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ABSTRACT

Due to the undesirable environmental consequences of R12, there is a great challenge to find a suitable substitute for R12. A new refrigerant blend R134a/R218/R600a (88%/ 9%/ 3%) called ISCEON 49 has been developed as a long term, zero ozone depletion-potential replacement for R12. A computer program was written to calculate the cycle characteristics, such as cooling capacity, coefficient of performance (COP), pressure ratio, system pressure and temperature, and compressor power for design and part-load performance which is also our base of comparison. A series of tests has been completed to validate the computer program and to study the behaviour of ISCEON 49. Experimental results indicate that the performance of a system does not degraded on using ISCEON 49, relative to that obtained with R12. The results of the computer program shows that R12 may be replaced by the new blend without a significant loss in overall performance.

NOMENCLATURE

ρ = Density (kg/m^3)

η_{vol} = Volumetric efficiency %

α = Heat transfer coefficient $(kW/m^2\,{}^\circ C)$

Q = Rate of heat transfer (kW)

U = Overall heat transfer coefficient $(kW/m^2\,{}^\circ C)$

ε = Heat exchanger effectiveness

V_s = Compressor displacement (cm^3)

N = Compressor speed (RPM)

h = Enthalpy (kJ/kg)

$\dot{m}$ = mass flow rate (kg/s)

T = Temperature $({}^\circ C)$

A = Area (m^2)

Subscripts

c = Condenser cc = Coolant in the condenser ci = Condenser inlet co = Condenser outlet

e = Evaporator ce = Coolant in the evaporator ei = Evaporator inlet eo = Evaporator outlet

f = Refrigerant $cool$ = Coolant

INTRODUCTION

At the present time the preferred replacement refrigerant for R12 is HFC R134a. This has thermophysical properties that compare well with those of R12 and more importantly it has zero ozone depletion potential. However, R134a is not a 'drop-in' replacement for R12 because there is a need to change the compressor lubricant and expansion valve. Changing over from R12 to R134a can therefore be an expensive and time consuming procedure.

This paper investigates the use of ISCEON 49, a refrigerant blend R134a/R218/R600a (88% /9% /3%) which has thermophysical properties similar to those of R12 and R134a as shown in table 1. It is also compatible with compressor lubricants, the expansion valve does not need to be replaced and the fluid is non-flammable.

This paper compares the thermodynamic performance of ISCEON 49, R134a and R12 at both design-point and part-load conditions using results from a validated computer model of a practical refrigerator.

COMPUTER MODEL

Compressor

For modelling the compressor, it is assumed that compression follows a polytropic process with constant polytropic efficiency. The mass flow rate through the compressor is given by

$$\dot{m}_f = \rho V_s \eta_{vol} N$$

Condenser

The condenser is treated as having a constant overall heat transfer coefficient for the superheat, condensation and subcooling region. The heat transfer in the condenser is governed by the following equations:

$$\dot{Q}_c = \dot{m}_f \Delta h_c \qquad \dot{Q}_c = \dot{m}_{cc} C_p (T_{co} - T_{ci})$$

$$\dot{Q}_c = \dot{m}_{cc} C_p \varepsilon_c (T_c - T_{ci}) \qquad \varepsilon_c = 1 - \exp\left(\frac{-U_c A}{\dot{m}_{cc} C_p}\right)$$

$$\frac{1}{U_c A} = \frac{1}{\alpha_c A_f} + \frac{1}{\alpha_{cc} A_c} + \frac{x}{k A_m}$$

For a given heat exchanger when the coolant flows in the shell outside of tubes [Stoecker and Jones]:

$$\alpha_{cc} = (const)(\dot{m}_{cc}^{0.6})$$

Whereas for the case where the coolant is in the tube:

$$\alpha_{cc} = (const)(\dot{m}_{cc}^{0.8})$$

Evaporator

The evaporator follows the condenser model in the assumption of an overall heat transfer coefficient for the evaporation and superheat region. A evaporator energy balance is yields:

$$\dot{Q}_e = \dot{m}_f \Delta h_e \qquad \dot{Q}_e = \dot{m}_{ce} C_p (T_{ei} - T_{eo})$$

$$\dot{Q}_e = \dot{m}_{ce} C_p \varepsilon_e (T_{ei} - T_e) \qquad \varepsilon_e = 1 - \exp\left(\frac{-U_e A}{\dot{m}_{ce} C_p}\right)$$

$$\frac{1}{U_e A} = \frac{1}{\alpha_e A_f} + \frac{1}{\alpha_{ce} A_e} + \frac{x}{k A_m}$$

Heat transfer coefficient for coolant, α_{ce}, in the evaporator follows heat transfer coefficient of coolant, α_{cc}, in the condenser model.

Refrigerant Thermodynamic Properties

The refrigerant thermodynamic properties of R12, R134a and ISCEON 49 were determined using the Martin, J. equation of state for superheat region [Martin, J.]. The refrigerant data for the liquid and two-phase region was obtain through curve fitting to their thermodynamic data.

SIMULATION MODEL

Design point model

This model neglected heat losses and pressure drop from refrigerant pipelines and heat exchanger shells. The model input data include evaporating, condensing, subcooling and superheating temperatures as well as compressor power characteristics. Coefficient of performance (COP), cooling capacity, compressor power, volumetric refrigeration effect, refrigerant mass flow rate and system pressure ratio are given outputs.

Part-load model

For a given refrigeration system the coolant approaching conditions are the main input data. The simulation was initiated by assuming a starting value for T_e and T_c. The heat balance was applied to the heat exchangers and an iteration process followed using the Wilson plot [Stoecker and Jones] relationship as follows:

$$\frac{1}{UA} = (Const_1) + \frac{(Const_2)}{(\dot{m}_{cool})^n A}$$

Where

$n = 0.6$ for coolant inside shell and outside tubes and $n = 0.8$ for coolant inside tubes.

Constant, $Const_1$ and $Const_2$ were derived from the experiments.

Experimental test conditions

In order to validate the computer model experiments were carried out using a laboratory vapour compression refrigerator. To achieve a realistic comparison of a refrigerant blend to a single refrigerant, the refrigeration cycle's operating conditions need to be defined. The evaporating and condensing temperatures are defined as the dew point evaporator and condenser temperatures. Superheat and subcooling temperatures were measured relative to the suction dew point temperature and discharge bubble point temperature receptively. The condensing temperature was kept constant at 38 °C by controlling the coolant flow and temperature of the coolant to the condenser while the evaporating temperature was varied over a range of -15 °C to 0 °C by using an electric heater. All test readings were taken under steady-state conditions.

EXPERIMENTAL RESULTS

The test facility uses an open drive, two-cylinder compressor which is equipped with an oil sightglass in the crankcase so that the behaviour of the oil/refrigerant can be observed during start-up and the oil level can be monitored during operation to ensure that there is proper oil return from the system. The evaporator and condenser were shell and tube types with secondary refrigerant fluid of an ethylene glycol/ water solution at 40% mixing ratio.

Experiments were carried out to estimate the performance of the test refrigerator operating on both R12 and ISCEON 49. Results showed that ISCEON 49 produced no observable differences in crankcase foaming characteristics of the oil/refrigerant during initial start-up. It also demonstrated good oil return to the compressor from the system. Figures 1-2 present an assessment of ISCEON 49 performance trends. Figure 1 clearly indicates that the cooling effect of ISCEON 49 is similar to that of R12. A maximum 6-8 % error was found in the computer prediction in both cases (with R12 and ISCEON 49) which was partly due to the fact that the model did not take into account the heat losses from the refrigerant line and heat exchanger shell. The computer prediction for pressure ratio has less than 5% discrepancy as shown in Figure 2.

It is believe that the computer model could be improved should heat losses be taken into account by the model. Nevertheless, the agreement obtained can be regarded as acceptable, especially if one compare the performance of different refrigerants.

SIMULATION RESULTS

For the purpose of analysis and comparison with experimental data the refrigerant condensing temperature was kept at 38°C while the evaporating temperature was varied between -30°C and 0°. The degree of subcooling at the condenser outlet was kept to 6°C while the degree of superheating was 6°C. The compressor volume was chosen constant at 1100 (cm^3) with an overall compressor efficiency of 55%. Figure 3 provides a comparison of the volumetric refrigeration effect for ISCEON 49, R12 and R134a. The volumetric refrigeration effect of ISCEON 49 has a little deviation from that of R12 but it is very close to that of R134a. This indicates that for the same refrigeration load, the required compressor volume would be practically the same for these refrigerants. The pressure ratio of ISCEON 49 is very close to that of R134a and is around 4-16% higher than that of R12 as shown in Figure 4. However experiments have shown that these increases in operating pressure with ISCEON 49 should not present a major problem for a compressor that is designed for R12. Figure 5 shows the value of the compressor discharge temperature measured and it indicates that ISCEON 49 had a discharge temperature that was 4-10 °C lower than that of R12 and 2-6 °C lower than that of R134a. The lower discharge temperature of ISCEON 49 will be advantageous in applications that require high system compression ratios or that have high compressor inlet superheat. Compressor powers are compared in Figure 6 and the graph shows that ISCEON 49 had a higher compressor power than R12 for higher evaporator temperatures (0-9%) and lower compressor power than R12 for lower evaporator temperatures (0-12%). ISCEON 49 had an increased compressor power requirement ranging from 2-7 % as compared to R134a. Figure 7 shows that ISCEON 49 has the same cooling capacity as R134a and both cooling capacity were shown to be very close to R12 at higher evaporator temperature. The coefficient of performance (COP) was derived by dividing the cooling capacity by the compressor power. The result shown in Figure 8, indicates that ISCEON 49 had 8-10 % lower COP than R12 and 5-7% lower than R134a.

CONCLUSION

The new refrigerant blend ISCEON 49 R134a/R218/R600a (88%/ 9%/ 3%), was found to be suitable as a substitute for R12. A series of experimental tests was conducted comparing the performance of ISCEON 49 with R12 and a computer model was validated. This was used to compare the performance of ISCEON 49 with that of R!2 and R134a. Experiments showed that ISCEON 49 appears to be a satisfactory drop-in replacement and has comparative performance with that of R12. The computer model showed that for 38°C condensing temperature, 6°C subcooling and superheating, ISCEON 49 had the same cooling capacity as R134a and both had the same cooling capacity to that of R12 at higher evaporator temperature. The volumetric refrigeration capacity of ISCEON 49 showed a little deviation from that of R12 but was the same as that of R134a, which indicates that, the required compressor volume would be similar for ISCEON 49, R12 and R134a. The pressure ratio of ISCEON 49 is very close to that of R134a and 4-10 % higher than that of R12. It is concluded, however, that this should not be a major problem for a compressor designed for R12 compression. The discharge temperature of ISCEON 49 is lower than that of R12 and R134a, which will be beneficial when the compressor is working in a system which has high compression ratios or that has high compressor inlet superheat. The COP of ISCEON 49 was found to be 8-10% lower than that of R12 and 5-7% lower than that of R134a.

The results show the new refrigerant blend was compatible with the standard oil used in R12 systems. Importantly ISCEON 49 offers an environmentally acceptable replacement for R12 and can provide satisfactory thermodynamic performance as a long term replacement.

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Physical Properties		I49	R12	R134a	Physical Properties		I49	R12	R134a
Molecule Weight		103.96	120.93	102.04	Ratio of specific heats (cp/cv) at 1 Atms., at 25°C		1.112	1.197	1.119
Bubble point at 1 Atms.	°C	-35	-29.79	-26.5	Heat of vapoursiation at b. pt.	kj/kg^3	211.6	165.1	216.4
Vapour pressure at 25°C	Bar	7.85	6.57	6.63	Thermal conductivity liquid at 25 °C	W/m °C	0.052	0.041	0.048
Critical temperature	°C	101.3	112	101.2	Viscosity vapour at 25 °C, 1Atms.	cp	0.012	0.013	0.012
Critical pressure	Bar	41.1	41.15	40.64	Viscosity, liquid at 25 °C	cp	0.23	0.26	0.2
Density of liquid at 25°C	kg/m^3	1188	1311	1206	Ozone depletion potential (relative to CFC11=1.0)		0	1.0	0
Density, saturated vapour at b. pt.	kg/m^3	3.9	6.3	5.2	HGWP (relative to CFC11=1.0)		0.44	2.09	0.35
Specificheat, liquid at 25°C	j/mol °k	158	121.4	145.8	Flammable		No	No	No
Solubility of water in refrigerant at 25 °C	wt. %	0.12	0.009	0.12					

Table 1. Physical properties

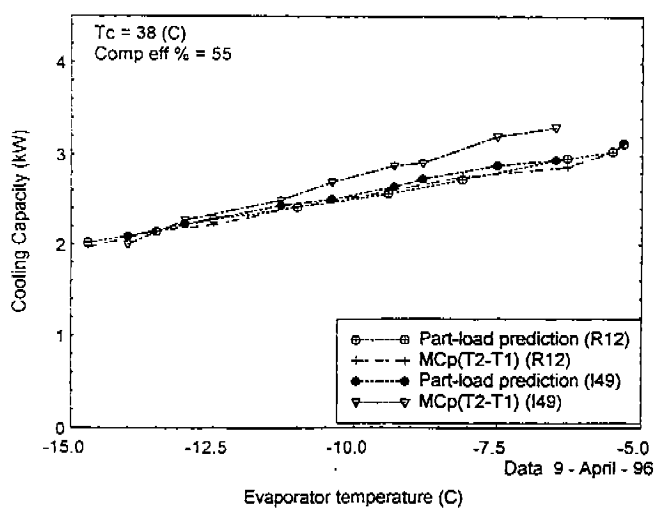

Figure 1 Evaporator cooling Capacity (T_c =38 °C)

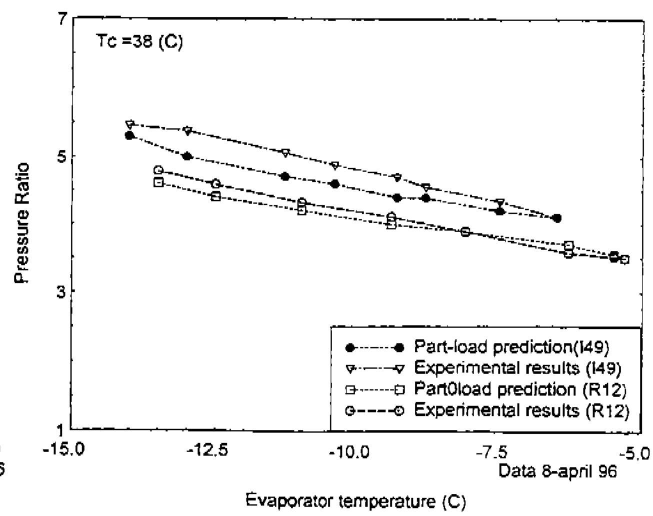

Figure 2 System pressure ratio

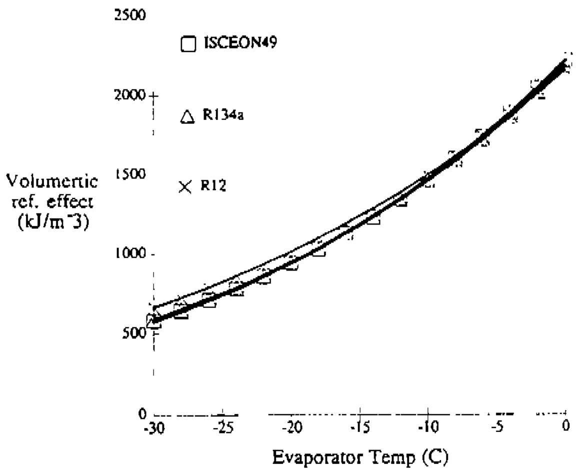

Figure 3 Volumetric refrigeration effect.

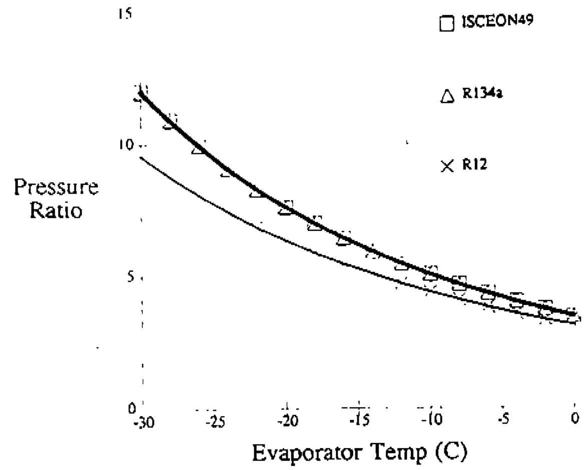

Figure 4 System pressure ratio

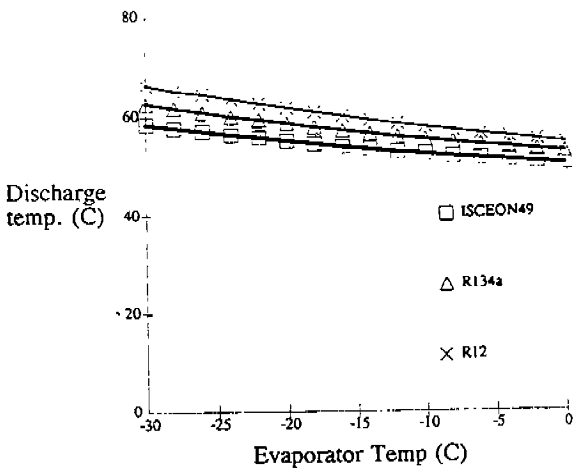

Figure 5 Compressor discharge temperature.

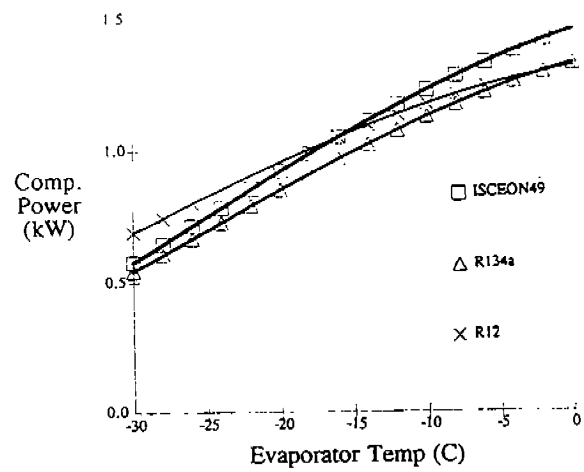

Figure 6 Compressor power

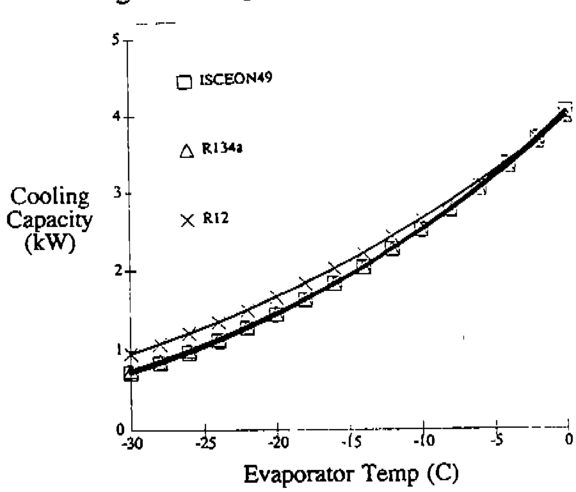

Figure 7 Evaporator cooling capacity

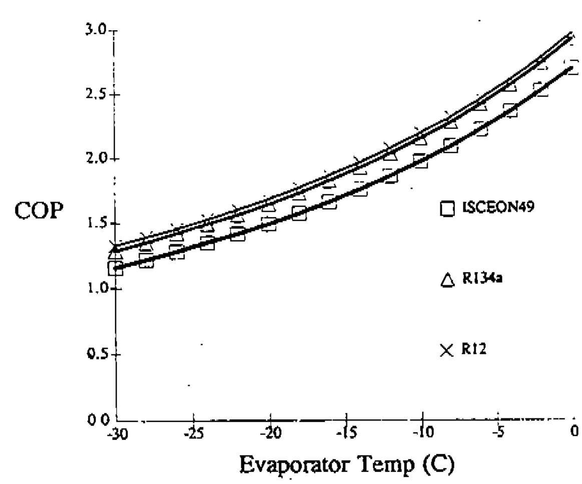

Figure 8 Coefficient of Performance