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Experimental investigation of a rolling piston compressor operated with a hydrocarbon refrigerant mixture

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Abstract

Theoretical considerations indicate enhanced system performance of household appliances when using zeotropic mixtures compared to R134a. This paper describes the experimental investigation of the performance behavior of a rolling piston compressor used in a heat pump tumble dryer and operated with a zeotropic zeotropic blend.

The experimental investigations demonstrate the change in operation conditions and compressor performance of the substitute refrigerant in comparison with the hydrofluorocarbon R134a. The compressor is tested on a compressor test rig according to DIN EN 13771-1 to measure and determine power consumption, cooling resp. condensing capacity, pressure ratio and cycle temperatures. The measured data of the reference setup with R134a and the substitute setup with a hydrocarbon refrigerant mixture will be compared with each other for a better understanding of their influences under different operation conditions.

Keywords: Rolling Piston Compressor; Compressor Performance;Liquid Slugging; Zeotropic Refrigerant Blend; Drop-in Refrigerant

1. Introduction

The Kigali Amendment to the Montreal Protocol [1] entering into force on January 1, 2019 regulates the global phase down of hydrofluorocarbons (HFC) with a high global warming potential. It confirms the European F-Gas Regulation from 2015 that already controls the phase down of fluorinated substances, e.g. refrigerants, based on their global warming potential (GWP). Consequently, hydrofluorocarbons (HFCs), for instance R134a, are limited in their availability and alternative solutions need to be investigated. Such solutions can include the development of new refrigeration systems designed to the specifics of the chosen low GWP refrigerant or retrofitting an existing system to said low GWP refrigerant. The latter is widely favored in terms of investment cost. The difficulty in retrofitting is to match the thermo-physical properties of one refrigerant with an entirely different substance not compromising the efficiency.

Having identified a refrigerant matching the boundary conditions, refrigeration cycle calculations follow to assess the system performance. However, the effects and consequences of a refrigerant change have to be tested in real systems. The influences of geometry or the compatibility with the lubricant and the behavior in the existing components are parameters difficult to appraise analytically. Therefore, an experimental investigation is necessary to achieve a deeper understanding e.g. of the effects on the compressor performance of an alternative solution.

A series of measurements on a test rig according to the DIN EN 13771-1 was carried out using a commercial rolling piston compressor designed for household heat pump applications. This paper presents the results of the performance measurements of the reference set-up with R134a and the substitute refrigerant blend of RE170 and R290. Three test conditions were examined corresponding to the operating range of the compressor.

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2. Experimental Method

2.1. Experimental Setup

The present test rig setup is designed for performance tests of small capacity refrigerant-compressors in accordance with the ASHRAE standard 23.1 [2], resp. DIN EN 13771-1 [3]. Based on a standard vapor compression cycle the test bench includes a calorimeter evaporator and a flow meter to determine the refrigerant mass flow rate. Besides the main components like the compressor, the condenser, and an electronic expansion device, the cycle is additionally equipped with an oil separator, a filter dryer, sight glasses, and an accumulator (see Fig. 1). The compressor and the condenser are each integrated in the circuit with a detachable screw connection ensuring the adaption of the test bench to varying compressor capacities. Additionally, a small 12 V-fan installed next to the compressor may optionally cool it with ambient air to prevent too high discharge temperatures.

There are five measuring points for temperature and pressure, one of them monitoring the state of the secondary fluid in the calorimeter. The numbering of the locations refers to the designated point of state of the refrigerant in the cycle. All temperature sensors are PT100 resistance thermometers with class A accuracy [4]. The pressure transducers have a measurement inaccuracy of ±0.1 %FS. Table 1 lists the measuring range of the installed pressure sensors, which is necessary to calculate the absolute value of the uncertainty. A power meter separately plots the electrical input power of the compressor and the resistance heating element of the calorimeter. Finally, the mass flow rate and the ambient temperature complete the set of measured data.

In this specific study, the compressor mounted onto the test bench is a rolling piston compressor with the following characteristics:

- displacement of 7.8 cm^3/rev,
- 2 pole induction motor, grid frequency of 50 Hz,
- voltage range of 187 to 264 V.

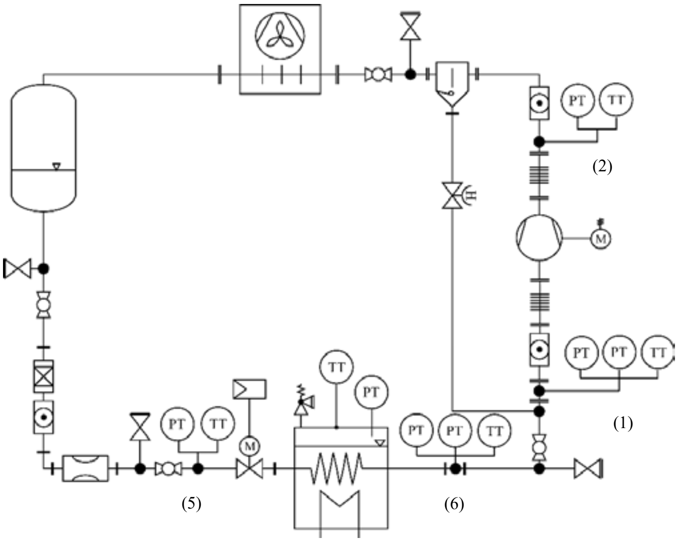


Fig. 1. Process and instrumentation chart of the test bench

Table 1. Installed Pressure Transducers

Measuring point	Range	Output
discharge line (2), inlet of the expansion device (5)	0 … 25 bar	4 … 20 mA
outlet of the evaporator (6), inlet of the compressor (1)	0 … 16 bar and 0 … 3 bar	4 … 20 mA
inner volume of calorimeter	0 … 25 bar	4 … 20 mA

Besides the pressure range of the sensors, the design of the evaporator calorimeter limits the operation range of the test bench. The maximum heat input capacity is 2 kW that is equivalent to the maximum cooling capacity. Secondly, the opening pressure of the safety relief valve is 20 bar, which leads to a maximum calorimeter temperature of 67.48 °C with liquid R134a as a secondary fluid.

2.2. Test Conditions

This paper investigates a rolling piston compressor used in a heat pump tumble dryer application. The objective is to compare the nowadays-used R134a with the substitute refrigerant blend RE170/ R290 at certain operating conditions. Besides the performance data supplied by the manufacturer, American and European standard rating conditions were considered to identify consistent parameters for the tests. The American AHRI standard 540, table 1, [4, p. 5] defines rating conditions for positive displacement refrigerant compressors. Next, the DIN EN 12900, table 5, [5, p. 10] states rating conditions of performance data as well but differing from the AHRI conditions. However, both standards distinguish three refrigeration appliances:

- "H" – high evaporation temperature,
- "M" – medium evaporation temperature,
- "L" – low evaporation temperature.

Figure 2 shows the manufacturer's data on the compressor operating pressure range with R134a. Considering those limits, the number of possible rating conditions decreases eminently leaving four rating conditions.

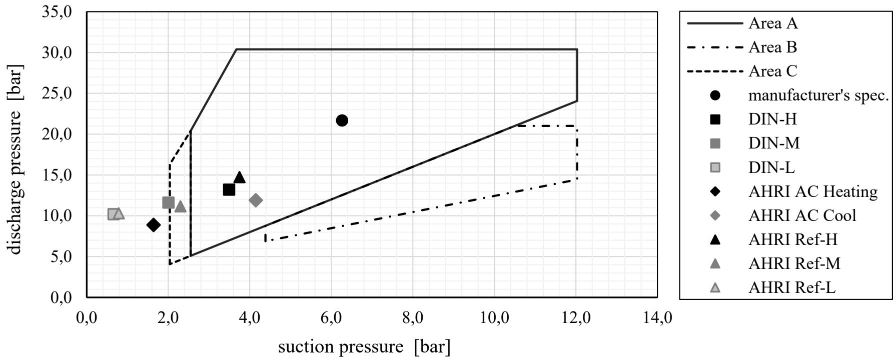


Fig. 2. Compressor operating range for R134a (Area A: normal operating zone; Area B: High density flow zone - running time less than 3 minutes; Area C: Low pressure zone - running time less than 3 minutes) with possible rating conditions

Out of the four points within area A, the "AHRI AC Cool" condition is neglected in the tests because it refers to an air-conditioning application not corresponding to the compressor's appliance. Since an adjustment of the refrigerant subcooling temperature and the ambient temperature is not possible with the described set-up of the test bench, those parameters are not specified in the rating conditions. Instead of the "AHRI Ref-H" point, the manufacturer defines a condition called "ASHRAE" condition. This state has similar temperature levels but different superheating which is included in the experimental investigation. Each test condition has a distinctive feature giving it merit for testing. The manufacturer specification refers to a heat pump application. The "DIN-H"-condition derives from a refrigeration appliance and the "ASHRAE"-condition adds a large superheating to similar refrigeration characteristics. The test conditions retained for the comparison of the two refrigerants are given in Table 2.

Table 2. Performance rating conditions for the rolling piston compressor in accordance with the manufacturer's data

	Suction dew point T_0 [°C]	Discharge dew point T_C [°C]	Suction gas T_1 [°C]
manufacturer's specification	23	71	35
ASHRAE	7.2	54.5	35
DIN-H	5	50	20

2.3. Experimental proceeding

The preparation for the test consists of evacuating the circuit and charging the cycle with refrigerant and the compatible oil (Table 3). The charged mass is measured indirectly by weighing the refrigerant bottle. Uncertainties result from the non- measurable liquid level of the receiver (see Table 3).

Table 3. Tested refrigerants, their charged amounts and the corresponding compressor oil

Refrigerant	Composition [%w]	refrigerant charge	Oil charge
R134a	100	434.8±20 g	prefilled, POE VG65
RE170/R290	69.86/30.14	383.8±20 g	200 ml of POE VG 100

The DIN test procedure [3] stipulates the recording of the measured data once the cycle reaches steady state. Therefore, the expansion device and the calorimeter heat input are adjusted according to the set temperature and the pressure at the suction side of the compressor. The condenser fan speed, respectively the swept air-volume flow, is the regulating variable for the compressor discharge pressure. When using a flow meter to determine the mass flow rate one must ensure that the minimum subcooling of the refrigerant is $\Delta T_{C,sc} = 3\,K$. Increasing or decreasing tendencies larger than 50% of the allowed deviation of the test parameter, as listed in Table 4, indicate that no steady state has been reached. The minimum duration of data recording is 15 minutes and at least one set of measurements per minute is required. Having established steady state conditions, several consecutive measurements of 15 minutes each are recorded. The same test is repeated in another session.

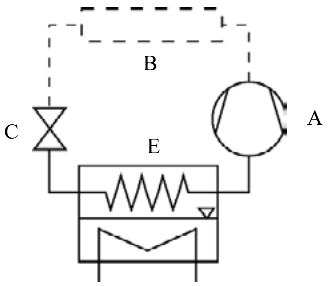
Table 4. Permitted deviation of measured values for the duration of a test series

Measured parameter	allowed deviation
pressure (absolute)	±1%
temperature	±3.0 K
calorimeter heat input	±1% of cooling capacity

2.4. Evaluation of the Data

The general processing of the acquired data for this experimental investigation follows the European standard on compressor rating [3]. If the measurement meets the criteria of the steady state, the mean values of the logged parameters are the basis for the calculation. Due to the diversion between the experimental data and the ideal rating condition, the consideration of corrective factors is vital for an objective comparison of the rated compressor performance. In addition to the COP, the isentropic efficiency is also determined using this test procedure.

The described data evaluation applies on the average values of a 15-minute-measurement. The first step is to determine the refrigerant mass flow rate. Therefore, the present test rig is equipped with the evaporator calorimeter (method A) and a refrigerant flow meter (method E), thus representing two standard methods [3] indicated in Fig. 3.



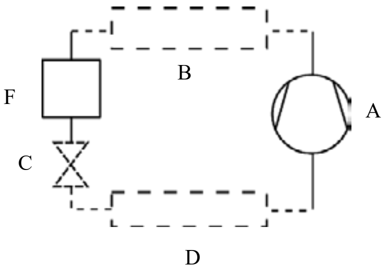


Fig. 3. Schematic of DIN 13771-1 test methods A (left) and E (right): A – compressor; B – condenser; C – expansion device; D – evaporator; E – evaporator calorimeter with heat source; F – mass flow meter

According to method A, the mass flow rate $\dot{m}_A$ is calculated using a calorimeter energy balance (1) considering the calorimeter heat loss F, the electrical input power of the heater $\dot{Q}_n$, the saturation temperature of the secondary fluid t_s'' and the enthalpy difference of the inlet (h_6) and outlet (h_5) state. The measured refrigerant mass flow $\dot{m}_E$ is assumed to be oil-free due to the installed oil separator (2). Subsequently, the average of those two values is referred to as the actual mass flow rate $\dot{m}_a$ (3). The multiplication with a corrective factor regarding the ratio of the actual density at the compressor inlet $v_{1,a}$ and the ideal density v_1 leads to the refrigerant mass flow at the specified rating conditions (4).

$$\dot{m}_A = \frac{\dot{Q}_n + F * (t_{amb} - t''_s)}{h_6 - h_5} \tag{1}$$

$$\dot{m}_E = \dot{m}_{flow\ meter} \tag{2}$$

$$\dot{m}_a = \frac{1}{2}(\dot{m}_A + \dot{m}_E) \tag{3}$$

$$\dot{m} = \dot{m}_a * \frac{v_{1a}}{v_1} \tag{4}$$

The compressor's cooling capacity $\dot{Q}$ is obtained by multiplying the mass flow with the enthalpy difference between the ideal state at the compressor inlet h_1 and the boiling point enthalpy h_{f2} at the ideal compressor discharge pressure.

$$\dot{Q} = \dot{m} * \left(h_1 - h_{f2}\right) \tag{5}$$

The calculation of the power consumption at the ideal rating condition regards the ratio of the isentropic compressor power P_i at the specified rating conditions (6) and the isentropic power at the measured conditions P_{ia} (7). This ratio multiplied with the measured compressor power consumption P_a equals the input power at the specified rating conditions P.

$$P_i = \dot{m} * (h_{i1-2} - h_1) \tag{2}$$

$$P_{ia} = \dot{m}_a * (h_{i1-2a} - h_{1a}) \tag{7}$$

$$P = \frac{P_i}{P_{ia}} * P_a \tag{8}$$

Finally, the characteristic values are determined putting the previously calculated figures into relation.

$$COP_R = \frac{\dot{Q}}{P} \tag{9}$$

$$\eta_i = \frac{P_{ia}}{P_a} \tag{10}$$

The evaluation of a 15-minute-measurement solely gives a momentary impression. The average of the consecutive measurements and the results of the repetition experiment performed on another day as well as the corresponding standard deviation of each parameter represent the performance characteristics categorised by the refrigerant and the test condition.

3. Results and Discussion

The investigation of the compressor performance consists of three aspects. First, the difference between the two refrigerants at the theoretical nominal rating condition is examined, second the values calculated from the measured data are investigated and finally the compressor performance with regard to the refrigerant substitution is discussed.

3.1. Nominal Conditions

The experimental investigation comprises three operating points defined by the suction pressure dew point temperature, the discharge pressure dew point temperature, and the actual suction gas temperature as described above. Applying two refrigerants calls for a closer examination of the fluids properties regarding the specific refrigeration capacity, the specific volume of the compressor inlet state, and the pressure ratio assuming an ideal refrigeration cycle.

Fig. 3 (left) illustrates the specific refrigeration capacity of both refrigerants at the three operating points. Generally, the refrigerant mixture of RE170 and R290 possesses a distinctively larger specific refrigeration capacity than R134a. It indicates that less refrigerant mass flow is required to obtain the same cooling capacity. The large superheat of the “ASHRAE”-condition leads to a slight increase of capacity while the high dew point temperatures of the manufacturer’s condition significantly decreases the specific refrigeration capacity. The latter observation is rather unsurprising as the latent heat decreases with rising temperature, respectively pressure.

As a rule, the volumetric refrigeration capacity (see Fig. 4, right) is an indicator for the compressor size. The mixture of RE170 and R290 exhibits a slightly larger volumetric capacity, indicating that an adjustment of the compressor size is necessary for future endeavours. Overall, the prospective reduction of the refrigerant mass flow due to the increased specific refrigeration capacity in combination with comparable volumetric refrigeration capacity leads to the conclusion that using the zeotropic blend reduces the refrigerant charge maintaining the compressor size or rather the compressor model. This observation corresponds to the actual refrigerant charge of the test bench in Table 3 and further supports the substitution of R134a.

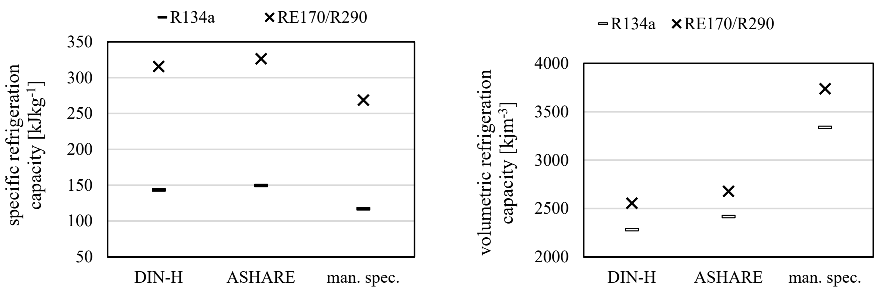


Fig. 4. Specific refrigeration capacity (left) and volumetric refrigeration capacity (right) of R134 and RE170/ R290 at the specified rating conditions

The last nominal parameter to discuss in terms of compressor performance is the pressure ratio depicted in Fig. 5. This value mainly influences the required compression work and gives an idea of the discharge temperature too. Ideally calculated according to the specified rating conditions, the pressure ratios of the two refrigerants are in a similar range supporting a substitution. Yet, it even suggests a slightly lower work input in the case of the zeotropic blend. Still, there are more influencing factors to the power consumption like the mass flow rate, the pressure ratio, and internal friction that should be analysed.

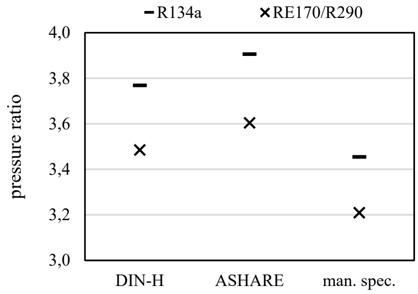


Fig. 5. Ideal pressure ratio of R134 and the mixture RE170/R290 at the specified rating conditions

3.2. Experimental Data

The evaluation of the experimental raw data follows the calculation described in section 2.4. All discussion points of the experimental results refer to the average value over the number of consecutive measurements at the same nominal conditions performed on one day and their repetition on another day. The standard mean error is shown to depict the deviation within the measured or calculated figure.

In general, the rating consists of the refrigerating capacity, the power input and the refrigerant mass flow rate. However, this paper refrains from discussing the refrigeration capacity, as it is the product of the ideal refrigeration capacity and the determined refrigerant mass flow rate. Boundary conditions not specified upfront or different from the standard condition must be stated as well (see Table 5).

The boundary conditions such as the ambient and the subcooled refrigerant temperature affect the compressor performance comparability but cannot be influenced at the present state of the test bench. The experiments were performed during the summertime causing fluctuating and, in some cases, elevated ambient temperatures. As it solely matters in terms of the compressor-shell cooling and the condenser fan speed, the latter being of low importance to the compressor performance, the relatively small divergence is negligible. The subcooled temperature of the refrigerant-mixture tends to be around 2 K cooler than the baseline refrigerant tracing back to the temperature glide of the refrigerant blend. As stated before, the subcooled state is a boundary condition that influences the calculation of the mass flow rate (1). There is no direct link to the rating of the compressor power due to the mentioned corrective factors.

Table 5. Boundary conditions for the performed compressor test

Test condition		DIN-H		ASHRAE		manufacturer's specification	
Refrigerant		R134a	RE170/R290	R134a	RE170/R290	R134a	RE170/R290
ambient temperature T_{amb}	[°C]	26.04±0.34	24.84±0.36	27.19±1.45	25.0±0.31	28.77±0.99	25.11±0.09
subcooled refrigerant T_5	[°C]	45.41±0.29	43.90±0.33	50.03±0.07	48.22±0.33	66.50±0.13	64.94±0.06

The discussion of the experimental data regards the determined mass flow rate as depicted in Fig. 5 (left diagram) and the standard mean error shown in Table 6. It supports the suggestion of a reduced mass flow of the alternative refrigerant. Given the fact that the compressor runs at a constant speed thus having a constant suction volume flow, the mass flow determined from the measured data according to equations (1) to (4) of the zeotropic mixture is on average 47 % lower than the one of R134a. In terms of the comparability of this value with other testing facilities, one must bear in mind that the mass flow rate is highly dependent on the expansion device and its inlet state. The measured mass flow rate of the test bench is the result of the valve lift and the expansion device's inlet state resp. the subcooled temperature, the priority of the valve lift being the setting of the suction pressure. Consequently, the measured flow rate might differ for other test benches for other subcooled temperatures. The examination of the product of the mass flow rate and the previously compared specific refrigeration capacity of both refrigerants combined lead to an on average 17.7 % higher refrigeration capacity with the zeotropic blend.

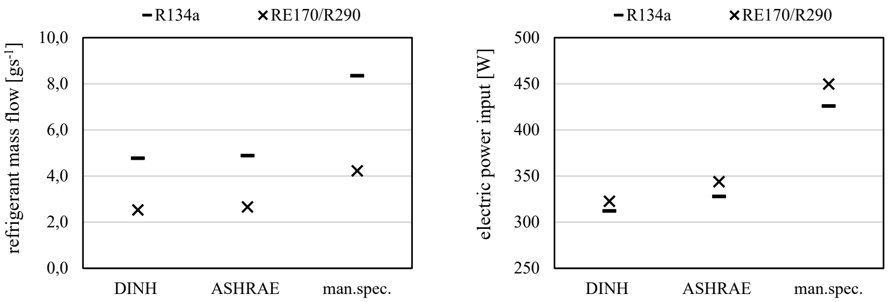


Fig. 6. Calculated refrigerant mass flow rate and electric power input at the specified rating conditions

Table 6. Relative standard mean error of averaged results in percentage [%]

Test condition	DIN-H		ASHRAE		manufacturer's specification	
Refrigerant	R134a	RE170/R290	R134a	RE170/R290	R134a	RE170/R290
power consumption	0.80	1.00	1.17	0.55	0.26	0.38
mass flow rate	1.42	0.46	0.98	0.09	1.24	0.59

In spite of the favorable pressure ratio and mass flow rate as stated before, the power consumption of the compressor (see Fig. 6, right diagram) in the case of the zeotropic mixture is on average elevated by 4.6 %. However, the trend for both refrigerants at the specified test conditions is the same. The refrigeration conditions “DIN-H” and “ASHRAE" have comparable pressure levels while the suction gas temperature is considerably higher for the latter condition. As a result, the power consumption increases while the mass flow rate remains rather constant. This observation testifies that a large superheating of the refrigerant is not beneficial to the compressor power consumption. The analysis of some compressor characteristics such as the discharge temperature, the oil sump temperature, and the isentropic efficiency in Fig. 7 helps to understand the mechanism behind this difference in the compressor input power for both refrigerants. The actual compressor input power is subject to thermal and mechanical losses. While thermal losses of the refrigerant gas affect the isentropic efficiency, friction causes the compressor material temperature to be elevated thus decreasing the heat loss of the compressed gas to the compressor shell. This chain of cause and effect helps to understand the observations made in Fig. 7. The left diagram reveals a higher isentropic efficiency at all operating conditions using RE170/ R290 compared to R134a. Meanwhile, the oil sump temperature of R134a is considerably below the one occurring during the tests with the refrigerant blend as the discharge temperature ranges among the same high values. The rather large thermal gradient over the shell of the compressor respectively the oil sump and the compressed gas supports the low isentropic efficiency. In return, the oil sump temperature of the tests with RE170/ R290 is close to the measured discharge temperature resulting in small to no thermal losses of the compressed gas. However, the elevated oil sump temperature indicates friction of the mechanical parts. Given the fact that the described behavior can be observed throughout all operating conditions it is the evidence that the lubricant is not favorable for the compressor although it was selected for the refrigerant blend due to the higher solubility.

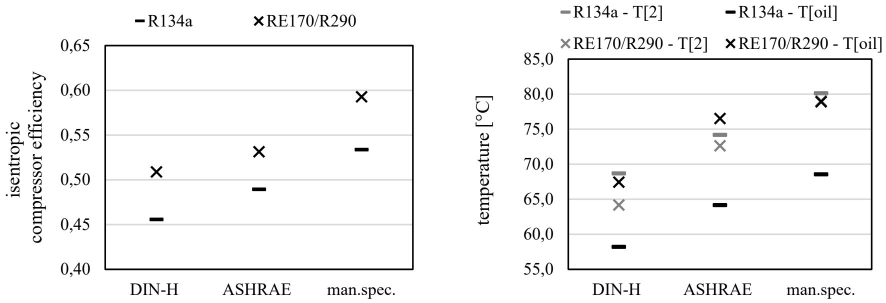


Fig. 7. Compressor characteristics – calculated isentropic efficiency based on measured values (left), refrigerant discharge temperature and oil sump temperature (right)

Finally, all evaluated data at the reference rating conditions come together when discussing the rated coefficient of performance COP_R depicted in Fig. 8. Despite the elevated power consumption of the compressor with the refrigerant blend, its coefficient of performance is greater than of the base line refrigerant at the three tested rating conditions. On the other hand, the results of the R134a tests reveal an upward trend throughout the rating conditions. Given the fact that the compressor is designed for the heat pump application, the observation of the highest COP using R134a-at the manufacturer's condition is not surprising. However, the COP of the substitute refrigerant is on average 12.6 % higher than the baseline.

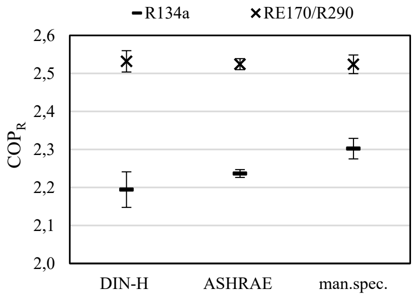


Fig. 8. Rated coefficient of performance at the specified rating conditions

4. Conclusion

In comparison to the high GWP refrigerant R134a, the tested natural refrigerant blend of RE170 and R290 shows favorable results e.g. the COP is significantly higher and the volumetric cooling capacity is in the same level of magnitude. Under this precondition, the experimental outcome supports the substitute refrigerant blend. The reduction of the charge and the increased performance at three different rating conditions are the core findings. The standard mean deviation proves the repeatability of the tests with respect to the stated boundary conditions like the ambient temperature. However, the increase of input power testing the alternative refrigerant raises interest for further optimization of the lubrication.

To conclude, the performed compressor tests reveal the potential of the natural refrigerant blend RE170/ R290 directly substituting R134a. It has merit to regard it for future cycle designs because the adaption of the compressor size, the heat exchangers or, the expansion device might affect the cycle efficiency positively. Further, the general acceptance of the natural yet flammable refrigerants must grow respecting the appropriate security measures since these refrigerants are inevitable for sustainable solutions.

Nomenclature

symbols		*Unit*
COP_R	rated coefficient of performance (cooling)	
F	thermal loss coefficient of calorimeter	$W\,K^{-1}$
h	specific enthalpy	$kJ\,kg^{-1}$
$h_{i1\text{-}2}$	specific enthalpy of the refrigerant gas at the compressor discharge port assuming an isentropic compression (specified rating conditions)	$kJ\,kg^{-1}$
$\dot{m}$	mass flow rate	$kg\,s^{-1}$
P	power	*W*
p	pressure (absolute)	*bar*
Q	cooling capacity at rating conditions	*W*
Q_n	electrical power input of the heater (calorimeter)	*W*
T	temperature	*°C or K*
V_{SW}	theoretical swept volume flow of the compressor	$m^3 s^{-1}$
v_1	specific volume	$m^3 kg^{-1}$
η	efficiency	

subscripts	
0	evaporation
1	compressor inlet
2	compressor outlet
f2	discharge bubble point of specified rating conditions
5	inlet expansion device
6	outlet evaporator
a	actual, measured
amb	ambient
i	ideal isentropic
s	secondary fluid
v	volumetric

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References

[1] UNEP, "Amendment to the Montreal Protocol on substances that deplete the ozone layer," 15 October 2016. [Online]. Available: https://treaties.un.org/doc/Publication/CN/2016/CN.872.2016-Eng.pdf.

[2] ASHRAE, *ASHRAE Standard 23.1-2010: Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant,* 2010.

[3] DIN - Deutsches Institut für Normung e.V., *DIN EN 13771-1: Compressors and condensing units for refrigeration - Performance testing and test methods – Part 1: Refrigerant compressors,* 2017.

[4] AHRI - Air-Conditioning, Heating and Refrigeration Institute, *Performance Rating of Positive Displacement Refrigerant compressors and Compressor Units,* 2015.

[5] DIN - Deutsches Institut für Normung e.V. , *DIN EN 12900 - Rating Conditions, tolerances and presentation of manufacturer's performance data,* 2013.