COMPARISON OF R134a AND R12 REFRIGERANTS IN A VAPOUR-COMPRESSION SYSTEM

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ABSTRACT

The performance of a vapour-compression system was examined with both R12/mineral oil and R134a/mineral oil charges. The mineral oil was then removed from the system using a multiple flushing method and the experiments repeated using a charge of R134a and an ester-based lubricant to establish the effect of the oil on the performance of the system. Results were compared with theoretical data for R134a and R12 refrigerants.

1. INTRODUCTION

Chlorofluorocarbons (CFCs) are used as working media in vapour-compression systems. Growing awareness of the danger to the earth's protective ozone layer, caused by the build-up of CFCs has stimulated industry's effort to produce "ozone friendly" alternatives. First commercial production of R134a, a zero ozone-depletion-potential (ODP) refrigerant, has been undertaken by I.C.I.[1]. However, there have been conflicting reports regarding the energy efficiency of systems using R134a.

Studies have indicated that a 5-11% drop in capacity and performance would result if R134a were used instead of R12 in a vapour compression system. These investigations were either based on theoretical analyses [2, 3] or experimental work [4, 5] and failed to take into consideration several important factors including the viscosity of the compressor lubricant, suitability of the compressor and degree of vapour superheat.

Refrigerants are soluble or miscible in lubricating oils to varying degrees, depending on the refrigerant and type of oil used. Previous studies have indicated that use of an unsuitable oil can reduce the capacity and performance of vapour compression systems by up to 30% [6]. A range of refrigeration lubricants has been specifically designed for use with R134a. A large family of such lubricants is represented by the Polyol-Esters which can be produced with different miscibility, solubility and viscosity depending on the molecular structure. Initial investigations indicate that where critical lubricant properties are altered, the efficiency of the cycle is affected. This work examines the suitability of R134a as a "drop in" replacement for R12 in vapour compression systems.

2. VAPOUR COMPRESSION CYCLE

The vapour compression system, illustrated in Fig.1, consists of two heat exchangers, a compressor, an expansion valve and a working fluid (e.g. R12 or R134a). In the evaporator heat exchanger, the working fluid evaporates by extracting heat from a source. The vapour is then compressed and gives up its heat at a higher temperature in the condenser heat exchanger. The condensed liquid is then expanded through an expansion valve and returned to the evaporator. This completes the cycle.
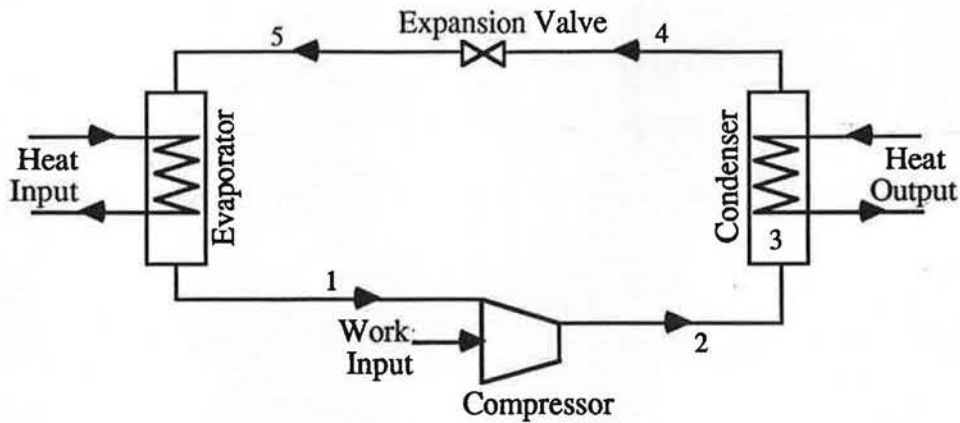
Figure 1. Vapour compression cycle.

The theoretical Rankine cycle can be illustrated with reference to a pressure-enthalpy diagram, as shown in Fig.2. The cycle is as follows:

Process 1-2:

The refrigerant vapour enters the compressor in a dry saturated state before being compressed to a high temperature, high pressure, superheated vapour. This process is assumed to be isentropic.

Process 2-3-4:

The superheated vapour is cooled at constant pressure before it begins to condense at point 2. Between points 3 and 4, condensation continues at constant pressure and temperature.

Process 4-5:

The high temperature, high pressure liquid at point 4 is expanded through the throttling valve to a low pressure and temperature in the evaporator. This process is isenthalpic expansion.

Process 5-1:

The low temperature liquid and vapour mixture passes through the evaporator where evaporation takes place at constant pressure and temperature.

The efficiency of the system is expressed in terms of Coefficient of Performance (C.O.P.):

For a heat pump system: $\text{C.O.P.}_H = \dfrac{\text{Heat rejected at condenser}}{\text{Work of compression}} = \dfrac{h_2 - h_4}{h_2 - h_1}$

For a refrigeration system: $\text{C.O.P.}_R = \dfrac{\text{Heat absorbed by evaporator}}{\text{Work of compression}} = \dfrac{h_1 - h_4}{h_2 - h_1}$

where h is the enthalpy of the refrigerant.

The relation between the two coefficients of performance is given by:

$$C.O.P._H = C.O.P._R + 1$$

Practical vapour-compression systems using the cycle described above differ in several ways from the ideal. The practical system has a pressure drop in the condenser and evaporator coils. The liquid refrigerant leaving the condenser is usually subcooled and the vapour leaving the evaporator is slightly superheated.
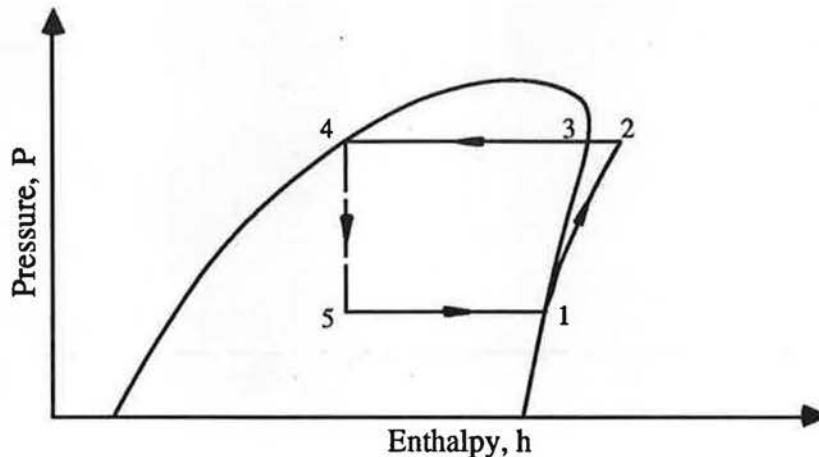

Figure 2. Pressure-enthalpy diagram, ideal Rankine cycle

3. EXPERIMENTAL SYSTEM AND TEST PROCEDURE

A schematic layout of the vapour-compresion system is shown in Fig.3. The experimental system was built using commercially available components designed for use with R12.

The compressor, manufactured by A.P.V. Baker Ltd, U.K., was a twin cylinder, air-cooled, semi-hermetic reciprocating type. The condenser was a finned air-cooled heat exchanger. The evaporator consisted of a concentric coil heat exchanger and was of the counterflow type in which water flowed through the outer tube whilst refrigerant passed through the inner tube. The chilled water then passed through a duct-mounted air cooling coil, returning to the evaporator heat exchanger via a constant level header tank. The airflow over the cooling coil was controlled by a variable speed centrifugal fan which had a maximum capacity of 1.0m3/s. The degree of refrigerant superheat at the outlet of the evaporator was controlled by an internally equalised thermostatic expansion valve. The valve was manufactured by Danfoss, Denmark, and was specifically designed for use with R12. The system employed a liquid receiver at the outlet of the condenser to ensure that there was an uninterrupted supply of liquid refrigerant to the expansion valve. Copper tubing was used to connect the components and a sight glass was used to observe the flow of refrigerant in the system.

Temperatures at various points in the system were measured using type-K thermocouples and the pressures at the inlet and outlet of the compressor were monitored using pressure gauges. The thermocouples were connected to a data logger, type MM300 Multilog, manufactured by E.L.E. International, U.K.. The data logger was in turn connected to an Elonex P.C. 320 personal computer, which managed the data input using "Dialog 300" software supplied by E.L.E. International. The power consumption of the compressor was monitored using a digital kilowatt-hour meter.
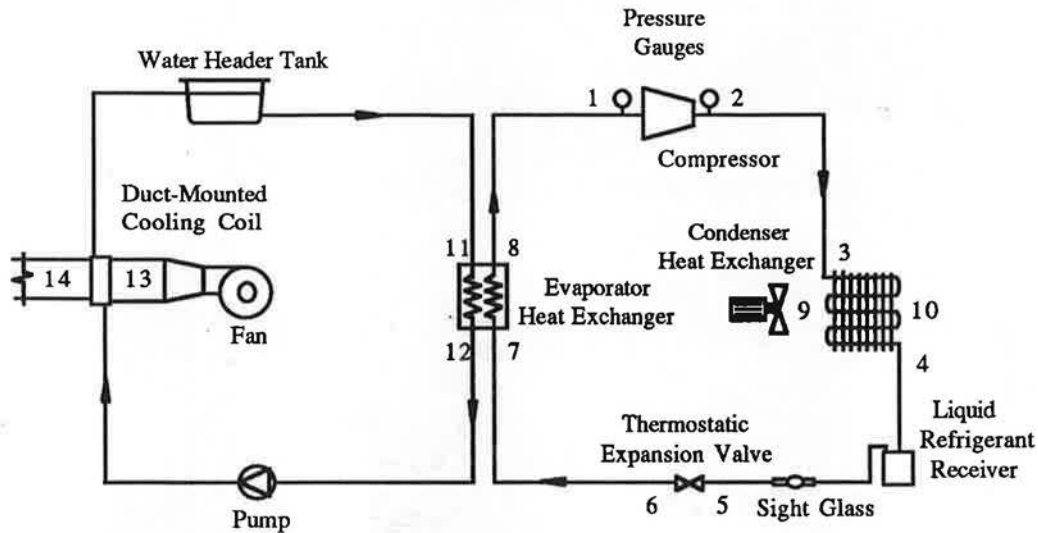
Figure 3. Experimental vapour compression system.

The experimental test procedure was as follows: The system was initially charged with R12 and a mineral oil (Shell Clavus N° 68). With the water pump and fan running, the compressor and data logger were switched on. After steady state temperatures and pressures were reached, the system was left running for a further 5 minutes to allow average steady state conditions to be determined. The test procedure was carried out for two expansion valve superheat settings, with fan speeds between 30% and 100% of the maximum for each superheat value. With the R12/mineral oil charge, steady state conditions were reached in approximately 15 minutes.

The R12 was then purged from the system and replaced with R134a, leaving the original mineral oil in the compressor. The above test procedure was then repeated, with steady state conditions being reached in approximately 20-25 minutes.

After completing the tests with the R134a/mineral oil charge, a multiple flushing method, recommended by I.C.I. Chemicals and Polymers Ltd, was used to replace the mineral oil with an ester based lubricant (I.C.I. Emkarate DE 184). The compressor was first isolated from the system by closing the service valves and the mineral oil was then drained from the compressor and replaced with the ester-based lubricant. The service valves were re-opened and the compressor allowed to run for about one hour to ensure thorough mixing of the ester-based lubricant with any remaining mineral oil. The resulting ester/mineral oil mixture was drained from the compressor and a small sample mixed with ten volumes of methanol. The mineral oil precipitated out of the sample, allowing the proportion of mineral oil to ester-based lubricant to be determined. The compressor was re-filled with fresh ester-based lubricant and the flushing process repeated until less than 1% mineral oil was present. For this system, four flushings were required. As some of the refrigerant was allowed to escape and air allowed to enter the system during the flushing procedure, the system was evacuated and re-charged with R134a.

The test procedure was then repeated with the R134a/ester-based lubricant.charge In this case, steady state conditions were reached in approximately 15-20 minutes.

4. RESULTS AND DISCUSSION

4.1 Theoretical Results

The vapour-compression system was analysed thermodynamically for various condenser and evaporator temperatures. Fig. 4 shows the variation of the Rankine cycle coefficient of performance (C.O.P.$_H$) with condensing temperature (T_c) for evaporating temperatures (T_e) of 260K and 280K. The C.O.P.$_H$ for R134a, over the condensing temperature range of 300K to 345K, was found to be between 2.6% and 9.4% lower than for R12 with an evaporating temperature of 260K, and between 0.3% and 15.6% lower with an evaporating temperature of 280K.
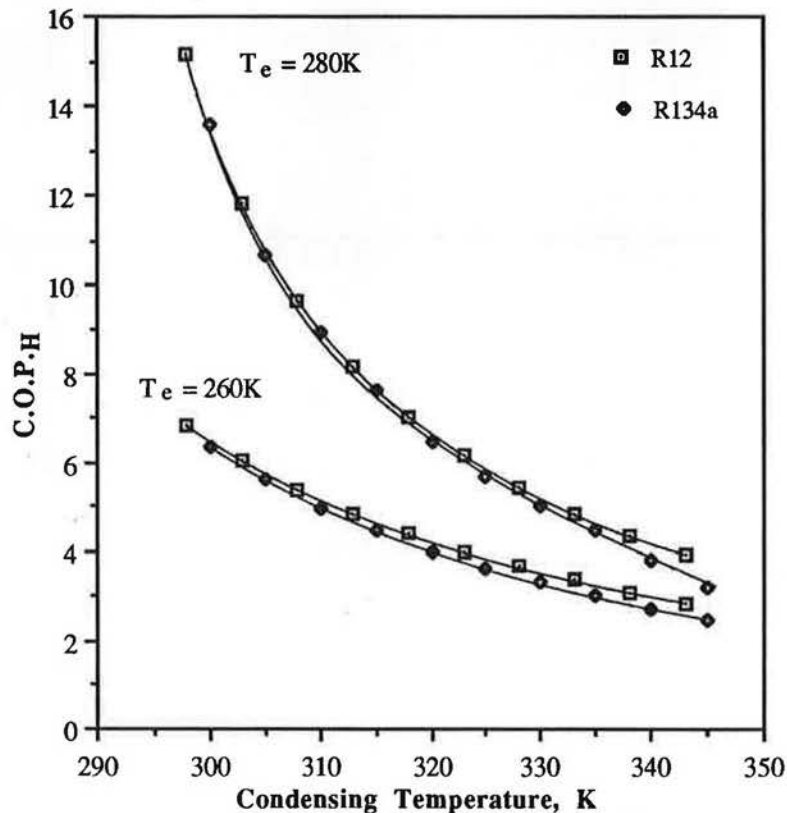

Figure 4. C.O.P.$_H$ vs condensing temperature for R12 and R134a with
evaporating temperatures of 260K and 280K.

4.2 Experimental Results

The average experimental values of C.O.P.$_H$, obtained over eight tests, for the various refrigerant/lubricant combinations and superheat settings are given in Table 1. These values of C.O.P. are based on the ratio of the enthalpy change of the air passing over the condenser coil to the electrical power consumption of the compressor, rather than on the enthalpy changes of the refrigerant; i.e., with reference to Fig.3:

$$C.O.P._H = \frac{m_{air}.(h_{10} - h_9)}{P_{comp}}$$

where m_{air} is the mass flowrate of air passing over the condenser heat exchanger and P_{comp} is the electrical power consumption of the compressor.

Table 1. Experimental coefficients of performance

Refrigerant/lubricant combination	Coefficient of Performance (C.O.P.$_H$)	
	Low Superheat Setting	High Superheat Setting
R12/Mineral Oil	2.152	2.262
R134a/Mineral Oil	1.988	2.140
R134a/Ester Lubricant	2.587	2.592

The experimental test results indicate that in a practical vapour compression system, an increase in performance is possible when the R12/mineral oil charge is replaced by the R134a/ester-based lubricant combination.

Further improvement in the performance of the system may be achieved if the compressor, condenser, evaporator and expansion valve are specifically designed for use with R134a.

5. CONCLUSIONS

The differences in the theoretical C.O.P.$_H$ calculated for R12 and R134a are consistent with values quoted in the literature.

The experimental values of C.O.P.$_H$ obtained with the R134a/ester charge were 20.2% higher and 14.6% higher on the low and high superheat settings, respectively, when compared with the R12/mineral oil charge. This increase in performance can be attributed to the effect of the lubricant, as up to 6.4% less compressor power was required when the system was operated with R134a and the ester-based lubricant.

An increase in performance of a vapour-compression system can be achieved when R134a is employed, providing that the correct lubricant is used.

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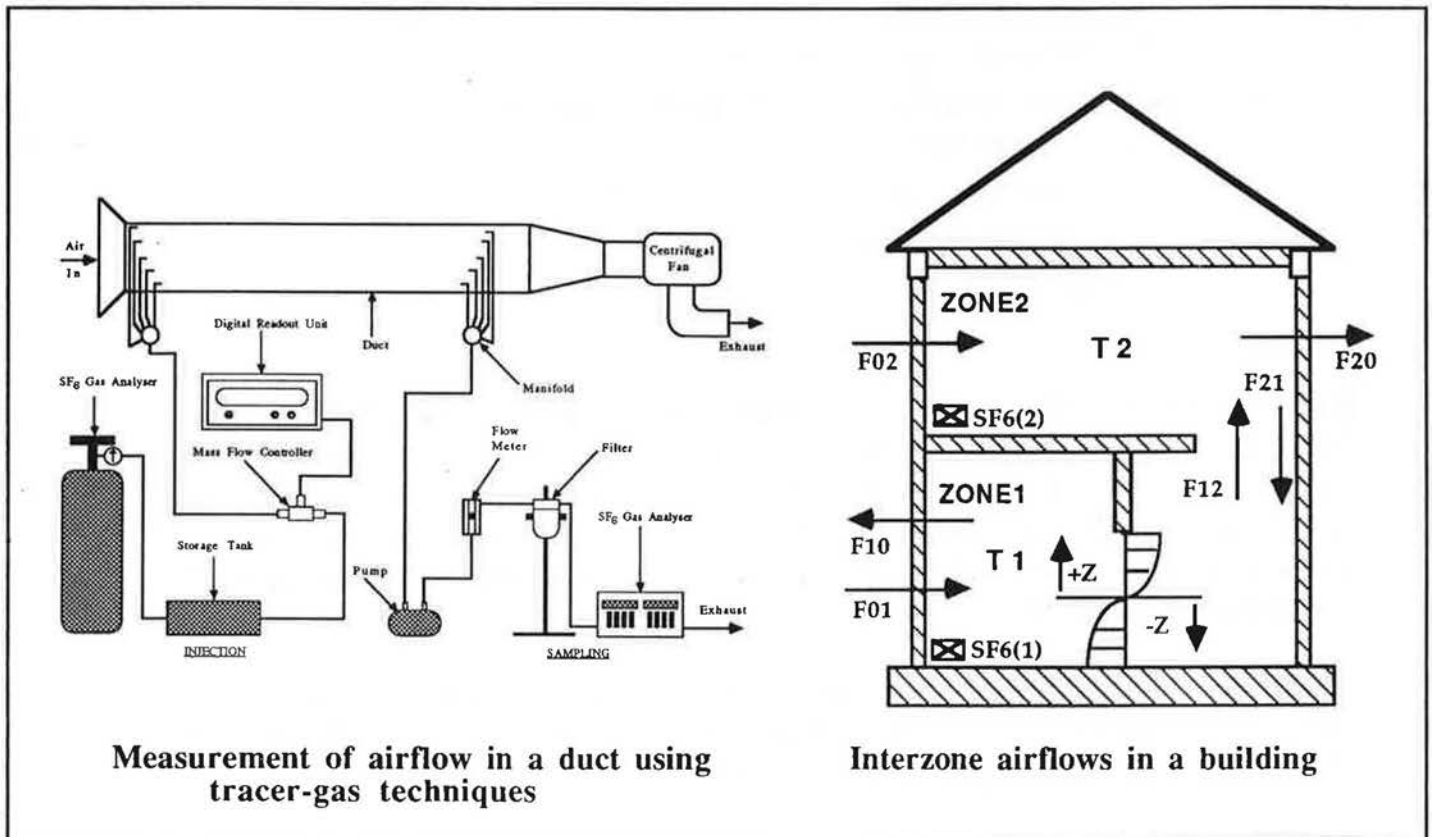
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