

## THE PERFORMANCE OF THE REFRIGERANTS R-134a, R-290, R404A, R-407c AND R-410A IN AIR CONDITIONERS AND REFRIGERATORS

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### ABSTRACT

A wide range of measurements was performed on the performance of the refrigerants R134a, R-290 (propane), R-404A, R-407c, R-410A, and R-22 in a vapor compression experimental set-up. The results were used to evaluate and compare the following performance criteria: cooling capacity, coefficient of performance, compressor discharge temperature, input power, refrigerant mass flow, pressure ratio, and volumetric refrigeration capacity. The measurements were taken over a wide range of evaporating temperatures from -20°C to 20°C at a condensing temperature of 55°C on the same experimental set-up. This wide selection of evaporating temperatures makes it possible to predict the performance of air-conditioning and refrigeration equipment. The results were used to compare the performances of the different refrigerants as possible substitutes for R-22. It was concluded that propane is a good long-term replacement for R-22.

### INTRODUCTION

Environmental concerns repeatedly forced the air conditioning and refrigeration industry into the selections of new and "more appropriate" refrigerants. This was an expensive and complex choice with little actual environmental benefits once ozone depletion was addressed. In this regard, during the last decade, the air-conditioning and refrigeration industry has gone through substantial changes due to the Montreal Protocol (Cavallini 1996; Rowland 1997; Dugard 1997). The Montreal Protocol and Kyoto Accord forced the climate control industry to substitute refrigerants for those that were commonly used. The aim of the Protocol is to replace traditional refrigerants, which are CFCs and HCFCs. The import of CFCs was already banned to most countries in 1996 while imports of HCFCs are progressively restricted, with complete phase-out early in the next century. Although numerous studies have revealed various possibilities, the search for more efficient, cheaper, environmentally friendly, and safe refrigerants is most probably a never-ending one.

One of the most widely used HCFCs is R-22, which represents the largest amount of refrigerant sold (Weiss and Goguet 1997). R-22 is to be phased out relative to a 1989 reference level for developed countries. Production was frozen at the reference level on January 1, 1996. Production will be limited to 65% of the reference level by January 1, 2004; to 35% by January 1, 2010; to 10% by January 1, 2015; and to 0.5% of the reference level by January 1, 2020. Complete cessation of the production of HCFCs is called for by January 1, 2030 (ASHRAE 1997). In addition to the international

agreement, individual countries may have domestic regulations for the earlier phase out of R-22. For example, Sweden and Germany have set the most stringent regulations for the phase out of HCFCs, i.e. in Germany for R-22 by January 1, 2000 (Kruse and Tiedemann 1997).

The search for a replacement for R-22 is a challenging task since no one substitute fits all needs. A few of the medium-term possible HFC replacements mentioned in literature are: R-134a, (Sanvorderdenker 1997), R-404A (Weiss and Goguet 1997), R-407C (Kruse and Tiedemann 1997), and R-410A (Kruse and Tiedemann 1997; Keller *et al.* 1997). However, the unknown long-term future of the HFCs due to their global warming potential has caused the manufacturers of unitary equipment in central Europe to employ natural refrigerants, such as R-290, which is propane (Keller, *et al.* 1997), in unitary air-conditioning and small heat pumps. The most important concern regarding the adoption of propane, is its flammability. While this is an emotive subject which conjures up visions of fireballs and dramatic explosions, safety concerns can be addressed (Ritter and Chem 1999). More and more studies have also shown propane to be an excellent long-term replacement for R-22 (Meyer 1999; Douglas *et al.* 1999; Keller *et al.* 1997; Purkayastha and Bansal 1998).

The purpose of this paper is to compare experimentally the performance of the following refrigerants to R-22 as base: R-134a, R-290, R-404A, R-407c, and R-410A. Many studies have been conducted previously (Keller *et al.* 1997; Baskin *et al.* 1997; Richardson *et al.* 1996; Wei *et al.* 1997; Spatz and Zheng 1996; Payne *et al.* 1999), where some of these refrigerants have been compared with each other or to other refrigerants. This study differs from these previous studies in the large number of refrigerants tested on exactly the same experimental set-up, the accuracy of the experiments as well as the wide range of evaporating temperatures tested, which are representative of many air-conditioning and refrigeration operating conditions.

### EXPERIMENTAL SET-UP

A schematic outlay of the experimental set-up is given in Figure 1. It was a vapor compression refrigeration and/or heat pump cycle. The compressor was a hermetically sealed, reciprocating type with a nominal cooling capacity of 4 kW. The condenser was a water-cooled fluted tube heat exchanger. A coriolis mass flowmeter with an error of  $\pm 0.1\%$  was used for the measurement of the refrigerant mass flow. Sight-glasses were installed before and after the coriolis flowmeter to

ensure that only liquid flows through it. A filter drier followed and a hand-controlled expansion valve. A water-heated fluted tube evaporator was used and a suction accumulator at the compressor inlet on the low-pressure side to complete the refrigerant loop.

Two main water loops were used, one flowing through the condenser and one flowing through the evaporator. On the condensing side the water loop was connected to a 1 000 liter insulated storage tank connected to a 20 kW chiller. The water flow rate through the test section could be controlled with a hand-controlled valve. The flow rate through the condenser was measured with a coriolis mass flowmeter with an error of  $\pm 0.02\%$ . A similar flow loop was used on the evaporating side, also with a coriolis mass flowmeter and an insulated 1000 liter storage tank, but connected to a 24 kW resistance heater. The water temperatures in both loops could be thermostatically controlled at a constant temperature with an error of  $\pm 1^\circ\text{C}$ .

Temperatures were measured with RTDs calibrated to measure temperature differences with errors less than  $\pm 0.03^\circ\text{C}$ . Temperatures were measured at all the refrigerant and water inlets and outlets of both the condenser and evaporator as shown in Figure 1. At each of these locations, four PT100s were located at the top, sides and bottom of the tube to take care of any circumferential temperature variation. The average temperature of the four values was used as the temperature measurement. Pressures were measured as indicated in Figure 1 with 160 mm dial pressure gauges. On the high-pressure side the gauges were calibrated to an error of  $\pm 5$  kPa and to an error of  $\pm 2$  kPa on the low-pressure side. A kilowatt hour meter was used to measure the electric power input to the compressor to an accuracy of  $\pm 0.5\%$  of the reading.

## EXPERIMENTS

The system was pressurized first with nitrogen up to a pressure of 3 MPa and placed in a hot-water bath for 48 hours at  $70^\circ\text{C}$  and checked very well for leaks. The test area was also kept well ventilated. Tests were initially started with R-22 to set up the base reference under identical operating conditions before the other refrigerants considered were loaded. Every time after one of the other refrigerants was tested, the tests were repeated with R-22 to ensure that the base reference was still constant. The refrigerant charge was loaded until  $5^\circ\text{C}$  of subcooling occurred at ARI conditions ( $7.2^\circ\text{C}$  evaporating and  $54.4^\circ\text{C}$  condensing). A polyolester ISO 68 oil was used for compressor lubrication.

Tests were conducted at different evaporating temperatures from  $-20^\circ\text{C}$  to  $20^\circ\text{C}$  in steps of  $5^\circ\text{C}$  while maintaining the condensing temperature at  $55^\circ\text{C}$ . When  $20^\circ\text{C}$  was reached the tests were repeated in the reverse direction. These cycles were completed five times after which the average of ten values was taken as the measured values. The temperatures were obtained by manual adjustments of the water flow rates and different water temperatures. Care was taken to ensure that no pinch points occurred in the condenser and evaporator. The superheat of the refrigerant was kept constant for all tests at  $10^\circ\text{C}$ . It was possible to keep the evaporating and condensing temperatures constant at  $\pm 0.5^\circ\text{C}$  during tests.

The 400 series refrigerants had temperature glides (typically  $7.2^\circ\text{C}$  for R-407c,  $0.8^\circ\text{C}$  for R-404A and  $0.1^\circ\text{C}$  for R-410A) during condensation and evaporation. In these cases the condensing/evaporating temperatures were calculated by averaging the bubble point and dew point temperatures. The heat transfer to the evaporator and condenser was calculated with the product of the refrigerant mass flow and enthalpy

change of the refrigerant determined with thermodynamic properties obtained from REFPROP (1993). This was compared, for each experiment, to the heat loss from the waterside, which was determined from the water temperature change and water mass flow. The energy balance for all experiments was within  $\pm 3\%$ .

The refrigerants were evaluated and compared by making use of the following parameters: cooling capacity, compressor input power, cooling coefficient of performance (COP), compressor input power, compressor discharge temperature, pressure ratio over the compressor and the volumetric refrigeration capacity. The volumetric refrigeration capacity is the evaporator capacity divided by the swept volume of the compressor.

## RESULTS AND DISCUSSION

The results of the measurements are given in Figures 2 to 8. From Figure 2 it follows that the cooling capacity of R-410A is the highest. It is followed by R-22 and R-404A whose cooling capacities are almost identical, but approximately 6% lower than that of R-410A. Then R-407c follows with R-290, which have similar cooling capacities except at high evaporating temperatures where R-407c has a slightly better capacity than R-290. R-134a has the lowest cooling capacity, which is on average 56% lower than the cooling capacity of R-410A.

Measurements on the compressor input power (Figure 3) show that R-134a needs the lowest energy input followed by R-290, R-407c, R-22, R-404A and R-410A. The compressor inputs required by R-407c, R-22 and R-404A are very similar.

The energy input of R-134a is on average 27% lower than that of R-290 and 61% lower than R-410A.

The cooling coefficients of performance (COP) are given in Figure 4. The heating COPs are not given, as they would theoretically be equal to the cooling COPs plus one. It was found from the measurements that for all cases the heating COPs were equal to the cooling COPs plus a factor of between 0.98 to 1.02. From Figure 4 it follows that the COP of R-134a is the highest although the differences in cooling COP between R-134a, R-290 and R-22 are on average less than 2%. It is followed by R-404A and R-410A, which have cooling COPs of approximately 12% lower than that of R-134a. The cooling COP of R-407c is the lowest, on average 17% lower than that of R-134a.

In Figure 5 the refrigerant mass flows are given as function of different evaporating temperatures while the condensing temperature is being kept constant at  $55^\circ\text{C}$ . The mass flows of R-404A and R-410A are very similar and are the highest when compared to the other refrigerants. R-22, R-407c, R-134a and R-290 follow them. The refrigerant mass flow of R-290 is on average 65% lower than that of R-404A and R-410A.

The compressor discharge temperatures are given in Figure 6. R-22, R-410A and R-407c have high discharge temperatures while the discharge temperatures of R-134a, R-290 and R-404A are much lower.

From Figure 7 it follows that the compressor pressure ratio for R-290 is the lowest. R-404A, R-410A and R-22 follow it closely. On average the pressure ratio of R-22 is only 9% higher than the pressure ratio of R-290. The pressure ratios of R-134a and R-407c are much higher than those of the other refrigerants. The pressure ratio of R-407c is 37% higher than that of R-290. High-pressure ratios are in general detrimental to compressor life as they cause high stresses on the compressor bearings and crankcase. In general the volumetric

efficiency also decreases with an increase in pressure ratio (Stoecker and Jones 1982).

The volumetric refrigeration capacities are a measure of the size of the compressor required for particular operating conditions and are given in Figure 8. The higher the volumetric capacity of the refrigerant, the smaller the size of the compressor will be. The volumetric refrigeration capacity of R-410A is the highest by far. It is followed by R404A and R-22, which are almost the same, and on average 29% lower than the volumetric refrigeration capacity of R-410A. R-407c and R-290 follow with R-134a that has the lowest volumetric refrigeration capacity. The volumetric refrigeration capacity of R-134a is on average 56% lower than that of R-410A.

Although the compatibility of the refrigerants with oils and materials has not been investigated in detail it was found to be compatible with all components of the system. Adverse effects of small traces of corrosive substances (e.g. moisture or sulfur) were not noted on either the compressor or the heat exchangers.

The results of the experiments given in Figures 2 to 8 are summarized in Table 1, together with other relevant criteria such as: effect on the environment, cost, toxicity (ASHRAE 1997), flammability and compatibility with compressor oil and refrigeration materials. The costs of the different refrigerants relative to R-22 are given in Figure 9.

It can be concluded from Table 1 as well as from Figures 2 to 8 that not one of the refrigerants outperformed all the other refrigerants on all the criteria considered. In the short term R-134a and the R-400 series refrigerants are excellent replacements (especially R-404A and R-410A) and in many instances are better than R-22. Propane (R-290) performed marginally better than the other refrigerants considered. Taking into consideration that it has an excellent ozone depletion potential (ODP) and global warming potential (GWP) it is a good long-term alternative for R-22. However, it is important that its flammability potential is addressed (Ritter and Chem 1999).

## CONCLUSIONS

All the refrigerants considered in this paper (R-134a, R-290, R404A, R-407c, R410A) can be used as alternatives for R-22. In the short term R134a is an excellent replacement. However, the global warming potential of R134a is high. Other alternatives are R-404A, R-410A and R407c (in the order given) with R-404A and R-410A performing the best. Propane in general performed marginally better than all the refrigerants considered. It is a good long-term replacement for R-22 as it is very environmentally friendly. The flammability of propane should, however, be addressed.

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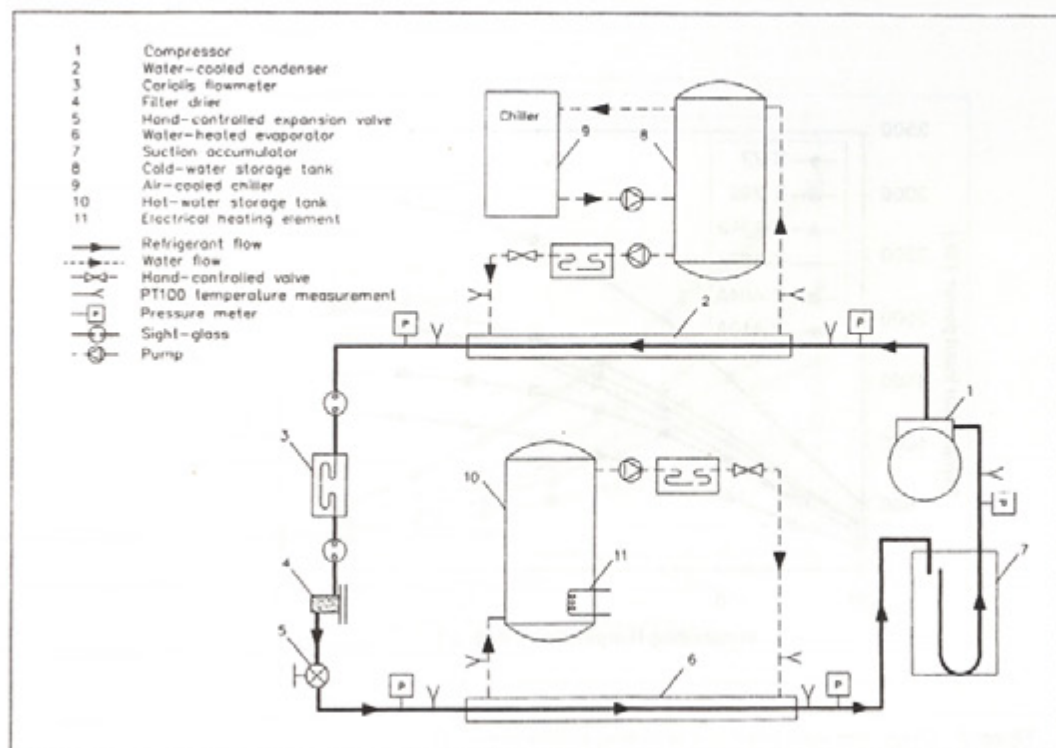


Figure 1: Schematic representation of the experimental set-up.

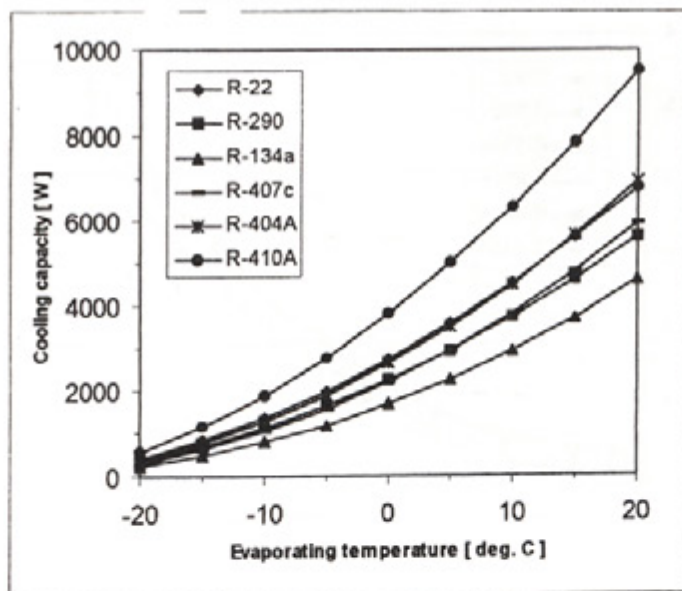


Figure 2: Cooling capacity at a condensing temperature of 55°C.

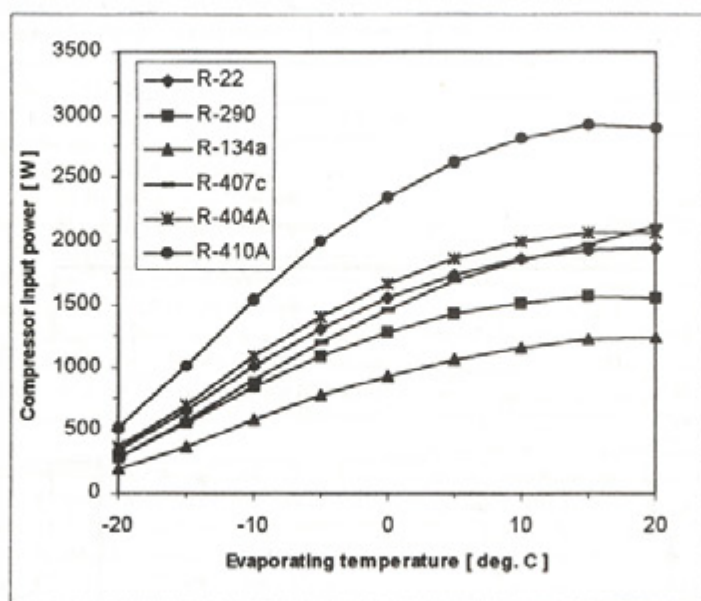


Figure 3: Compressor input power at a condensing temperature of 55°C.

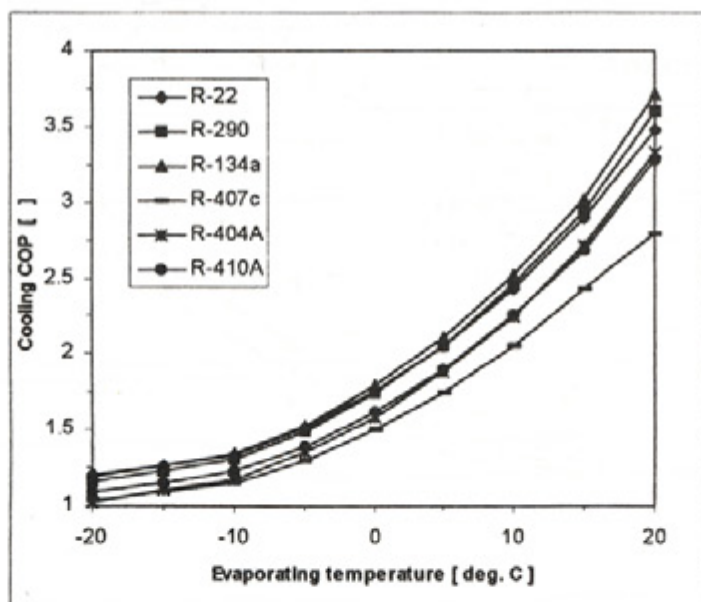


Figure 4: Cooling coefficient of performance at a condensing temperature of 55°C.

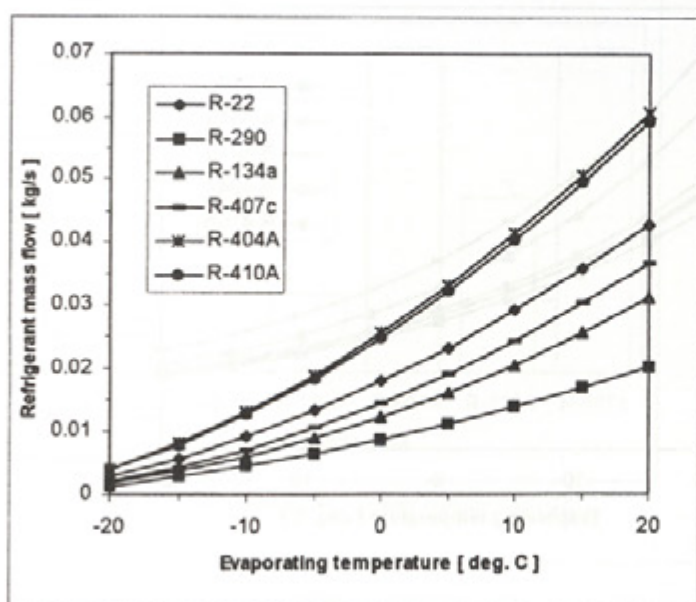


Figure 5: Refrigerant mass flow at a condensing temperature of 55°C.

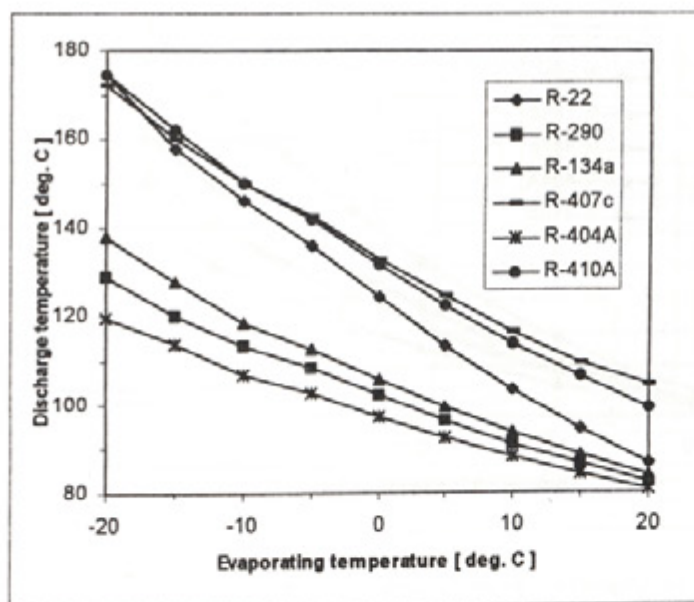


Figure 6: Compressor discharge temperature at a condensing temperature of 55°C.

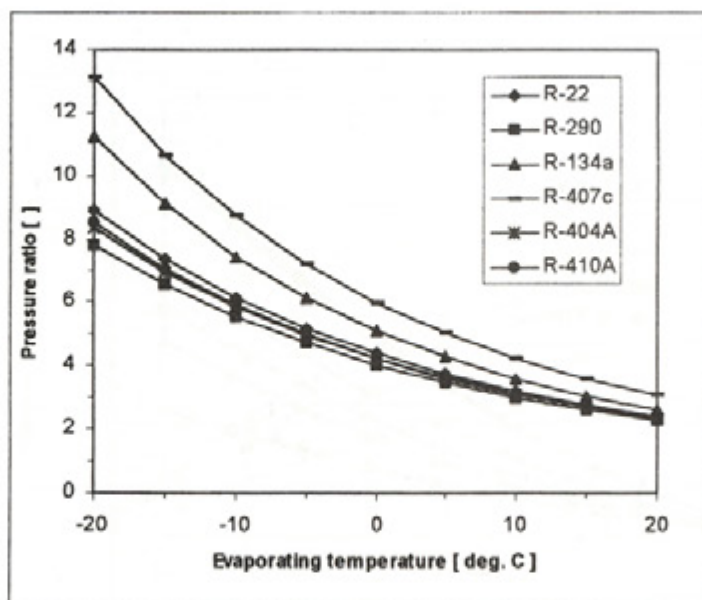


Figure 7: Compressor pressure ratio at a condensing temperature of 55°C.

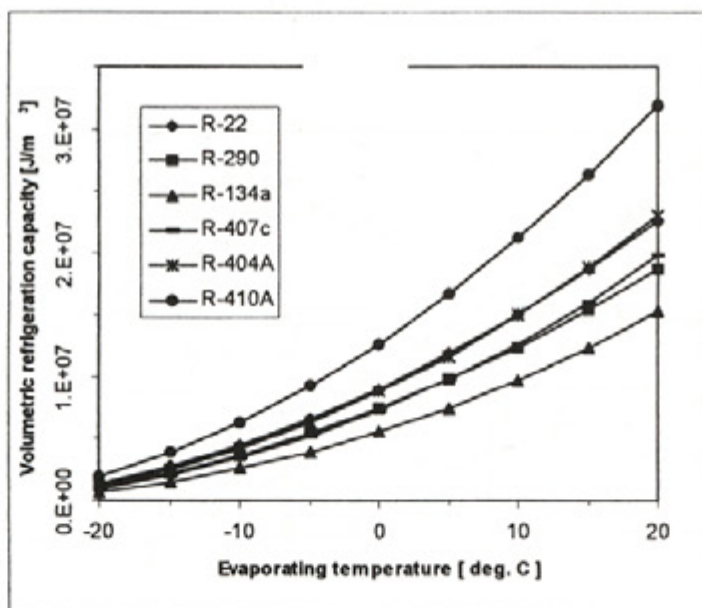


Figure 8: Volumetric refrigeration capacity at a condensing temperature of 55°C.

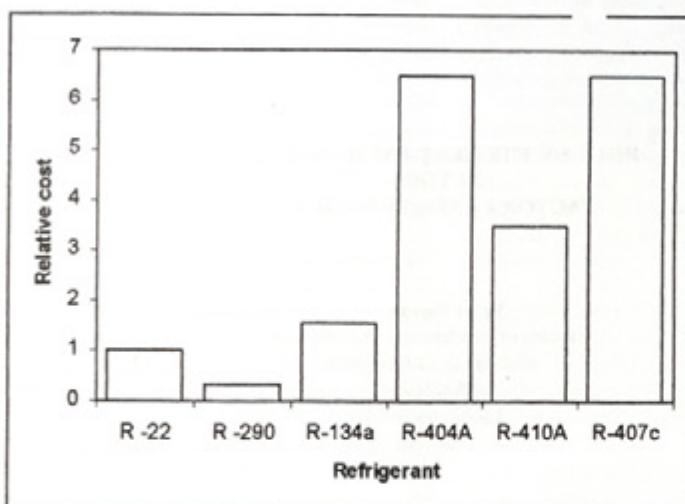


Figure 9: Relative cost per kilogram of refrigerants compared to R-22.

Criteria 1 = poor 5 = excellent	Refrigerant					
	R-22	R-290	R-134a	R-404A	R-407c	R-410A
Cooling capacity	3	2	1	3	2	5
Input power	3	4	5	2	3	1
Cooling COP	4	5	5	3	2	3
Refrigerant mass flow	4	1	2	5	3	5
Discharge temperature	3	5	5	5	3	3
Pressure ratio	5	5	2	5	1	5
Vol. refig. capacity	4	3	1	4	3	5
ODP	1	5	5	5	5	5
GWP	2	5	3	1	2	2
Cost	3	5	2	2	1	1
Non-toxicity	5	5	5	5	5	5
Non-flammability	5	1	5	5	5	5
Compatibility	5	5	5	5	5	5
<b>TOTAL</b>	<b>47</b>	<b>51</b>	<b>46</b>	<b>50</b>	<b>40</b>	<b>50</b>

Table 1: Evaluation and Comparison of Refrigerant Properties