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"THEORETICAL COMPARISON OF R-123, R-124, R-125, R-134A, R-143A, AND R-152A IN A SINGLE-STAGE VAPOUR COMPRESSION REFRIGERATION SYSTEM AT LOW EVAPORATING TEMPERATURES WITH EXERGY ANALYSIS OF COMPRESSION"

Dinesh Choudhary [1], Prof. Bhupendra Singh Sikarwar [2]

[1] PG scholar, Department of Mechanical Engineering, SAGE University, Indore, Madhya Pradesh, India

[2] Assistant Professor, Department of Mechanical Engineering, SAGE University, Indore, Madhya Pradesh, India

ABSTRACT

This study compares halocarbon refrigerants in a single-stage vapour compression refrigeration cycle by a theoretical model and relates the results to a small experimental unit that uses R-134a. The cycle was evaluated for R-123, R-124, R-125, R-134a, and R-143a at evaporating temperatures of −30, −40, −50, and −60 °C, a condensing temperature of 40 °C, and 5 K of superheat, and the effects of condenser pressure and of subcooling on the COP of R-134a and R-152a were studied for an evaporator pressure of 100 kPa and condenser pressures of 600 to 1200 kPa. At −30 °C the COP was 2.76 for R-134a, 2.7 for R-123, 2.68 for R-143a, 2.18 for R-124, and 1.75 for R-125, and all of them fell as the evaporating temperature was lowered, by 37.0% (R-123) to 50.3% (R-125) at −60 °C. R-123 overtook R-134a below about −45 °C. The second-law efficiency, evaluated against the Carnot COP between the evaporating and condensing temperatures, was 0.80 to 0.77 for R-134a and 0.50 to 0.41 for R-125. At an evaporator pressure of 100 kPa the COP fell by about 45.9% for R-134a as the condenser pressure rose from 600 to 1200 kPa, R-152a gave a COP 6.6% higher than R-134a at 1200 kPa but 1.2% lower at 600 kPa, and 5 K of subcooling raised the COP by 4.6 to 6.6% for R-134a. The practical COP of the R-134a test unit was 85–91% of the theoretical value. An exergy analysis of the compression process showed an exergy destruction of 4.6 to 8.1 kJ/kg at a discharge superheat of 15 K, which rose to 14.5–25.1 kJ/kg at 25 K. Data for R-152a in the cycle tables were found to be thermodynamically inconsistent and were excluded, and these checks are reported.

Key Words: *Vapour compression refrigeration; R-134a; R-152a; R-123; COP; second-law efficiency; subcooling; exergy destruction; low evaporating temperature.*

I. INTRODUCTION

Refrigeration systems transfer heat from a low-temperature space to a higher-temperature surrounding by means of a work input, and the vapour compression cycle is used for almost all applications because a large amount of heat is absorbed in the latent heat of vaporisation and the evaporation is isothermal [1], [2]. In the cycle, low-pressure liquid refrigerant evaporates in the evaporator and leaves slightly superheated, the compressor raises the pressure of the vapour, the condenser desuperheats, condenses, and subcools it, and the expansion device reduces the pressure and controls the flow to the evaporator. The performance is measured by the coefficient of performance (COP), the ratio of the refrigerating effect to the compressor work, which depends on the refrigerant and on the evaporating and condensing conditions [1], [2].

The phase-out of chlorofluorocarbons has led to the use of hydrofluorocarbon and hydrochlorofluorocarbon refrigerants, and R-134a, which is widely used, and R-152a, which has a lower global warming potential, are frequently compared [3], [4]. Bolaji [5] reported that the average COP of R-152a in a domestic refrigerator designed for R-134a

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was 4.7% higher than that of R-134a, and in a study of subcooling the same author found that subcooling raised the COP and the capacity for all the alternatives to R-12 that were tested [6]. Exergy analysis, which quantifies the irreversibility of the cycle components, has been applied to several refrigerants and shows the compressor and the condenser to be the main sources of loss [7], [8]. However, comparisons are usually made at evaporating temperatures typical of domestic refrigerators, and the behaviour of several halocarbons at −30 °C and below, with a check against the second-law limit, is less documented.

This paper therefore compares R-123, R-124, R-125, R-134a, and R-143a at evaporating temperatures from −30 to −60 °C, studies the effect of condenser pressure and subcooling on R-134a and R-152a, compares the theoretical and practical COP of an R-134a test unit, and examines the exergy destruction in the compression process. A consistency check of the data against the Carnot limit is also reported.

II. Problem Identification

The refrigerant and the operating conditions of a vapour compression system must be selected together, but the effect of the evaporating temperature, the condenser pressure, and the degrees of superheat and subcooling differs between refrigerants. A refrigerant that gives a high COP at a moderate evaporating temperature may perform poorly at a lower temperature, and the ranking of refrigerants can change over the temperature range. For example, a refrigerant that requires sub-atmospheric pressure in the evaporator at low temperature is difficult to use even if its COP is attractive.

A second problem is the reliability of theoretical comparisons. The enthalpies of different refrigerants are tabulated on different reference states, and an error in a single enthalpy value can give a COP that violates the second law of thermodynamics. Such checks should be part of any comparison. The problem addressed in this study is to compare the performance of six halocarbon refrigerants in a single-stage cycle with a consistent set of conditions, to quantify the effect of condenser pressure and subcooling on R-134a and R-152a, and to relate the theoretical COP to that of an experimental unit.

III. Research Objective

The main objectives of the present study are as follows:

- To calculate the COP, refrigerating effect, compressor work, and second-law efficiency of R-123, R-124, R-125, R-134a, and R-143a at evaporating temperatures of −30 to −60 °C.
- To study the effect of condenser pressure and of 0 to 5 K of subcooling on the COP of R-134a and R-152a.
- To compare the theoretical and practical COP of a small R-134a vapour compression test unit.
- To estimate the exergy destruction in the compression process of R-134a for different discharge pressures and discharge superheat.
- To check the data against the Carnot limit and to identify inconsistent values.

IV. Modeling and Simulation

4.1 Cycle model

The cycle was modelled as a single-stage vapour compression cycle with isenthalpic expansion. The state 1 is the vapour leaving the evaporator with 5 K of superheat, state 2′ is the isentropic discharge state, state 2 is the actual discharge state, and state 3 is the saturated or subcooled liquid leaving the condenser, with $h_4 = h_3$. The refrigerating effect q_e, the compressor work w, the COP, and the Carnot COP between the evaporating temperature T_e and the condensing temperature T_c are

$$q_e = h_1 - h_4 w = h_2 - h_1 \qquad \mathrm{COP} = q_e / w \qquad \mathrm{COP_C} = T_e / (T_c - T_e) \qquad (1)$$

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and the second-law efficiency is η_{II} = COP / COP_C. The model was set up in a spreadsheet using enthalpies from refrigerant property tables. The cycles were evaluated for the five refrigerants at T_e = −30, −40, −50, and −60 °C (243, 233, 223, and 213 K), a condensing temperature of 40 °C (313 K), and 5 K of superheat. The liquid enthalpy of R-134a used, 256.41 kJ/kg, is that of the saturated liquid at 40 °C. In the source calculation the Carnot COP was evaluated with a condensing temperature of 318 K, which is inconsistent with the 40 °C state; the Carnot COP and the second-law efficiency in this paper were therefore recomputed with T_c = 313 K.

4.2 Condenser pressure and subcooling

For R-134a and R-152a the COP was calculated for an evaporator pressure of 100 kPa, condenser pressures of 600, 700, 800, 900, 1000, 1100, and 1200 kPa, and 0 to 5 K of subcooling.

4.3 Exergy analysis of the compression process

For an adiabatic compressor the exergy destroyed per unit mass equals the product of the dead-state temperature T_0 and the entropy generated, $T_0(s_2 - s_1)$, and the minimum work for the change of state is $\Delta\psi = (h_2 - h_1) - T_0(s_2 - s_1)$, so the exergetic efficiency of the compression process is $\eta_{ex} = \Delta\psi / (h_2 - h_1)$ [9]. The suction state was saturated R-134a vapour at 101 kPa (−26.06 °C, h_1 = 383 kJ/kg, s_1 = 1.7481 kJ/kg·K), the discharge pressure was 250 to 1000 kPa, the discharge superheat above the saturation temperature at the discharge pressure was 15 to 25 K, and T_0 = 300 K.

V. Design Of The Experimental Unit

A small vapour compression test unit working with R-134a was assembled. It consists of a DC rotary compressor, an air-cooled condenser, an expansion device, an evaporator, a thermostat with a relay circuit for on–off control, and a voltmeter, an ammeter, and an energy meter for the measurement of the electrical input. The specification of the compressor and the refrigerant is given in Table 5.1. R-134a (1,1,1,2-tetrafluoroethane, CH_2FCF_3) has a molar mass of 102.03 g/mol and a normal boiling point of −26.3 °C, and it has zero ozone depletion potential.

Table 5.1: Specification of the compressor and refrigerant of the test unit

Item	Specification
Compressor	DC rotary compressor, 12–36 V
Rotational speed	2500 / 2750 / 3000 rpm
Cooling capacity	60–110 W
Input power	60 W
Rated COP of the compressor	1.3
Refrigerant	R-134a (CH_2FCF_3)
Molar mass; normal boiling point	102.03 g/mol; −26.3 °C

VI. Result and Discussion

6.1 COP at low evaporating temperatures

Table 6.1 gives the refrigerating effect, the compressor work, and the COP of the five refrigerants, and Figure 6.1 shows the COP against the evaporating temperature. All the refrigerants show the same trend: as the evaporating temperature is lowered from −30 to −60 °C the refrigerating effect decreases and the compressor work increases, so the COP falls. The fall is smallest for R-123 (37.0%), followed by R-134a (40.6%), R-124 (41.3%), R-143a (44.4%), and R-125 (50.3%). At −30 °C the order is R-134a (2.76), R-123 (2.7), R-143a (2.68), R-124 (2.18), R-125 (1.75), but R-123 overtakes R-134a between −40 and −50 °C and has the highest COP at −60 °C (1.7 against 1.64). The advantage of R-123 should not be over-interpreted, because its normal boiling point is close to 28 °C, so that at these evaporating

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temperatures it works at pressures far below atmospheric, which is a practical disadvantage. R-125 has the lowest COP and its COP falls below 1 at −60 °C.

Table 6.1: Refrigerating effect (kJ/kg), compressor work (kJ/kg), and COP at condensing temperature 40 °C and 5 K superheat

Quantity	Te (°C)	R-123	R-124	R-125	R-134a	R-143a
q_e (kJ/kg)	-30	123.1	97.22	62.94	123.9	106
w (kJ/kg)	-30	45.5	44.5	35.9	44.8	39.6
COP	-30	2.7	2.18	1.75	2.76	2.68
q_e (kJ/kg)	-40	117.3	91.29	57.36	117.6	99.82
w (kJ/kg)	-40	51.3	50.4	41.5	51.2	45.7
COP	-40	2.29	1.81	1.38	2.3	2.18
q_e (kJ/kg)	-50	111.6	85.39	51.68	111.2	93.54
w (kJ/kg)	-50	57	56.3	47.2	57.5	52
COP	-50	1.96	1.52	1.09	1.93	1.8
q_e (kJ/kg)	-60	106.1	79.55	45.93	104.9	87.17
w (kJ/kg)	-60	62.5	62.2	53	63.8	58.3
COP	-60	1.7	1.28	0.87	1.64	1.49

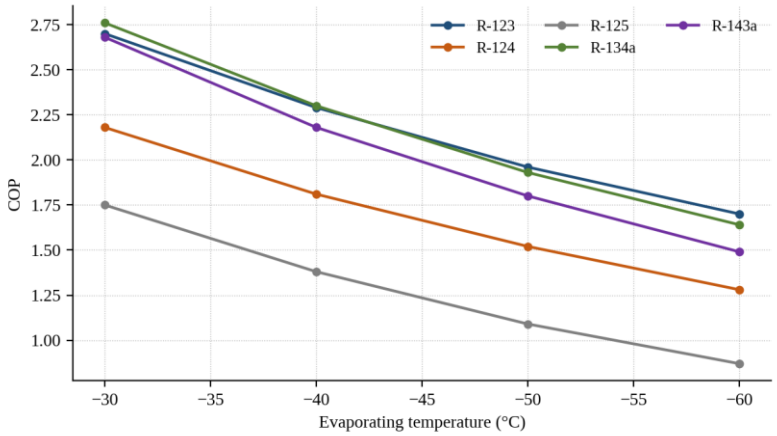


Figure 6.1 COP of the refrigerants at different evaporating temperatures

6.2 Second-law efficiency and consistency check

The Carnot COP between 313 K and the evaporating temperature is 3.47, 2.91, 2.48, 2.13 at −30, −40, −50, and −60 °C. The second-law efficiencies in Table 6.2 and Figure 6.2 are all below one, as they must be. R-123 (0.78–0.80) and R-134a (0.80–0.77) have the highest efficiencies and R-143a is close behind, whereas the efficiency of R-125 falls from 0.50 to 0.41. The efficiencies of R-123 rise slightly as the evaporating temperature is lowered, which means that its COP decreases less steeply than the Carnot COP, whereas those of the other refrigerants fall. The efficiencies in the source tables, which used a condensing temperature of 318 K, are higher by about 0.04 to 0.06 for R-134a (0.85 to 0.81), and the ranking is the same.

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Table 6.2: Second-law efficiency (Carnot COP at 313 K condensing temperature)

Refrigerant	−30 °C	−40 °C	−50 °C	−60 °C
R-123	0.778	0.786	0.791	0.798
R-124	0.628	0.621	0.613	0.601
R-125	0.504	0.474	0.440	0.408
R-134a	0.795	0.790	0.779	0.770
R-143a	0.772	0.748	0.726	0.700

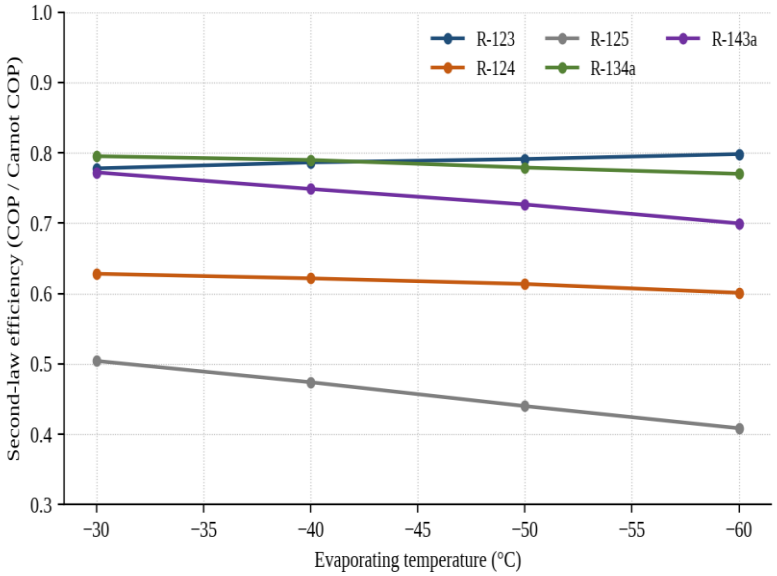


Figure 6.2 Second-law efficiency of the refrigerants

The cycle data for R-152a in the source were checked in the same way. At −30 °C and 40 °C the tabulated values give a refrigerating effect of 214.2 kJ/kg and a compressor work of 52.6 kJ/kg, and hence a COP of 4.07, which is greater than the Carnot COP of 3.47 and gives a second-law efficiency greater than one, which is impossible. The compressor discharge enthalpy of R-152a in that table is therefore inconsistent with the other enthalpies, and the R-152a results of that table were excluded from this paper. The R-152a results in Section 6.3 come from a separate calculation at a fixed evaporator pressure and show values that are consistent with R-134a.

6.3 Effect of condenser pressure and subcooling: R-134a and R-152a

Table 6.3 and Figure 6.3 show the COP of R-134a and R-152a at an evaporator pressure of 100 kPa. For both refrigerants the COP decreases strongly as the condenser pressure rises, by 45.9% for R-134a and 41.6% for R-152a between 600 and 1200 kPa, because the pressure ratio and the compressor work increase. The COP of R-152a is 6.6% higher than that of R-134a at 1200 kPa and 1.7% higher at 900 kPa, about the same at 800 kPa, and 1.6% to 1.2% lower at 700 and 600 kPa. The advantage of R-152a at high condenser pressures is of the same order as the 4.7% found experimentally in a domestic refrigerator [5]. Subcooling of 5 K raises the COP of R-134a by 4.6% at 600 kPa to 6.6% at 1200 kPa and that of R-152a by 3.7% to 4.9%, and the gain is larger at the higher condenser pressure, which agrees with the experimental finding that subcooling improves the COP of all the alternatives tested [6]. One table in the source (a summary of the same comparison) gives different R-152a values at 600 to 800 kPa from the subcooling tables; the subcooling tables, which are complete and internally consistent, were used.

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Table 6.3: COP of R-134a and R-152a (evaporator pressure 100 kPa)

Pc (kPa)	R-134a, 0 K	R-152a, 0 K	R-152a vs R-134a	R-134a, 5 K sub.	R-152a, 5 K sub.
600	4.138	4.090	-1.2%	4.329	4.242
700	3.594	3.538	-1.6%	3.772	3.677
800	3.218	3.227	+0.3%	3.387	3.360
900	2.903	2.953	+1.7%	3.065	3.080
1000	2.643	2.710	+2.5%	2.799	2.832
1100	2.429	2.596	+6.9%	2.580	2.720
1200	2.240	2.387	+6.6%	2.388	2.504

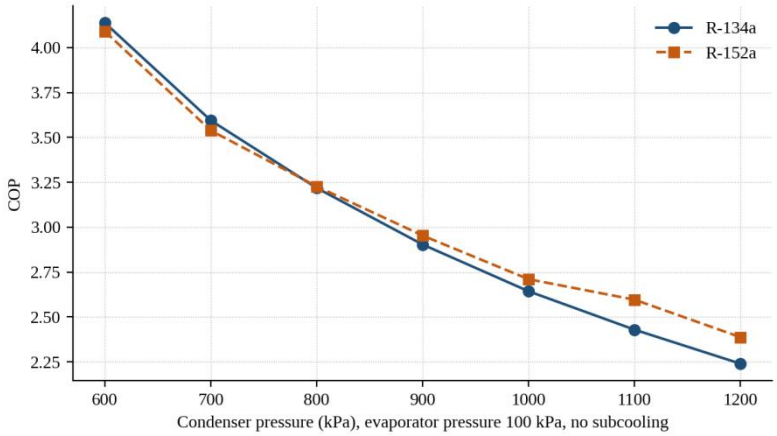


Figure 6.3 COP of R-134a and R-152a against condenser pressure

6.4 Theoretical and practical COP of the R-134a unit

For seven operating points of the R-134a test unit the theoretical COP lay between 4.1 and 4.7 and the practical COP between 3.5 and 4.3, so that the practical COP was 85% to 91% of the theoretical value, with a mean of 88% (Table 6.4). The ratio is plausible for a small unit and shows the losses in the compressor, the pressure drops, and the heat gains. The operating variable that changes between the seven points is not stated in the source, and the compressor of the unit is rated at a COP of 1.3, much below the values in the table, so the comparison should be treated as indicative.

Table 6.4: Theoretical and practical COP of the R-134a unit

Point	Theoretical COP	Practical COP	Ratio
1	4.1	3.5	0.85
2	4.2	3.58	0.85
3	4.3	3.7	0.86
4	4.4	3.9	0.89
5	4.5	4	0.89
6	4.6	4.2	0.91
7	4.7	4.3	0.91

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6.5 Exergy destruction in compression

Table 6.5 and Figure 6.4 give the results for R-134a with a discharge superheat of 15 K. The work Δh increases with discharge pressure from 26.3 kJ/kg at 250 kPa to 52.8 kJ/kg at 1000 kPa, whereas the exergy destroyed $T_0(s_2 - s_1)$ decreases from 8.07 to 4.56 kJ/kg, so that the exergetic efficiency of the compression process rises from 0.69 to 0.91. For a given discharge pressure the exergy destruction increases with the discharge superheat: at 250 kPa it rises from 8.07 to 25.05 kJ/kg when the superheat rises from 15 to 25 K, and at 1000 kPa from 4.56 to 14.46 kJ/kg. A larger superheat at the discharge corresponds to a larger deviation from isentropic compression, so the irreversibility grows with the superheat. The trend with pressure is a consequence of the definition of the cases (a fixed superheat above a saturation temperature that rises with pressure); it should not be read as a decrease of the irreversibility of a real compressor with the pressure ratio.

Table 6.5: Exergy destruction in the compression of R-134a (suction at 101 kPa, discharge superheat 15 K, T0 = 300 K)

P_2 (kPa)	$h_2 - h_1$ (kJ/kg)	$T_0(s_2 - s_1)$ (kJ/kg)	Exergetic efficiency
250	26.3	8.07	0.693
300	29.6	8.04	0.728
350	32.3	7.44	0.770
400	34.8	6.93	0.801
450	37.0	6.54	0.823
500	38.3	6.24	0.837
600	42.5	5.58	0.869
700	44.6	4.38	0.902
800	48.3	5.07	0.895
900	50.7	4.77	0.906
1000	52.8	4.56	0.914

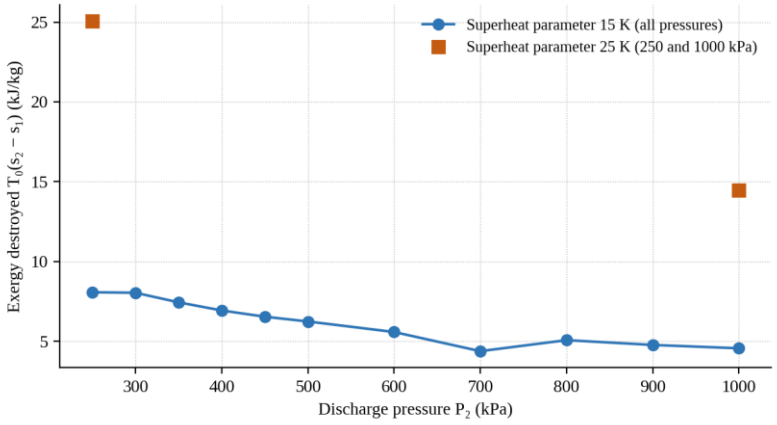


Figure 6.4 Exergy destroyed in the compression of R-134a

In the source tables a further column adds the work Δh and the term $T_0\Delta s$, and the formula in the text subtracts them; the sum has no standard thermodynamic meaning, and only the exergy destroyed and the exergetic efficiency defined in

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Section 4.3 are used here. The values for the 25 K superheat at pressures other than 250 and 1000 kPa show irregularities in the source and are not used.

6.6 Limitations

The following limitations apply. The refrigerant property values were taken from tables and were not independently verified, and the R-152a cycle data in the observation tables were inconsistent. The model is a simple single-stage cycle with no pressure drops, heat gains, or compressor efficiency specified, so that the COP values are theoretical. The degree of superheat of 5 K, the condensing temperature of 40 °C, and the ambient temperature of 300 K are assumed. The experimental data are limited to a theoretical and a practical COP for seven unspecified points, so that no validation of the low-temperature results was possible. Environmental properties such as the global warming potential were not included in the comparison.

VII. CONCLUSION

A single-stage vapour compression cycle was evaluated for six halocarbon refrigerants. At a condensing temperature of 40 °C the COP at −30 °C was 2.76 for R-134a, 2.7 for R-123, 2.68 for R-143a, 2.18 for R-124, and 1.75 for R-125, and it fell by 37.0% to 50.3% when the evaporating temperature was lowered to −60 °C. R-123 had the highest COP at −60 °C, but it works at sub-atmospheric pressure, and R-125 gave the lowest COP, which fell below 1. The second-law efficiency, recomputed with a Carnot COP consistent with the 40 °C condensing state, was 0.80–0.77 for R-134a and 0.50–0.41 for R-125, and the R-152a data of one observation table violated the second law and were excluded. At an evaporator pressure of 100 kPa the COP of R-134a and R-152a decreased with condenser pressure by 45.9% and 41.6% over 600 to 1200 kPa, R-152a was better than R-134a by up to 6.9% at high condenser pressure, and 5 K of subcooling raised the COP by 4.6–6.6% for R-134a. The practical COP of the R-134a unit was 85–91% of the theoretical COP. The exergy destruction in the compression of R-134a was 4.6 to 8.1 kJ/kg at 15 K discharge superheat and increased about threefold at 25 K. Future work should use a single, verified property source for all the refrigerants, include pressure drops and compressor efficiency, validate the low-temperature results experimentally, evaluate the volumetric capacity and the environmental properties of the refrigerants, and extend the exergy analysis to the condenser, evaporator, and expansion device.

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