Energy and exergy analysis of a VCRS working with R600a within a thermodynamic analysis

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Abstract. The most significant concerns related to vapour compression refrigeration systems (VCRS) refer to environmental behaviour and energy efficiency. The present study is taking into consideration the refrigerant R600a due to its very good thermodynamic properties and environmental position. On the other hand, the maximum performance of such a system will be identified by the help of the laws of thermodynamics. This is due to the fact that the analysis based exclusively on the energy analysis will not reveal where the performance of the system is degraded and how intense. After the formulation of the specific mathematical background, within the analysis have been varied the evaporation temperature in the range (–22, –8) °C. Exergy losses in the main components of VCRS were assessed; the energy and exergy efficiencies of VCRS – as well. Resulted that the system is working at its highest efficiency for the highest evaporation temperature considered: for \( t_e = -8^\circ C \), the energy efficiency is 2.7 while the exergy efficiency is 55%. The most inefficient component part was found to be the compressor, while the most efficient is the evaporator. Such an analysis is very useful when aiming the design, the optimization and performance assessment of VCRS.

1. Introduction
Marine refrigeration is a technology used for the heat evacuation from spaces or substances to be cooled on board the ships, in order to achieve and to keep lower temperature than the environment.
In the last decades, the reefer shipping sector is featured by a constant increase – a result of the consumption growth all over the world and of globalization [1].
Marine refrigeration applications involve chilling and freezing processes for which are suitable mechanical refrigeration technologies [2].
Besides the benefits brought by this technology to the society, the system is also known for high energy demand at the compressor, ozone layer depletion and global warming – if the refrigerant is not environmental friendly [3].
In this respect, efforts are made to enhance the performance of these systems correlated with the use of ecological refrigerants [4].
Refrigerants are the working medium in refrigeration systems. Nowadays, these substances are regulated by the application of environmental requirements amining the operation of modern refrigeration with less emission.
The adoption of a refrigerant has to comply with several aspects [5]:
• thermodynamic characteristics: normal boiling point, critical temperature and heat capacity should respond to the application in use,
chemical stability: assurance that the plant will work long time,
personnel safety and environment protection: no negative influence on the personnel health,
null ODP (Ozone Depletion Potential) and low GWP (Global Warming Potential),
thermo physical characteristics: good heat transfer coefficients, low viscosity, etc.

In maritime refrigeration were used CFCs and HCFCs – which deplete the ozone layer. HFCs came
to substitute the above mentioned since they present similar properties of CFCs and HCFCs. But their
disadvantage is the contribution to global warming making them substitutes for a short period.

This is why natural refrigerants Hydrocarbons (HCs), such as isobutene (R600a), are preferred due
to null ODP, low GWP and good plant performance [6].

The most obvious concern regarding HCs is their flammability because of leaks, but with
appropriate safety measures this obstacle can be overcome in order to exploit their very good
thermodynamics properties [7].

HCs refrigerants, such as R600a, are good substitutes for traditionally refrigerants met in marine
applications (R12, R22, R502, R134a) since they are available, cheap and environmental friendly;
morden they ensure a high performance of the system and low power consumption [8, 9].

Only the energy balance is not able to identify always the system effect, the exergy analysis being
an effective tool for the identification of system imperfections; in other words, the short coming of the
First Law is exceeded by the Second Law of Thermodynamics [10].

Due to the fact that this paper deals with the energy and exergy analysis of a vapour compression
refrigeration system (VCRS) working with the refrigerant R600a, some aspects pointing out the
properties of this HC are given bellow [11-13]:

- molecular mass: 58.1 g/mol
- boiling point: –12°C
- critical temperature: 134.66°C
- critical pressure: 3.64 MPa
- atmospheric lifetime: 0.016 years
- ODP: 0
- GWP: <20
- flammability: yes
- specific requirements: small charge, safety standards and guides, certified personnel.

2. Methods and materials

The simplest vapour compression refrigeration system (VCRS) includes an evaporator, a compressor,
a condenser and a throttling valve; its cycle comprises the following processes: evaporation (p = const.
and T = const.), compression (s = const.), condensation (p = const. and T = const.) and lamination
(h=const.) (see figure 1) [14,15].

Working of vapour compression refrigeration cycles involve the release of significant amount of
heat into the environment so that the heat transfer between the system and its environment occurs at
finite temperature difference, meaning that irreversibility takes place [16].

A realistic analysis of these systems is possible by the use of exergy analysis since it allows the
assessment of the maximum performance of the system and the identification of the sites of exergy
destruction as well [17].

Irreversibility destroys the exergy of a system -- which is the maximum useful work that can be
extracted from a system when it is brought into thermodynamic equilibrium with its environment, the
exergy analysis being a support in achieving more equitable distribution of resources worldwide [18].

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Figure 1. Vapour compression cycle with no superheating and sub cooling in (p-h) diagram.

The energy and exergy performance assessment of a VCRS working with R600a is based on the assumptions stated below; the mathematical formulations on which the analysis is developed are provided in the following [19, 20]:

- in all the component parts of VCRS, are kept steady state conditions
- pressure losses in pipes are ignored
- heat gains / losses from the system are neglected
- negligible change in kinetic and potential energy and exergy.

The evaporator load is estimated with equation (1):

\[ Q_e = q_m (h_i - h_4) \]  

where:
\( q_m \) – mass flow, (kg/s)
\( h \) – specific enthalpy, (kJ/kg)

Exergy losses in the evaporator is calculated with equation (2):

\[ E_{x_e} = q_m (e_4 - e_1) + Q_e \left( 1 - \frac{T_0}{T_e} \right) \]  

where:
\( e \) – specific exergy, (kJ/kg)
\( e = (h - h_0) - T_0 (s - s_0) \)
\( T_0 \) – surrounding ambient temperature, (K)
\( T_e \) – evaporation temperature, (K)
\( s \) – specific entropy, (kJ/KgK)

The work input is found with equation (3):

\[ W_c = q_m (h_2 - h_1) \]  

Exergy losses in the compressor are evaluated with equation (4):

\[ E_{x_c} = q_m (e_1 - e_4) + W_c \]  

(4)
where:

\[ W_{el} = \text{electrical power at the compressor, (kJ)} \]

\[ W_{el} = \frac{W_c}{\eta_m \cdot \eta_{el}} \]

\[ \eta_m = \text{mechanical efficiency of the compressor} \]

\[ \eta_{el} = \text{electrical efficiency of the motor} \]

The condenser load is found with equation (5):

\[ Q_{cd} = q_m (h_2 - h_3) \]  

(5)

Exergy losses in the condenser are found with equation (6):

\[ Ex_{cd} = q_m (ex_2 - ex_3) - Q_{cd} \left(1 - \frac{T_0}{T_c} \right) \]  

(6)

where:

\[ T_c = \text{condensation temperature, (K)} \]

Exergy losses in the throttling valve are calculated with equation (7):

\[ Ex_{TV} = q_m (ex_4 - ex_5) \]  

(7)

The Coefficient of Performance is assessed with equation (8):

\[ COP = \frac{Q_c}{W_{el}} \]  

(8)

The overall exergy losses are assessed with equation (9):

\[ Ex_0 = Ex_c + Ex_{cd} + Ex_{TV} \]  

(9)

The exergy efficiency is found with equation (10):

\[ \eta_{ex} = \frac{Ex_1 - Ex_4}{W_{el}} \]  

(10)

3. Results and discussions

The results of this thermodynamic analysis are obtained for an ambient temperature of 20°C, a condensation temperature of 40°C and a cooling capacity of 1000W; the mechanical efficiency of the compressor and the electrical efficiency of the motors are considered to be 85%, both of them.

In the following figures are given the influences of evaporation temperature variations on the specific work input, COP, specific exergy losses in the main components of the system and exergy efficiency (figures 2, 3, 4, 5, 6, 7 and 8). My paper deals with this analysis due to the fact that the working refrigerant should ensure the following requirements: a low work input and a high value of the Coefficient of Performance- for an efficient running of the plant, from energetic point of view; low specific exergy losses and high values for exergy efficiency- for the minimisation of losses in the energy quality; the evaporation temperature of the working refrigerant should be low, in order to avoid air infiltration in the plant.
The evaporation temperature varies in the range \((-22^{\circ}C-8^{\circ}C)\).

Figure 2. Influence of evaporation temperature on specific work input.

Figure 3. Influence of evaporation temperature on COP.

Figure 4. Influence of evaporation temperature on specific exergy losses in the compressor.
Figure 5. Influence of evaporation temperature on specific exergy losses in the throttling valve.

Figure 6. Influence of evaporation temperature on specific exergy losses in condenser.

Figure 7. Influence of evaporation temperature on specific exergy losses in evaporator.
Figure 8. Influence of evaporation temperature on exergy efficiency.

The assessment of figures 2 and 3 shows that the lowest value of the specific work input and the highest value of the Coefficient of Performance are obtained for the maximum considered value of the evaporation temperature (-8°C). The analysis of figures 4-7 indicates that exergy losses, in the components of the plant, decrease with the increase of the evaporation temperature in all the components of the system. The lowest values of exergy losses are get for the maximum considered value of the evaporation temperature (-8°C). From figure 8, results that the best value of the exergy efficiency is obtained for the maximum considered value of the evaporation temperature (-8°C).

The motivation of the results obtained in this paper it is given below.

In the evaporator and the condenser, the increment in temperature differences between the heat exchanges and ambient leads to exergy losses accentuation.

An increase in the evaporation temperature has a positive effect on the irreversibility’s occurred, in the sense that irreversibility decrease.

The diminishment in the exergy losses at higher evaporation temperature has as a result the improvement of exergy efficiency of VCRS.

The same trend is sein also in the case of the energy efficiency (COP).

My main observation is that, for the range taken into discussion for the evaporation temperature \( t_e = -22, -16, -12, -8 \)°C, VCRS reaches its optimum performance for the highest value of the evaporation temperature \( t_e = -8 \)°C.

Also, it was found that the most inefficient component of the systems resulted to be the compressor, while the most efficient is the evaporator.

4. Conclusions

This paper highlights the benefit of using the R600a refrigerant in marine refrigeration: due to its thermo-physical characteristics and good environmental behaviour, the refrigerant R600a may substitute traditionally used refrigerants in marine applications, together with the implementation of proper safety measures on board.

Besides the environmental aspect related to VCRS, their performance enhancement is of a great concern within a thermodynamic analysis.

In this paper, accurate results are obtained based on the laws of Thermodynamics, by the assessment of energy efficiency (COP) and exergy efficiency (\( \eta_{ex} \)).
Exergy analysis is able to reveal the hierarchy of inefficiencies in VCRS. This approach has been used in this paper.

In this study were obtained important results for a sustainable running of the marine refrigerating plant. Thus, when increasing the evaporation temperature in the range (–22 ÷ –8°C), the compressor is consuming less energy, so that the COP is increasing.

Also, the increment of the evaporation temperature has a positive effect on exergy losses in all the components of the plant, due to the irreversibility decrement.

This is why the exergy efficiency is highest for the highest evaporation temperature considered.

The most performant part of the plant is the evaporator, followed by the condenser, the throttling valve and the compressor.

The plant reaches its highest performance for the highest evaporation temperature considered in the study (–8°C), when the energy efficiency is the highest (2.7) and the exergy efficiency as well (55%).

The conclusions of this paper are useful in the maritime sector, because it is indicated the manner in which are accomplished environmental and energetic requirements, related to the maritime refrigeration.

5. References

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