AN EXPERIMENTAL STUDY OF A REFRIGERATING PLANT WHEN REPLACING R22 WITH HFCs REFRIGERANTS

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Abstract
This paper presents the results of an experimental analysis comparing the performance of a vapour compression refrigerating unit operating with R22, and its performance in comparison to some HFCs fluids, substituting the former according to reg. n. 2037/2000. In particular, the plant working efficiency was first tested with R22 and then with three HFC fluids: R417A, R407C and R404A. The investigation verified that the performance with HFCs refrigerants did not result as efficient as when using R22. An environmental analysis also was performed.
Keywords: *R22; R417A; R407C, R404A.*

1. INTRODUCTION
Chlorofluorocarbons (CFCs) are being phased out, even if recovery is still a major problem. Hydrochlorofluorocarbons (HCFCs) will be phased out according to the time frame defined by the Montreal Protocol and next international obligations [1].
Among all the fluids that can replace the HCFCs there are the Hydrofluorocarbons (HFCs) synthetic fluid refrigerants. The HCFCs replacement with HFCs does not have damaging actions on the ozone layer, because they do not contain chlorine, yet they introduce other problems, this is because HFCs are greenhouse gases.
The HCFC most widely used in refrigeration and air conditioning plants was R22. Its ban imposes the need to resort to alternative solutions which go from the redesigning of systems that run on natural fluids to that of continuing to use the same plants with substitute fluids that do not harm the ozone layer (ODP = 0), with low or zero GWP and able to operate efficiently in the existing plants running on R22 without having to revert to laborious refitting operations requiring a change in lubricating oil and components in the refrigeration plant.
There are already numerous publications, both theoretical and experimental, relating to the substitution of R22 in existing plants and on the subsequent results of its substitution [2-8].
The aim of this work is to experimentally verify the validity of R22 being substituted by three HFC fluids: R417A, R407C and R404A. Test comparing these new fluids' performance with R22's were carried through an experimental facility at the Cold Energy Technology Laboratory, Department of Energy, University of Palermo.

2. EXPERIMENTAL APPARATUS
Figure 1 shows the experimental vapour-compression refrigeration plant. Essentially, it consists of a vapour-compression plant having a four cylinder semi-hermetic reciprocating compressor with a volumetric flow of 32.54 m^3/h. The fluid condensation is accomplished by transferring heat to the outside through a condenser consisting of a finned heat exchanger.
The refrigerant load consists of a water-ethylene glycol mixture (10% in mass) being continuously heated by a gas boiler and then sent to an evaporator. The pressure and temperature of the working fluid are measured and recorded at the inlet and exit of each device of the plant. The mass flow rate is measured at the outlet of the liquid receiver placed downstream from the condenser.

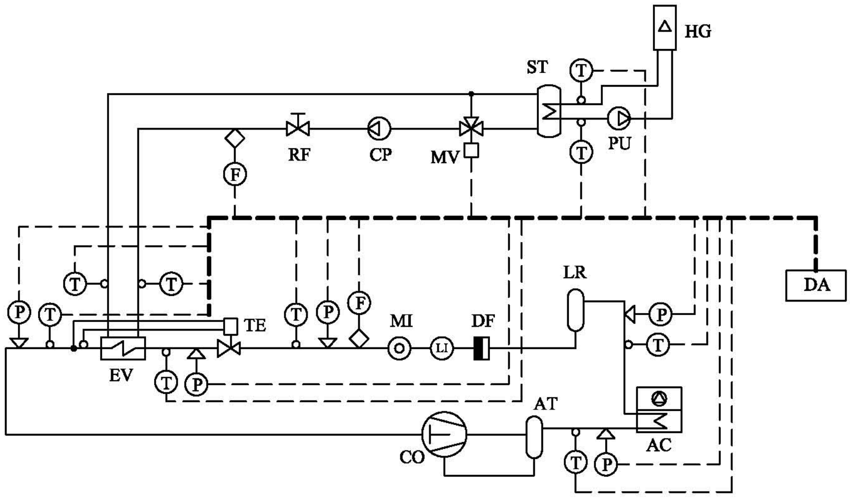


Fig.1: *Layout of the test setup.*

The values of the different parameters were recorded and processed by a data acquisition system that includes a thirty channel data logger and a personal computer. The heat transfer rate discharged in the evaporator is simulated through the water-ethylene glycol mixture loop. Therefore the mass flow rate and the relative temperature are measured at the evaporator inlet section and at the evaporator outlet section. Table 1 shows the characteristics of the main components of the refrigerator, while Table 2 lists the most significant parameters of the tested fluids: R22, R417A, R407C and R404A.

Table 1: *Main characteristics of some of the components.*

Dorin compressor, model: K750CC	Type	Reciprocating semi-hermetic four cylinders
	Volume risen up to 50 Hz	32.54 m^3/h
	Bore gauge	61 mm
	Stroke	32 mm
	Oil load	2.5 kg
	Oil type	Mineral 32 cSt
	Electric feeding	220-240V(Δ)-380-420V(Y)/3/50Hz
Dorin air condenser model 1004	Type	In copper pipes with aluminium finning
	Surface	67.4 m^2
	Volume	12.8 dm^3
	Number of fans	4
	Absorbed power	375 W a 50 Hz
Danfoss thermostatic expansion valve model TES5	Type	With outside pressure equalization
	Work range	-40 °C to 10 °C
	Allowed work pressure	22 bar
ALFA LAVAL evaporator model EC 38	Type	Coaxial pipes
	Surface	3.78 m^2
	Thermal power	25 kW

Table 2: *The main characteristics of the tested refrigerants.*

Refrigerant	R22	R417A	R404A	R407C
Molar mass	86.48	106.75	97.60	86.20
Boiling temperature [°C]	- 40.80	- 39.12	- 46.60	- 43.80
Critical temperature [°C]	96.15	87.04	72.14	86.05
Critical pressure [kPa]	4980	4036	3740	4630
Certainty class	A1	A1	A1	A1

3. TEST PROCEDURES

The objective of the tests was to determine how the vapour-compression refrigeration plant actually behaved when the different types of refrigerant fluids were used. This way it was possible to obtain adequate information on the plant with replacement fluids other than R22 while working under different conditions.

The tests are intended to provide indications on the performance of the refrigeration system designed and built to operate with R22 when the original fluid has been replaced with other HFCs fluids.

The same thermostatic expansion valve with an exchangeable orifice was used for all the tests. The orifice component was changed with the different fluids in order to avoid swinging during trials. The thermostatic expansion valve was regulated in order to set certain overheating parameters of the fluids (Δt_s = 5 °C). The condensation temperature was set at 38 °C.

The experimental tests simulated a real running consistent with the technical specifications of the evaporator. The inlet for the water-glycol mixture was maintained at the constant temperature of 12 °C. The mass flow rate was chosen within a specific range in order to have the outlet glycol-water mixture temperature between 4 and 10 °C.

The mass flow rate was kept constant at a chosen value, and the inlet and outlet temperatures were recorded in order to determine the exchanged thermal power.

The tests were performed after reaching steady operating conditions. The control parameters indicating the attainment of the above conditions were: pressure and temperature in the inlet and outlet evaporator sections. The working conditions were considered steady when the temperature changes were smaller than 0.3 °C and the pressure variations were 5 kPa and 20 kPa in the low and high pressure circuit respectively.

The pressure and temperature values at the fundamental points in the cycle were experimentally taken and read for each tested fluid. After, in the table of the thermodynamic properties of each of the tested fluids we have read the corresponding enthalpy values, necessary to be able to calculate the COP.

The setup of the thermostatic expansion valve was changed by replacing the orifice depending on the different fluids being tested (R22, R417A, R407C and R404A).

4. EXPERIMENTAL RESULTS

The plant was run on R22 under various working conditions in order to verify its efficiency. The evaporator energy balance was found by evaluating the thermal power transferred in it by determining the enthalpy difference between the terminal sections of the exchanger of both the water-glycol solution and the R22. Seven different test conditions were run to compare the exchanged thermal power of water-ethylene glycol mixture and the change in the working fluid enthalpy.

The tests show that although the plant did not have any inconveniences while running after the drop-in operations, from the energy point of view this substitution involves substantial drawbacks.

Figure 2 shows the trend of COP in function of the water–ethylene glycol mixture load for the tested fluids. As far as the comparison among the different refrigeration fluids is concerned, it can be noted that the R22 results are better than those reported with R417A, R404A and R407C, which resulted more modest.

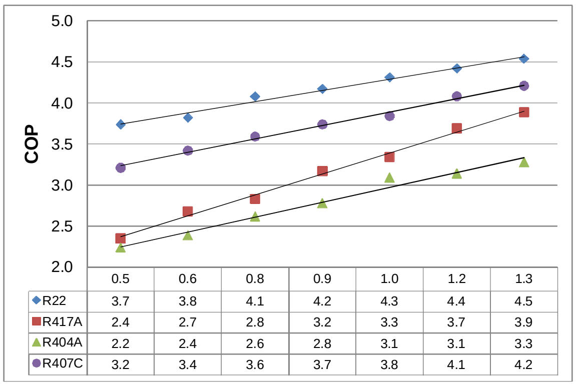


Fig. 2: *COP trend in function of the load of the water–ethylene glycol mixture.*

This implies a higher consumption of energy that would be reflected in greater environmental impact; therefore we compared the impact on the greenhouse effect caused by the use of different refrigerants, by calculating the TEWI, that is a parameter that takes into account the greenhouse effect caused by the direct release when using a plant with a specific refrigerant and the indirect greenhouse effect caused by CO_2 release in the plant where the electricity for the refrigerating plant is produced.

The direct greenhouse effect in our working system was evaluated on the amount of working fluid loaded in the system and by estimating the annual refrigerant loss equal to 7% of the load. Table 4 shows the values of the direct greenhouse effect of the working fluids, reported as amount of CO_2 released in the environment.

Table 4 – Direct greenhouse effect for each fluid

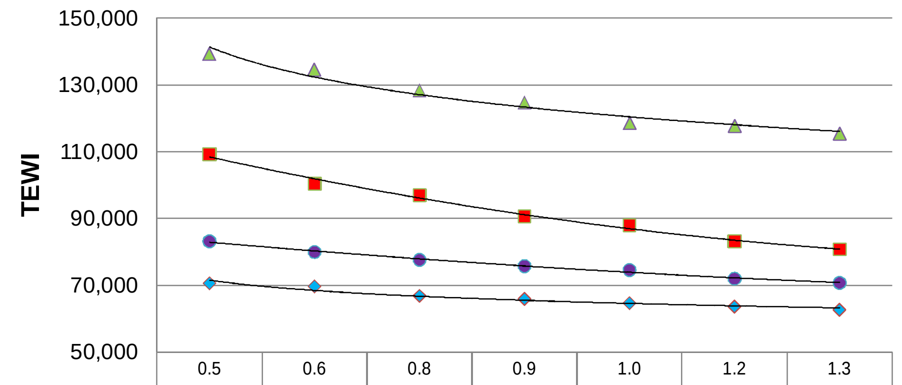
Refrigerant	Load [kg]	GWP	Direct Contribution [kg CO_2]
R22	16.15	1500	25,436
R417	18.25	1950	37,367
R404A	18.70	3260	64,010
R407C	19.10	1526	30,603

The indirect greenhouse effect is:

$$CO_{2,\ i.e.e} = \alpha \cdot Pot / COP \cdot N \cdot SL \quad (1)$$

where α = kg CO_2 emitted per generated kWh, *Pot* = plant refrigerating power, *COP* = coefficient of performance, *N* = hours of equipment operation per year, *SL* = service life (years of equipment operation). To evaluate $CO_{2,\ i.e.e}$ the following values were adopted for the terms specified above α = 0.45 [kg CO_2 / kWh], N = 1000 [h/year], SL = 15 [year]. Furthermore this contribution depends on the COP system.

Finally by adding the two types - direct and indirect contributions - we calculated the TEWI for the fluids and obtained the values in Figure 3 where they are plotted in function of the secondary fluid flows.



	0.5	0.6	0.8	0.9	1.0	1.2	1.3
R22	70,556	69,611	66,796	65,904	64,589	63,615	62,606
R417A	109,176	100,333	96,996	90,600	87,891	83,099	80,747
R404A	139,345	134,617	128,418	124,711	118,622	117,752	115,427
R407C	83,173	79,945	77,609	75,723	74,548	71,963	70,686

Fig. 3: *Trend of the TEWI in function of the glycol water load.*

5. CONCLUSIONS

Several series of tests were performed on R22 (the reference fluid) and the alternative fluids (R417A, R407C and R404A).

Unfortunately, the COP experimental measurements show that substituting a pure HCFC fluid such as R22 is disadvantageous since the different replacement fluids give a less satisfactory rendering due to their particular thermo-physical properties in a system sized for the given original fluid. This should lead to a reflection on the use of alternative fluids in existing systems.

Also from an environmental point of view the TEWI increment may be far more damaging than the low ODP of the original fluid. The European Regulation EC 842/2006 established that the GWP impact when calculating the TEWI is to be taken into account among the various directives issued to improve our planet environment. The main priority in this context becomes the use of the proper refrigerants in order to facilitate the achievement of the set objectives that is: incrementing energy efficiency and lowering the overall environmental impact.

References

[1] Regolamento CE. No. 2037/2000, 29 June, 2000.

[2] Cavallini A. Working fluids for mechanical refrigeration. Int. J. Refrigeration 1996;19: 485–96.

[3] Zheo Y., Yitai M., Ye L. New alternatives for R22. Energy 1997; 22 (7): 669–76.

[4] Gabrielli C., Vamling L. Replacement of R22 in tube-and-shell condensers: experiments and simulations. Int. J. Refrigeration 1997; 20: 165–78.

[5] Aprea C., Greco A. An exergetic analysis of R22 substitution. Appl. Therm. Eng. 2002; 22:1455–69.

[6] Aprea C., Mastrullo R., Renno C., Vanoli G.P. An evaluation of R22 substitutes performances regulating continuously the compressor refrigeration capacity. Appl. Therm. Eng. 2004; 24: 127–39.

[7] Calm J.M., Domanki P.A. R22 replacement status. ASHRAE J. 2004; 45(8): 29–39.