Experimental Evaluation of Propane, Propylene and HFC438A as Drop-in Replacements for an R22 System

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Abstract

This paper describes an experimental investigation of the application of hydrocarbons propane and propylene as well as HFC438A as drop-in replacements for R22 in a 15 kW refrigeration system. The experimental system was composed basically of a semi hermetic compressor operated by a frequency inverter, tube in tube heat exchangers and an electronic expansion valve. The tests were performed by replacing the refrigerant without changing any components, not even the lubricant oil, as typical in a drop-in process. The main parameters were varied to verify the range and performance of each refrigerant and then compared to the reference, R22. The natural refrigerants presented the best coefficient of performance–COP–while HFC438A yielded the worst performance, below that of R22. Using TEWI as a benchmark for environmental impact, propane and propylene presented the best results while HFC438A had the worst impact.

Keywords: R22, R290, R1270, R438A, Drop-in, COP, TEWI.
Introduction

The refrigerant HCFC22, or R22, is the most widely used refrigerant in refrigeration systems in developing countries such as Brazil, China, India, among others. Since the Montreal (1987) and Kyoto (1997) protocols, many efforts were made to evaluate replacements for R22 in its various applications. Several obstacles were identified in the search for a new substitute, since the new candidate must be environmentally friendly, compatible with mineral oil and the refrigerant properties should be close to R22 to avoid major system changes.

Due to the latest agreement on the elimination of R22, the price is increasing gradually and the chemical industries are offering a wide range of possible substitutes; however, many of these options have higher GWP values and lower performance than R22. Recent investigations show that the HCFCs and HFCs are being gradually replaced by HFC blends or natural refrigerants (Mohanraj et al., 2009). In general, natural refrigerants such as water (R718), ammonia (R717) and CO2 (R744) have zero ODP and very low GWP values. Furthermore, these natural refrigerants are found in abundance in nature; while the hydrocarbons tend to be less available.

Electrical energy savings are directly related to refrigeration system performance. Rational energy use, combined with control techniques, condition automated refrigeration systems to operate continuously and intelligently for many hours. The use of an electronic expansion valve (EEV) contributes to higher thermal efficiency (Schmidt, 1999). Bandarra Filho et al. (2012) conducted experimental tests in a 9 kW refrigeration system operating in air conditioned temperatures. In this study, the automated system operating with an electronic expansion valve and a frequency inverter presented higher efficiency than the system operating at nominal frequency with a thermostatic expansion valve. The experimental results showed a higher COP when the system operates with R1270 (propylene).
Several studies comparing the thermal performance of synthetic and natural refrigerants were published in recent years. Domanski and Yashar (2006) presented an analytical study on R600a, R290, R134a, R22, R410A and R32 in a refrigeration system used for comfort cooling applications. The study used the system operating with R22 as the basis for comparison, and was carried out without replacing any components from the original system. The authors optimized the heat exchanger models and the system COP was the highest when operating with R290 and R32.

The goal of this experimental study is to compare the performance of R22, R290, R1270 and R438A in an evaporating temperature range from -15°C to -5°C in a system using an electronic expansion valve and a variable speed driver (VSD) linked to the compressor. Preliminary tests with R22 were conducted in order to create a base of comparative data and, after that, only the refrigerant was replaced in the system. The environmental impact was evaluated in terms of TEWI (the Total Equivalent Warming Impact).

**Experimental System**

The experimental system consists of a semi hermetic piston compressor, two concentric tube heat exchangers (refrigerant/water), an electronic expansion valve (EEV) and all the appropriate instrumentation. The analog signals of temperature, pressure and flow rate were converted to digital with a Programmable Logic Controller (PLC).

The data was monitored and managed through an interface created with the software LABVIEW. The secondary fluid, water, circulates through the condenser and a cooling tower. On the other side, the heat transferred to the refrigerant in its evaporation process is purposely generated in a thermal storage tank that simulates a thermal load through electric resistance, which has the function of keeping the desired
water temperature stable at the evaporator inlet. Figure 2 depicts schematically the experimental system used in this study.

The refrigeration capacity was calculated using the First Law of Thermodynamics, Equation 1, considering a steady state condition in which the refrigerant is the only substance present in the control volume delimited by the evaporator.

\[
\dot{Q}_{\text{REF}} = \dot{m} (\Delta h_{\text{EVAP}})
\]

\(\Delta h_{\text{EVAP}}\) is the enthalpy difference between the evaporator outlet and inlet. Piezoresistive pressure transducers (with an average measurement uncertainty of 25.0 kPa) and PT-100 resistance temperature detectors (with an average measurement uncertainty of 0.15°C) were used to measure these properties, thereby enabling the determination of the thermodynamic state of the refrigerant at each point of interest in the vapor compression cycle. A Coriolis flow meter was used to measure the mass flow rate of refrigerant \(\dot{m}\) in the main circuit, with an average measurement uncertainty of 0.0015 kg/s. The compressor's power consumption was measured and the data has an average uncertainty of 0.003 kW. The propagation of uncertainties for the refrigeration capacity, \(\dot{Q}_{\text{REF}}\), and for COP can be estimated and these can be viewed in the results phase.

The experimental facility operated originally with R22 in normal refrigeration applications. The incremental effect of opening the electronic expansion device, the compressor operating speed, the cooling tower operation and the refrigerant charge are controllable parameters in the operation of the system.

The R22 charge used for all tests was 3.2 kg.

Many tests were performed in steady state conditions and the comparisons were made in terms of refrigeration capacity, power consumption, mass flow and COP. The four refrigerant fluids remained in three different evaporation temperatures, -15°C,
-10°C and -5°C. These temperatures were achieved by controlling pressure at the evaporator outlet by modulating the electronic expansion valve opening.

The three selected refrigerants can replace the R22 in this system via a drop-in process, without changing any component nor the lubricant oil. It is important to mention that HFC 438A—a blend of R32, R125, R134a, R600 and R601a—has a hydrocarbon component in its formula, therefore it is soluble in mineral oil. According to Solomon et al. (2007), the GWP for this refrigerant is 2264. Allgood and Lawson (2010) conducted tests with R438A and the results showed good applicability of this refrigerant as a substitute for R22. As for the hydrocarbons, propane and propylene, both can be applied at low and medium temperatures and they have similar thermodynamic characteristics. Colbourne and Suen (2000) showed the advantages of using hydrocarbons when compared with the halogenated refrigerants. The hydrocarbon systems yielded an improvement of 6% in thermal performance for domestic refrigeration, 15% for a commercial application and 8.8% for a comfort cooling application. Park and Jung (2007) concluded that the R290 and R1270 systems had an 11.5% higher COP than R22.

Figure 2 shows the different thermodynamic properties of these refrigerants superimposed on a pressure versus enthalpy diagram. The four isothermal curves (-10°C) present the differences in the evaporation temperature conditions. The latent heat of evaporation values of the hydrocarbons are higher than those of the other refrigerants. This difference has a direct influence on the refrigerant mass flow values.

The first drop-in operation was performed with the hydrocarbon R290. Only 47% of the refrigerant charge adopted for R22 was used, 1.5 kilograms. The second system drop-in, R1270, used the same amount as the R290 system. The final system operation, R438A, worked with 94% of the original system charge, i.e. 3.0 kilograms.

The optimum charge of each refrigerant was previously defined in an arbitrary manner and these charges represent the conditions in which the heat exchangers are
fed correctly. All the tests were performed with approximately two liters of mineral lubricant oil, respecting the level indicated on the reciprocating compressor itself.

During the tests, the parameters responsible for the simulation of thermal load were kept constant for all refrigerants; in other words, the temperature and mass flow rate of the water at the evaporator inlet were maintained at 20°C and 0.35 kg/s.

Safety Criteria

The New Zealand Fire Service publication (2008) refers to an investigation of an explosion and fire in cold rooms of the Icepak company in the city of Tamahere on April 5, 2008. Icepak began to use propane based refrigerants in the installation in January 2003. About 400 kg of LPG, containing approximately 95% propane and 5% ethane, were used at this facility. The injured firefighters’ description confirmed that the origin of the flammable atmosphere in the plant room was a refrigerant leak. The ignition event probably had an electric cause.

The case described above highlights the fact that safety is the most important aspect to observe from the design to the installation, operation and maintenance of any facility, especially with the application of natural refrigerants.

The investment costs for installations using natural refrigerants are typically 20% higher than those using synthetic fluids, depending on the refrigeration system application and capacity.

ASHRAE Standard 34-2010 classifies these hydrocarbons as Class 3 (low toxicity and high flammability). According to standards EN 378 2008 and DIS ISO 5149 2014 parts 1 to 4, covering safety and environmental requirements for refrigeration systems and heat pumps, the HCs are classified as A3 working fluids.
Corberán et al. (2008) summarize in their paper the principal safety standards adopted for the use of hydrocarbon refrigerants. Most of these standards cover the following topics: classification of the type of refrigerant (toxicity and flammability); authorized locations; maximum quantities of refrigerant; construction requirements for the mechanical system and external resources associated with the system (like ventilation and detection of HCs).

Limiting the quantity of refrigerant charge within a refrigerant circuit is one approach to achieve a certain level of safety. Standards EN 378, IEC 60335-2-40 and DIS ISO 5149 2014 seek to incorporate this methodology. In this paper, standard EN378-1 (2008) was used as a reference for calculating the maximum refrigerant charge. The conditions described in Table 1 are sufficient for the adaptation to the test system.

The direct expansion refrigeration system is installed in a lab room, occupied by people, which is not characteristic of a machine room. In addition, the occupancy and supervision are restricted to a particular group of people who are aware that the system is charged with an HC. The maximum refrigerant charge is equivalent to 2.5 kg, while the permissible refrigerant mass must represent the PLV product, where: V is the volume of the room (approximately 180 m$^3$) and PL represents the practical limit of the refrigerant.

The lower flammability limit (L) for R1270 is 0.043 kg/m$^3$, while the limit for R290 is equivalent to 0.038 kg/m$^3$. The minimum ignition energy (MIE) needed is 0.25 mJ. The PL parameter (0.0086 kg/m$^3$) represents the practical limit for avoiding dangerous concentrations, typically equivalent to 20% of L. Therefore, the permissible charge equals 1.548 kg of HC for the test system.

Generic mechanical ventilation limits are offered and the minimum emergency ventilation flow as a function of the refrigerant charge can be calculated with Equation 2.
\[ \dot{V} = 0.0014m^{2/3} \]  \hspace{1cm} (2)

The ventilation flow rate of the laboratory room must be above 0.002 m\(^3\)/s. An installed axial fan provides 20 renewals of the internal air per hour; in other words, it delivers a volume of air equal to 0.85 m\(^3\) per second to the room. Therefore, the safety conditions of the location satisfy the requirements of the standards.

**Results and Discussion**

The correct operation of any refrigeration system operating according to the vapor compression cycle requires that some thermodynamic parameters be monitored and controlled; among them are the evaporation and condensation temperatures, the degree of superheating (measured immediately after the evaporator outlet) and the degree of subcooling.

The power consumed by the compressor was considered a limiting parameter for the operation of the four refrigerants tested. The value of this parameter should not exceed 3.8 kW, thereby preserving the service life of the variable speed drive used on the compressor.

*First Stage Results*

The term drop-in, widely used in the refrigeration sector, refers to the process of substitution of the original refrigerant fluid with another fluid with a different chemical composition. The original fluid must be recovered and a legally correct destiny for this substance must be adopted. The system must also undergo a vacuum process in which a vacuum pump must provide a gauge reading of at least 400 mmHg which must be maintained for four hours. Any alteration of the pressure reading may represent a point of leakage in the line.
The objective of this first analysis is to detail the three drop-in processes applied to the original system. This prior analysis begins with initial standard working conditions in the R22 system, represented in Table 2. Note that these conditions include the compressor operating at a fixed frequency of 60 Hz and the EEV kept at a fixed opening of 90%.

From these initial standard conditions begins the transition to the application of the remaining refrigerants. Table 2 illustrates the results obtained for the three alternative refrigerants replacing R22. All the tests were carried out at a 60 Hz compressor frequency and a fixed 90% opening in the expansion mechanism. Note that the refrigeration capacity values are higher for the hydrocarbons.

**Second Stage Results**

A second analysis of results portraying a drop-in/retrofit type operation became necessary to establish the comparative basis as well as the limits and capabilities of each alternative refrigerant to the HCFC.

In this sense, the four refrigerant fluids operated under different evaporation conditions. Observe in Table 3 that that the cooling capacity values at -15°C stabilize at 6.2 kW for all the refrigerants. When the system operated at -10°C, this capacity level rose to 7.5 kW. Finally, the cooling capacity reached 9.0 kW for an evaporation temperature of -5°C.

Figure 3 illustrates the condensation conditions for the different refrigerants. The estimated condensation temperatures are very similar for the systems containing R22, R1270 and R438A; note that the evaporating temperature and cooling capacity conditions are fixed. The condensation temperatures for R290 are higher than those of the other refrigerants, but this difference is not greater than 5°C. In general, the control performed in the heat exchangers by the EEV/VSD pair made possible a
thermodynamic comparison of the four refrigerants operating practically among the same thermal reservoirs.

The experiments with R438A were performed at 65Hz (1900 RPM). This condition results in the maximum cooling capacity for this refrigerant operating in the system. The R22 and R290 systems operated at 55 Hz, a sufficient level to achieve the same capacity limits of R438A. Finally, propylene worked with 45Hz and satisfied the comparative conditions of this methodology. Table 3 presents the results of the experimental testing stage. The spread of uncertainties due to the accuracy of the sensors was estimated and accompanies the cooling capacity and COP results.

The estimated COP behavior and the (measured) power consumption of the compressor are presented in Figure 4. These figures demonstrate that the systems operating with HCs have an excellent applicability in the evaporation temperature range analyzed, exceeding the COP values obtained with the original system. Regarding the worst power consumption result, a trend was observed in which the system containing R438A operates more efficiently at lower evaporating temperatures; in other words, the COP values of the system with the HFC blend deviate from the COP values of the R22 system as the capacity and evaporation temperature increase.

Figure 5 illustrates the behavior of the superheat and subcooling levels of the four refrigerants. It is important to note that the superheat and subcooling values are distinct for each system. The experiments did not follow a specific technical standard for refrigerants and/or component testing. The methodology adopted in this investigation involved maintaining the same evaporation and condensation conditions, and from there, comparing the system operating with different refrigerants.

Clearly, higher levels of superheat are consequences of lower evaporation conditions, in which the EEV was modulated to reduce the opening of the orifice. Observing
The mass flow rate behavior of the R22 system is similar to that of the R438A system. The R1270 and R290 systems also presented an increase in mass flow values with a five degree increase in the evaporation temperature.

Figure 6 complements the analysis of the heat exchangers, since it is possible to compare the compressor suction and discharge temperatures for each system and test condition.

As mentioned above, the testing methodology does not follow a specific standard; given that, the suction temperatures were not the same for all the refrigerants.

The fluid temperature at the compressor outlet proved to be higher for the system containing R290 in all ranges of evaporation.

The highest discharge temperature values for each refrigerant type are observed specifically at a cooling capacity of 6.2 kW and a -15°C evaporating temperature. This is due to the increase in the pressure ratio as the evaporation temperature is reduced. Despite the reduced power consumption of the compressor, the COP of the system is minimal when compared to all the other evaporation conditions, Figure 4.

**TEWI analysis**

The damage to the environment was calculated based on the TEWI. This method considers the direct and indirect impacts associated with the use of a refrigerant for applications in the HVAC&R sector. Equation 3 represents the methodology used in this calculation.
The part associated with the direct effect can be calculated with equation 4.

\[
\text{CO}_2\text{equ}_\text{DIRECT} = M_{\text{REF}} L_{\text{RATE}} L_{\text{TIME}} GWP + M_{\text{REF}} (1 - \alpha) GWP
\]  

Where:

- \(M_{\text{REF}}\) = Mass of refrigerant in the system, [kg];
- \(L_{\text{RATE}}\) = Annual rate of refrigerant emitted (replacement and leakage), [%];
- \(L_{\text{TIME}}\) = Service life of the system, [years];
- \(GWP\) = Refrigerant specific index, [-];
- \(\alpha\) = End of service life recovery/recycling, [%].

For the calculations, the service life of the equipment was set at 10 years for all of the refrigerants. According to the TEWI calculation methods (Methods of Calculating Total Equivalent Warming Impact, 2012), the annual leaks (from normal operation, catastrophic losses and during maintenance services) were set at 12.5% for a centralized system. 70% was the percentage used for the liquid refrigerant recovery rates (\(\alpha\)). This value represents equipment with a refrigerant charge below 100kg.

According to the IPCC (2007), Intergovernmental Panel on Climate Change (Fourth assessment report), GWP measures the impact of a substance as a greenhouse gas relative to the global warming effect of a similar mass of carbon dioxide over a specified time interval.

The amount associated with the indirect effect can be calculated with equation 5.

\[
\text{CO}_2\text{equ}_\text{INDIRECT} = \beta E_{\text{ANNUAL}} L_{\text{TIME}}
\]  

\(\beta\) = Energy consumption factor.
Where:

\[ E_{\text{ANNUAL}} = \text{Electric power consumption of the equipment, [kWh/year]}; \]
\[ \beta = \text{CO}_2 \text{ emissions for electricity generation, [kg CO}_2/kWh]. \]

The electrical power of the equipment was measured and is shown in Table 3. The total hours of operation were given as the same for all the systems. The CO\(_2\) emissions per kWh of electricity generated is a value taken from the document *CO\(_2\) Emissions from fuel combustion* (International Energy Agency, 2011). Figure 7 shows the analysis of the total equivalent global warming impact for each configuration of the test system.

Energy savings are important to reduce the indirect impact and, consequently, the TEWI. This can be observed by the lower TEWI for the R290 and R1270 systems as compared with the R22 system. It is very important to observe the parameters of CO\(_2\) emissions for electricity generation. Some countries, such as the USA (0.531 kgCO\(_2\)/kWh), have elevated emissions values as a consequence of their energy matrix. The use of a lower GWP refrigerant reduces the environmental damage (direct impact); this fact is observed when comparing the TEWI value for the R438A system (GWP = 2264) with the R1270 system (GWP = 1.8). The energy matrix values for the European Union and Brazil used in the Figure 7 are, respectively, 0.356 kgCO\(_2\)/kWh and 0.075 kgCO\(_2\)/kWh.

**Conclusions**

When comparing the thermodynamic properties of R22 to the proposed alternative fluids, little can be concluded regarding the operability of an immediate drop-in. Some mathematical models contribute positively to the determination of the best replacement choice for R22.
The preliminary analysis of the results demonstrated the real limits of the drop-in process. The tools developed in the second stage of results permitted the application of fluids with distinct thermo-physical properties from the project conditions (capacity, evaporation and condensation) adopted when the original system was designed. These results, obtained through the drop-in/retrofit methodology, show that the use of R1270 and R290 resulted in the highest COP values.

The behavior of the refrigerants (with the same cooling capacity) showed that the R1270 system presented the highest COP for the whole range of evaporation explored in this investigation, i.e. -15°C, -10°C and -5°C. It is interesting to observe that the COP values obtained for R438A are lower than those corresponding to R22.

The use of HCs when compared to R438A and R22 can provide several advantages such as: reduced electricity consumption; reduced refrigerant charge; lower cost of the HCs; considerably low GWP levels and reductions in carbon taxes.

The TEWI comparison showed that superior performance and environmentally friendly processes can be applied simultaneously in order to reduce the direct and indirect effects of global warming. The use of a low GWP refrigerant reduces the direct environmental impacts.

Finally, the most significant technical contribution of this paper to refrigeration systems was to show that, regardless of the refrigerant charge or system capacity, the functionality of the EEV / VSD pair is an essential tool for the adaptation of an alternative refrigerant to the original vapor compression cycle; and in the process, reducing the environmental impact.
Acknowledgments

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References

Allgood, C. C., Lawson, C. C., 2010, “Performance of R-438A in R-22 refrigeration and air conditioning systems”, International Refrigeration and Air Conditioning Conference at Purdue, USA.


ISO 5149 parts 1 to 4, under the general title Refrigerating systems and heat pumps — Safety and environmental requirements; 2014.


## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$\text{CO}_2\text{equ}$</td>
<td>CO$_2$ equivalent, TEWI</td>
</tr>
<tr>
<td>EEV</td>
<td>Electronic expansion valve</td>
</tr>
<tr>
<td>F-gas</td>
<td>Fluorine gas</td>
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<tr>
<td>GWP</td>
<td>Global warming potential</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone depletion potential</td>
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<tr>
<td>VSD</td>
<td>Variable speed driver</td>
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### Special characters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Opening</td>
</tr>
<tr>
<td>$\Delta h$</td>
<td>Specific enthalpy difference</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>Temperature difference</td>
</tr>
<tr>
<td>F</td>
<td>Frequency</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy</td>
</tr>
<tr>
<td>L</td>
<td>Lower flammability limit</td>
</tr>
<tr>
<td>LP</td>
<td>Practical refrigerant limit</td>
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<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
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<tr>
<td>P</td>
<td>Pressure</td>
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<tr>
<td>$\dot{Q}$</td>
<td>Refrigerating capacity</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>Minimum ventilation flow rate</td>
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<tr>
<td>$\dot{W}$</td>
<td>Energy consumption</td>
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Subscripts

CD    Condensation
DC    Discharge
EV    Evaporation
EVAP  Evaporator
LI    Liquid
LV    Liquid-Vapor (latent heat of evaporation)
REF   Refrigerant
SC    Subcooling
SU    Suction
SH    Superheating
Figure 1. Pressure-enthalpy diagram comparing the refrigerants
Figure 2. Schematic diagram of the experimental facility
Figure 3. Experimental results for R22, R290, R1270 and R438A, in terms of saturation temperatures in the heat exchangers
Figure 4. Experimental results for R22, R290, R1270 and R438A, in terms of COP and power consumption of the compressor.

![Figure 4. Experimental results for R22, R290, R1270 and R438A, in terms of COP and power consumption of the compressor.](image)

Figure 5. Experimental results for R22, R290, R1270 and R438A, degree of superheating and subcooling.

![Figure 5. Experimental results for R22, R290, R1270 and R438A, degree of superheating and subcooling.](image)
Figure 6. Experimental results for R22, R290, R1270 and R438A, discharge temperature and suction temperature of the compressor

Figure 7. Comparative analysis of the TEWI for the systems with different operating conditions and locations
Table 1. Limits for the utilization of hydrocarbon fluid refrigerants

<table>
<thead>
<tr>
<th>Location of system parts containing refrigerant</th>
<th>System’s expansion type</th>
<th>Type of occupation (RHPAC supervised)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Space occupied by people, not characteristic of a machinery room.</td>
<td>Direct</td>
<td>Allowable mass: Maximum mass: 2.5 kg or 1 kg (underground)</td>
</tr>
<tr>
<td>Compressor and phase separator tank in unoccupied machine room or outside.</td>
<td>Direct</td>
<td>Allowable mass: Maximum mass: 2.5 kg or 1 kg (underground)</td>
</tr>
<tr>
<td>All the parts containing refrigerant in an unoccupied machinery room or outside.</td>
<td>Indirect</td>
<td>Allowable mass equal to maximum mass: 10 kg or 1 kg (underground)</td>
</tr>
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</table>

Table 2. Experimental results of the drop-in processes, at a fixed frequency of 60 Hz and a fixed 90% opening of the EEV

<table>
<thead>
<tr>
<th>System</th>
<th>$T_{EV}$ [°C]</th>
<th>$T_{CD}$ [°C]</th>
<th>$T_{DC}$ [°C]</th>
<th>$\Delta T_s$ [°C]</th>
<th>$\Delta T_{sr}$ [°C]</th>
<th>$\dot{m}$ [kg/s]</th>
<th>$\dot{W}_{CVV}$ [kW]</th>
<th>$\dot{Q}_{REF}$ [kW]</th>
<th>COP [-]</th>
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<tbody>
<tr>
<td>R22</td>
<td>-3.1</td>
<td>39.1</td>
<td>66.9</td>
<td>8.5</td>
<td>6.2</td>
<td>0.0665</td>
<td>3.45</td>
<td>11.27±0.26</td>
<td>3.27±0.07</td>
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<tr>
<td>R290</td>
<td>1.1</td>
<td>36.9</td>
<td>54.3</td>
<td>2.8</td>
<td>3.9</td>
<td>0.0407</td>
<td>2.92</td>
<td>11.93±0.44</td>
<td>4.09±0.15</td>
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<td>R1270</td>
<td>5.6</td>
<td>35.9</td>
<td>49.0</td>
<td>0.9</td>
<td>5.4</td>
<td>0.0402</td>
<td>2.72</td>
<td>12.30±0.46</td>
<td>4.52±0.17</td>
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<tr>
<td>R438A</td>
<td>-3.1</td>
<td>37.5</td>
<td>74.0</td>
<td>2.1</td>
<td>7.9</td>
<td>0.0654</td>
<td>3.20</td>
<td>9.11±0.19</td>
<td>2.85±0.26</td>
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Table 3. Experimental results of the drop-in/retrofit processes

<table>
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<tbody>
<tr>
<td>R22</td>
<td>55</td>
<td>32.7</td>
<td>95.1</td>
<td>30.8</td>
<td>6.5</td>
<td>0.0328</td>
<td>2.35</td>
<td>6.17±0.28</td>
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<td>R290</td>
<td>55</td>
<td>34.7</td>
<td>82.6</td>
<td>34.1</td>
<td>9.7</td>
<td>0.0174</td>
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<td>R1270</td>
<td>45</td>
<td>32.1</td>
<td>96.0</td>
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<td>R438A</td>
<td>65</td>
<td>31.8</td>
<td>89.8</td>
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Notes:

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