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First Law Analysis of a Two-stage Ejector-vapor Compression Refrigeration Cycle working with R404A

The traditional two-stage vapor compression refrigeration cycle might be replaced by a two-stage ejector-vapor compression refrigeration cycle if it is aimed the decrease of irreversibility during expansion. In this respect, the expansion valve is changed with an ejector. The performance improvement is searched in the case of choosing R404A as a refrigerant. Using the ejector as an expansion device ensures a higher value for COP compared to the traditional case. On the basis of the ejector approach it possible to identify the highest COP value for a given condensation temperature, when the evaporation temperature varies.

Keywords: compression, ejector, performance, cycle.

1. Introduction

An important application of thermodynamics is refrigeration, which deals with the heat transfer from a "cold body" to a "hot body", through devices called refrigerators. The cycles on which refrigerators operate are known as refrigeration cycles; the working fluids used in refrigerators are called refrigerants. The most often met refrigeration cycle is the vapor compression refrigeration cycle in which the refrigerant is vaporized and condensed alternately and the refrigerant in gaseous phase is compressed.

Some industrial applications ask for moderately low temperatures and their specific temperature range may be high for a single stage vapor-compression refrigeration cycle. This situation leads to a high pressure range in the cycle and a poor performance for the reciprocating compressor included in the plant. The solution is to carry on the refrigerating process in stages.

The two-stage vapor compression system consists of a low pressure compressor, a flash intercooler, a high pressure compressor, a condenser, expansion valves and an evaporator. In order to reduce irreversibility during expansion, it is propsed the substitution of the expansion valve with an ejector [Yari, 2009]. In the following will be energetically assessed a two-stage ejector-vapor compression refrigeration cycle replacing a traditional two-stage vapor compression refrigeration cycle. Such a replacement, together with the use of a HFC refrigerant leads to a performance improvement and an environmentally friendly running [Bergander et al, 2009].

2. Thermodynamic analysis based on the first law

In Figure 1 it is presented the scheme of a two-stage ejector-vapor compression refrigeration cycle. The gaseous phase is leaving the internal heat exchanger with the pressure "p_s" to be compressed in the first stage of the plant in order to be reached the pressure "p_i", with an isentropic efficiency "η_c" (6 - 6a). In the intercooler the vapor suffers an isobar cooling; next, in the second stage of the compressor, it is compressed till a new pressure – "p_c", with an isentropic efficiency "η_c" (6b - 1). In the condenser, the gaseous phase is first cooled and then condensed ($p_c = ct, T_c = ct$); next, the liquid is cooled in the internal heat exchanger (2 - 3).

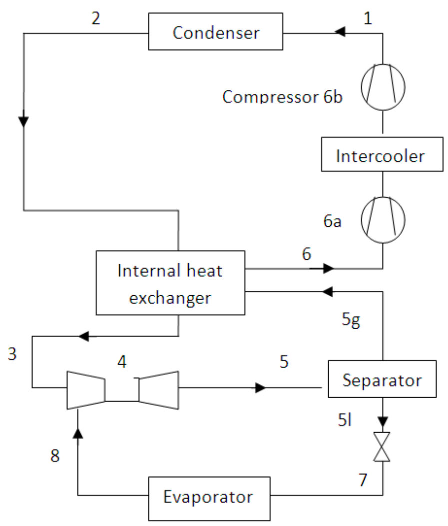

Figure1. Scheme of a two-stage ejector-vapor compression refrigeration plant.

The cooled liquid expands in the ejector with a nozzle efficiency "η_n". Saturated vapors having the pressure "p_e" enter in the ejector with the state noted "8". In the ejector are mixed the two streams, at constant pressure, resulting the state "4". This mixture has the pressure "p_s" in the state "5", the diffuser efficiency being noted with "η_d". The mixture is separated in the separator into saturated liquid (5l) and saturated vapor (5g). The saturated liquid expands till it is reached the pressure "p_e" from the evaporator (5l – 7), while the saturated vapors are superheated in the internal heat exchanger (5g – 6).

In Figure 2 is given the (p – h) diagram for the above described cycle.

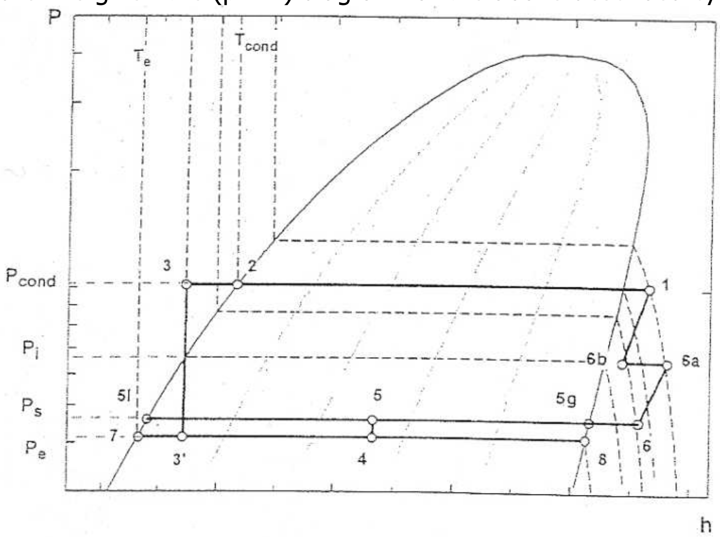

Figure2. (p – h) diagram of the two-stage ejector-vapor compression refrigeration cycle.

In the following it is assumed that [Sarkar, 2009] kinetic energies of the refrigerant at the ejector inlet and outlet are negligible, are neglected flow losses in pipes and heat exchanges, the analyzed system is in thermodynamic equilibrium, processes in the compressor and ejector are adiabatic, process in the expansion valve is izoenthalpic, there is no heat transfer with the environment for the system excepting the condenser.

The overall energetic performance of a refrigerating plant is assessed by the help of the coefficient of performance (COP) [Zubair et.al. 1996].

$$COP = \frac{desired.output}{required.input} = \frac{cooling.effect}{working.input} \quad (1)$$

In our case:

$$COP = \frac{q_e}{w_{c1} + w_{c2}} \quad (2)$$

Above where noted: q – specific refrigeration output per unit mixture mass flow rate, (kW / kg), w – specific power, (kW / kg).

Next it is introduced the ejector entrainment ratio, μ, as the ratio of the ejector suction mass flow rate at (8) to the motive mass flow rate at (3), this parameter being related to the vapor quality. For 1 kg/s refrigerant mixture in the ejector, the suction mass flow rate is $\mu/(1+\mu)$ kg/s and the motive mass flow rate is $1/(1+\mu)$ kg/s [Deng and al, 2007].

Thus:

$$COP = \mu \frac{h_8 - h_7}{h_1 - h_6 + h_{6a} - h_{6b}} \quad (3)$$

In equation (3) "h" is the enthalpy.

3. Overview of the working refrigerant

Refrigerants belonging to chlorofluorocarbon family (CFCs) have been banned since 1996; while from January 1, 2010 the use of virgin hydro-chlorofluorocarbons have been prohibited in the maintenance and servicing of refrigeration and air-conditioning equipment existing in EU at that date. All hydrochlorofluorocarbons shall be prohibited starting with 1 January 2015.

The selection of a refrigerant in the vapor compression system must be done also according to the absence of chlorine atoms in the molecule (see the ozone depletion potential, ODP, of the refrigerant) and their low contribution to the greenhouse effect (see the value of the global warming potential, GWP). Hydrofluorocarbons are potential substitutes for CFCs and HCFCs since these refrigerants do not contain chlorine.

R404A is a blend of HFC refrigerants: R125, R143a and R134a (44/52/4 mass fractions). This refrigerant has a pressure and a thermal critical point high enough to be used in best conditions as a substitute for R502 (CFC 502) and R22 (HCFC 22) in the industrial refrigeration plants for cooling and freezing [Apostol et al, 2007]. R404A is not flammable, its ozone depletion potential is null (ODP=0), but its global warming potential is rather high (GWP=3260). A lubricant miscible with HFC refrigerants must be used with R404A, such as polyolester (POE), since miscibility is important for oil return to the compressor.

Table 1 presents physical properties of R404A, the HFC selected as the working fluid in the two stage ejector-vapor compression refrigeration cycle discussed before.

Table1. R404A – Physical properties

Molecular formula	$CHF_2CF_3/CH_3CF_3/CH_2FCF_3$
Molecular weight	97,6
Boiling point, ^oC	-46,3 (at 101,3 kPa)
Critical temperature, ^oC	72,2

Critical pressure, kPa	3668,6
Critical density, kg/m^3	483,7
Saturated liquid density at 26,7^oC, kg/m^3	1034,7
Heat of vaporization at boiling point, kJ/kg	35,8
Specific heat of vapor at constant pressure at 26,7^oC and 1 atm	0,09

4. Results and discussions

It is evaluated the COP of the previously given cycle, for an evaporation temperature in the range (-40, -15) ^oC and a condensation temperature of 40^oC. We found the values shown in Table 2.

Table2. COP values for a condensation temperature of 40^oC

Evaporation temperature, ^oC	-40	-35	-25	-15	-5
COP	1,67	1,75	2,5	3,12	4

COP values are increasing together with the increase of the evaporation temperature values (higher values of the evaporation temperature lead to lower values for the cooling capacity and compressor power) [Bilge, Temir, 2004].

Calculating COP values for the traditional two-stage refrigeration cycle (COP_{trad}) and the two-stage ejector-vapor compression refrigeration plant (COP_{ev}), obtained for a condensation temperature of 40^oC results:

$$COP_{trad} = 1{,}67$$

$$COP_{ev} = 1{,}78$$

COP value of the two-stage ejector-vapor compression refrigeration plant is about 6% higher than the one obtained for the traditional one.

5. Conclusions

The refrigerating process might be developed in stages by using an ejector instead of an expansion valve. The performance comparison of the traditional two-stage vapor compression refrigeration cycle and the two-stage ejector-vapor compression refrigeration cycle shows a COP increment of 6,58%.

The analysis of the plant with an extension valve in terms of energetic performance, for a given condensation temperature (40°C) and a range of the evaporating temperature (-40°C, -5°C), shows that the highest COP value (4) is obtained for the lowest evaporation temperature (-5°C). This means that COP values are increasing with the increase of the evaporation temperature values.

References

[1] Apostol V., Popescu Gh., Pop H., Prodan M., Popescu T., *Thermodynamic analysis of a new eco-refrigerant – R404A and Dimethylether blend*, Proceedings "Termotehnica 1-2", 2007.

[2] Bergander M., Butrymowicz D., Karwacki J., Wojciechowski J., *Application of two-phase ejector as second stage compressor in refrigerating cycles,* Proceedings "7th World Conference on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics", Krakow, Poland, 03 July, 2009.

[3] Bilge D., Temir G., *On the optimum numbers of stages in vapor compression refrigeration systems*, Proceedings "American Journal of Applied Sciences 1(2):71-75, 2004.

[4] Deng J.Q., Jiang P.X., *Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector*, Proceedings "Applied Thermal Engineering 27", 2007.

[5] Yari M., *Theoretical analysis of a two-stage ejector vapor compression refrigeration cycle*, Proceedings "ECOS", Panama, Brasil, 2009, 741-752.

[6] Sarkar J., *Performance characteristics of natural- refrigerants – based ejector expansion refrigeration cycles*, Proceedings "IMechE", Vol. 23, Part A: J."Power and Energy", 2009, 543-550.

[7] Zubair S.M., Yakula M., Khan S.H., *Second law based thermodynamics analysis of two stage and mechanical subcooling refrigeration cycles*, Proceedings Int. J. Refrig., Vol 19", 1996.

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