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A REVIEW OF HOUSEHOLD COMPRESSOR ENERGY PERFORMANCE

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ABSTRACT

There are many published articles discussing questions related to the performance of vapor compression refrigerators. Several aspects are dealt with but it is not easy to understand the limit that such kind of systems can reach, compared to the maximum value, or the Carnot efficiency. In fact the limit is very well known for the ideal cycle, but so many last minute improvements were made to the real system that sometimes it is difficult to say how far real vapor compression systems will be from the ideal ones in the near future. This paper presents an energy efficiency comparison between the ideal Carnot refrigerator and the performance of a vapor compression system applying high efficiency compressors. The Rankine cycle and compressor losses are evaluated in detail showing how far the application is from the ideal energy performance. For the compressor, which nowadays can reach up to 50%of the ideal efficiency, the losses are presented and some ways of improving performance are discussed. A general overview is also given on further developments in this area.

1. INTRODUCTION

The development of high energetic efficient systems has been common practice in the refrigeration industry for the past 35 years, to meet both market and legislative requirements. Since the beginning of the 70's, or possible even before, with the advent of this era of the "Energy Crisis", a huge amount of work has been done towards improving energy efficiency. Coates (1972) describes commonization and performance improvement by changing some geometrical parameters in a family of compressors while Johnson (1974) points out the importance of the motor efficiency over the EER for an air conditioning system. According to Cohen *et al* (1974), products could be made more energy effective on a life cycle basis through the use of variable capacity compressors.

Shaffer (1976) and Jacobs (1976), present the energy losses of hermetic compressors, where mechanical, valve, suction gas heating, mean manifold pressure drop, leakage and heat transfer losses are described. An EER as high as 4.78 BTU/Wh for 1067 BTU/h refrigerating capacity according to ASHRAE LBP conditions, is shown by Riffe (1976). Schroeder (1976) comments that the use of PSC motor with start assisted PTC improving the electrical motor from 73 % to 80%. Peruzzi (1980), mentions a correlation between EER and the compressor running frequency, showing benefits for lower frequencies, however, experiments have not confirmed that effect. Direct suction was the best solution for EER improvement.

During the 80's and 90's lots of articles were published dealing with EER improvements by reducing losses related to superheating, friction, suction and discharge valves and manifolds and electrical motor. There are still some non conventional solutions as a condenser integrated to the compressor like that presented by Giuffrida (1984), or high pressure inside the shell for reciprocating compressors, according Duane Fry (1992), is in fact usual for rotary compressors. From that time, till now, numerical simulation has been widely applied for optimization of EER, noise and reliability issues (see for instance Deschamps *et al* (1988), Prata *et al* (1988) and Todescat *et al* (1994)).

The 90s was also a period in which different refrigerant gases were analyzed and implemented for environmental concerns such as R12 replacement for household refrigeration. Gases like R134a and R600a had some impact over the energetic performance, as Fagotti (1994) has shown. Currently, variable capacity compressors and linear mechanism are being indicated as the most promising solutions for energy efficiency, according to Krueger *et al*

(1994) and Lamantia *et al* (2002).

The major question is how far all those kinds of improvements can go. Riffe (1992) wrote: "Considerable experimentation will be required in order to determine how far we can go". On the other hand, it is well known that the limit is the Carnot COP. This work intends to compare the ideal limit with the efficiency of present day compressors, by analyzing the main energy losses found in hermetic compressors for household applications.

2. CARNOT REFRIGERATION CYCLE

In 1824 a French engineer called Sadi Carnot published a treatise called "Reflections on the Motive Power of Heat". In this paper, Carnot showed that the maximum possible efficiency of a thermal machine, working between two temperatures, depends only on these temperatures. Therefore, Carnot's efficiency is independent of the way the refrigerator operates and of the refrigerant in use, or even the refrigeration principle applied.

The Carnot refrigeration cycle is shown in figure 1:

1 - 2: reversible adiabatic compression
2 - 3: isothermal heat release
3 - 4: reversible adiabatic expansion
4 - 1: isothermal heat admission

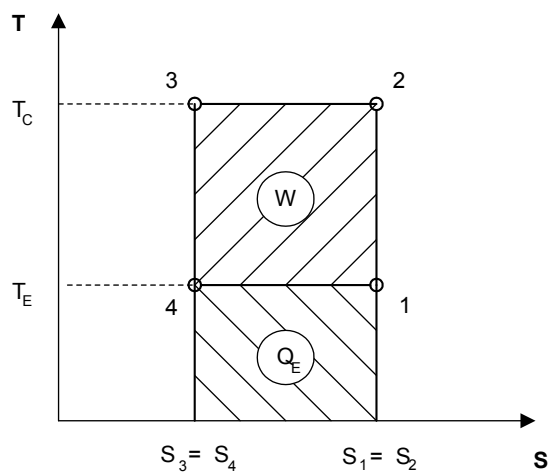

Figure 1: T-S diagram for the Ideal Carnot cycle

The performance coefficient is the relationship between the quantity of heat removed in the evaporator and the net work performed:

$$\varepsilon_c = \frac{Q_E}{W}; \quad W = Q_C - Q_E \quad (1^{st} \text{ law}) \tag{1}$$

$$\varepsilon_c = \frac{Q_E}{Q_C - Q_E} \tag{2}$$

$$\varepsilon_c = \frac{T_E(S_1 - S_4)}{T_C(S_2 - S_3) - T_E(S_1 - S_4)}; \quad S_1 = S_2 \text{ and } S_3 = S_4 \tag{3}$$

$$\varepsilon_c = \frac{T_E}{T_C - T_E} \tag{4}$$

To achieve the maximum performance coefficient there must be no difference between the cold environment inside the refrigerator and the refrigerant fluid that is removing heat from this environment. In a similar way, there should be no difference between the refrigerant fluid and the environment in which the heat is being released.

Table 1 shows the Carnot Coefficients of Performance for two typical refrigeration system operating conditions in the European market and the North American market respectively. Those numbers are the maximum reachable energy efficiency, with the principle of producing cold not being important.

Table 1: Carnot COP considering no difference between refrigerant and environment

T_E	T_C	ε_C (W/W)
255 K(-18°C)	298 K (+25°C)	5.93
258 K (-15°C)	305 K (+32 °C)	5.49

3. VAPOR COMPRESSION PROCESS AND COMPRESSOR LOSSES DISTRIBUTION

In practice it is impossible to remove or reject heat without temperature differences between the means. These differences can be reduced but never eliminated entirely. According to Gosney (1982) the temperature differences commonly encountered are in the range of 5 – 15 K, both in the evaporator and the condenser.

Those differences depend on project characteristics such as heat exchange area in the evaporator and condenser, the use or not of forced ventilation, gas load, capillary dimension, among others. The T-S diagram presented in figure 2 illustrates this fact. A typical vapor compression system diagram is presented in figure 3.

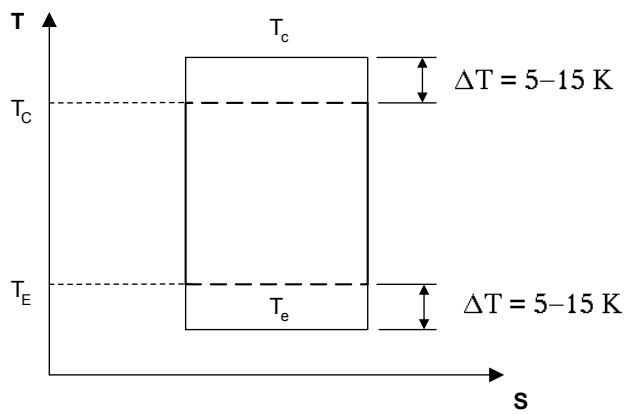

Figure 2: T-S diagram for the Ideal Carnot cycle with the application ΔT

If one considers a temperature difference of 12 K in the evaporator and 10 K in the condenser, which is the average of some of the systems tested at EMBRACO's facilities with on-off standard compressors, it is possible to obtain the Carnot Coefficients of Performance that are presented in table 2.

Table 2: Carnot COP considering the application ΔT between refrigerant and environment

T_e	T_c	ε (W/W)
243 K(-30°C)	308 K (+35°C)	3.74
246 K(-27°C)	315 K (+42 °C)	3.57

This means a reduction in the range of 35-37% in the performance coefficient of a refrigeration system operating in a reversible mode compared to the Carnot cycle for the regulation required temperatures. The performance coefficient reduction shown above indicates the importance of an adequate sealed unit project in such a way as to

minimize the ΔT. Increasing the heat transfer areas or the global heat transfer coefficient are two ways of reducing the ΔT.

Another way that has been in use more recently is the application of variable capacity compressors. In these compressors the refrigeration capacity is adjusted according to the application demand. Most of the time this capacity is much lower than the maximum compressor refrigeration capacity and the result is a compressor most of the time operating at low speed for a longer period of time.

Consequently, the main benefits of a variable capacity compressor are: greater compressor efficiency, since at lower speed the friction losses and the flow losses are lower than the velocities of the on/off compressors; reduced start-up and heat reflow losses and lower temperature difference (ΔT) because the amount of heat to be removed or rejected is the same and the compressors with variable capacity remain in operation longer.

In the vapor compression process, the function of the compressor is to remove the vapor from within the evaporator, increase the vapor pressure and release it to the condenser. The reciprocating hermetic compressors are those most commonly found, especially in refrigerator and freezer type applications.

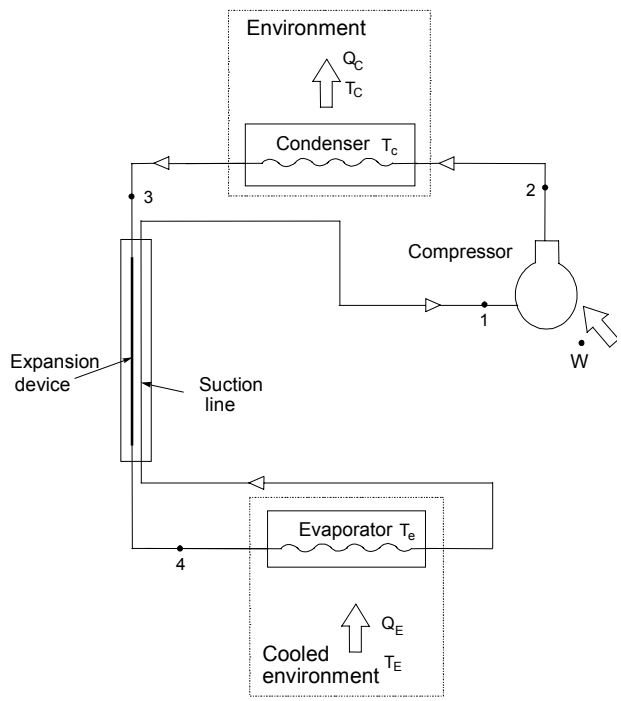

Figure 3: Typical vapor compression system

Table 3 represents a comparison between the two previously calculated efficiencies and the current compressor efficiency operating under the same condition:

Table 3: Comparison between ideal COP and real compressor COP

Cycle	**Condition**	**Efficiency (W/W)**
Ideal cycle (Carnot cycle)	T_E = -15°C and T_C = 32°C	ε_c = 5.49
Ideal cycle with ΔT	T_e = -27°C and T_c = 42°C	ε = 3.57
Compressor COP (measured R134a at 60Hz)	T_e = -27°C and T_c = 42°C	ε_{comp} = 1.85

As shown in table 3 the compressor efficiency is approximately 50 % of the Ideal cycle with ΔT. During the calorimeter compressor evaluation, the cycle's sub cooling and superheating temperatures were kept as 32°C. The

cooling capacity for this testing condition is very similar to the one that occurs in a system without sub cooling and superheating, but with a quality of 15 % at the evaporator entrance, as presents figure 4.

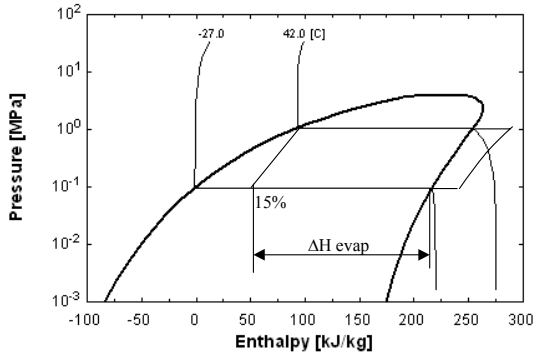

Figure 4: Evaporator enthalpy difference in a vapor compression system for R134a

The main energy losses found in reciprocating hermetic refrigeration compressors are:

- cycle losses (compression and expansion);
- thermodynamic losses;
- friction losses;
- electrical losses.

In the last 20 years a lot of effort has been made to increase compressor efficiency. As a result compressor efficiency has increased approximately 60% and figure 5 shows how the coefficient of performance has been improved in the last decades and how the losses affect the COP for current high efficient compressors for R134a at 60Hz.

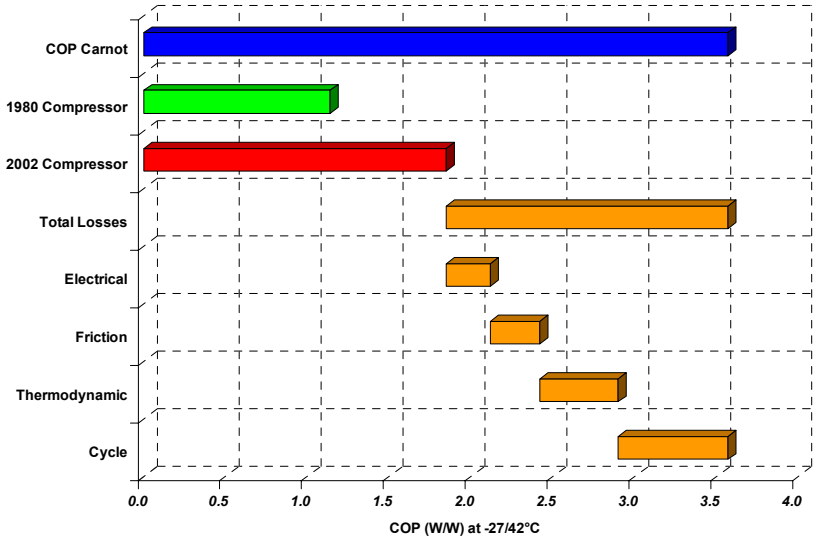

Figure 5: Losses distribution for hermetic compressors

In the Carnot cycle, the work produced in the expansion is used, while in the vapor compression cycle no expansion work is used. Added to this, in the ideal Carnot cycle, there is an isentropic compression until the condensing temperature and then compression so that it will be isothermal. On the other hand, in the vapor compression cycle, there is an isentropic compression process up to the condensation pressure. Adding these two losses is what we define as cycle losses of the vapor compression cycle in relation to the ideal Carnot cycle. Actually, the real process is polytropic, with a polytropic coefficient that is strongly influenced by gas thermodynamics and the compressor design.

The thermodynamic losses are linked to the flow of refrigerant gas inside the compressor. In the suction system the main losses are the superheating in the suction muffler, the flow losses in the muffler and the suction valve and the backflow in the suction valve. In the discharge system the main losses are the losses through the discharge valve and the discharge muffler, and the backflow in the discharge valve. Another loss under consideration here is leakage through the clearance between the piston and the cylinder. The plastic suction muffler, optimization of the suction and discharge valves have provided considerable improvements to the efficiency over recent years.

The reciprocating compressors have a mechanism with connecting rod to transform the rotating movement of the motor into alternate movement of the piston. The main friction losses of this mechanism occur in the following components: journal bearing; thrust bearing and piston vs. cylinder. Over the past few years friction losses have been reduced by the use of the low viscosity oils, thrust ball bearing, lower diameter shafts, pistons with undercut and optimized geometrical design.

The electric losses can be divided into losses in the electric motor and in the starting device. The losses in the electric motor of a compressor can be divided into: copper losses, steel losses and rotor losses. Steel with lower losses, higher fill factors, improvements in the manufacturing process, optimized lamination design and the electronic starting device are the main improvements that enable motor efficiency of around 87%.

Table 4 presents the efficiencies related to the above described losses. It is interesting to note that there is no longer an expressive loss to be reduced. In fact all of them have similar values and there will be improvements to all the internal parts in the future generations of high efficiency compressors resulting in a significant COP benefit.

Table 4: Compressor efficiencies

Losses	**Efficiency**
Cycle	81.2 %
Thermodynamic	83.4 %
Friction	87.6 %
Electrical	87.3 %

Over the coming years, electrical motors with permanent magnetic rotor will reach higher efficiencies. Friction losses may be reduced with linear compressors. Surely new engineering solutions based on numerical optimization and experimental tools will reduced the thermodynamic losses somehow. However, it is very difficult to say much about the cycle efficiency as this depends on machine design and refrigerant properties.

4. CONCLUSIONS

In the previous chapters a comparison was made between Carnot's ideal cycle and the compression vapor cycle. The temperature difference in the heat exchangers (evaporator and condenser) commonly found in the refrigeration systems represents a loss of approximately 35 % of efficiency when compared to the Carnot cycle for the required working temperatures.

In our opinion, one way of improving system efficiency is to reduce the ΔT between the evaporation temperature and the refrigerated environment and the condenser ΔT with the outer environment. As previously stated, the use of variable capacity compressors is one of the ways of reducing this ΔT.

The data shows that compressor efficiency has increased by about 60% in the last 20 years. In spite of this effort only 50% of the efficiency of an ideal cycle has been achieved taking into account the evaporator and condenser temperatures.

In order to increase compressor efficiency all compressor losses were reduced. Thus, there is no preponderant power loss in the compressor today. Future efficiency improvements will therefore come from improvements on each of the components, and not from one specific area.

Based on this study, we can conclude that the vapor compression cycle and the reciprocating hermetic compressors still have potential for significant efficiency increases. Emphasis must be given to the development of new materials, the use of more efficient mechanisms and the use of compressors with the concept of variable capacity if compressor efficiency is to be improved over the next few years.

NOMENCLATURE

T	Temperature (K)	S	Specific entropy
T_E	Inner environmental temperature	T_e	Evaporating temperature
T_C	Outer environmental temperature	T_c	Condensing temperature
Q_E	Quantity of heat removed from inner environment by evaporator	Q_C	Quantity of heat transferred from condenser to outer environment
EER	Energy Efficiency Ratio	W	Net work performed
ε_c	Performance coefficient of a Carnot cycle operating between T_E and T_C	ε	Performance coefficient of a refrigeration system operating between T_e and T_c
ε_{comp}	Compressor efficiency	ΔT	Temperature difference between the refrigerant fluid and the internal or the outer environment of the refrigerator

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