Why Hydrocarbons Save Energy*

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Abstract

Various measurements on hydrocarbon refrigerants have shown they sometimes use over 20% less energy than fluorocarbons. To explain these measurements, we propose new parameters which are functions of well known refrigerant properties. These parameters show that R600a has about half the leakage, pressure loss and condenser pressure and double the heat transfer coefficient of R12 and R134a. Future designs may take advantage of refrigerant properties to minimize energy consumption. They will likely use R600a for car air-conditioners and below about 5 kW capacity, R717 over 50 kW and R290 in between.

1 Environmental Impacts

Air, ammonia, carbon dioxide, hydrocarbons and water are the only refrigerants with no ozone depletion, negligible global warming and low environmental impacts in production. These refrigerants are acceptable to environmental groups and all national governments except the Environmental Protection Agency of the United States. China and Germany have made most progress with these refrigerants but Denmark, Luxembourg and Sweden are now requiring all other refrigerants be phased out.

Global warming from refrigeration equipment occurs both directly and indirectly. When the refrigerant leaks into the atmosphere, the global warming is direct and proportional to the global warming potential of the refrigerant. When carbon dioxide is emitted from burning fossil fuels to provide all or part of the work or

*Published in *AIRAH Journal*, June 1997, Volume 51 No. 6, pp. 33–37.

Table 1: Energy consumption of domestic refrigerators to ISO 7371 with internal temperature 5°C and ambient 25°C.

Make	Model	Refrigerant	Capacity (L)	Consumption (kWhr/24 hr)
UK	A	R12	129	0.75
UK	B	R12	160	0.71
Liebherr	KT1580	R600a	155	0.38
Siemens	KT15RSO	R600a	144	0.52

electricity to drive compressors, the global warming is indirect and depends on energy consumption. If refrigerant leakage is low as in hermetic equipment, energy consumption is the most important parameter in comparing the environmental impact of refrigerants. If refrigerant leakage is large as in car air-conditioners, the global warming potential dominates but energy costs are still of interest.

Air, water and carbon dioxide use least energy with cycles other than the reversed Rankine cycle. It is difficult to separate cycle effects from refrigerant effects so these refrigerants will not be compared here. Ammonia and hydrocarbons however use the reversed Rankine cycle and so will be compared with popular fluorocarbon refrigerants.

Traditionally the equation of state and thermodynamic properties have been used in selecting refrigerants. The energy consumption differences between refrigerants predicted using the equation of state and thermodynamic properties in Section 3 are much smaller than the measured differences in Section 2. Section 3 and 4 show that differences in transport properties and the effect of gauge pressure on required metal thicknesses and leakage are very large and consistent with measurements (Section 2).

2 Experimental comparisons of refrigerants

German manufacturers now use hydrocarbon refrigerant in all domestic refrigerators. Table 1 compares test results by EA Technology, UK (Strong 1994) on UK R12 and German R600a refrigerators.

In February 1995, Email released Australia's first R600a refrigerators with a 16% energy saving over the previous R134a models. They are the Westinghouse Enviro RA142M and Kelvinator Daintree M142C, both 140 L bar refrigerators.

The energy savings obtained by a conversion to hydrocarbons vary considerably with design (Lohbeck 1995).

Petz and Wolf (1995) tested to DIN 8977 a reciprocating piston compressor mass-produced for car air-conditioners with about 5 kW cooling using R12. Their measurements were for 20–50 Hz rotational speed and 58–75°C condensing temperature with refrigerants 12, 134a, 290 and 600a. With R134a the cooling capacity and energy consumption were within 1% of that for R12. With R290 the cooling capacity was about 20% higher and the energy consumption about 5% higher than for R12. With R600a, the cooling capacity was just over half that of R12 but the energy consumption was about 10% lower. These results assume that pressure losses and temperature drops in the evaporator and condenser are the same for all refrigerants. The capacity of car air-conditioners is an order of magnitude larger than for refrigerators. The better efficiency of R600a and worse of R290 is consistent with vapour leakage through valves and piston seals still being important in these compressors.

Abboud (1994) and Parmar (1995) measured the performance of natural HC refrigerants relative to R12 on ten typical Australian cars. The cars were stationary with engines idling and in a shaded and sheltered outdoor position. The superheats measured were smaller for HC as low as 1 K and for some mixtures the condenser pressure was 8% higher. The compressor drive shaft torque was not measured. Instead the relative cooling capacity of the HC mixture to R12 was calculated from the return and supply air states in the passenger compartment and from the compressor speed, pressures and temperatures in the refrigerant circuit. The two measures of the ratio of HC to R12 capacity disagreed sometimes by 20%. The average ratio of HC to R12 cooling capacity was 1.00 with the average energy consumption for HC cooling 13% less than R12. The scatter from differences in charge, ambient and instrumentation was considerable for these results.

German refrigeration mechanics had used commercial propane surreptitiously to replace R22 in heat pumps for many years. Rheinisch/Westfählische Elektrizitätwerke Essen (RWE) field tested several heat pumps with R22 replaced by R290 for two heating seasons. In 1993, RWE emphatically recommended replacing R12, R22 and R502 with R290 in all domestic and small commercial heat pumps for its power savings (Döhlinger 1993). RWE is Germany's largest electricity supplier.

RWE also tested (Frehn 1993) commercial 20 kW water to water and 15 kW brine to water heat pumps in the laboratory with R22 replaced by R290. Table 2 shows R290 reduced heating capacity but increased coefficient of performance (COP) reducing energy consumption. R22 does not benefit from heat exchange between liquid from the condenser and vapour to the compressor but R290 does significantly at the test conditions.

Table 2: Capacity and coefficient of performance increase on substituting R290 for R22 in typical German heat pumps (Frehn 1993).

Type	Temp. (°C)		R22 Performance		R290 % Increment	
	Evap.	Cond.	Heat kW	COP	Heating	COP
WI 24	10	35	22.5	4.2	-10.6	+9.6
WI 24	10	55	20.5	3.1	-16.0	+3.2
SI 17	0	35	15.5	3.4	-9.0	+5.0
SI 17	0	55	13.95	2.49	-15.1	+1.0
With liquid-suction heat exchange for R290						
SI 17	0	35	15.5	3.4	-5.8	+16.2
SI 17	0	55	13.95	2.49	-10.3	+11.6

3 Refrigerant property parameters

Refrigerant properties are known accurately because they are based on physical laws, repeated experiments and physical relations between properties measured independently. Comparing refrigerants by tests on equipment includes all relevant factors and weights them but the results depend on the apparatus. If one piece of equipment is used with all refrigerants, the equipment might favour one refrigerant. Even if the equipment is optimized for each refrigerant before test, the equipment might still favour one refrigerant over another. Comparing refrigerants by comparing a range of property parameters related to the performance of various components avoids these difficulties.

Tables 3 and 4 compare refrigerant properties (Huber *et al.* 1996) and parameters affecting COP for small systems. The REFPROP version used was 5.12 with the highest accuracy equations selected. Saturated vapour is assumed to enter an ideal compressor and saturated liquid to enter the expansion valve except for calculating COP with 20 K suction superheat. ASHRAE (1993), Table 7 on page 16.7 also uses these assumptions. Tables 3 and 4 include the four refrigerants currently used in mass-produced domestic refrigerators and air-conditioners and R290 and R717 which were used in the past and are otherwise environmentally acceptable. Comparing the two tables shows that for different temperatures the relative differences between refrigerants change only slightly.

The paragraph numbers in the following discussion of the ten parameters are the same as the line numbers in Tables 3 and 4:—

1. In the 1920s, open-drive compressors were common for refrigerators (Ta-

ble 3) and a below atmospheric evaporator would cause ingress of air through shaft seals reducing reliability. Car air-conditioners are the only small system with open drives but they operate with evaporators about 0°C (Table 4).

2. When the refrigerator is in storage, the evaporator must withstand pressures which normally occur only in the condenser. The condenser gauge pressures determine metal thickness and hence heat transfer resistance and capital cost. For car air conditioners, the flexible hoses connecting the compressor mounted on the engine to the body components are very sensitive to this parameter.

3. This COP is calculated for a simple reversed Rankine cycle with zero subcooling of liquid, zero superheat of suction vapour and ideal heat transfer and compression. This COP is appropriate for car air-conditioners which are without liquid-suction heat exchange.

   All these theoretical COPs are close to the maximum thermodynamically possible COP for the temperatures (5.74 for Table 3 and 6.07 for Table 4). On small systems, measured COPs are typically less than half the theoretical COP. The other parameters considered here explain a part of this difference.

4. Liquid-suction heat exchange increases COP for some refrigerants and reduces it for others. Domestic refrigerators always use a capillary tube in close thermal contact with the compressor suction line so this COP is the most appropriate.

5. Low compressor discharge temperature allows a cheaper and more efficient design of electric motor in hermetic systems. High discharge temperature means that inlet vapour must cool the motor reducing the refrigerating effect and cycle COP.

6. A large effective displacement implies a larger compressor and higher specific speed. For small systems, the specific speed is very low and any increase improves compressor efficiency. The compressor will still be much smaller than the driving electric motor for small systems.

7. Small refrigerators usually have a serpentine condenser with laminar flow at the beginning of condensation. For condensers of the same length and tube mass but differing diameter and wall thickness, the condenser loss parameter includes all refrigerant properties which contribute to COP loss caused by the pressure drop.

8. Heat transfer by forced convection in the condenser and evaporator tubes of small units occurs mainly by conduction through the thin liquid film on the wall. The usual correlations for this heat transfer (ASHRAE 1993) depend mainly on the ratio of the thermal conductivity of the liquid to its dynamic viscosity, k/μ.

9. For hermetic compressors diffusion through the sealing compounds is a major part of the refrigerant loss which occurs. The size of the molecule increases with liquid molar volume. Large molecules diffuse more slowly so the charge which leaks during the life of the equipment is smaller. With large molecules the charge can be closest to that giving maximum COP throughout the life of the equipment.

10. Significant refrigerant leaks occur typically by laminar isothermal flow through pinholes or cracks. The leakage speed is approximately inversely proportional to the time a complete charge of a given refrigerant takes to leak out.

4 Energy consumption and refrigerant properties

The basic physical differences between the six refrigerants compared in Tables 3 and 4 are their molecular masses and boiling points. R290, R600a and R717 have lower molar mass than the traditional refrigerants R12, R22 and R134a. R600a has another difference from the other five, a high boiling point or low vapour pressure.

For higher capacity systems, R22 was frequently preferred to R12 and in Table 3 the main advantage is the smaller compressor, line 6 in the tables. The significant advantage of R290 over R22 is a liquid k/μ about double, line 8. This appears sufficient to produce a COP increment of about 10% (Table 2).

Car air-conditioners commonly use open drive compressors with evaporators about 0°C (Table 4). R600a has the lowest evaporator pressure but at 0°C it is sufficiently above atmosphere to prevent air ingress. Australian winters are too warm but some countries may require the air be purged from an R600a system once early in spring.

R600a has a condenser gauge pressure, line 2, of about half the lowest of the other five refrigerants. This allows containing metal thickness to be halved and heat exchange area increased for the same capital cost. Such design improvements take time so it is unlikely the full benefit was realized for the tests in Section 2.

For car air-conditioners, R600a has an idealized COP about 2% higher than R134a, line 3 in Table 4, and for refrigerators about 3% higher, line 4 in Table 3. R600a has the lowest compressor discharge temperature, line 5, of the six refrigerants allowing more efficient motors on hermetic systems. For R717 this is 86°C

Table 3: Comparison of refrigerant properties and parameters affecting the measured energy consumption of small systems for an idealized reversed Rankine cycle operating between -15°C and 30°C saturation temperatures. Leading numbers identify comments in the text.

Refrigerant number	12	22	134a	290	600a	717
Chemical classification	CFC	HCFC	HFC	HC	HC	NH_3
x_1 Molar mass (g/mol)	120.9	86.47	102.0	44.10	58.12	17.03
x_2 Refrigerating effect (J/g)	114.5	160.6	148.1	277.9	263.5	1102
x_3 30°C sat. liquid volume (L/kg)	0.774	0.854	0.842	2.064	1.837	1.680
x_4 30°C sat. vapour volume (L/kg)	23.03	19.46	26.67	42.59	95.61	110.5
x_5 30°C sat. vapour viscosity (μPas)	12.79	13.19	12.33	8.68	7.77	10.53
x_6 30°C Condenser pressure (kPa)	746.8	1192	770.1	1079	404.4	1167
1. -15°C Evaporator pressure (kPa)	183.1	296.2	163.9	291.5	88.5	236.2
2. x_7 Condenser gauge $x_6 - 101.3$ (kPa)	645.5	1091	668.8	977.7	303.1	1066
3. COP 0 K suction superheat	4.67	4.66	4.63	4.54	4.67	4.77
4. COP 20 K suction superheat	4.68	4.58	4.68	4.62	4.80	4.56
5. Compressor discharge temp. (°C)	39.1	52.3	36.6	36.2	30.0	86.0
6. Effective displacement (L/kJ)	0.79	0.48	0.82	0.55	1.52	0.46
7. Cond. loss par. $x_7^2 x_4 x_5/(x_2 x_6)$ (μPas)	1.44	1.59	1.29	1.18	0.64	1.03
8. -15°C sat. liquid k/μ (kJ/kgK)	0.292	0.409	0.289	0.785	0.455	2.469
9. Liquid molar volume $x_1 x_3$ (mL/mol)	93.6	73.9	85.9	91.0	106.8	28.6
10. Leakage speed $x_3 x_7/(x_4 x_5)$ (1/ns)	1.70	3.63	1.71	5.46	0.75	1.54

Table 4: Comparison of refrigerant properties and parameters affecting the measured energy consumption of small systems for an idealized reversed Rankine cycle operating between 0°C and 45°C saturation temperatures. Leading numbers identify comments in the text.

Refrigerant number	12	22	134a	290	600a	717
Chemical classification	CFC	HCFC	HFC	HC	HC	NH_3
x_1 Molar mass (g/mol)	120.9	86.47	102.0	44.10	58.12	17.03
x_2 Refrigerating effect (J/g)	105.7	146.6	134.8	251.7	245.3	1046
x_3 45°C sat. liquid volume (L/kg)	0.811	0.904	0.889	2.183	1.907	1.750
x_4 45°C sat. vapour volume (L/kg)	15.59	13.02	17.37	29.23	64.55	72.47
x_5 45°C sat. vapour viscosity (μPas)	13.61	14.11	13.13	9.29	8.23	11.12
x_6 45°C Condenser pressure (kPa)	1088	1729	1160	1534	604.0	1782
1. 0°C Evaporator pressure (kPa)	309.4	498.0	292.7	474.3	156.4	429.4
2. x_7 Condenser gauge $x_6 - 101.3$ (kPa)	986.7	1628	1059	1433	502.7	1681
3. COP 0 K suction superheat	4.78	4.73	4.73	4.60	4.82	5.03
4. COP 20 K suction superheat	4.84	4.70	4.80	4.72	5.01	4.83
5. Compressor discharge temp. (°C)	52.4	64.4	49.8	49.6	45.0	104.3
6. Effective displacement (L/kJ)	0.518	0.319	0.514	0.383	0.961	0.277
7. Cond. loss par. $x_7^2 x_4 x_5/(x_2 x_6)$ (μPas)	1.80	1.92	1.63	1.44	0.91	1.22
8. 0°C sat. liquid k/μ (kJ/kgK)	0.324	0.441	0.326	0.853	0.521	2.795
9. Liquid molar volume $x_1 x_3$ (mL/mol)	98.0	78.2	90.7	96.3	110.8	29.8
10. Leakage speed $x_3 x_7/(x_4 x_5)$ (1/ns)	3.77	8.01	4.13	11.52	1.80	3.65

in Table 3, so a hermetic compressor would require an auxiliary cooling system or cool the motor with inlet vapour. The COP reduction from this depends on the motor efficiency but it could easily be 10%.

The effective displacement is highest for R600a. For car air conditioners and small systems where compressor valve and piston leakage dominate, the high displacement gives R600a a better measured compressor efficiency (Section 2). For systems over 10 kW cooling, a large effective displacement is a disadvantage unless the system is large enough for centrifugals.

For serpentine condensers, line 7, the COP loss due to pressure drop is significantly less for R600a than for the other five. R600a, R290 and R717 are all less than the lowest of R12, R22 and R134a. The k/μ ratio, line 8, is proportional to evaporator and condenser film conductance and is best for R717 but R600a and R290 are both better than R12, R22 or R134a. A doubling of k/μ may give less than a 10% COP improvement with air condensers and evaporators because the R22-R290 comparison above was with water and brine which may have had a larger temperature drop in the refrigerant film.

Lines 2, 9 and 10 of the tables show that leakage with R600a will be less than half of the other five refrigerants. The excess charge can be less than half. For small systems there is a charge which gives an optimum COP and with R600a the charge can be closer to this optimum throughout the life of the equipment. The typical charge for a small R600a refrigerator is only 25 g (Döhlinger 1993, Lohbeck 1995).

For systems with refrigerant reservoirs, like car air-conditioners, the improvement in COP from reduced leakage may be very small. However lower leakage means lower direct emissions and extended service intervals. Lower running cost means more can be spent on capital cost to improve COP.

A refrigerant which can replace another without any hardware changes is sometimes referred to as a 'drop-in'. Blends of R290, R600a and other hydrocarbons are commercially available to replace R12 and R134a and frequently not even the expansion valve needs adjustment. Such 'drop-in' replacements sometimes achieve remarkable energy savings and sometimes not (Section 2). Zeotropic mixtures give temperature glide which often saves energy but is possible with all refrigerants and so does not effect the results of the above comparisons. The full energy savings possible with R290, R600a and R717 can only be achieved with significant hardware changes. This especially applies to R600a and lines 2, 6 and 7 in the tables.

5 Conclusion

Measurements comparing small systems using hydrocarbon and fluorocarbon refrigerants have shown hydrocarbons sometimes save over 20% of energy consumption. Comparison of refrigerant property parameters have shown that R600a is superior to the three most popular fluorocarbons on nine parameters affecting consumption of systems optimized for the refrigerant.

For car air-conditioners and simple systems below about 5 kW capacity, the properties of R600a offer lower energy consumption than R290 and R717 as well as the fluorocarbons. Systems above 50 kW capacity may justify the compressor designs required to achieve lower energy consumption with R717.

Increasing concern about global warming emissions makes the design of new small systems using fluorocarbons seem extraordinarily brave. Corporate planning should instead be how and when to phase these systems out.

6 Acknowledgement

Alan Tayler of the Refrigeration and Air Conditioning Laboratory at The University of New South Wales assisted with refrigerant handling and instrumentation design and construction for the measurements by Abboud (1994) and Parmar (1995).

7 References

Abboud, B., 1994, *Field Trials of Propane/Butane in Automotive Air-Conditioning*, B.E. thesis, School of Mechanical and Manufacturing Engineering, The University of New South Wales, Sydney, 300 p.

ASHRAE 1993, *1993 ASHRAE Handbook Fundamentals*, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta.

Döhlinger, M., 1993, *Comparative Energy Efficiencies of Hydrocarbon Refrigerants*, Ozone Safe Cooling Conference, 18–19th October, Washington DC.

Frehn, B., 1993, *Propan als Arbeitsmittel für Wärmepumpen — die beste Alternative zu R22*, Ki Klima – Kälte – Heizung 10, pp. 402–405.

Huber, M., Gallagher, J., McLinden, M., and Morrison, G., 1996, *NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (REFPROP), Version 5.0 Users' Guide*, NIST Standard Reference Database 23, U.S. Department of Commerce, National Institute of Standards and Technology, Gaithersburg MD.

Lohbeck, W., 1995, editor *Hydrocarbons and other progressive answers to refrigeration*, Proceedings of the International CFC and Halon Alternatives Conference, 23–25th October, Washington DC, published by Greenpeace, Hamburg.

Parmar, A. S., 1995, *Performance of Hydrocarbon Refrigerants in Motor Car Air-Conditioning*, B.E. thesis, School of Mechanical and Manufacturing Engineering, The University of New South Wales, Sydney.

Petz, M. and Wolf, R., 1995, *Performance tests of a vehicle refrigerant compressor with R12, R134a, R290 and R600a*, Hydrocarbons and other progressive answers to refrigeration, editor W. Lohbeck, Greenpeace, Hamburg, pp. 185–194.

Strong, D., 1994, *Natural refrigerants: the next revolution?*, Refrigeration and Air Conditioning, August, pp. 26–27.