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AN EXERGY ANALYSIS OF VAPOR COMPRESSION HEAT RECOVERY SYSTEM

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Abstract

Energy analysis based on exergy method, was carried out for R-12 refrigeration machine, for cooling and heat recovery modes. The analysis of the individual elements and the system as a whole showed that, the exergy coefficient of performance of heat recovery mode is higher than that of cooling mode by 40 % while the energy saving exceeds 20 %. The survey of losses in individual elements of the system and ways of improvements were clarified.

Nomenclatures

t	temperature	^{o}C	W	Input power	kJ/kg
T	Temperature	K	E	Exergy	kJ/kg.
P	Pressure	mPa	A	Anergy	kJ/kg.
h	Enthalpy	kJ/kg	S	Entropy	kJ/(kg.K)
q	Heat	kJ/kg	L	Exergy losses	kJ/kg
η	Exergy coefficient.		Δ	difference	
Y	Coefficient of specific energy consumption.				

Subscripts

		HE	Heat exchanger
0	Outside air	Cm	Cooling mode.
e	Evaporation	Hm	Heat recovery mode
c	Condensation	Comp	Compressor
R	Cooled space	th	Throttling
H	Heated space	cd	Cooling demand
S	Surface	Hd	Heating demand
		CHd	Cooling-heating demand

1. Introduction

In the refrigeration system, heat is transferred from a low to high temperature level, by means of mechanical work. The losses due to the irreversible processes, in condenser can be utilized for heating. Utilization of the waste heat of condensation has received a great attenion with the development of heat pump [1]. The vapor compression system can be used for cooling and heat recovery (cooling - heating) modes. Researches of this field have two groups: first is directed to find ways of utilization of heat of condensation [2-6], second is dealing with evaluation of the system based on the coefficient of performance (COP).[1,2,4,7].The coefficient of performance (COP)includes the effect of condensation and evaporation temperatures, and does not express the effect of outside air temperature and losses associated with the processes. The aim of this paper is to clarify - by exergy method[8] the effect of outside air temperature, the individual losses of vapor compression system, and consequently the consumption of energy.

Case Study

Fig. 1.a, Fig. 1.b and Fig. 1.c show a schematic flow diagram of refrigerating machine, operating in cooling and heat recovery modes, and the corresponding Pressure-enthalpy and temperature - entropy plot for

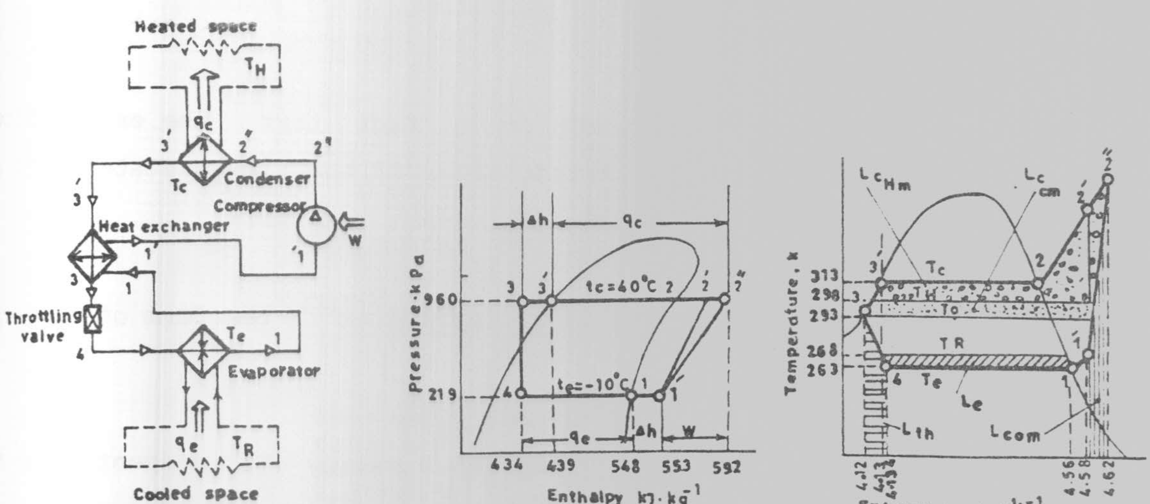

Fig.(1-a) Fig.(1-b) Fig.(1-c)

Fig.(1-a) Schematic flow diagram of heat recovery system

Fig.(1-b),(1-c) The pressure enthalpy and temperature - entropy plot for R-12 refrigerant circuit.

R-12 refrigerant circuit. The thermal conditions and assumed efficiencies are listed in table. I.

Table I. Thermal Condition and Assumed Efficiencies

Temperatures, K	
Outside air	283-293
Evaporation	253-273
Condensation	303-323
Heated space	298
Cooled space	258-283
Efficiencies, %	
Isentropic compression	70
Mechanical	90

3. The Exergy Analysis

The exergy analysis is based on a fact that, the exergy flow is determined unambiguously by the parameter of the flow's state (P,T) and the parameter of the surrounding (P_o, T_o).

The input exergy is supplied to the system in the form of electrical power to the compressor.

For cooling mode Fig. 2.a. the input exergy (W) is spent to provide exergy demand for cooling E_{cd} (available energy that necessary to absorb the anergy of the cooled space A_R), and to cover the irreversible losses of the cycle ΣL_{cm}.

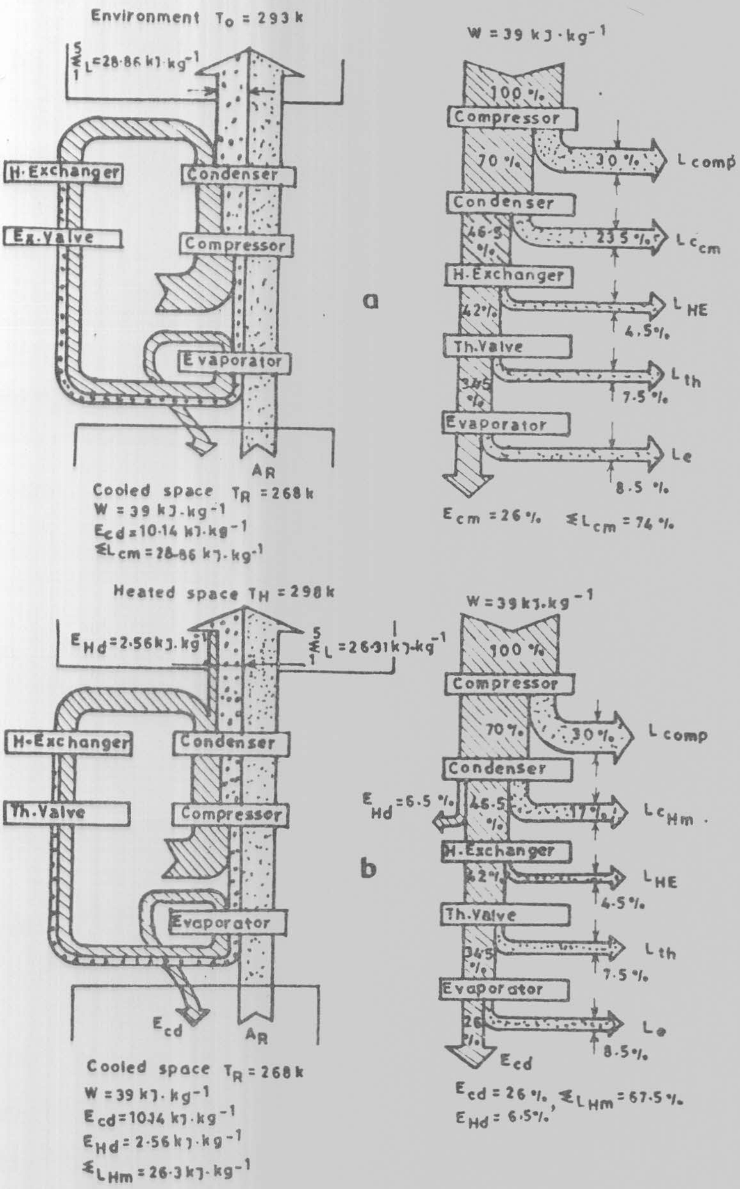

Fig.2 The distribution of exergy and losses through the main elements of the system.
a - Cooling mode.
b - Heat recovery mode.

$$W = E_{cd} + \Sigma L_{cm} \tag{1}$$

For heat recovery mode Fig. 2.b. the input exergy (W) is spent to provide exergy demand for cooling E_{cd}, heating E_{Hd} and to cover associated losses ΣL_{Hm}.

$$W = E_{cd} + E_{Hd} + \Sigma L_{Hm} \tag{2}$$

The exergy demand for cooling and heating may be determined as follows

Exergy demand for cooling, kJ/kg.

$$E_{cd} = q_e \, (T_o - T_R)/T_R \tag{3}$$

Exergy demand for heating, kJ/kg.

$$E_{Hd} = q_c \, (T_H - T_o)/\, T_H \tag{4}$$

The anergy of cooled space, kJ/kg.

$$A_R = q_e \cdot T_o/T_R \tag{5}$$

The exergy analysis showed that, for cooling mode E_{cd} is about 14-25 % of the input exergy while the remaining part (86-75%) to cover compression, throttling and heat transfer losses. The input exergy of the heat recovery mode, is spent to provide cooling-heating exergy demand $E_{c.H.d} = E_{cd} + E_{Hd}$ (20-40%) of the input exergy and the residual part (80-60%) to cover the above mentioned losses. Referring to Fig. (1-b), the exergy coefficient of performance may be determined as follows:

The exergy coefficient of performance for, cooling mode, is given by

$$\eta_{cm} = E_{cd}/W \tag{6}$$

The exergy coefficient of performance for heat recovery mode, is given by

$$\eta_{Hm} = E_{cHd}/W \tag{7}$$

Fig. 3, shows the exergy coefficients of performance η_{cm}, η_{Hm} versus the cooled space temperature t_{R}, for various condensing temperatures. From Fig. 3, it is clear that, the behaviour of the coefficients η_{cm} and η_{Hm} is the same.

The average value of the coefficient η_{Hm} is greater than the coefficient η_{cm} by 23-38 % at an outside air temperature T_o = 293 K. This is due, to the decreasing temperature difference through the condenser.

At an outside air temperature T_o = 283 K the ratio η_{Hm}/η_{cm} reaches about 200 %. This is due to the increasing effect of exergy demand of heating $q_c(T_H-T_o)/T_H$ and decreasing losses of the system as a whole.

The useful utilization of energy can be evaluated by the coefficient of specific energy consumption Y, expressing the relation of input power (W) to the useful exergy demand (E). Figure (4), shows the coefficient (Y) versus the cooled space temperature for various modes. The coefficient (Y_{Hm}) of heat recovery mode, is considerably lower than that of cooling mode (Y_{cm}). This is due to increasing the magnitude of useful portion of exergy, utilized in both cooling and heating. The

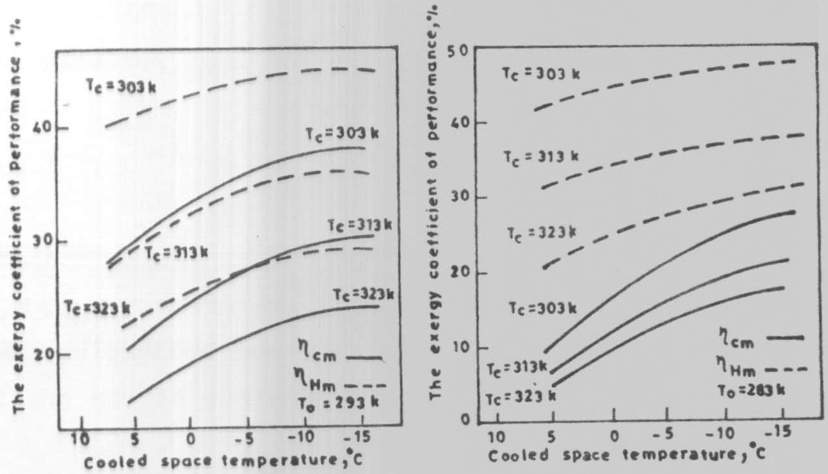

Fig.3 The effect of outside air temperature on the coeffieient η_{cm}, η_{Hm} .

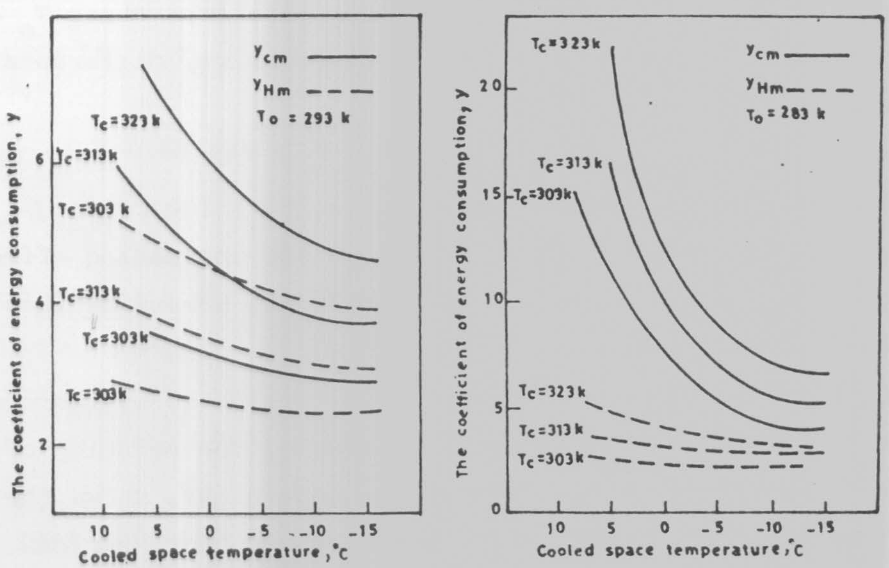

Fig. 4 The effect of outside air temperature on the coeffieient of energy consumption .

average percentage of energy saving $(Y_{cm} - Y_{Hm}) / Y_{cm}$ % is 21 and 60 at T_o = 293 and 283K respectively. The analysis of the main individual losses, specified as percentage of input power (W) shows that, the maximum losses of the liquid to suction heat exchanger, does not exceeds 5%, while the most dominating losses occur in the compressor, expansion device, evaporator and condenser are shown in Fig.5.

Compressor losses due to friction, clearance, and internal heat exchange of vapor with cylinder wall of compressor as shown in Fig. 5 a, were obtained referring to Fig. 1-c from the following relation,

$$L_{comp} = T_o \, (S_2'' - S_1') \tag{8}$$

These losses are connected with the compression ratio Pc/Pe, and condenser-evaporator temperature difference, and may be reduced by stagging compression with intercooler. As well as two cascade cycle with different working media operating in two levels.

The losses of expansion device are calculated as follows, in kJ/kg,

$$L_{th} = T_o \, (S_4 - S_3) \tag{9}$$

As shown in Fig. 5b throttling losses increase with the decrease of cooled space temperature and whereas the condensing temperature (t_c) increases. These losses can be reduced in number of ways: recovery of expansion work, subcooling of liquid before expansion, and step wise throttling and recompression of flash gas. From Fig. 5b the average reduction of throttling losses due to the subcooling of liquid is 30 %. Although the power is mainly consumed in driving the compressor in

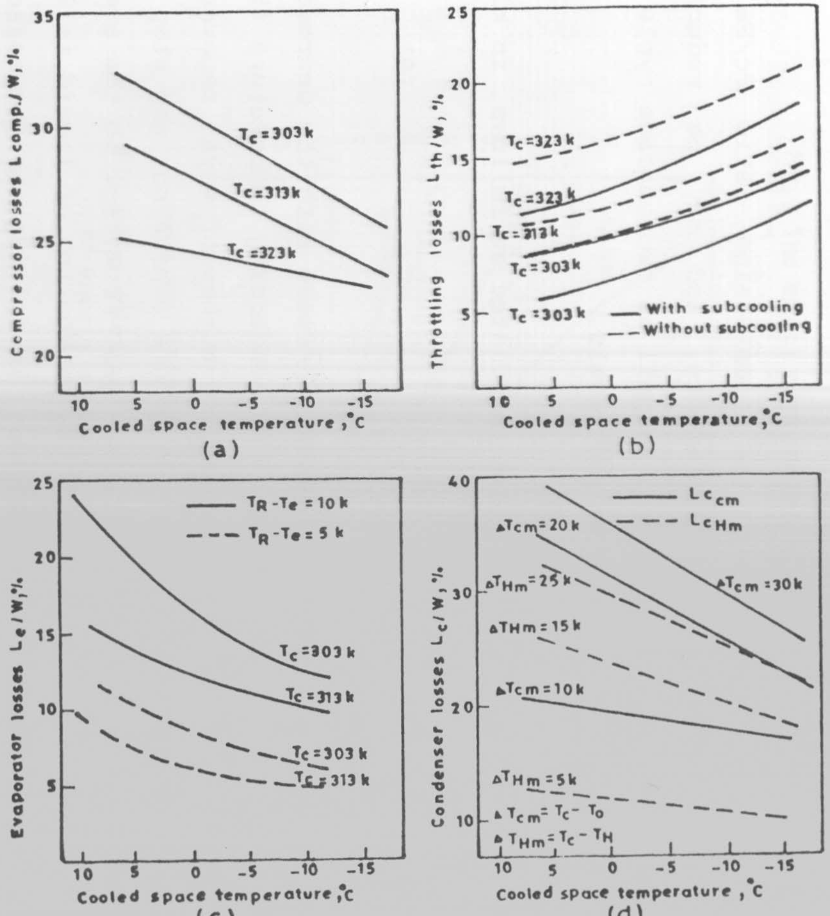

Fig.5 The relative exergy losses in the main elements of the system.

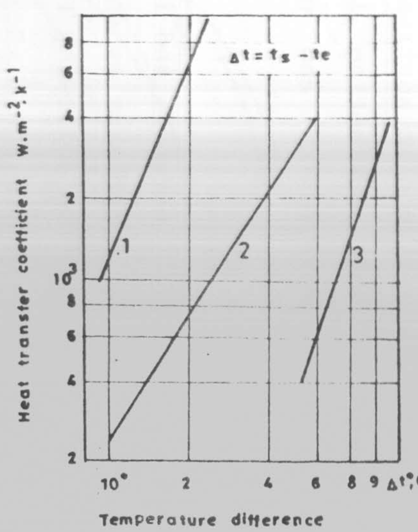

Fig. 6 The coefficient of heat transfer of boiling R-12 versus temperature difference at $t_e = -20\,^\circ C$, /9/ .

1- Surface with porous metallic coating
2- Finned surface 3- Smooth surface

common refrigerating plant, adominating part of the losses are due to inefficient heat exchange in evaporator and condenser. The losses of evaporator are determined referring to figures (1-b) and (1-c), by the following equation in kJ/kg:

$$L_e = [\ E_{4-1} - E_{cd}] \tag{10}$$

$$L_e = [\ T_o\ (S_1 - S_4) - q_e - E_{cd}] \tag{11}$$

These lesses are mainly due to the temperature difference (T = (T_R - T_e), and illustrated by Fig. 5c, from which it is clear that, these losses decrease as temperature T_R drops, i.e as the temperature difference (T_R - T_e) decreases.

Condenser losses due to the temperature difference, were considered for cooling and heat recovery modes. These losses are shown in Fig. 5d, and calculated referring to Fig. (1-b), (1-c) by the following equations in kJ/kg.

For cooling mode ($T_{cm} = T_c - T_o$)

$$Lc_{cm} = E_{2''-3'} = q_c - T_o(S_2'' - S_3') \tag{12}$$

For heat recovery mode ($T_{Hm} = T_c - T_H$)

$$Lc_{Hm} = E_{2''-3'} - E_{Hd} = q_c - T_o\ (S_{2''} - S_{'}) - E_{Hd} \tag{13}$$

From Fig. 5d, it is clear that, the behaviour of the losses of both modes is the same, regarding the temperature (t_R) and temperature difference. They decrease with the decrease of both cooled space temperature and the temperature difference. In the case of heat recovery mode condenser losses are about 30 to 50% less than that of cooling mode, this is due to the small temperature difference of heat recovery mode compared with that of cooling mode.

The analysis showed that, the evaporator is more affected by the temperature difference than condenser. For example, an increase of temperature difference by $5C^{o}$ leads to increasing of evaporator losses by 70 to 80% against 30 to 50% for condenser for the same thermal conditions. The influence of the temperature difference can be reduced by introducing apparatus of new design. Considerable improvements, to the heat transfer surface, and consequently to the coefficient of heat transfer may be obtained using evaporators with porous metallic coating [9], to the evaporator surface. Fig. 6, illustrates the dependence of the coefficient of heat transfer, of boiling R-12 on temperature difference, at an evaporating temperature $t_e = -20^{o}C$, for various type of evaporators. The comparison shows that, evaporator with surface covered by porous coating, is less affected by temperature difference. For heat transfer coefficient equals 2000 $W/(m^2k)$ the corresponding Δt are 1.2, 4 and 8.5 ^{o}C for porous coated, finned and smooth surfaces respectively.

4. Conclusion

Analysis based on exergy method, has the advantage to evaluate the perfection of individual elements and the system as a whole. This indicates ways to reduce losses and energy consumption. One improvement is, to reduce the temperature head of the cycle by application of multiple stage compression and throttling to expand operating range. Other possibility is the reduction of temperature difference in heat exchanging apparatus specially in evaporator. This can be achievd by introducing apparatus of new design with good heat transfer conditions.

The above points do not exhaust all the possible ways of economizing on energy consumption for vapor compression heat recovery system, as they are still in an early stage of development.

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