirique pour les compresseurs hermétiques à piston roulant. International Journal of Refrigeration, 79, 226–237. https://doi.org/10.1016/j.ijrefrig.2017.04.015 (2017.).

Müller, J. Energy Efficiency Improvement in the Refrigeration Cycle Using Internal Heat Exchanger - a New Approach. Thermal Engineering. https://doi.org/10.4467/2353737XCT.16.199.5948 (2016.).

Oruç, V. in Devecioğlu, A. G. Experimental assessment of the retrofit of an internal heat exchanger in refrigeration systems: The effect on energy performance and system operation. Applied Thermal Engineering, 180(July). https://doi.org/10.1016/j.applthermaleng.2020.115843 (2020.).

Palm, B. Refrigeration systems with minimum charge of refrigerant. Applied Thermal Engineering, 27(10), 1693–1701. https://doi.org/10.1016/j.applthermaleng.2006.07.017 (2007.).

Palm, B. Hydrocarbons as refrigerants in small heat pump and refrigeration systems - A review. International Journal of Refrigeration, 31(4), 552–563. https://doi.org/10.1016/j.ijrefrig.2007.11.016 (2008.).

Ramaraj, S., Yang, B., Braun, J. E., Groll, E. A. in Horton, W. T. Experimental analysis of oil flooded R410A scroll compressor. International Journal of Refrigeration, 46, 185–195. https://doi.org/10.1016/j.ijrefrig.2014.08.006 (2014.).

Roskosch, D., Venzik, V. in Atakan, B. Modèle thermodynamique pour compresseurs à piston, se concentrant sur les rendements dépendant du fluide. International Journal of Refrigeration, 84, 104–116. https://doi.org/10.1016/j.ijrefrig.2017.08.011 (2017.).

Roy, R. in Mandal, B. K. Exergy analysis of cascade refrigeration system working with refrigerant pairs R41-R404A and R41-R161. IOP Conference Series: Materials Science and Engineering, 377(1). https://doi.org/10.1088/1757-899X/377/1/012036 (2018.).

Sun, Z., Liang, Y., Liu, S., Ji, W., Zang, R., Liang, R. in Guo, Z. Comparative analysis of thermodynamic performance of a cascade refrigeration system for refrigerant couples R41/R404A and R23/R404A. Applied Energy, 184, 19–25. https://doi.org/10.1016/j.apenergy.2016.10.014 (2016.).

Tanveer, M. M. in Bradshaw, C. R. Quantitative and qualitative evaluation of various positive-displacement compressor modeling platforms. International Journal of Refrigeration, 119, 48–63. https://doi.org/10.1016/j.ijrefrig.2020.07.009 (2020.).

Wang, L., Li, H., Wu, J. in Qiu, K. Feasibility of using near-azeotropic refrigerant mixture R290/R1234ze(E) as substitute for R22. Environmental Progress & Sustainable Energy, 40(3). https://doi.org/10.1002/ep.13574 (2021.).

NOMENCLATURE

w:	Specific compressor work [kJ/kg]
$\dot{W}$:	Compressor power [W]
h:	Enthalpy [kJ/kg]
cp:	Specific heat capacity [kJ/(kg K)]
ρ:	Density [kg/m3]
P:	Pressure [kPa]
T:	Temperature

Subscripts

1:	Inlet of compressor
2:	Outlet of compressor
eva:	Evaporator
con:	Condenser
comp:	Compressor
fg:	Latent heat
vap:	Vapor
liq:	Liquid
sh:	Superheat
sc:	Subcooling