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EFFECTS OF VAPORIZATION TEMPERATURE AND SUB COOLING VARIATION ON THE PERFORMANCE OF A VAPOUR COMPRESSION REFRIGERATION CYCLE WORKING WITH R134A, MET ON NEWER SHIPS

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Abstract: R134a is a refrigerant met in several marine refrigeration applications, such as fishing vessels, passenger and cargo ships. In 2014, 26% of the international commercial fleet was using R134a. Although R134a shows a null Ozone Depletion Potential, it has a quite high Global Warming Potential (1300). R134a is a greenhouse gas and, even if it is present on newer ships, the future will be marked by its replacement with substitutes having low GWP. Still, because its GWP is less than 2500, R 134a will continue to be used. Due to the fact that vapour compression refrigeration systems are dominant on board the ships and knowing that these technologies are high energy consumers, analysing their performance in the contemporary energetic context, is imperious required. This paper presents a theoretical analysis of a single stage vapour compression cycle, working with R134a, based on the laws of thermodynamics. The analysis will reveal the influence of the evaporator temperature on the Coefficient of performance and on exergy efficiency, and also the influence of sub cooling on these two efficiency terms, on the refrigerant mass flow rate and compression rate. It was considered a variation of the evaporator temperature in the range (-40÷ -10)ºC and of the sub cooling in the range (0÷10)ºC. The increase of the evaporator temperature will contribute to a COP increment (50%) and an exergy efficiency decrease (34%). The sub cooling will lead to both COP and exergy efficiency increase (11%). Higher sub cooling degree will provide an increment in the refrigerant mass flow (18%) and a decrease of the compression rate (76%) meaning lower work consumption at the compressor.
Key words: *vaporization, refrigeration cycle, evaporator temperature, ship.*

1. INTRODUCTION

Regardless the destination of nowadays ships, refrigeration plants are mandatory on board. Their role is to cool and freeze the catch-on board of fishing vessels, to storage the perishable food or to ensure comfort for crew and passengers throughout air conditioning systems, [1].

The most spread marine refrigeration method is vapour compression refrigeration, [2]. Due to the fact that refrigeration systems work with electric energy consumption and because refrigerants are associated with environmental issues, such as ozone depletion and global warming, modern refrigeration has to face with energy saving and environment protection, [3].

In time, freons like R12 (dichlorodifluoromethan), R22 (chlorodifluoromethan) and R134a (1,1,1,2-tetrafluoroethane), are noticed to be among the most popular refrigerants adopted by marine refrigeration.

Because of its chemical composition, R12 (or CFC12) contributes to the stratospheric ozone depletion. For this reason, R12 was a subject of Montreal Protocol and, as a consequence, it was phased out by 1996 – in developed countries and by 2010 – in developing countries, according to an established schedule.

The phenomena of stratospheric ozone layer depletion have a negative impact on human health and climate, since allows a higher amount of radiation to reach the surface of our planet. By the use of ODP, which is a number indicating the amount of stratospheric ozone damage produced by a substance, R12 is included in Class I ODS, while R22 (or HCFC22) is included in Class II ODS. Because R22 shows a less ODP compared with the one of R12, the phase-out of this refrigerant is in progress (ODP_{R22} is approximately 20 less than ODP_{R12}), [4]. Due to its thermodynamic properties and null ODP, R134a (or HFC 134a) replaced successfully R12, on board the ships [5].

In the light of F-gas Regulation and Kigali Amendment, R134a is a greenhouse gas, but it is currently in use due to its GWP less than 2500, until low GWP refrigerants will replace it [6,7]. However, the present is marked by unsureness given by a phase down schedule, costs of the substitutes or specific new refrigeration systems, [8].

The lifetime of a modern vessel is between 25 and 30 years. Refrigeration systems have a 25-year lifetime, [9]. Eight years ago, about 26% of the international commercial fleet was using R134a. In this context, the analysis of the performance of vapour compression systems working with R134a, on board of newer ships, is in perfect consonance with the contemporary energetic context.

Therefore, this paper deals with a theoretical analysis of a single stage vapour compression cycle, working with R134a, based on the first and second laws of thermodynamics. The analysis will reveal the influence of the evaporator temperature on the Coefficient of Performance and on exergy efficiency and the influence of sub cooling on these two efficiency indicators, as well. Results regarding the influence of sub cooling on the refrigerant mass flow rate and compression rate are also obtained, within this theoretical study.

2. SYSTEM DESCRIPTION

During cyclic processes occurring in vapour compression refrigeration systems, refrigerants absorb heat from a place and release it to another, with a higher temperature; the main component parts of the simplest system being the compressor, the condenser, the throttling valve and the evaporator – in which take place isentropic compression, isenthalpic expansion and phase change of the refrigerant, as seen in figure 1 [10-13]. The two heat exchangers (the evaporator and the condenser) present a superheating region and a sub cooling region.

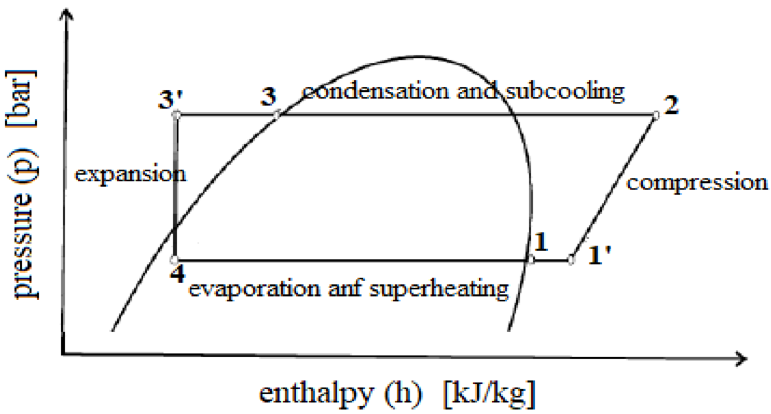

Fig. 1. Pressure-enthalpy diagram of a cycle with superheating and sub cooling, [13]

R134a enters in the compressor at vapour state 1', presenting a superheating degree with respect to the evaporator temperature. Vapours are compressed at constant entropy – in the case of the theoretical process (the actual compression is irreversible and the entropy increases during this process). The compression takes place in the superheated vapour zone. With the pressure increased, superheated vapours at state 2 enter in the condenser, where are condensed and sub cooled, until became sub cooled liquid of state 3'. Condensed and sub cooled R134a liquid is led to the throttling valve, where it is throttled at the constant enthalpy, resulting a liquid-vapour mixture of state 4. Thus, R134a reaches a lower pressure: the evaporator pressure. With the vaporization and superheating processes (4 1'), the cycle is closed. The sub cooling of liquid R134a has a benefit influence on the performance of the cycle because the refrigerating effect per unit mass is greater and no energy consumption increment at the compressor is required.

Greater refrigerating effect per unit mass is obtained in the case of superheating, sub cooling and superheating processes being applied for the gain in efficiency of the system, [14].

3. FIRST AND SECOND LAWS BASED ANALYSIS

First law based analysis of thermal systems is based on the conservation of energy and do not provide information on how, location and the intensity of degradation of the system performance; on other hand, exergy analysis enables the estimation of the maximum performance of the system and the identification of locations of energy destructions, [15].

A strong analysis of refrigeration systems involves the evaluation of exergy used in its component parts. Concepts like exergy and irreversibility are introduced by the second law of thermodynamics. Irreversibilities are encountered in refrigeration cycles, in real conditions, because the heat transfers in evaporator and in condenser take place at finite temperature difference and the compression process in the compressor is an irreversible adiabatic process – because of the frictions, [16].

The exergy analysis is a tool used to approach both the quantity and quality of the forms of energy in discussion, [17].

The thermodynamic analysis for the discussed cycle rely on the following assumptions, [18]:

•the flow of refrigerant is analysed in steady state and the friction losses are neglected;
•the change of kinetic and potential energies of the whole system are neglected and chemical, magnetic and nuclear reactions as well;
•the heat transfers occur only in the heat exchangers;
•the refrigerant flows at constant pressure through the heat exchangers;
•the refrigerant flow through pipes, compressor and throttling valve is considered to be adiabatic.

The mathematic model of the energy and exergy analysis comprises of the following equations [19, 20]:
The first law of thermodynamics applied for a control volume:

$$\frac{dE_{cv}}{dt} = \sum_i \dot{m}_r\left(h + w^2/2 + gz\right) - \sum_o \dot{m}_r\left(h + w^2/2 + gz\right) + \dot{Q}_{cv} - \dot{W}_{cv} \quad (1)$$

where:
E – energy of the system, (J)
t – time, (s)
h – specific enthalpy, (J/kg)
w – speed, (m/s)
g – gravitational acceleration, (m/s^2)
z – altitude, (m)
$\dot{m}_r$ – mass flow rate of the refrigerant, (kg/s)
$\dot{Q}_{cv}$ and $\dot{W}_{cv}$ – energetic changes of the control volume with its surroundings in form of heat flux and power consumed, (W)
i and o – inlet and outlet states.

The above equation, applied to steady state operation becomes:

$$\dot{Q}_{cv} = \sum_o \dot{m}_r\left(h + w^2/2 + gz\right) - \sum_i \dot{m}_r\left(h + w^2/2 + gz\right) + \dot{W}_{cv} \quad (2)$$

Considering the above-mentioned assumptions, we get equation (3), as follows:

$$\dot{Q}_{cv} - \dot{W}_{cv} = \sum_o \left(\dot{m}_r h\right) - \sum_i \left(\dot{m}_r h\right) \quad (3)$$

The last equation will be applied to the components of the system, resulting:

- the refrigeration capacity of the system, (W):

$$\dot{Q}_E = \dot{m}_r\left(h_{1'} - h_4\right) \quad (4)$$

- the condenser capacity, (W):

$$\dot{Q}_c = \dot{m}_r\left(h_{3'} - h_2\right) \quad (5)$$

- the rate of work input to the compressor, (W):

$$\dot{W}_c = \dot{m}_r\left(h_{1'} - h_2\right) \quad (6)$$

The overall performance of the plant, or the first law efficiency, expressed by the ratio between the refrigeration capacity and the work input is:

$$COP = \dot{Q}_E / \dot{W}_c \quad (7)$$

Being known that exergy (ψ) is a concept indicating the maximum theoretical work possible to be obtained from a given amount of heat at the reversible unit, the exergetic balance equation for a control volume is provided below:

$$\frac{dX_{cv}}{dt} = \sum\left(1 - T_o/T_r\right)\dot{Q}_r - \left(\dot{W} - P_E\frac{dV_{cv}}{dt}\right) + \sum_i \dot{m}_r\psi - \sum_o \dot{m}_r\psi - \dot{X}_{destr} \quad (8)$$

where:
T_o – absolute temperature of the surrounding, (K)
T_r – absolute temperature of the heat source, (K)
$\dot{Q}_r$ – heat flux evacuated at the condenser
$\dot{W}$ – work input, (W)
P_E – heat flux absorbed at the evaporator, (W).

For steady state operation, equation (8) enables the founding of the irreversibility rate, (W):

$$\dot{X}_{destr} = \sum\left(1 - T_o/T_r\right)\dot{Q}_r - \dot{W} + \sum_i \dot{m}_r\psi - \sum_o \dot{m}_r\psi \quad (9)$$

Performing the exergetic balance equation to each device of the systems, results:

- for the evaporator, (W):

$$\dot{X}_{destr.E} = \left(1 - T_o/T_r\right)\dot{Q}_E + \dot{m}_r\left(h_4 - T_o s_4\right) - \dot{m}_r\left(h_{1'} - T_o s_1\right) \quad (10)$$

where:
s – specific entropy, (J/kgK)

- for the compressor:

$$\dot{X}_{destr.C} = \dot{m}_r\left(h_{1'} - T_o s_{1'}\right) + \dot{W}_C - \dot{m}_r\left(h_2 - T_o s_2\right) \quad (11)$$

- for the condenser, (W):

$$\dot{X}_{destr.cd} = \dot{m}_r(h_2 - T_o s_2) - \dot{m}_r(h_{3'} - T_o s_{3'}) \quad (12)$$

- for the throttling valve, (W):

$$\dot{X}_{destr.Thv} = \dot{m}_r(h_{3'} - T_o s_{3'}) - \dot{m}_r(h_4 - T_o s_4) \quad (13)$$

Since $h_{3'} = h_{4'}$, results that:

$$\dot{X}_{destr.Thv} = \dot{m}_r T_o (s_4 - s_{3'}) \quad (14)$$

The total exergy destruction rate is given by the sum, (W):

$$\dot{X}_{destr.TOT} = \dot{X}_{destr.E} + \dot{X}_{destr.C} + \dot{X}_{destr.cd} \dot{X}_{destr.Thv} \quad (15)$$

The overall exergy efficiency or the second law efficiency is:

$$\eta_{ex} = \left(1 - \dot{X}_{destr.TOT} / \dot{W}_c\right) \times 100\% \quad (16)$$

4. RESULTS AND DISCUSSIONS

Figure 2 shows the influence of evaporator temperature on COP values. The figure reveals that COP increases with the increment of the evaporator temperature.

Figure 3 shows that the exergy efficiency decreases with increase in evaporator temperature.

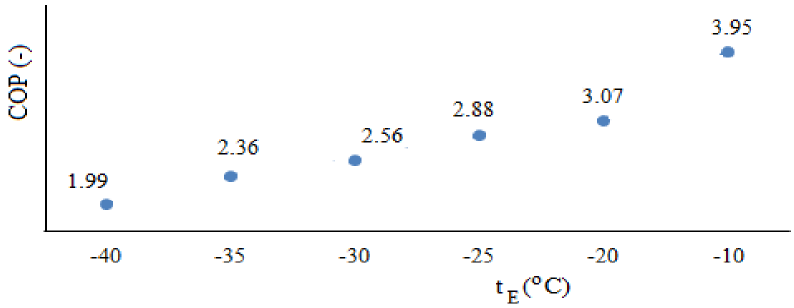

Fig. 2. COP variation with evaporator temperature

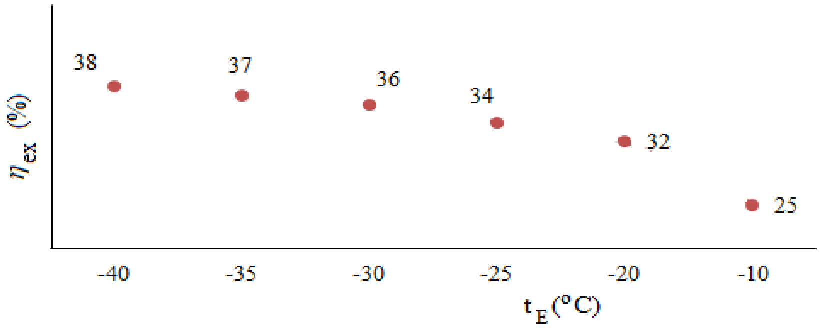

Fig. 3. Exergy efficiency variation with the evaporator temperature

Figures 4, 5, 6, 7 are useful to see the influence of degree of sub cooling increment on values of COP, exergy efficiency, refrigerant mass flow rate and pressure ratio – an indicator of compressor performance given by the rate between condensation pressure and evaporator pressure.

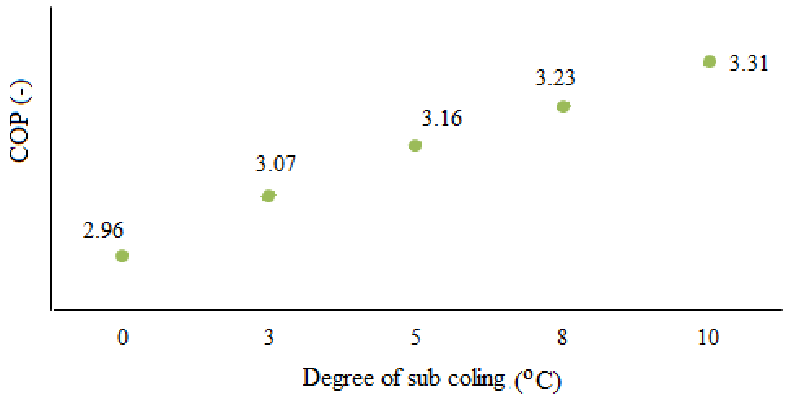

Fig. 4. COP variation with degree of sub cooling.

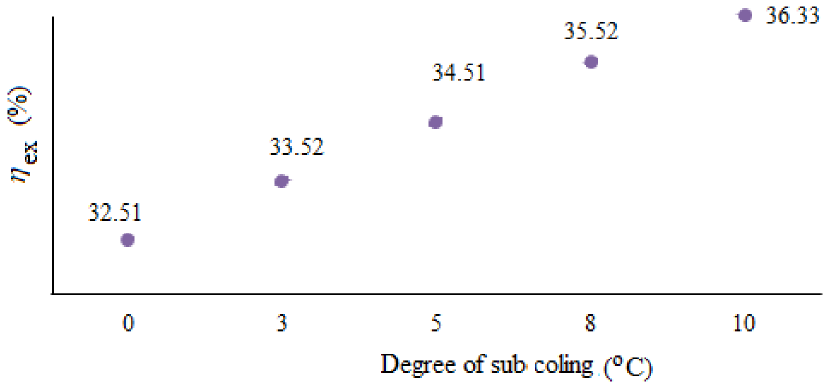

Fig. 5. Exergy efficiency variation with degree of sub cooling

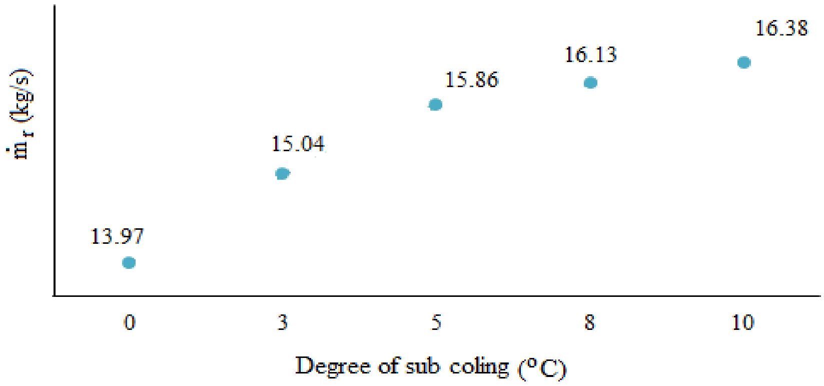

Fig. 6. Refrigerant mass flow rate variation with degree of sub cooling

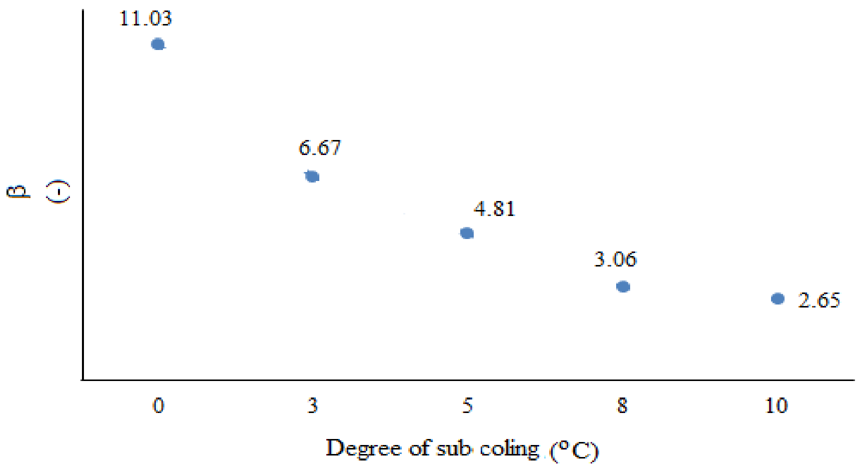

Fig. 7. Pressure ratio variation with degree of sub cooling

The above provided graphs show that the increase in degree of sub cooling leads to increment of COP, exergy efficiency and refrigerant mass flow values and to lower pressure ratio values.

Increasing the evaporator temperature in the considered limits will be found in a gain of COP value of 50% and a diminishment of exergy efficiency value of 34%. Increasing the sub cooling degree will show gains in the performance of the system, for the limits considered, the gain in COP is of 11%, while the gain in exergy efficiency is also 11%. Higher values for sub cooling degree are reflected in higher mass flow rate values (18%) and lower pressure ratio values (76%).

The increase of evaporator temperature is responsible for a higher refrigeration effect and a lower pressure ratio, this combination leading to a gain in COP.

The same increase leads to a decrement in the difference between the evaporator temperature and the temperature of the cooled space. This fact is found in the increase of exergy destruction, with the increase in the evaporator temperature. Thus, for higher evaporator temperatures, the exergy efficiency is lower.

Higher evaporator temperatures correspond to higher COP values, which is equivalent with lower values for the exergy intake to perform the given task. This is the

reason for which exergy efficiency decreases.
Sub cooling has a positive influence on COP, due to the fact that sub cooling increase leads to an increase in the refrigeration effect.
Sub cooling has also a positive influence on the exergy efficiency, due to the fact that sub cooling increase leads to the decrease in total exergy losses.
Sub cooling reduces losses occurring in the throttling valve and also reduces the specific work consumed by the compressor. Sub cooling affects in opposite ways the values of refrigerant mass flow – which will increase with the increase of sub cooling, and the values of pressure ratio – which will decrease.

5. CONCLUSIONS

An energy and exergy analysis of a single stage vapour compression refrigeration cycle was carried for R134a refrigerant. Operation and performance was targeted, based on evaporator and sub cooling degree variation. Increasing of evaporator temperature leads to a gain in the first law efficiency, while higher values of sub cooling degree offers gains in the both efficiencies (first and second law efficiencies) and in less energy consumption.
Since future potential bans would be, probably, applied to new equipments, newer existing ships will continue to use R134a.
For these equipments, the performance analysis is of great interest for the assessment of maritime refrigeration sector relying on this refrigerant.

6. ACKNOWLEDGMENTS

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