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THE R-134A ENERGY EFFICIENCY PROBLEM: FACT OR FICTION

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ABSTRACT

This paper examines the controversy over the relative energy efficiencies of R-134a and R-12, from a theoretical thermodynamic perspective. In this regard, we have used an in-house process flowsheeting program which allows us to simulate the complete thermodynamic cycle, and investigate the effects of superheat and subcooling. Special attention is given to the suitable basis for comparing the energy efficiencies of different refrigerants calculated from thermodynamic data.

Modelling experiments demonstrate the relative extents to which R-12 and R-134a respond differently to superheat and subcooling. With appropriate superheat and subcooling taken into consideration, such as applied in standard practice in the home appliance industry, R-134a can provide COP values essentially equivalent to that of R-12.

INTRODUCTION

Following the Montreal Protocol, there has been considerable effort in identifying replacement chemicals for commonly used refrigerants. Obviously, in choosing a replacement refrigerant, many factors must be considered. The primary task of the refrigerant producers and equipment manufacturers is the development of safe, energy efficient, reliable, and yet affordable appliances which exhibit minimal effects on the environment.

For several industry sectors, R-134a appears to be the replacement of choice for many R-12 applications. For the American home appliance industry, though, the situation still appears to be fluid, largely because of concerns about energy efficiency. Recently revised DOE standards will require refrigerator/freezers to use 25% less energy in 1993. To meet this challenge, improvements have to be made in every facet of these appliances. The intrinsic thermodynamic characteristics of the refrigeration cycle also affect the total system efficiency, albeit to a lesser degree than is generally publicly cited. Despite its relatively small impact on total performance, it has come under scrutiny. The industry does not want to have to find efficiency elsewhere in the total system to make up for any slip in energy performance of this cycle.

Many calculations based on a simple Rankine cycle have concluded that R-134a is less energy efficient than R-12 [1]. These claims were apparently confirmed by preliminary experimental results which showed efficiencies for R-134a which were 6-10% less than those for R-12, under comparable conditions [2]. The prevailing view was summarized in the UNEP Report [3], which concluded that "choosing R134-a would...incur energy consumption increases estimated to be 8-12% initially and 5-10% after optimization of designs".

However, to our knowledge, there is insufficient evidence to relate the two data sets (theoretical and practical). Generally, the experimental "verifications" were made with unmodified refrigeration systems, which did not take advantage of the specific compression and heat transfer characteristics of the two refrigerant systems. The choice of non-optimal lubricants for the R-134a system, in retrospect, probably also has contributed to poor efficiency performances [2,4]. The agreement of the conclusions of the experimental data and theoretical calculations appear to have been accidentally coincidental.

In 1989, one experimentalist [5] brought together theoretical and practical experimental evidence which challenged this concept of energy penalty. His results, in both laboratory calorimeters and refrigeration units, appeared to demonstrate that R-134a was actually superior to R-12 in terms of energy efficiency [6,7]. Furthermore, he supported his claims with an explanation based on calculations of the modified Rankine cycle which is actually used in hermetic compressor systems. In particular, he included the effect of superheating suction line vapor.

More recently, another experimentalist has also presented theoretical evidence [8] based on "corresponding states". His conclusions were that, differences in heat capacities of two refrigerants could change the relative ordering of energy efficiencies of two refrigerants, especially when subcooling and superheats were considered. In fact, while R-12 appears to be more efficient in optimum ideal (no superheat/subcool) conditions, R-134a appears more efficient in optimum real cycle conditions.

In an attempt to better understand the energy efficiency issues in this field, we have begun to explore both the theoretical and practical aspects of it. This paper describes our initial results in understanding the extent to which the different input parameters can effect the efficiency of the refrigeration cycle. Research programs are also in place which will allow us to experimentally verify the effects of these input parameters, by calorimetry, as well as investigate the effect of lubricants on energy efficiency and wear performance. The results of these latter efforts will be described separately.

MODELLING PACKAGE

An in-house process flowsheeting program was used to calculate the thermodynamic efficiencies of R-12 and R-134a in typical refrigeration cycles. This program, in addition to having a library of unit operations with which to model processes, also has the ability to link in separate programs. In its present use, the program allowed the convenient thermodynamic calculation of the isenthalpic flash, evaporation, compression, and condensation stages (Figure 1). Moreover, it allows for the evaluation of variables, like superheat, subcooling, isentropic efficiency, and time related variables, like flowrates.

The program is linked to a physical property calculation system, which enables the user to select from a range of physical property (estimation) methods and to easily update the physical property data. In the present analyses, the Martin Hou equation of state was used. Together, the package leaves scope to develop the refrigeration model to a more sophisticated one.

CONSIDERATIONS IN THE PRESENT MODEL

This model is used to understand the factors which effect the refrigeration cycle from purely a thermodynamic perspective. While it is within the capability of the modelling package to quantitatively incorporate, for example, heat transfer within the heat exchangers, this has not been done in the present analysis. Obviously, these features can have significant impact on the energy balance and efficiency of the total system. In many cases, this impact has been demonstrated, and provided impetus for improvements in component design. It is recognized that re-design of components will be necessary for R-134a to match the performance of R-12, even under conditions where thermodynamic parity exists.

An isentropic efficiency of 100% was used for both refrigerants throughout these calculations. The actual value used for isentropic efficiency was found not to influence the relative Coefficient of Performance (COP) values, as long as those efficiencies are equal. This view is consistent with recent conclusions of McLinden [8], in which he demonstrated that introduction of isentropic efficiencies does little to affect the relative COP values for different refrigerants, when directly compared.

During the calculations presented in this work, we have assumed that the total superheat energy transferred from various sources to each refrigerant is the same. This assumption is particularly relevant for the examples involving constant cooling capacity, and will be discussed further in that section. At this point, though, it is unclear how far individual systems deviate from this limit, or whether this limit would favor either refrigerant in the present analysis. These are likely to be system dependent. Future calorimetric investigations will be used to examine the validity of this assumption.

RESULTS AND DISCUSSIONS

Comparison at Equal Mass Flowrate

The first basis for comparing the two refrigerants to be considered is on a constant mass or mass flow basis. While easiest to visualize and work through, it takes no account for the different cooling capacities offered by the individual systems. Nevertheless, it does provide a convenient starting point for further discussion. In particular, this section will be used to introduce the effects of superheat.

Many simple calculations do not take account of superheat energy, despite its constant presence in real systems. In the case of hermetic compressors, superheat energy is added to suction vapor from several sources, including suction line heat exchangers, heat from the compressor motor, and heats from the discharge cylinder conducted to the casing to the inlet chamber.

The calculations to compare the thermodynamic efficiencies of R-12 and R-134a are based on the following refrigeration cycle conditions (Figure 2):

Evaporator Temperature:	-23 deg C (-10 deg F)
Condenser Temperature:	+54 deg C (130 deg F)
Mass Flowrate :	1.0 kg/hr (2.2 lb/hr)

These temperatures are consistent with accepted experimental calorimetric conditions, as prescribed by the Association of Home Appliance Manufacturers (AHAM). In many laboratories, refrigerants are judged experimentally under these conditions.

Figure 2 shows the results of calculations made on this basis. Several features concerning the effect of superheat are apparent from this Figure, and it is worthwhile addressing them. First, when no superheating exists (i.e., simplest Rankine conditions), the COP value for R-134a is about 7% less than that of R-12. This is in agreement with published reports which indicate this level of "energy penalty" under these conditions.

Figure 2 also shows that energy efficiency decreases with increasing superheat for both refrigerants. This is a well-known phenomenon within the industry. Higher efficiency compressors are, in fact, designed to transfer less superheat energy to the suction vapor. Beyond this, though, Figure 2 points out rather clearly that R-134a is less effected by inputting superheat energy than is R-12. This results from the differences in heat capacity of the two vapors. The actual range of superheats shown spans the calculated range of superheat energies, calculated from available unpublished calorimetric information. It is interesting to point out here that at some specific superheat energy input, there is a crossing in relative performance, where R-134a becomes relatively more energy efficient that R-12. Under the conditions of this experiment, this occurs at 40 kJ/kg (17.2 Btu/lb).

Comparison at Equal Cooling Duty

Refrigerants are used in appliances designed to achieve a specific cooling duty. Therefore, for a more realistic comparison of refrigerants that are to be used in the same application, the flowrates must be adjusted so that the cooling duties are equal. This enables a comparison of the effect of superheating on the same scale.

For the case of R-12 and R-134a, this means that the flowrate of R-134a must be reduced to match the cooling duty of R-12. For the conditions used in this comparison, the flowrates of 1.131 kg/hr (2.49 lb/hr) of R-12 and 0.954 kg/hr (2.10 lb/hr) of R-134a are required to produce an equivalent cooling duty of 100 kJ/hr (94.8 Btu/hr). This requires an 18% higher volumetric flowrate for R-134a than R-12, based on the saturated evaporator suction conditions (the thermal boundaries are the same as given in the previous section). Current physical property data shows that this difference does not change significantly with increased superheat. This relative volumetric flowrates are also consistent with currently accepted practices for comparing these refrigerants.

The degree to which each refrigerant is superheated will be directly related to the factors discussed previously. For this particular comparison, it is assumed that the superheat energy transferred to each refrigerant is equal. Considering the nature and sources of superheat, and the thermal characteristics of the refrigerants, the amounts of energy transferred in the two cases are likely to be very similar. Fixing this limit does represent a first approximation limit in heat transfer correlations.

Also, under the refrigeration cycle conditions and the range of superheats investigated, the heats of compression (Figure 1, H4/5 - H3) for the two refrigerants are within 5% of each other (the real differences are actually directly reflected in the COP values for the case of equivalent cooling duties). This approximate parity allows us to compare the performance of the two systems on the same graph.

The results of calculations done on this basis and with these limits are given in Figure 3. In this Figure, we also consider the effect of subcooling. Data is given for calculations done with (upper set) and without (lower set) consideration of of this effect. For the sake of reference, our available refrigeration cycle data suggests that, under these conditions, superheat values of 50 to 80 kJ/hr (47.4 to 75.8 Btu/hr) are normally applied in domestic units.

Many of the general observations made previously which relate to the inclusion of superheat are apparent here. The effect of subcooling is also significant. In the absence of subcooling, the relative energy performance has changed. There is a slight change in the slopes of the two COP vs. superheat lines (lower curves), attributable to a slight change in heat capacity of the two flows (lesser R-134a flows are expected to carry proportionately greater superheat loads). The result of this is that, while the two performance curves close significantly over the region of interest, they begin to cross only at superheat energies approximating 100 kJ/hr (94.7 Btu/hr).

The presence of subcooling not only raises absolute efficiencies, but the change is proportionately greater for R-134a than it is for R-12. In this case, the COP values for the two refrigerants are essentially identical over the entire useful superheat range.

The reason for this difference relates to the higher liquid heat capacity of R-134a relative to R-12. In contrast to superheat transfer, subcooling of both refrigerants is considered to be driven to a constant temperature, related to ambient. For any given temperature of subcooling, R-134a will release more energy (the isenthalpic flash line of Figure 1 will move more to the left) than R-12. In this particular example, the upper curves of Figure 3 show the effect of subcooling the liquid condensate to 40 deg C (104 deg F) from 54 deg C (130 deg F; corrections have been made in the flow rates to maintain constant cooling capacities). This is well above the 32 deg C (90 deg F) constraint imposed by the DOE standard; a 14 deg F temperature differential would appear to be sufficiently large to adequately drive the heat transfer operation.

Effect of Operating Temperatures

To further explore the effect of operating conditions on the thermodynamic efficiency of refrigeration cycles, condenser and evaporator temperatures were varied. The results are given in Figures 4 and 5.

Figure 4 was generated to directly consider the effect of the test conditions on the relative energy efficiency. It differs from the lower curves of Figure 3, in that the evaporator and condenser temperatures have been changed. Flow rates of the two refrigerants have also been changed to maintain constant cooling duties.

		Figure 3	Figure 4
Evaporator Temperature:		-23 deg C (-10 deg F)	-21 deg C (-5 deg F)
Condenser Temperature:		+54 deg C (130 deg F)	+43 deg C (110 deg F)
Mass Flowrate	R-12 :	1.13 kg/hr (2.49 lb/hr)	0.99 kg/hr (2.18 lb/hr)
	R-134a :	0.95 kg/hr (2.10 lb/hr)	0.81 kg/hr (1.79 lb/hr)

It should be recognized that there is debate as to whether the AHAM test conditions still accurately reflect the standard operating conditions of most American home appliances. These appliances will be judged by DOE energy efficiency standard, which fix only the ambient (32 deg C/90 deg F) and freezer compartment (-15 deg C/5 deg F). Using currently available heat exchangers, it may be possible to drive the heat transfers with much smaller temperature gradients than those implied by the AHAM test conditions. Taking this into account the performance of the two refrigerants are essentially equivalent. Further, taking into account the presence of additional subcooling (not shown here) would tend to favor R-134a even more.

The effect of operating temperatures is elaborated more fully in Figure 5. These data were generated assuming 75 kJ/hr (71.1 Btu/hr) superheat added to the suction vapor; flow rates were calculated in every case to provide a constant cooling duty. As expected, there is a significant improvement in the COP values of both refrigerants if the condenser temperature is dropped with the evaporator temperature remaining constant. However, this improvement is relatively larger for R-134a than it is for R-12.

Condenser temperatures are not likely to drop much below 40 to 45 deg C (104 to 113 deg F) as long as appliances are being judged at 32 deg C (90 deg F) ambient conditions. Even so, under these conditions, the performance of R-134a appears to compare very favorably with that of R-12.

ACKNOWLEDGMENTS

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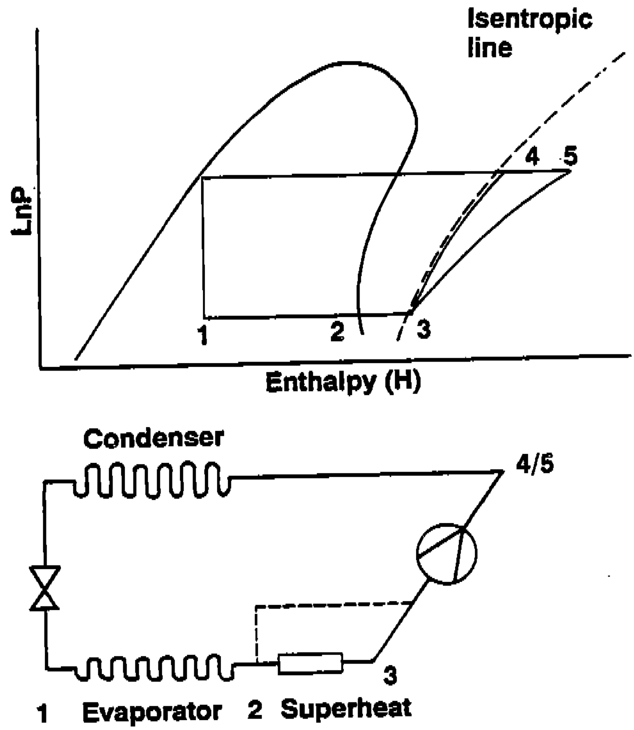
Figure 1. Typical Refrigeration Cycle



$$COP = \frac{H_2 - H_1}{H_4 - H_3} \quad \text{(100\% Isentropic Efficiency)}$$

Figure 2. Comparison of COP values for R–134a and R–12 at equal mass flow rates (see text for conditions)

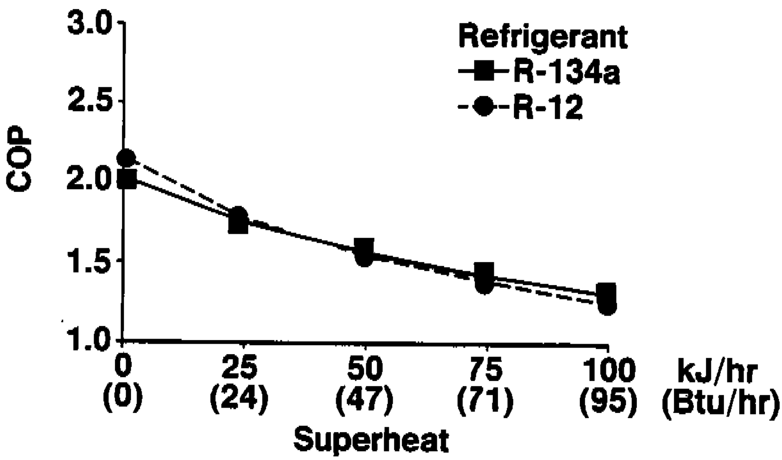


Figure 3. Comparison of COP values for R–134a and R–12 at equal cooling duty (–23°C/+54°C)

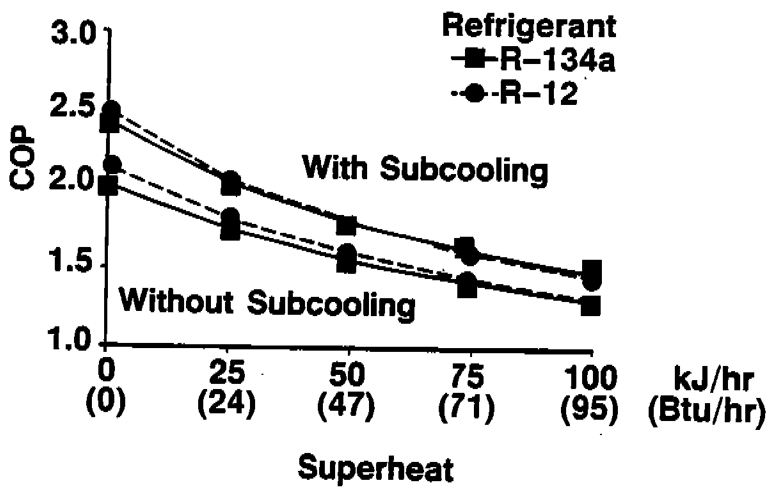


Figure 4. Comparison of COP values for R–134a and R–12 at equal cooling duty (-21°C/+43°C)

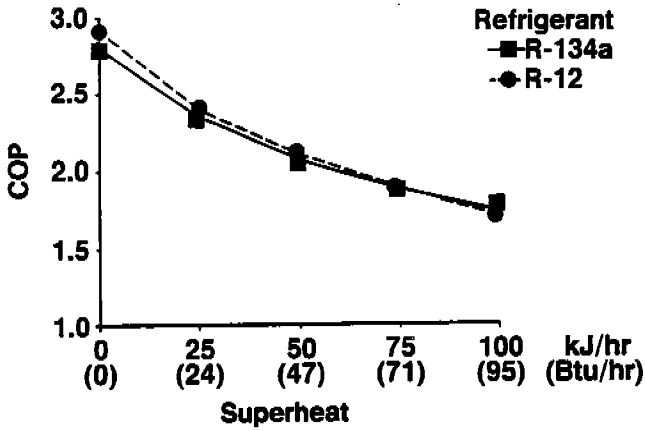


Figure 5. Effect of evaporator and condenser temperatures on the COP values of R–134a and R–12 (with 75 kJ/hr superheat)