1996

Optimization of the Refrigeration Machinery With One Stage of Compression Using R152a

D. Aradau
University "Dunarea de Jos" of Galati

L. Costiuc
University "Dunarea de Jos" of Galati

OPTIMIZATION OF THE REFRIGERATION MACHINERY WITH ONE STAGE OF COMPRESSION USING R152a

Dumitru Aradau and Liviu Costiuc
Department of Mechanical Engineering
University "Dunarea de Jos " of Galati
ROMANIA

Abstract

A method of the exergetic optimization of the refrigerating machinery with one stage of compression using R152a and reciprocating compressor, starting from the indicated diagram of compressor, is set in this paper. Also some practical relations for the calculation of the exergetic efficiency that brings in evidence the influence of the real reciprocating compressor and the nature of the working refrigerating agent, are set in this work.

1.Introduction.

The meetings of specialized commissions of the I.I.R. at London (1990), at Copenhagen, Ghent (1992) and Hanovra (1994) have brought significant changes for the Montreal Protocol (1987) regarding the rise of restrictions for the production and the use of the polluting refrigerants. It is now acknowledged that in order to satisfy these environmental concerns, the research level have increased for finding ecological alternatives for working refrigerants. For small refrigerating machineries an alternative ecological working fluid for R12 has imposed R152a.

The aim of this study is to assess the performance of one stage of compression refrigerating machinery using R152a. There are some parameters that determine the optimization for a given set of conditions, starting with environmental conditions (the evaporator temperature, the condenser temperature) and the indicated diagram of the compressor.

2.Simulation model

For one stage of compression refrigerating machinery (Figure 1) the exergetic efficiency η_{ex}, it is carry out using the global exergetic balance and the two combined laws of thermodynamics (Figure 2), have been used looking at in-out exergetic flows, $(Ex)^-$ and $(Ex)^+$:

$$\eta_{ex} = \frac{\left(E_x^-\right)}{\left(E_x^+\right)} = \frac{Q_0\left(\frac{T_a}{T_r}-1\right)}{L} \tag{1}$$

where: Q_0 - cooling capacity [kW]; $L = P_i + P_f$ - compressor consumed power (P_i -compressor indicated power, P_f -compressor dissipated power due to friction).

The compressor indicated power P_i depends on the shape of compression and expansion curves, but also with the variation law of the compressor pressure among the aspiration and discharge stages. In Figure 3, the mechanical indicated work L_i is given by 3 components $(L_i)_a$, $(L_i)_b$, $(L_i)_c$ and in the same way will be the compressor indicated power $(P_i)_a$, $(P_i)_b$, $(P_i)_c$. The main part $(L_i)_a$ could be given by:

$$\left(L_i\right)_a = \frac{n}{n-1}p_1V_a\left[\left(\frac{p_2}{p_1}\right)^{\frac{n}{n-1}}-1\right]-\frac{n'}{n'-1}p_1V_d\left[\left(\frac{p_2}{p_1}\right)^{\frac{n'}{n'-1}}-1\right] \tag{2}$$

This equation can be written using the correcting coefficient of indicated diagram λ_0 [1,2], if it is admitted that the same politropic compression and expansion coefficient, $n = n'$, and both equal with the adiabatic coefficient k. In these conditions, equation 2 becomes:

$$\left(L_i\right)_a = \frac{k}{k-1} p_1 \frac{\left(V_a - V_d\right)}{\lambda_0}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k}{k-1}} - 1\right] \tag{3}$$

The volume difference (V_a-V_d) is expressed in relation to the volume of the compressor stoke V_s and the indicated volumic efficiency $\lambda_i = \lambda_1\ \lambda_2$, where λ_1 and λ_2 are the volumic efficiency due to dead-space and the volumic efficiency due to isenthalpic expansion at aspiration [1,2], expressed with the following relations:

$$(V_a - V_d) = \lambda_i\ V_s = \lambda_1\ \lambda_2\ V_s \tag{4}$$

Replacing the equation 4 into the equation 3, that becomes:

$$\left(L_i\right)_a = \frac{k}{k-1} p_1 \frac{\lambda_1 \lambda_2}{\lambda_0} V_s\left[\left(\frac{p_2}{p_1}\right)^{\frac{k}{k-1}} - 1\right] \tag{5}$$

In case of adiabatic reversible compression the specific compressor work (neglecting kinetic and potential energy change) is given by the equation:

$$\left(i_2 - i_1\right)_s = \int_1^2 v\, dp = \frac{k}{k-1} p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{k}{k-1}} - 1\right] \tag{6}$$

If the equation 6 is replaced into the equation 5 then:

$$\left(L_i\right)_a = \frac{\lambda_1 \lambda_2}{\lambda_0 v_1} V_s \left(i_2 - i_1\right)_s = \frac{\lambda_i V_s}{\lambda_0 v_1}\left(i_2 - i_1\right)_s \tag{7}$$

where subscript 1 means real state at aspiration, and subscript s it is used for final state of compression at the same entropy value as on the entropy value on the state 1. So, for the indicated power $(P_i)_a$ that will be:

$$\left(P_i\right)_a = \frac{\lambda_i V_t}{\lambda_0 v_1}\left(i_2 - i_1\right)_s \tag{8}$$

where V_t it is the compressor theoretical volumic flow.

The equation 8 can be rewrite according to [3,4]:

$$\left(P_i\right)_a = \frac{\lambda_1 \lambda_2}{\lambda_0} \frac{1}{\left(\eta_{ex}\right)_{ad}} Q_0\left(\frac{T_a}{T_r} - 1\right) \tag{9}$$

where: $\left(\eta_{ex}\right)_{ad} = \frac{Q_0}{P_{ad}}\left(\frac{T_a}{T_r} - 1\right)$, exergetic adiabatic efficiency; $P_{ad} = \frac{\lambda V_t}{v_1}\left(i_2 - i_1\right)_s$ - adiabatic compressor power ; λ - compressor volumic efficiency ;

Taking into account that the compressor volumic efficiency is : $\lambda = \lambda_1\ \lambda_2\ \lambda_3\ \lambda_4$, and assuming that $\lambda_4\lambda_0 \approx 1$ [1], the equation 9 becomes:

$$\left(P_i\right)_a = \frac{1}{\lambda_3} \frac{1}{\left(\eta_{ex}\right)_{ad}} Q_0\left(\frac{T_a}{T_r} - 1\right) \tag{10}$$

where λ_3 and λ_4 are the volumic efficiencies due to overheating at aspiration and leakage.

The indicated compressor power P_i is acquired adding the three parts, $(P_i)_a$, $(P_i)_b$ and $(P_i)_c$, corresponding to the compressor works $(L_i)_a$, $(L_i)_b$ and $(L_i)_c$. In general they are function of equivalent pressure loss:

$$\Delta p_b = \frac{\left(L_i\right)_b}{V_s} = \frac{\left(P_i\right)_b}{V_t}; \quad \Delta p_c = \frac{\left(L_i\right)_c}{V_s} = \frac{\left(P_i\right)_c}{V_t} \tag{11}$$

or function of the real mean pressure loss at aspiration Δp_1 and discharge Δp_2 (Figure 3), like on the expressions:

$$\Delta p_1\left(1 - \frac{X_a}{S}\right) = \frac{\left(L_i\right)_b}{V_s} = \frac{\left(P_i\right)_b}{V_t}; \quad \Delta p_2\left(\frac{X_r}{S}\right) = \frac{\left(L_i\right)_c}{V_s} = \frac{\left(P_i\right)_c}{V_t} \tag{12}$$

where: S - piston stroke ; X_a/S , X_r/S - relative position of the piston at the opening of the aspiration valve, respective of the discharge valve, and calculated by:

$$\left(\frac{X_a}{S}\right)=m\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{k}}-1\right];\ \left(\frac{X_r}{S}\right)=\left[(1+m)\left(\frac{p_2}{p_1}\right)^{-\frac{1}{k}}+m\right] \tag{13}$$

where m is the dead-space compressor coefficient.

The lost power due to friction P_f is written on the relation with a mean equivalent pressure loss ΔP_f:

$$\Delta p_f=\frac{P_f}{V_t} \tag{14}$$

Total pressure loss Δp becomes:

$$\Delta p=\Delta p_b+\Delta p_c+\Delta p_f=\left((P_i)_b+(P_i)_c+P_f\right)/V_t \tag{15}$$

The total mechanical power consumed by the compressor will be:

$$P=\frac{1}{\lambda_3}\frac{1}{(\eta_{ex})_{ad}}Q_0\left(\frac{T_a}{T_r}-1\right)+\Delta p\,V_t \tag{16}$$

For refrigerating machinery with recuperative heat-exchanger (SR), the compressor power $(P_i)_a$ is calculated by:

$$(P_i)_a=\frac{\lambda_1\lambda_2}{\lambda_0}\frac{V_t}{v_{1R}}\left(i_{2R}-i_{1R}\right)_s \tag{17}$$

and cooling heat flow Q_0 is calculated by:

$$Q_0=\lambda V_t q_v \tag{18}$$

where: $q_v=\frac{i_1-i_{5R}}{v_{1R}}=\frac{i_1-i_5+\eta(i_{1T}-i_1)}{v_1\left[1+\eta\frac{(T_k-T_0)}{T_0}\right]}$ - specific volumic cooling heat flow; v_{1R} - specific volume of vapor at the compressor aspiration; η - recuperative heat-exchanger efficiency.

The adiabatic power corresponding to Figure 1 will be get by the relation:

$$P_{ad}=\frac{\lambda V_t}{v_{1R}}\left(i_{2R}-i_{1R}\right)_s \tag{19}$$

Replacing the equation 18 and 19, the adiabatic exergetic efficiency becomes:

$$(\eta_{ex})_{ad}=\frac{(i_1-i_{5R})}{(i_{2R}-i_{1R})_s}\left(\frac{T_a}{T_r}-1\right)=\frac{i_1-i_5+\eta(i_{1T}-i_1)}{\left[1+\eta(T_k-T_0)T_0^{-1}\right](i_2-i_1)_s}\left(\frac{T_a}{T_r}-1\right) \tag{20}$$

The adiabatic exergetic efficiency $(\eta_{ex})_{ad}$ depends on the properties of the working refrigerant, the working regime (the evaporator temperature T_0 and the condenser temperature T_k) and the recuperative heat-exchanger efficiency defined by:

$$\eta=(i_{1R}-i_1)/(i_{1T}-i_1) \tag{21}$$

Using equation 16 and equation 18, and replacing into equation 1, this becomes [2,3]:

$$\frac{1}{\eta_{ex}}=\frac{1}{\lambda_3}\frac{1}{(\eta_{ex})_{ad}}+\frac{\Delta p}{\lambda q_v\left(\frac{T_a}{T_r}-1\right)} \tag{22}$$

The equation 22 obviously shows different effects of the compressor losses for the real exergetic efficiency. Also, for a null mean equivalent pressure loss Δp the exergetic efficiency (η_{ex}) will be smaller than the adiabatic exergetic efficiency $(\eta_{ex})_{ad}$ because $\lambda_3<1$.

According to this, it is important to remark the wholesome effect of the compressor volumic efficiency increase due to the overheating at aspiration, the total compressor volumic efficiency and the specific volumic cooling heat flow, for a given working regime of the refrigeration machinery.

The optimization option for one stage of compression refrigerating machinery using R152a is based on the equation 22. A computer simulation program has developed and tested using following relations for refrigeration cycle state-points:

- the saturation pressure estimated as function of temperature :

$$\ln\left(\frac{p}{p_{cr}}\right)=\frac{T}{T_{cr}}\left[c_1\left(1-T_r\right)+c_2\left(1-T_r\right)^{1.2}+c_3\left(1-T_r\right)^2+c_4\left(1-T_r\right)^3\right] \tag{23}$$

- the equation of state:

$$p_r=\frac{T_r\rho_r}{Z_{cr}}+\sum_1^{25}\frac{a_i\,\rho_r^{m_i}}{T_r^{n_i}} \tag{24}$$

- the specific enthalpy value for the saturated and overheated vapors:

$$i=\left(i_0-RT_0\right)+1000\,p_{cr}\sum_{i=1}^{25}\frac{a_i\left(n_i+1\right)\,\rho_r^{m_i}}{T_r^{n_i}\left(m_i-1\right)\,\rho}+1000\frac{p}{\rho}+RT_{cr}^{\oplus}\sum_{i=1}^{4}\frac{b_i}{i}\left[\left(\frac{T}{T_{cr}^{\oplus}}\right)^i-\left(\frac{T_0}{T_{cr}^{\oplus}}\right)^i\right] \tag{25}$$

- specific entropy value for the saturated and overheated vapors:

$$s=s_0+1000\frac{p_{cr}}{T_{cr}}\sum_{i=1}^{25}\frac{a_i n_i\;\rho_r^{m_i}}{T_r^{n_i+1}\left(m_i-1\right)\,\rho}+R\ln\left(\frac{\rho_0}{\rho}\right)+R\sum_{i=2}^{4}\frac{b_i}{(i-1)}\left[\left(\frac{T}{T_{cr}^{\oplus}}\right)^{i-1}-\left(\frac{T_0}{T_{cr}^{\oplus}}\right)^{i-1}\right]+Rb_1\ln\left(\frac{T}{T_0}\right) \tag{26}$$

- the saturated liquid density ρ‘:

$$\rho'=\rho_{cr}+\sum_1^4 e_i\left(1-\frac{T}{T_{cr}}\right)^{\frac{i}{3}} \tag{27}$$

where: T_{cr}, $T_{cr}^{\oplus}$, p_{cr}, ρ_{cr}, Z_{cr} are the critical properties; a_i, b_i, c_1, c_2, c_3, c_4, e_i, m_i, n_i - constants; p_r, T_r, ρ_r - reduced properties; R - specific gas constant; i_0,s_0 - enthalpy, entropy values at the reference state; T_0=273.15K [4,5].

The saturated liquid enthalpy value i' and entropy value s' are estimated using the Clapeyron equation:

$$i'=i''-T\left(\frac{1}{\rho''}-\frac{1}{\rho'}\right)\frac{dp}{dT}\;\;;\;\; s'=s''-\left(\frac{1}{\rho''}-\frac{1}{\rho'}\right)\frac{dp}{dT} \tag{28}$$

where superscripts ' and " are used for saturated liquid state and saturated vapor state and the function (dp/dT) is estimated using the equation 23.

The results are plotted in Figure 4 for exergetic efficiency $\eta_{ex}=\eta_{ex}(t_0,\ \eta,\ t_k)$, as function of the evaporator temperature change, recuperative heat-exchanger change η, at different condensing temperatures t_k. Each surface plotted represents exergetic efficiency behavior at constant condensing temperature, at steady-state simulation of the refrigeration cycle and the variation of both evaporator temperature and recuperative heat-exchanger efficiency.

3. Conclusions

It is possible to predict that the exergetic efficiency decreases when the evaporating and condensing temperatures are rising. The recuperative heat-exchanger efficiency have a slight influence on exergetic efficiency for one stage of compression refrigerating machinery. As a result, using the R152a as working refrigerant it is suggested that the recuperative heat-exchanger should not be used.

References

1.Daba,M.,Aradau,D.- Calculul si constructia compresoarelor . Compresoare cu piston. Institutul Politehnic Galati, 1971.

[2].Aradau,D.-Le rendement exergetique des installations frigorifiques a deux etages de compression utilisant des compresseurs a piston-Revue Generale du Froid,No.6,1976,pp.415-420.

[3].Aradau,D.-L'analise exergetique des installations frigorifiques a un et deux etages de compression utilisant des compresseurs a piston-Buletinul Universitatii din Galati,Anul II,Fascicula IV,1979, p.43.

[4] Takeshi,T.,Haruki,S.,Koichi, W. - An equation of state for 1,1-difluoroethane (R152a), IJR, no.5, vol.16, 1993, pg.347.

[5].xxx-R134a Thermodynamic and physical properties-L'Institute International du Froid,Paris,1992.

[6] xxx - ASHRAE Handbook, 1985 Fundamentals, SI Edition.

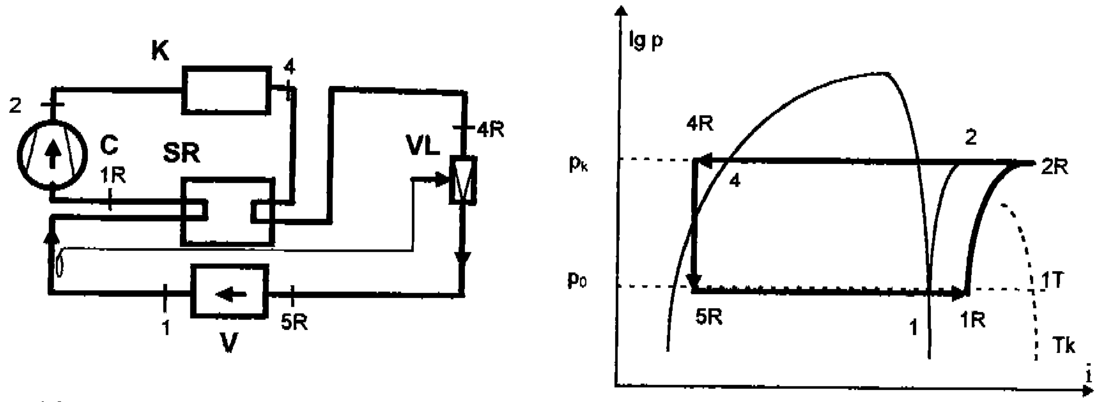

Figure 1. The scheme and the cycle for one stage of compression refrigerating machinery: C-compressor; K- condenser; SR- recuperative heat-exchanger; V- evaporator; VL- throttle

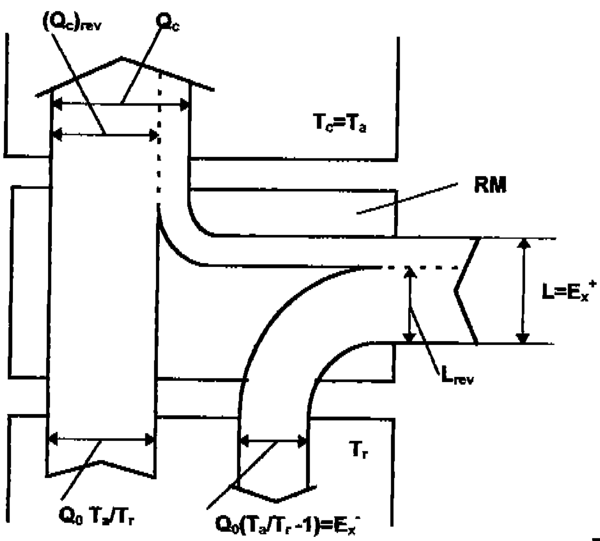

Figure 2. The exergetic flow chart for one stage of compression refrigeration machinery (RM) $T_c=T_a$ - hot-source temperature (environment); T_r - cold-source temperature (cold space)

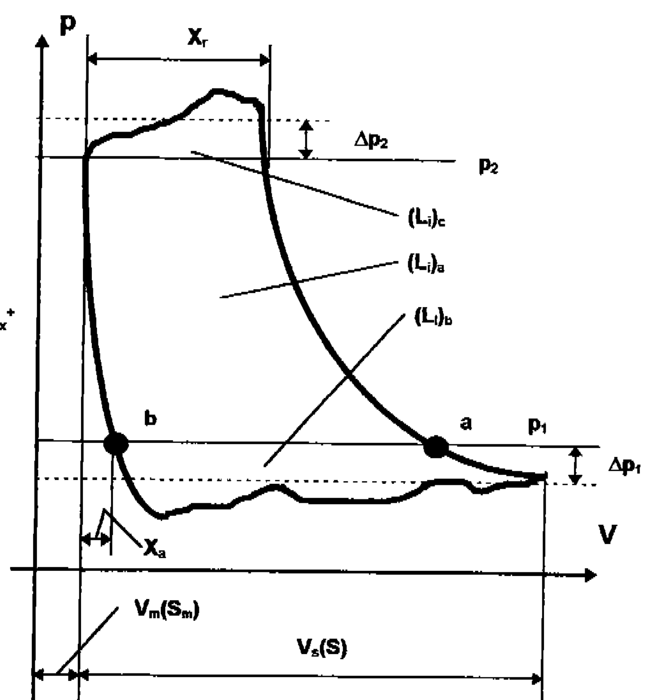

Figure 3. The compressor indicated diagram

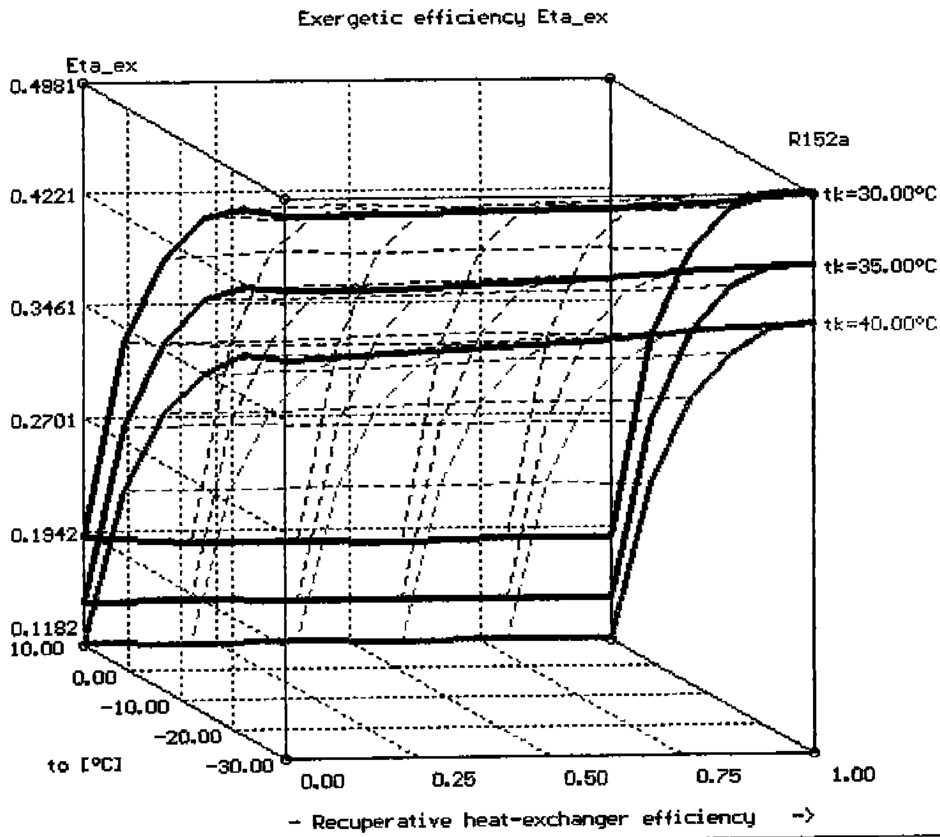

Figure 4. The exergetic efficiency dependence as function of the recuperative heat-exchanger efficiency η, the evaporator temperature t_0 and the condenser temperature t_k.