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THERMODYNAMIC COMPARISON OF R502 AND R125 AS VAPOR COMPRESSION PLANT WORKING FLUIDS

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ABSTRACT

In this paper most significant thermodynamic parameters of R125, based on the properties data presented in [1, 2], will be examined and compared with those one of R502. In fact, after the UNEP Montreal Protocol, at the present time the manufacturers and researchers have selected R125 as more probable long term substitute of R502 in refrigerating vapor compression plants operating with cold ambient at -40 °C.

The comparison should point out the improvements and/or the aggravation resulting from the use of R125. Referring to a standard reversed cycle an exergetic analysis is carried out for both the working fluids.

The exergy-enthalpy chart of R125 is presented too.

INTRODUCTION

In the near future R502 replacement problem can be solved by R22, in the high evaporation temperature applications, and by R69-S (near azeotropic mixture of 85% of R22, 6% of R290 and 9% of R218), in the low temperature installations.

This fluids are, at the moment the most suitable substitutes of R502, yet more R69-S is considered a drop-in alternative, but, although their Ozone-Depletion Potential are very low compared to that one of R502 (0.05 for R22, 0.04 for R69-S and 0.3 for R502), they are not completely ozone-safe and then R22 and R69-S will must be substituted in future.

In seeking a long-term alternative solution the search must be focused on ozone-safe working fluids selected among pure substances and/or mixtures.

Among pure substances, the research in progress to find and test alternative refrigerants points out R143a and R125 as interesting sustitutes of R502. The first one, in spite of a good thermodynamic behaviour, is not acceptable owing to its flammability, while the second, although its low thermodynamic performances, looks like the more accreditate solution.

Other possible R502 substitutes are those mixtures that use R125 as a component and that perform a nearly azeotropic behaviour with zero Ozone Depletion Potential.

In this paper a thermodynamic analysis about R502 and its substitute R125 is performed, particularly a R125 exergy-enthalpy chart is supplied together with most adopted thermodynamic properties diagrams. Furthermore, in order to make a comparison between R502 and R125, referring to a refrigeration vapor compression plant in its simplest version, some parameters affecting plant performances, like the exergetic efficiency, and physical dimension of the compressor and of the whole plant, like the Specific Compression Displacement, are evaluated.

Finally, by means of the exergetic analysis, the irreversibility of the whole plant are shared among the components one by one.

The theoretical-numerical analysis is carried out referring to [1] and [3] respectively for R125 and R502.

ANALYSIS AND DISCUSSION

The choice of the working fluid for a vapor compression refrigerating machine results from the weighing of the thermodynamic, technological, environmental and commercial constraints.

1. Usually in the actual applications, evaporating and condensing temperatures should observe these limits, referring to critical temperature [4]: $T_{ev} > 0.65\ T_C$, $T_{co} < 0.95\ T_C$. Furthermore the evaporating pressure, the lowest one in the plant, should never be below the ambient pressure in order to avoid air and water vapor going in the machine. However, according to a less restrictive limit, the evaporating pressure must not be less than 0.5 bar to limit the specific volume value at the compressor inlet [5]. Condensation pressure should be less than 21 bar for construction reasons. Finally, compression ratio low values allow good compressor performances.

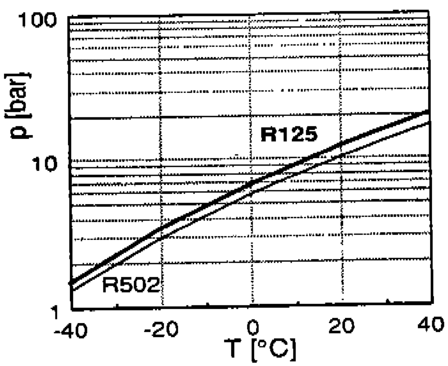

Fig. 1 Vapor-pressure vs. temperature

In fig. 1 R125 and R502 vapor pressur-temperature curves are showed. The curves are really close.

2. Latent heat of vaporization ought to be as high as possible in order to supply the required duty by means of mass flow rate as low as possible. The values, are plotted in fig. 2 versus temperature. There, a refrigerating capacity of R125 lower than that one of R502 can be noted; the average difference is about 20%.

3. The specific compressor dispacement, SCD, that is the ratio of the volumetric flow rate at the compressor inlet to refrigerating duty, ought to be as low as possible in order to reduce the plant

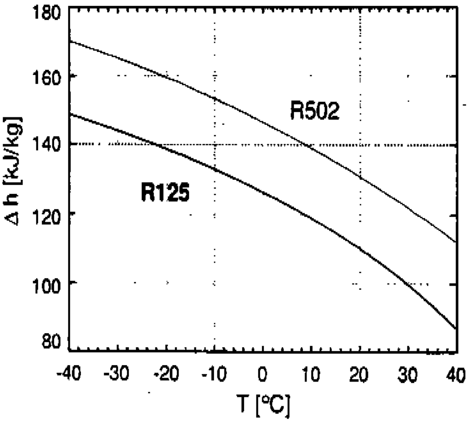

Fig. 2 Latent enthalpy vs. temperature

size and so plant costs. On the other side, high values of performance factor allows low plant running costs. Referring to a vapor compression refrigeration machine in its simplest version, as sketched in fig.3, both the SCD and the exergetic efficiency versus condensing temperature are showed, fig. 4. Following data have been assumed:

evaporating temperature	-40°C
superheating degree	15°C
subcooling degree	0°C
compressor isoentropic efficiency	72%

Pressure drops have been negleted.

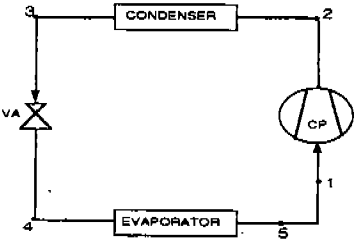

Fig.3 Schematic layout of the refrigerating cycle

R125 exhibits a more favorable SCD until T_{CO} = 35°C, but its exergetic efficiency is significatively less than that one of R502 in all the range examinated (-8% for T_{CO} = 20 °C, -20% for T_{CO} = 40 °C).

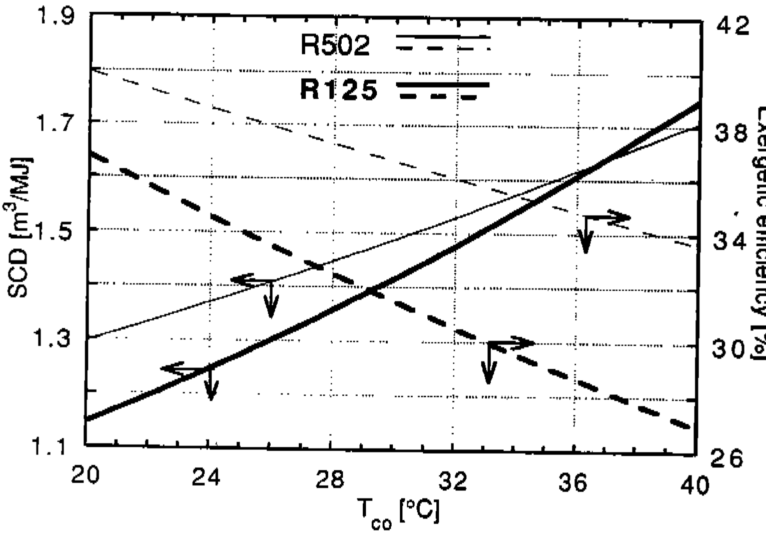

Fig. 4 Specific Compressor Displacement and exergetic efficiency vs. condensing temperature ; evaporating temperature at - 40°

4. An exergy analysis referred to the before mentioned compression machine supplying a refrigerating duty of 1kW to a cold cell at -35 °C, fig. 5, is presented. The environment, used as heat sink, is at 35 °C, temperature differences of 5 °C have been considered in both the evaporating and condensing processes. In fig. 6 exergy flow diagrams pointing out, for each component, the inlet and outlet exergy flow rates and the exergy destruction are reported.

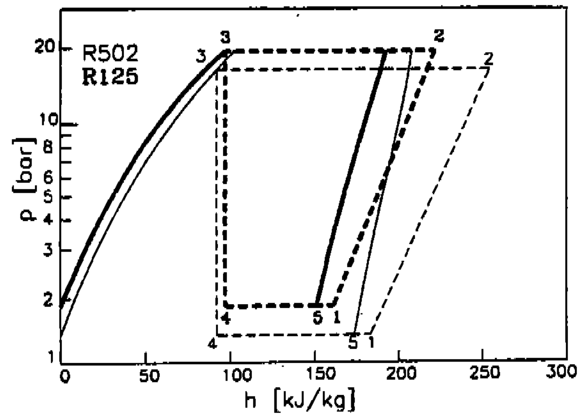

Fig.5 Refrigerating cycle sketched on pressure-enthalpy chart

The exergy flow rate corresponding to the refrigerating duty is 0.29 kW; the exergy input required by the plant operating with R125 to supply this desired flow is greater than that one required by operating with R502 (+28 %). Of course the exergy increase in input is required to balance the increase in destroyed exergy. The irreversibilities are partly caused by inefficiencies of plant devices and partly by the working fluid characteristics. According to the remark that the expansion and the compression processes are strongly related to the working fluid behaviour [5], the highest increases take place in the valve and in the compressor.

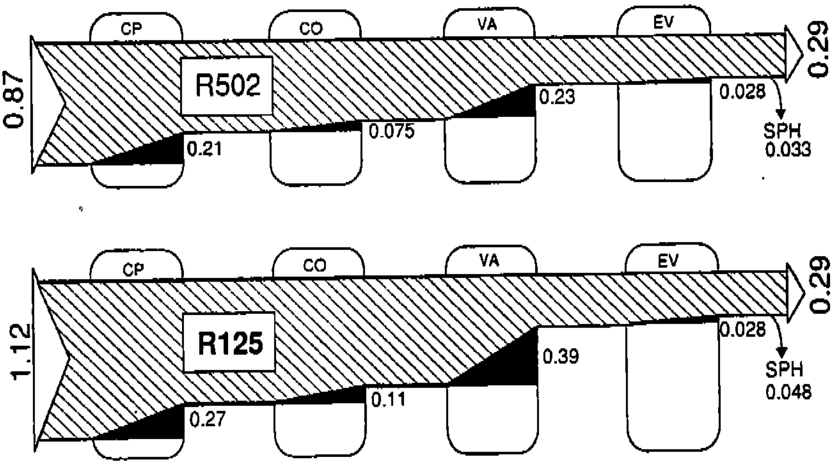

Fig. 6 Exergy flows though the whole cycle. CP compressor, CO condenser, VA expansion valve, EV evaporator, SPH superheating by exchanging with the environment

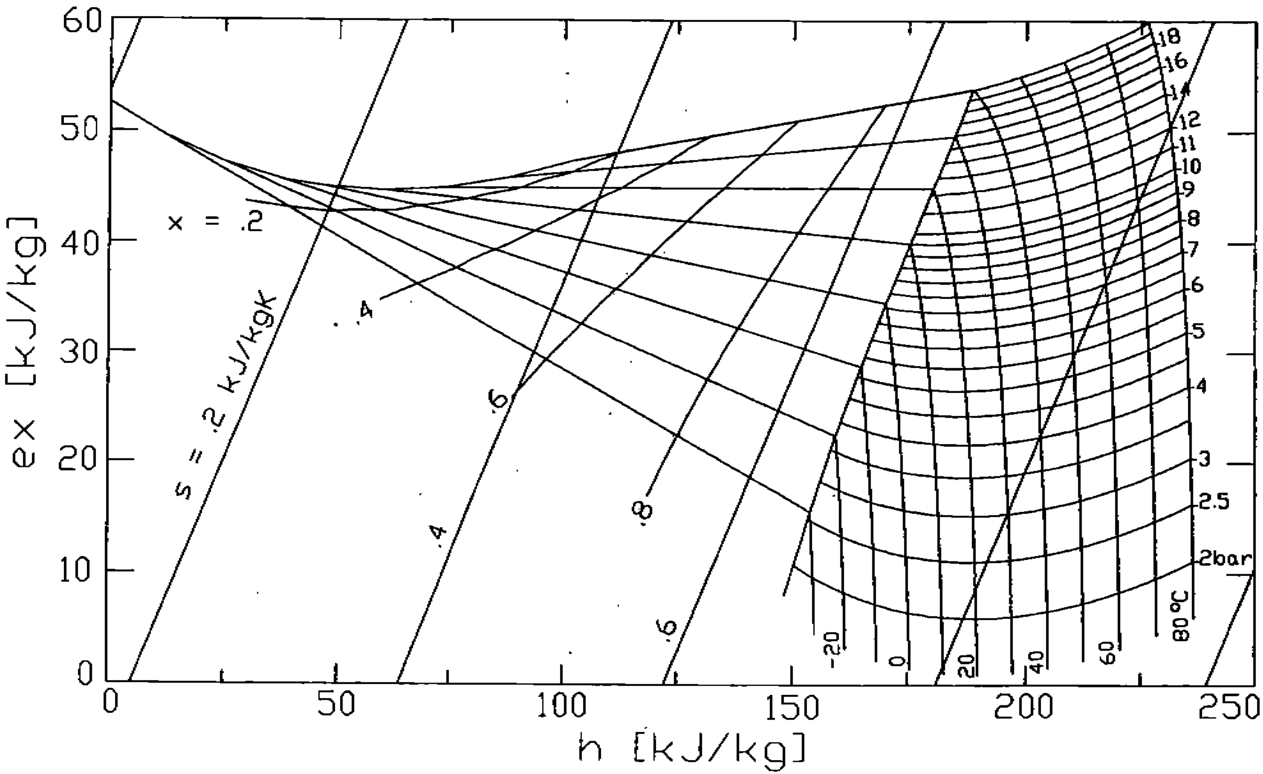

Fig.7 R125 exergy-enthalpy chart

To carry out the exergetic analysis an exergy-enthalpy chart is an helpful tool [6]; in fig. 7 it is presented for R125 with :

reference state: saturated liquid at -40 °C
h = 0 kJ/kg;
s = 0 kJ/kgK;

environmental state: t = 20 °C, p = 101.325 kPa.

CONCLUSIONS

In this paper a thermodynamic comparison between R502 and R125, as vapor compression plants working fluids, is carried out. The main result is that R125 exhibits a value of exergetic efficiency significatively lower in all the range examinated. It is mainly due to higher irreversibilities in expansion process depending on both the liquid specific heat capacity and the latent enthalpy values that provide higher rate of flash vapor production.

The exergy-enthalpy chart of R125 is presented too.

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