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AN EXPERIMENTAL INVESTIGATION OF TERNARY REFRIGERANT BLENDS PERFORMANCE PROPOSED AS SUBSTITUTES FOR CFC-12

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ABSTRACT

In this paper, the performance of various concentrations, of the proposed blends, as alternatives to CFC-12, are evaluated under different operating conditions from thermal capacity, as well as coefficient of performance point of view. Other cycle characteristics, such as pressure ratio, system pressures, temperatures and compressor power are also discussed. This has been accomplished with the help of a fully instrumented brine/water vapor-compression experimental rig.

Results obtained after extensive detailed experimental work, indicated that the performance of the proposed blends, has a strong dependency on the blends concentrations. Furthermore, the proposed blends appear to have superior performance over CFC-12, in both refrigeration and air conditioning applications. On the other hand, it has been observed that the proposed blends experience slightly higher pressures than CFC-12.

1. INTRODUCTION

Although the Montreal Protocol bans CFCs, by the year 2000, however, some nations are in favour of full phase-out, by the year 1997. This definitely calls for accelerated development of new alternative refrigerants. In the past few years, a number of fluids, that may serve as substitutes, either as pure fluids or as constituents of mixtures, have been developed or identified [1-2].

Non-azeotropic mixtures may produce environmentally sound superior working fluids, depending on their components and relative concentrations. At a given composition or concentration, the non-azeotropic mixture condenses and boils, over a temperature range, by an isobaric thermal process. Therefore, a non-azeotropic mixture has a temperature distribution parallel, to that of the surrounding fluid, with which the heat transfer takes place during evaporation and condensation processes. This leads to an improved thermodynamic performance.

Research on the thermodynamic and heat transfer characteristics, and the energy performance of non-azeotropic fluids, is still in its infancy [1-3].

Ternary blends have been suggested, as alternatives for CFC-12, in refrigeration and air conditioning applications. These blends are not "drop-in" replacements for CFC-12, for all applications, but they are viable alternatives, because they provide comparable performance in terms, of both capacity and energy efficiency. They also feature reduced ozone depletion and global warming potential.

Applications for these blends are similar to those targeted for HFC-134a, including: automotive air conditioning, home refrigerators and freezers, chillers, medium-temperature retail food and transportation, dehumidifiers, ice makers, and water fountains.

In addition to being energy-efficient replacements, for CFC-12, these blends offer advantages over other potential alternatives because they:

> Can be used with oils that are currently available; they feature improved capacity compared to HFC-134a, and even CFC-12, in some applications; and are compatible with most materials of construction in CFC systems, thus requiring minimal retrofitting, for use in existing systems.

Computer modelling of theoretical performance characteristics was used to identify the first blend for evaluation. A blend comprised of HCFC-22, HFC-152a and CFC-114 is currently available. It contains all commercially available materials, and offers near-azeotrope performance characteristics. In addition, it has very low toxicity (acceptable exposure limit [AEL] of 1,000 ppm); a reduced ozone depletion potential (ODP) compared to CFC-12 (0.42 vs. 1.0); and a reduced global warming potential compared to CFC-12 (1.61 vs. 2.8); making it a good interim replacement for CFC-12.

Another blend, which contains HCFC-22, HFC-152a, and HCFC-124, is currently proposed. Its refrigeration performance is similar to that of the aforementioned blend, however, it shows a more dramatic environmental improvement over CFC-12, with an ODP of 0.03, and a global warming potential of 0.16.

To the authors knowledge, limited information has been published on the performance characteristics of these refrigerant mixtures, inside enhanced surface tubing.

2. EXPERIMENTAL APPARATUS AND MEASUREMENTS

Figure .1, shows a schematic diagram of the experimental setup, which is a brine/water vapour compression heat pump, composed mainly of: an 8 KW compressor, oil separator, condenser, pre-condenser, pre-evaporator, adjustable expansion device, capillary tubes, and evaporator. The brine solution circulating in the evaporator loop was sodium chloride/water solution at 85/15% mixing ratio. The oil content in the refrigerant loop was estimated to be 1%. Pressure, temperature and flow rate measuring stations are shown in Fig .1. All pressures were measured using calibrated pressure transducers (0-800 kPa). The accuracy of pressure transducers was 2.5%. Differential pressure transducers were employed to measure the refrigerant flow rate. Temperatures were measured by thermo-couples type J and K. Temperature measurements accuracy was within ±1 °K.

All recorded measurements were obtained at a variable sink and source water entering temperatures. On the other hand, the capillary tubes were adjusted to optimize the system's performance with every tested refrigerant mixture concentration. This simulates the system performance using a variable thermal expansion valve.

A calibrated orifice installed in the liquid refrigerant line after a liquid receiver was used to measure the refrigerant mass flow rate. Pressure taps of both orifices were connected to a differential pressure transducer (0-250 kPa). Water mass flow rate was also measured by a calibrated orifice. The accuracy of the mass flow measurements was 3% of the nominal flow.

Power supplied to the compressor was measured because it is needed for the heat balance. An AC/DC clamp-on was calibrated for power measurements with accuracy of ±3%.

Refrigerant composition for each particular mixture was determined. In addition, a liquid sample of each mixture was expanded to superheated vapour, and analyzed by the gas chromatography to accurately determine the overall composition of the mixture prior to the testing.

Data collection was carried out using an AT/PC 286 equipped with a data acquisition system with a capacity of 112 channels. This enabled us to record, at a single scan, local properties such as pressure drops, pressures, temperatures, flow rates and power.

All tests were performed under steady state conditions. The data collection were scanned every one second and stored every 10 seconds. The experimental values were averaged over a period of 10 seconds.

The primary parameters observed during the course of this study were , mass flux, heat flux, and quality for pure refrigerants; R-12 and R-22, as well as non-azeotropic refrigerant mixtures R-22/R-152a/R-114 and R-22/R-152a/R124 at various concentrations. Mass flow rates ranged from 50 to 90 g/s.

During the course of this study, the performance characteristics data for pure refrigerant R-12 is presented, as reference data base for comparison purposes.

In order to evaluate the blends performance, the thermodynamic properties of pure and non-azeotropic refrigerant mixtures, should be known. The Carnahan-Starling-DeSantis (CSD) equation of State [4] was used to evaluate the mixture characteristics. On the other hand, the mixing rules suggested by Reid et al. [5] were employed, with caution, to determine the transport properties of the mixed refrigerants.

3. RESULTS AND DISCUSSION

In the following sections, the results of the performance of the ternary mixture at different conditions will be presented and discussed. The test conditions were: condenser pressure, varied between 500 to 2500 kPa, condenser refrigerant temperature was between 20-45° C. Evaporator pressure ranged from 50 to 600 kPa, and the refrigerant evaporator temperature was between -15 to 10° C. Under these conditions, the following parameters have been measured: thermal capacities at evaporator and condenser sides, power consumed by compressor, ternary mixture flow rates, ternary mixture concentration at various points of the heat cycle, and at particular the heat exchangers (condenser and evaporator), coolant flow rates, and mixture quality at evaporator side.

The aforementioned, measured/calculated parameters, such as consumed power and thermal capacities at the evaporator and condenser, are necessary for evaluating the COP, under heating and cooling loads:

$$COP = \frac{Heat\ Absorbed\ /\ Released}{Compressor\ Power} \tag{1}$$

The heat absorbed/released are calculated after:

$$Q_{a/r} = \dot{m}_f \times C_{p.f} \cdot \Delta T \qquad (2)$$

Where $\dot{m}$ and ΔT represent the coolant (water and/or brine) mass flow rate, and the difference of coolant temperatures across the evaporator/ condenser coils. $C_{p,L}$ is the specific heat for coolant fluid.

Upon completion, of the base line results of R-12, under the aforementioned conditions, the compressor and the system wall drained and evacuated. The system was then recharged, with the preferred mixture concentration. This procedure has been repeated, before conducting the series of tests for every single mixture concentration.

In the following, the direct dependence of the input power, thermal heating or cooling capacity, pressure ratio, relative compressor power, condenser pressure, relative heating capacity COP, and relative COP on the temperatures of evaporator, and condenser will be outlined.

In the first series of experiments, the sink water temperature was kept constant at 16° C, while the source coolant temperature varied from -15° C to 25° C, at the evaporator inlet. The obtained results during these runs were plotted in Figures 2 through 4, at various refrigerant mixture concentrations, and compared to R-12. As expected, the results show that cooling capacity and COP cooling, as well as the compressor power increase at higher brine solution temperature, at the evaporator. On the other hand, Figure 3, gives clear evidence that the pressure ratio decrease at higher temperatures at the evaporator. Furthermore, these results clearly indicate that the proposed refrigerant mixture concentrations have superior properties over R-12. In particular, the refrigerant mixture KC503020, R-22/R-152a/R-124 with concentrations 50/30/20 %, has the most promising performance as a R-12 substitute, among other concentrations and R-12. In addition, it has a permissible pressure ratio over the range investigated of the cooling fluid temperatures.

Another series of experiments have been carried out to study the impact of varying the temperatures of the cooling fluid of the condenser at the system performance. Results obtained during the course of these experiments, have been displayed in Figures 5 through 7, and compared with those of R-12. Under these conditions, it is quite clear from these figures, that the refrigerant mixture KC503020, R-22/R-152a/R-124 at concentrations of 50/30/20%, has superior performance characteristics over other mixture concentrations including that of R-12. Furthermore, Figure 7, demonstrates that this particular mixture concentration has also an acceptable pressure ratio, over the range investigated.

As this point, the relative coefficient of performance (COPR) cooling is defined as;

$$COPR = \frac{COP_x - COP_{R12}}{COP_{R12}} \qquad (3)$$

Where COP_x and COP_{R12}, is the coefficient of performance cooling for particular mixture under investigation and, R12 respectively.

The relative coefficient of performance represents the enhancement of the proposed mixture concentration, over that of R-12. In order to demonstrate on the functional dependence, of the inlet evaporator coolant temperature on the relative COPR cooling, Figures 8 and 9 have been constructed. The COPR of R-12, in these figures, has been assigned a value of zero. As shown in the figures, it is quite evident that at constant liquid condenser temperature (16° C), the mixture KC 53020, has the most promising performance and the higher relative COPR cooling over the investigated mixture concentrations, and R-12. However, Figure 9, shows that varying the condenser cooling coolant temperature results in degrading the performance of the mixture KC503020. Under higher cooling fluid temperature (>27° C), the mixture KCD94433A would be recommended as a substitute for R-12.

4. CONCLUSIONS

During the course of this experimental study, the performance characteristics of the newly proposed ternary refrigerant blends, as substitutes for R-12, have been investigated and compared to that of R-12. One particular mixture composition has demonstrated a superior performance over that of R-12, over the range investigated.

5. ACKNOWLEDGEMENT

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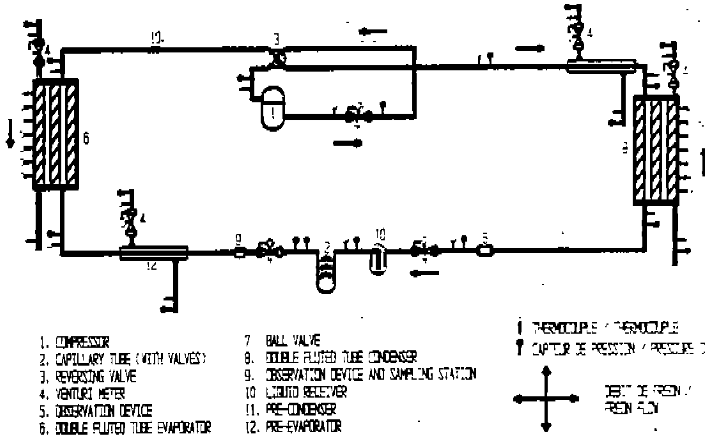
FIGURE 1 SCHEMATIC VIEW OF THE EXPERIMENTAL SETUP



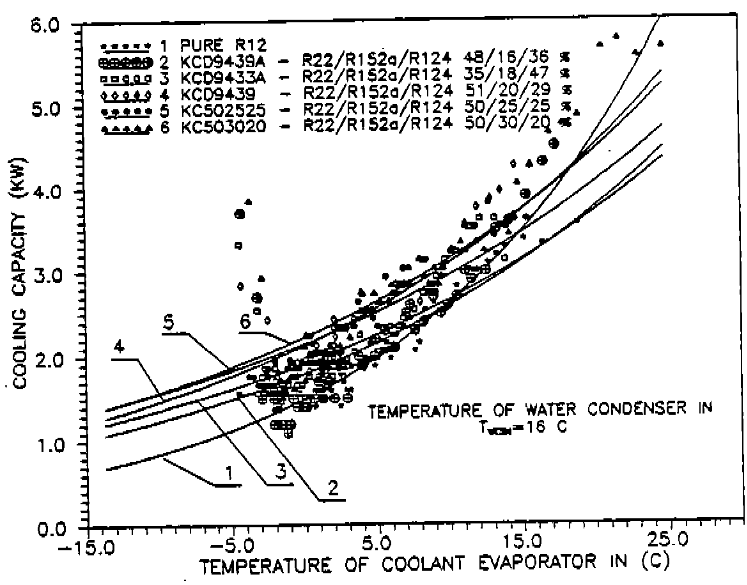


FIG.2 COOLING CAPACITY VS. TEMPERATURE OF COOLANT EVAPORATOR IN.

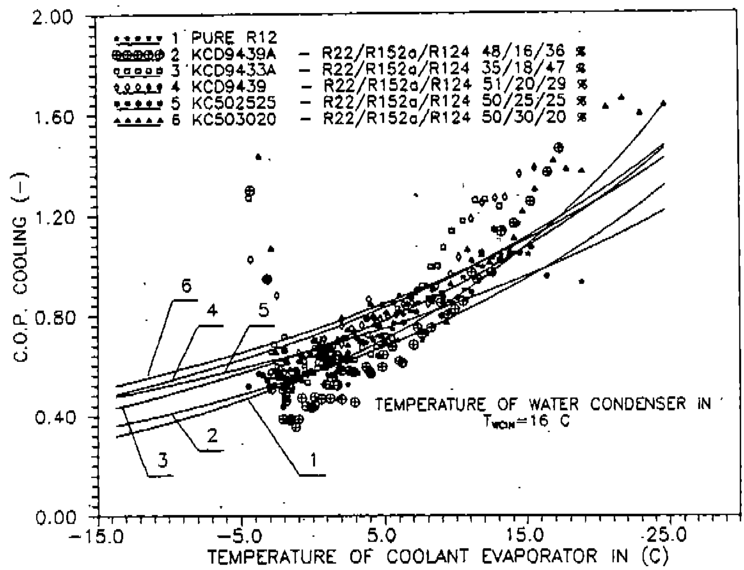


FIG.3 C.O.P. (COOLING) VS. TEMPERATURE OF COOLANT EVAPORATOR IN.

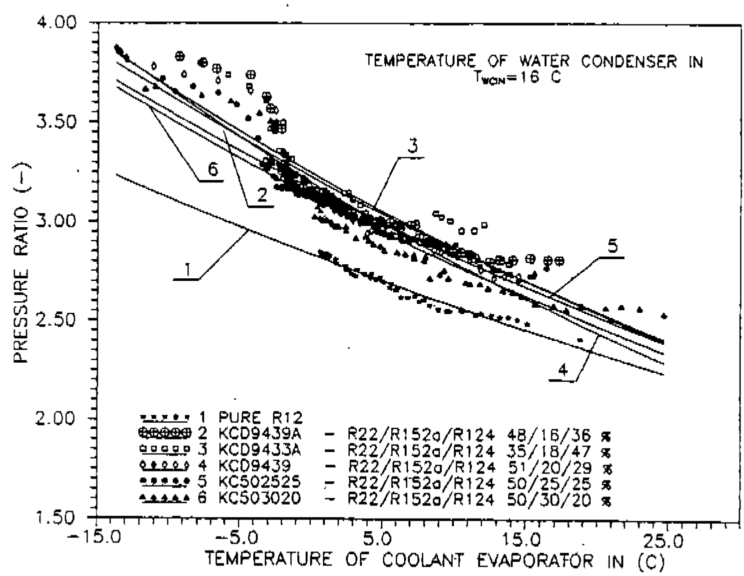


FIG.4 PRESSURE RATIO VS TEMPERATURE OF COOLANT EVAPORATOR IN.

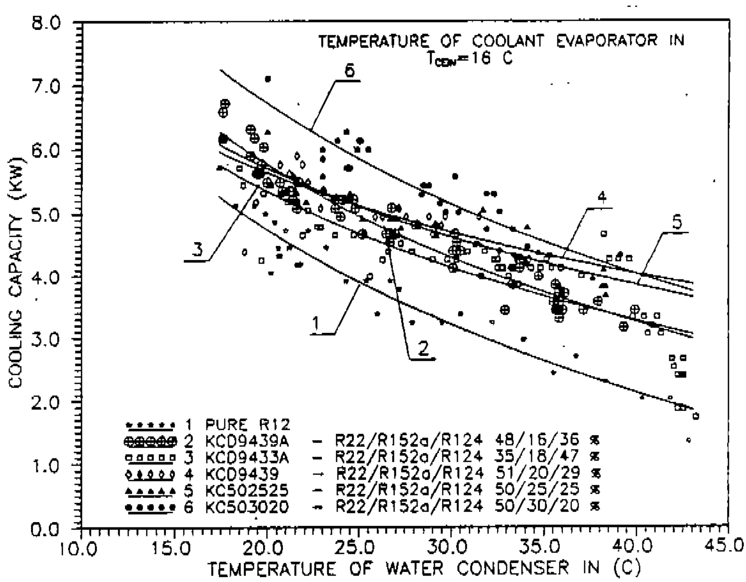


FIG.5 COOLING CAPACITY VS TEMPERATURE OF WATER CONDENSER IN

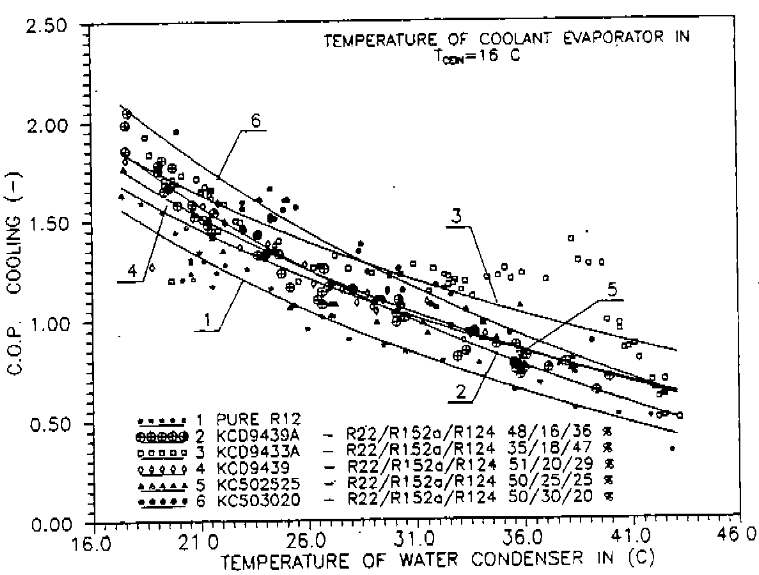


FIG.6 C.O.P COOLING VS TEMPERATURE OF WATER CONDENSER IN

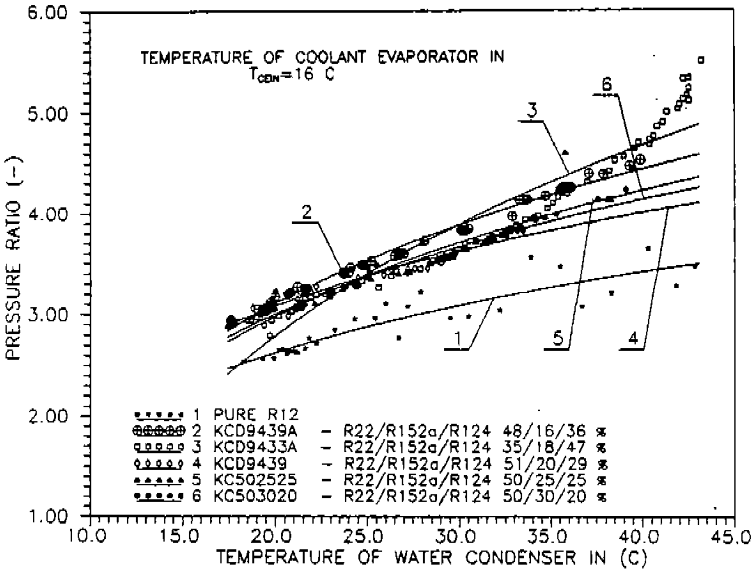


FIG.7 PRESSURE RATIO VS. TEMPERATURE OF WATER CONDENSER IN.

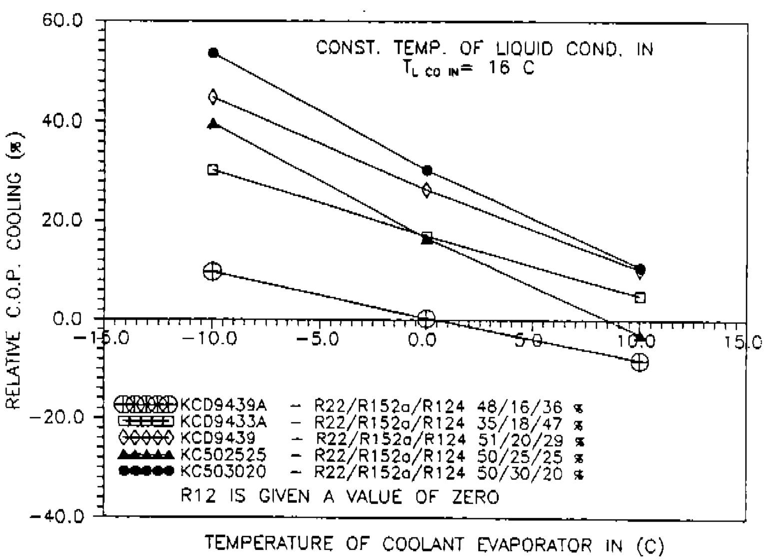


FIG.8 RELATIVE C.O.P. VS. TEMPERATURE OF COOLANT EVAPORATOR IN

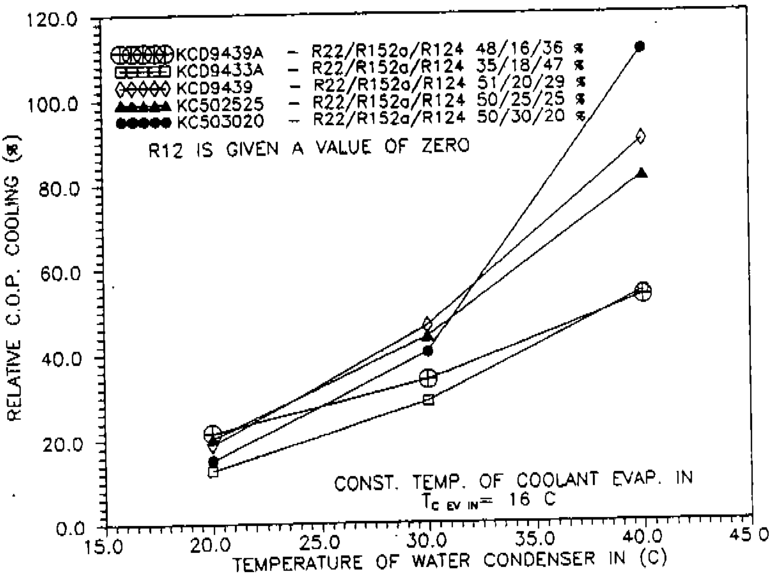


FIG.9 RELATIVE C.O.P VS. TEMPERATURE OF WATER CONDENSER IN.