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Alternative Refrigerants For Household Refrigerators

Mario Straub
Liebherr Hausgeraete Ochsenhausen GmbH, Germany, mario.straub@liebherr.com

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Mario STRAUB*

Liebherr Hausgeräte Ochsenhausen GmbH
88416 Ochsenhausen, Germany
Tel. +49 7352 928 4548
Email: mario.straub@liebherr.com

* Corresponding Author

ABSTRACT

In recent decades, the energy consumption of household appliances has been reduced continuously. This reduction leads to a change in the working conditions of the cooling circuit of domestic refrigerators and freezers. For instance the introduction of fans at the heat exchangers leads to an increased evaporating temperature and a decreased condensing temperature during normal use. In addition, the demand for cooling capacity is lower than in the past, due to improved insulation of the appliances (e.g. improvement of the gaskets, introduction of vacuum insolation panels). Since the beginning of the 1990s it is common in Europe to use R600a (Isobutane) as a refrigerant in household appliances. It is worth to evaluate, if R600a is still the most suited refrigerant for the changed boundary conditions. In this paper, a theoretical analysis of more than 100 refrigerants is carried out to identify possible alternative refrigerants for household appliances. The most promising refrigerants are tested inside different domestic appliances. Apart from the main scope of reducing the energy consumption, the impact on the environment (ozone depleting potential / global warming potential / toxicity) is taken into consideration. As a result, it is feasible to reduce the energy consumption in accordance with the European standard DIN EN 62552:2013-10 (DIN, 2013) and the EU-directive 1060/2010 (European Parliament and Council, 2010) up to 5% by using an alternative refrigerant. The impact of this alternative refrigerant on the environment is similar to the one of isobutane.

1. INTRODUCTION

During the last decades, the energy consumption of household appliances has been steadily reduced. The motivation for this development has higher power costs and stricter statutory regulations. To illustrate this trend, Figure 1 shows the decrease of the energy consumption of a fridge-freezer with a net volume of around 330 liters between 2002 and today (Liebherr, 2002, 2012, 2018) The measurements were carried out in accordance with the European standard DIN EN 62552:2013-10 (DIN, 2013) and the EU-directive 1060/2010 (European Parliament and Council, 2010). In 2002 the model was rated as energy efficiency class “B” (black area in Figure 1). Ten years later, in 2012, the energy consumption of this kind of appliances was decreased by around 50% and rated as energy efficiency class A++ (hatched area in Figure 1). In 2018 there is a further reduction of the energy consumption: a total of 72% less than the baseline appliance of 2002 (dotted area in Figure 1). To achieve these goals, it was necessary to improve and evolve each component of the cooling circuit as well as the insulation of the devices.

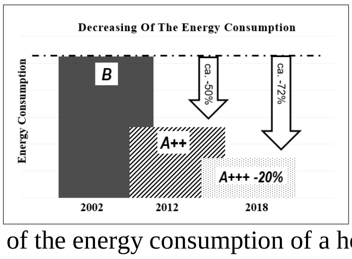

Figure 1 Development of the energy consumption of a household fridge-freezer

2. IMPROVEMENT OF THE COOLING SYSTEM

Each main component of the cooling circuit has been improved along with the insulation to reduce the energy consumption of domestic appliances during the last years. This developments are described in this chapter.

2.1 Heat Exchangers

A decreasing of the losses in the heat exchangers (evaporator and condenser) will result in a lower energy consumption of the appliance. One possibility to improve the heat exchangers is to minimize the temperature difference between refrigerant and the environment of the heat exchanger (Baehr, 1978). A lower temperature difference will lead to a lower heat transfer (Q). This is shown for the evaporator (T_{IN} – T_O) in equation (1) and for the condensers (T_C – T_{AT}) in equation (2). But if the heat to transfer is kept constant, it is required to increase the surface of the heat exchanger (A) or to improve the thermal transmittance (U). The U-value can be improved for instance by using a fan at the heat exchanger.

$$\dot{Q}_O = U_{EV} \times A_{EV} \times (T_{IN} - T_O) \tag{1}$$

$$\dot{Q}_C = U_{CO} \times A_{CO} \times (T_C - T_{AM}) \tag{2}$$

2.2 Compressor

In the past and up to today it has been common to use a single-speed compressor in household appliances with few requirements concerning energy efficiency. In Europe, this kind of compressors has an asynchronous motor with continuous speed of around 2950 rpm (50 Hz / 230 V Input, R600a). Due to the need to reduce the energy consumption, the variable speed compressor was introduced. It is possible to run this compressor with reduced speed (down to approximately 950 rpm), which decreases the volumetric flow rate of the refrigerant inside the cooling circuit. This leads to a reduced cooling capacity of about two thirds compared to a single-speed compressor with the same displacement. On the other hand it is also possible to increase the cooling capacity by boosting the speed by around 50% compared to a single-speed compressor with the same displacement. This difference in the cooling capacity of a single-speed and variable speed compressor at similar working conditions is shown in Figure 2. With a variable speed compressor the full cooling capacity will only be delivered when required (e.g. if the appliance has a high heat load). While a single-speed compressor lacks this flexibility. Another advantage of the variable speed compressor is the ability to decrease the cooling capacity when the appliance is running under stable conditions and with a low heat load. This lower cooling capacity leads to changed boundary conditions (higher evaporating and lower condensing temperature) in the heat exchangers (see also chapter 2.1 heat exchangers). Due to the dependence between pressure and temperature during the phase-change of the refrigerant in the heat exchangers, the evaporation pressure increases and the condensing pressure decreases. Berliner (1978) showed that by decreasing the pressure ratio for the compressor the losses inside the compressor also decreased.

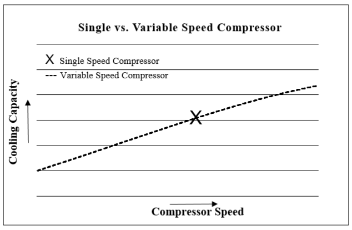

Figure 2 Comparison of the cooling capacity of single and variable speed compressors as a function of compressor speed.

2.3 Expansion Device

The use of capillary tubes in household appliances as expansion devices is common. Plank (1966) showed that an inner heat exchanger between the capillary tube and the suction line of the appliance reduces the energy consumption. The cooling capacity leaving the evaporator is recovered so the efficiency of the appliances increases. Over the course of the last years this inner heat exchanger was improved continuously. For instance by soldering the two tubes together, the heat transfer enhanced. The more effective this exchange is, the better the influence on the energy efficiency of the appliance.

2.4 Insulation of the cabinet

The insulation of the cabinet has been enhanced and therefore the heat flow inside the appliances has been reduced. One way to do this is to increase the thickness of the insulation. Another possibility is to improve the geometries of the gasket between the appliance housing and the door (Pereira, 2016). Furthermore vacuum insulation panels were introduced, which have a lower heat transfer coefficient than the normally used foam in household appliances (Thiessen, 2016). The thicker insulation together with the improvement of the gasket and the introduction of the vacuum insulation panel results in a lower heat transfer inside the appliance and therefore leads to a lower required cooling capacity.

2.5 Summary of improvement of the cooling circuit

The described improvements in chapter 2.1-2.4 lead to a lower energy consumption of the household appliances under normal working conditions. The condensing temperature decreased while the evaporating temperature increased. It is worth to evaluate if the common used refrigerant R600a (isobutane) is still the most suited one for this conditions. To evaluate this, two reference appliance were chosen: a refrigerator and a freezer.
The working conditions of these appliances provide the baseline for the theoretical analysis of different alternative refrigerants. At this points the alternative has to have an advantage in terms of thermodynamic properties, so that it is feasible to reduce the energy consumption of the appliances. The baseline tests are carried out in accordance with the European standard DIN EN 62552:2013-10 (DIN, 2013) and EU-directive 1060/2010 (European Parliament and of the Council, 2010). The working conditions of the two tested appliances are listed below (Table 1). The first appliance is a small built-in refrigerator from Liebherr. This appliance, an IKP 2350, has a net volume of 216 liter and an energy consumption of 70 kWh/a. This leads to an energy efficiency class A+++. The second appliance is a free-standing freezer also from Liebherr. This appliance, a GNP 2713 with a net volume of 221 liter has an energy consumption of 225 kWh/a. The freezer is rated as energy efficiency class A++.

Table 1 Baseline appliance data

Appliance		IKP 2350 (Liebherr)		GNP 2713 (Liebherr)	
Refrigerant		R600a		R600a	
Net volume	Liter	216		221	
Height	cm	122		164	
Depth	cm	55		63	
Width	cm	56		60	
Energy consumption	kWh/a	70		225	
Energy efficiency class		A+++		A++	
Compartment T	°C	5		-18	
Ambient T	°C	25		25	
Evaporating T	°C	-10		-25	
Condensing T	°C	40		33	
Subcooling	K	0		0	
Compressor inlet T / Superheating	°C	25		25	

3. THEORETICAL ANALYSIS OF REFRIGERANTS

The previous chapter described changes of the working conditions of a household appliance over the course of the last years. Since the early 1990s, R600a is used as a refrigerant in these appliances. In the following chapter various alternative refrigerants for R600a are analyzed and evaluated. Different criteria are used to compare serval refrigerants. These are the global warming potential (GWP), the ozone depleting potential (ODP), pressure ratio, specific volumetric cooling capacity and the isentropic compression COP (coefficient of performance). The thermodynamic properties of all analyzed refrigerants are determined using the Nist Refprop Version 9.1 database.

3.1 Potential refrigerants

The refrigerant R600a (Isobutane) is widely used in domestic refrigerators and freezers. Apart from R600a, a lot of refrigerants exist and are used in different cooling applications. Plank (1956), Janke (2015) and Bitzer (2016) provides a detailed overview of the various refrigerants, their chemical and thermodynamic properties and also their field of use. The chemical composition divides the refrigerants into 5 groups (Table 2).

Table 2 Groups of refrigerants (Herr, 2002)

Group	1	2	3	4	5
Name	Fluorocarbon with chlorines and / or bromine	Fluorocarbon without chlorines / bromine	Hydro-fluorocarbon with chlorine	Hydro-fluorocarbon without chlorine	Natural refrigerants
Components	carbon, fluorine, chlorine, bromine	carbon, fluorine	carbon, fluorine hydrogen, chlorine	carbon, fluorine hydrogen	carbon, hydrogen others (e.g. ammonia)
GWP	high	high	medium to high	medium to high	low
ODP	high	0	medium	0	0
Example	R11	R14	R22	R134a	R600a

3.2 Preselection

In Europe nowadays it is not allowed to use a refrigerant with a GWP > 150 for household appliances. This is related to the European Regulation 517/2014 (European Parliament, 2014). Additionally the ODP has to be zero (Montreal, 2000). So if a refrigerant is used inside a domestic appliance in Europe it is mandatory to fulfill the regulations concerning GWP and ODP. Refrigerants of all groups out of the Table 2 are analyzed concerning these criteria and those with an ODP higher than zero and/or a GWP higher than 150 are identified. These are mostly the refrigerants of groups 1, 2 and 3. In a preselection these refrigerants are excluded from further investigations. For instance common substances like R134a (GWP=1300) or R11 (ODP=1) will be not taken into further consideration.

With some refrigerants the pressure inside the system would be higher than 20 bar during normal operation. This would require a special construction of the tubing and heat exchangers to reach the required stiffness of the components. This type of refrigerant (e.g., R744) will therefore not be further investigated. Since water (R718) has a freezing point of 0°C, it is not suitable for domestic appliances and will not be considered further. After the preselection there are seven alternative refrigerants to substitute R600a, which are analyzed in detail. These refrigerants are listed in Table 3.

Table 3 Properties of alternative refrigerants (Herr, 2002)

Name	Group	Component	Molecular formular	GWP	ODP	Boiling point @1,013bar
R600a	natural refrigerant	isobutane	C_4H_{10}	3	0	-12°C
R290	natural refrigerant	propane	C_3H_8	3	0	-42°C
R600	natural refrigerant	butane	C_4H_{10}	3	0	-1°C
R717	natural refrigerant	ammonia	NH_3	0	0	-33°C
R1234ze	hydrofluorocarbon without chlorine	carbon fluorine hydrogen	$C_3H_2F_4$	7	0	-19°C
R1234yF	hydrofluorocarbon without chlorine	carbon fluorine hydrogen	$C_3H_2F_4$	4	0	-30°C
R1270	natural refrigerant	propene	C_3H_6	3	0	-48°C
Dimethyl ether	natural refrigerant	dimethyl ether	C_2H_6O	1	0	-25°C

3.3 Thermodynamic properties – pressure ratio

One important criterion for evaluating the refrigerant is the ratio between condensing and evaporating pressure. (Apart from the pressure ratio there are also other properties of the refrigerant e.g. viscosity, which influences the leakage between piston and cylinder, but this are not taken into account in this theoretical study.) The lower the pressure ratio, the lower the losses inside the compressor. For instance with a reciprocating compressor the leakage through the gap between the piston and the cylinder will become larger by increasing the pressure ratio (Berliner, 1978). To compare the different refrigerants the working conditions out of Table 4 are kept constant.

Table 4 Working condition

Condensing T	°C	40
Subcooling	K	0
Compressor inlet T / Superheating	°C	25

In Figure 3, the pressure ratio of all refrigerants out of Table 3 are shown as a function of evaporating temperature. The working conditions are used out of Table 4. It is clear that with R290 and R1270 the lowest pressure ratios can be achieved. This is advantageous for the refrigerant leakage inside the compressor. The highest pressure ratios are found for R600a, R600 and R717.

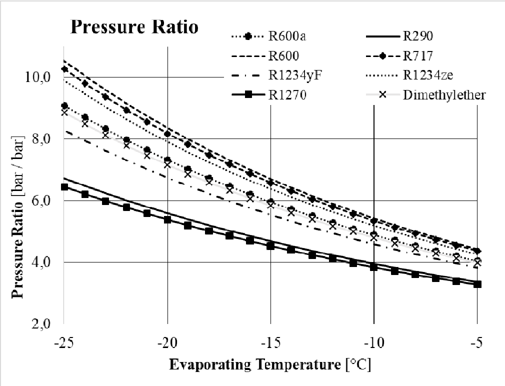

Figure 3 Pressure ratio of different refrigerants

3.4 Thermodynamic properties – specific volumetric cooling capacity

Another important criterion is the specific volumetric cooling capacity of the refrigerant. For this quantity it depends on the specifics of the cooling system whether a higher or lower volumetric cooling capacity is advantageous overall. To compare the different refrigerants the working conditions out of Table 4 are kept constant. In Figure 4, the specific volumetric cooling capacity of all refrigerants out of Table 3 are shown as a function of evaporating temperature. The lowest volumetric cooling capacity is reached with R600. The biggest cooling capacity is achieved with R290, R717 and R1270.

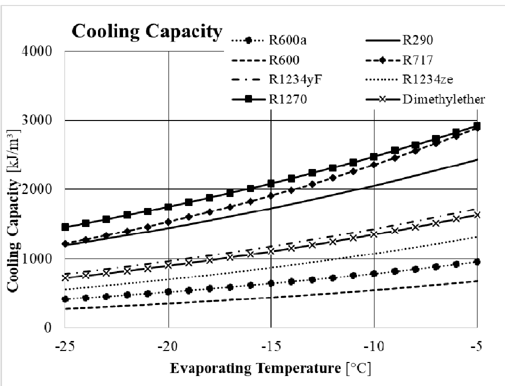

Figure 4 Volumetric cooling capacity of different refrigerants

3.5 Thermodynamic properties – isentropic compression COP

Equation (3) shows the calculation of the isentropic compression COP (COP_{iscom}). Therefore the cooling capacity from chapter 3.4 is used. Furthermore the isentropic work (W_{is}) for compressing the refrigerant from evaporating pressure to condensing pressure is taken into account. This work does not include any losses related to the compressor.

$$COP_{iscom} = \frac{Cooling\ capacity}{W_{is}} \tag{3}$$

To illustrate the isentropic work, two refrigerants (R600a and R600) are compared in the pressure - enthalpy diagram. The working conditions for both refrigerants are the same (Table 4). Also the evaporation temperature is kept constant at -10°C. The comparison of the cooling circuit of the two refrigerants is shown in Figure 5. The black line describes the cooling circuit of R600a and the dotted line the circuit of R600. The isentropic work for R600a is the enthalpy difference between point 2a and 1a in Figure 5. The isentropic work for R600 is the enthalpy difference between point 2b and 1b in Figure 5. It can be shown, that the isentropic works of the two refrigerants are different.

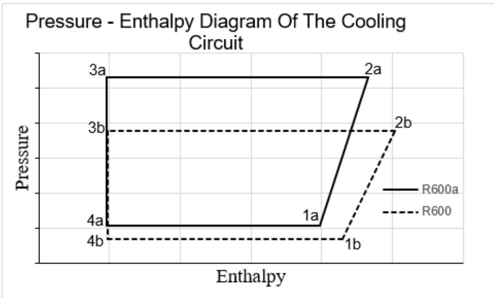

Figure 5 pressure - enthalpy diagram of the cooling circuit (R600a and R600)

In Figure 6 the isentropic compression COP of the refrigerants out of Table 3 are shown as a function of evaporating temperature. Therefore the working conditions out of Table 4 are used. The higher the COP_{iscom} the better the theoretical performance of the refrigerant. The best COP_{iscom} is achieved with R600a and R600, the lowest isentropic COP_{iscom} with R717.

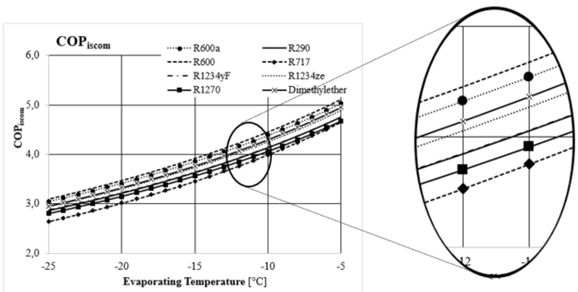

Figure 6 Isentropic compression COP of different refrigerants

3.6 Summary of the theoretical analysis

Due to its lower pressure ratio and its higher cooling capacity, R290 seems to be an alternative refrigerant for all domestic appliances with a high heat load and low evaporating temperature. It has to be taken into consideration that system conditions like condensing and evaporating temperatures will change. Nevertheless it is worth carrying out experimental tests with R290 in the big freezer baseline appliance (GNP 2713).

For appliances with a small heat load, R600 could be an alternative refrigerant. Although the specific volumetric cooling capacity of R600 is lower than the one of R600a, it should be sufficient to cool the baseline refrigerator appliance (IKP 2350). Despite the higher pressure ratio the system conditions (e.g. condensing and evaporating temperatures) should be improved by using R600. In addition the higher COP_{iscom}, is expected to provide a potential reduction in energy consumption of the small refrigerator. In conclusion, experimental tests with R600 in the IKP 2350 appliance are justified.

4. EXPERIMENTAL ANALYSIS OF REFRIGERANTS

As concluded in the theoretical analysis (chapter 3) the freezer GNP 2713 is tested with R290 and the IKP 2350 with R600 as an alternative refrigerant. All energy consumption tests are performed in accordance with the European standard DIN EN 62552:2013-10 (DIN, 2013). The baseline test with the IKP 2350 and GNP 2713 is done in the serial configuration with R600a as refrigerant. After the baseline test the refrigerant is changed to R600 in the IKP 2350 and to R290 in the GNP 2713. The configuration of the appliances apart from the refrigerant is kept as during the baseline test. So the displacement and the speed of the compressor are kept constant.

4.1 GNP 2713 – Freezer appliance

The experimental comparison of R600a and R290 in the GNP 21713 is summarized in Table 5. By changing the refrigerant from R600a to R290 the running time ratio of the compressor decreases. The reason is that the specific cooling power of R290 is higher than the one of R600a. The heat exchangers are the same for all tests. Therefore the U×A-value of the condenser and evaporator can be assumed as constant. Taking equation (1) and (2) into consideration, the constant U×A value and the higher cooling power of R290 result in a higher temperature difference between the refrigerant in the heat exchangers and the surrounding. So also the pressure ratio for the compression increases. This leads to a disadvantage in terms of energy consumption. It can be concluded that the theoretical potential of R290 due to the lower pressure ratio cannot be used in this appliances, because the system conditions change. In the end the energy consumption increases by using R290 instead of R600a in the tested freezer.

Table 5 Comparison of R600a and R290 in GNP 2713

Appliance		GNP 2713	
Refrigerant		R600a	R290
Compressor		Variable speed	Variable speed
Compressor running time ratio	%	65%	37%
Evaporating T	°C	-26,1	-35,8
Evaporating pressure	bar	0,50	1,23
Condensing T	°C	32,8	36,2
Condensing pressure	bar	4,51	10,66
Pressure ratio	-	8,9	8,7
Deviation in energy consumption	**%**	**Baseline**	**+41%**

4.2 IKP 2350 – Fridge appliance

The experimental comparison of R600a and R600 in the IKP 2350 is summarized in Table 6. By changing the refrigerant from R600a to R600 the running time ratio of the compressor increases. The reason for this is that the specific volumetric cooling power of R600 is lower than the one of R600a. The heat exchangers are the same for all tests. Therefore the U×A-value of the condenser and evaporator can be assumed as stable. Taking equation (1) and (2) into consideration, the constant U×A value and the lower cooling power of R600 cause a lower temperature difference between the refrigerant in the heat exchangers and the surrounding. So the pressure ratio for the compression decreases. This leads to an advantage in terms of energy consumption. In addition the higher COP_{iscom} has a positive influence on the performance of the appliance. In the end the energy consumption in the tested refrigerator decreases by using R600 instead of R600a.

Table 6 Comparison of R600a and R600 in IKP 2350

Appliance		IKP 2350	
Refrigerant		R600a	R600
Compressor		Variable speed	Variable speed
Ambient T	°C	25	25
Compressor On-Time	min	12,3	14,2
Compressor Off-time	min	29,6	21,4
Compressor running time ratio	%	29%	40%
Evaporating T	°C	-9,6	-6,9
Evaporating pressure	bar	1,10	0,79
Condensing T	°C	41,7	39,0
Condensing pressure	bar	5,55	3,68
Pressure ratio	-	5,0	4,7
Deviation in energy consumption	**%**	**Baseline**	**-5%**

5. CONCLUSIONS

Over 100 refrigerants were analyzed theoretically to substitute commonly used R600a for household appliances. It is obvious that after a preselection all refrigerants which do not fulfill legal requirements (for the European market) are not further analyzed. These include refrigerants with an ODP > 0 and/or a GWP > 150. Furthermore water is no alternative, because of its high freezing point of 0°C. Also refrigerants, which have a pressure higher than 20 bar at a condensing temperature of 40°C are not able to substitute R600a. For this kind of refrigerants a completely new design would be mandatory, especially because of the higher requirements of stiffness for the heat exchanges. In the end the focus was placed on seven alternatives to R600a: R290, R600, R717, R1234ze, R1234yF, R1270, Dimethyl ether. These refrigerants were compared theoretically by pressure ratio, specific volumetric cooling capacity and isentropic compression COP. As a conclusion of the theoretical study, R290 was found to be a potential alternative to R600a in a household freezer with a relatively high heat load and low evaporating temperature. In a domestic refrigerator with a small heat load and high evaporating temperature R600 is considered as a candidate for substituting R600a as a refrigerant.

In the experimental analysis that followed the theoretical considerations, these two refrigerants were tested and compared to R600a. In a freezer (GNP 2313), R290 results in a higher energy consumption compared to the baseline (~41%) due to the higher specific cooling capacity of R290 compared to R600a and the resulting increase in temperature difference in the heat exchangers.

In the refrigerator (IKP 2350), R600 was tested to substitute R600a. On the one hand the lower specific cooling capacity leads to a higher running time ratio of the compressor. On the other hand the system conditions (condensing and evaporating temperature) improve. In the end the pressure ratio for the compressor in the system is lower than with R600a. Also the higher isentropic compression COP has a positive influence on the energy consumption of the appliance. This results in a decrease of the energy consumption of 5% compared to the baseline system.

The present study shows that the energy consumption of appliances with low heat load and high evaporating temperatures (mainly small domestic refrigerators) can be reduced by around 5% when substituting the refrigerant R600a with R600.

NOMENCLATURE

A	area	(m^3)
COP	coefficient of performance	(–)
GNP 2713	domestic freezer from Liebherr Hausgeräte GmbH	(–)
GWP	global warming potential	(–)
h	specific enthalpy	(J/kg)
IEC	international electrotechnical commission	(–)
IKP 2350	domestic refrigerator from Liebherr Hausgeräte GmbH	(–)
ODP	ozone depleting potential	(–)
R11	refrigerant: fluorocarbon with chlorines	(–)
R12	refrigerant: fluorocarbon with chlorines	(–)
R14	refrigerant: fluorocarbon without chlorines	(–)
R22	refrigerant: hydrofluorocarbon with chlorine	(–)
R134a	refrigerant: hydrofluorocarbon without chlorine	(–)
R290	natural refrigerant: propane	(–)
R600	natural refrigerant: butane	(–)
R600a	natural refrigerant: isobutane	(–)
R717	natural refrigerant: ammonia	(–)
R718	natural refrigerant: water	(–)
R1234ze	refrigerant: hydrofluorocarbon	(–)
R1234yF	refrigerant: hydrofluorocarbon	(–)
R1270	natural refrigerant: propene	(–)
rpm	compressor speed (revolutions per minute)	(1/min)
T	temperature	(°C)
$\dot{Q}$	heat flow	(W)
ΔT	temperature difference	(K)
U	thermal transmittance	($W/m^2/K$)
W	isentropic work for compression	(W)

Subscript

AM	ambient
C	condensing
CO	condenser
EV	evaporator
O	evaporating
is	isentropic
iscom	isentropic compression

REFERENCES

Baehr, H.D. (1978). *Thermodynamik*. Springer-Verlag.

Berliner, P. (1979). *Kältetechnik: Theorie, Systeme, Komponenten, Anwendungen*. Vogel-Verlag.

Bitzer (2016). *Kältemittelreport-19.* Bitzer Kühlmaschinenbau GmbH.

DIN, Deutsches Institut für Normung e. V. (2013), *DIN EN 62552:2013-10, Household refrigerating appliances - characteristics and test methods.* VDE Verlag GmbH.

European Parliament and Council (2010). *No 1060/2010*. Official Journal of the European Union L314.

Herr, H. (2002). *Tabellenbuch Wärme Kälte Klima*. Europa Lehrmittel.

Janke I. (2015), *Alternative refrigerants for household appliances*. Liebherr Hausgeräte Ochsenhausen GmbH.

Liebherr (2002, 2012, 2018). *Domestic appliance brochures*. Liebherr Hausgeräte GmbH Ochsenhausen.

Montreal, 2000). *The Montreal protocol on substances that deplete the ozone layer*. Ozone secretariat united nations.

Pereira P., Sgrott A., Back L. Kohara D., Melo C. (2016). *A methodology for measuring the air infiltration rates into refrigerated compartments.* 16th International Refrigeration and Air Conditioning Conference at Purdue, USA, 2016.

Plank R. (1956). *Handbuch der Kältetechnik – Kältemittel - Vierter Band*. Springer Verlag.

Plank R. (1966). *Handbuch der Kältetechnik – Kältemaschinen - Fünfter Band*. Springer Verlag.

Thiessen S., Knabben F., Melo C., Gonçalves J. (2016). *An experimental study on the use of vacuum insulation panels in household refrigerators*. 16th International Refrigeration and Air Conditioning Conference at Purdue, USA, 2016.

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