

White paper

Thermodynamic limitations and opportunities for reaching high energy-efficient refrigeration, heat pump and air conditioning systems

In the refrigeration systems industry, energy consumption and energy efficiency are the subject of ongoing debate. In this paper, the theoretical limitations of the Carnot process, as well as the more realistic theoretical approach of the vapour compression refrigeration cycle are examined.

By comparing these theoretical processes to real refrigeration systems, we can identify ways to improve the energy efficiency of the vapour compression cycle. And by evaluating data from existing refrigeration systems, the energy efficiency of current state-of-the-art systems can be assessed.



Refrigeration processes

Often overlooked, the vapour compression refrigeration cycle deserves a reappraisal, and is therefore the focus of this paper. Energy-efficient as well as economical, the vapour compression process is widely used in refrigeration, heat pump and air-conditioning systems. Other energy conversion processes, such as sorption, magnetic and the Joule processes are mainly applied in niche applications.

The basics of the vapour compression cycle remain unchanged whether used in refrigeration systems to provide cooling below the ambient temperature,

in heat pumps to provide heat above the ambient temperature, or in air-conditioning systems to provide cooling at slightly below ambient temperatures. However, as definitions of energy-efficiency vary slightly from application to application, in the interests of consistency, this paper will use the refrigeration system definition.

Evaluating the energy efficiency of refrigeration cycles

The Carnot cycle

The Coefficient of Performance (COP) is used to evaluate the energy efficiency of the vapour compression refrigeration cycle. In a refrigeration system, COP expresses the relation between the refrigeration capacity $\dot{Q}_K$ (benefit) and the total power P (effort) required to achieve refrigeration.

$$COP = \frac{\dot{Q}_K}{P}$$

Although the COP has no upper limit, evaluating the energy efficiency of a specific refrigeration system does require a theoretical limit.

Nicholas L. S Carnot expressed this limitation in the Carnot cycle that he developed in the early 19th century. According to the Carnot cycle, the heat amount $\dot{Q}_K$ is withdrawn at a constant temperature T_K from the area to be refrigerated. And the heat amount $\dot{Q}_W$ is rejected to the ambient at a constant temperature T_W . To do this, work (P) is required. The energy balance is expressed by:

$$P = \dot{Q}_W - \dot{Q}_K$$

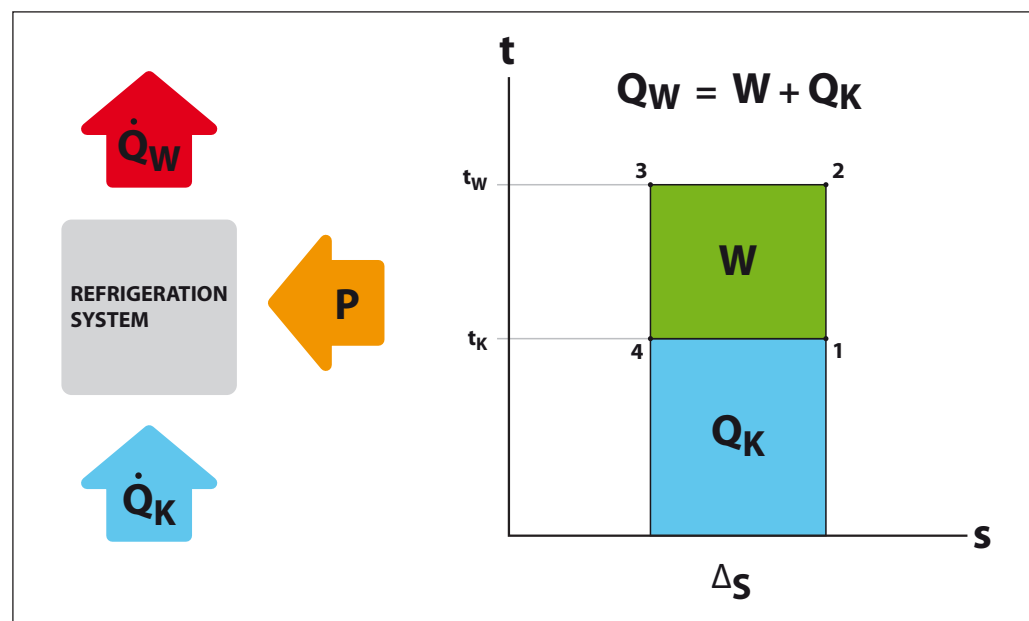


Figure 1: The Carnot refrigeration cycle

The Carnot cycle, which is reversible, is a perfect model for a refrigeration cycle that operates between two fixed temperature levels because it defines the maximum possible COP. The COP_{Carnot} of the reversible Carnot cycle depends on temperatures only.

$$COP_{Carnot} = \frac{T_K}{T_W - T_K}$$

The COP_{Carnot} values are quite high for small differences between temperature T_W and T_K as shown in Figure 2. But with large temperature differences and low refrigeration temperatures, the COP_{Carnot} value drops significantly.

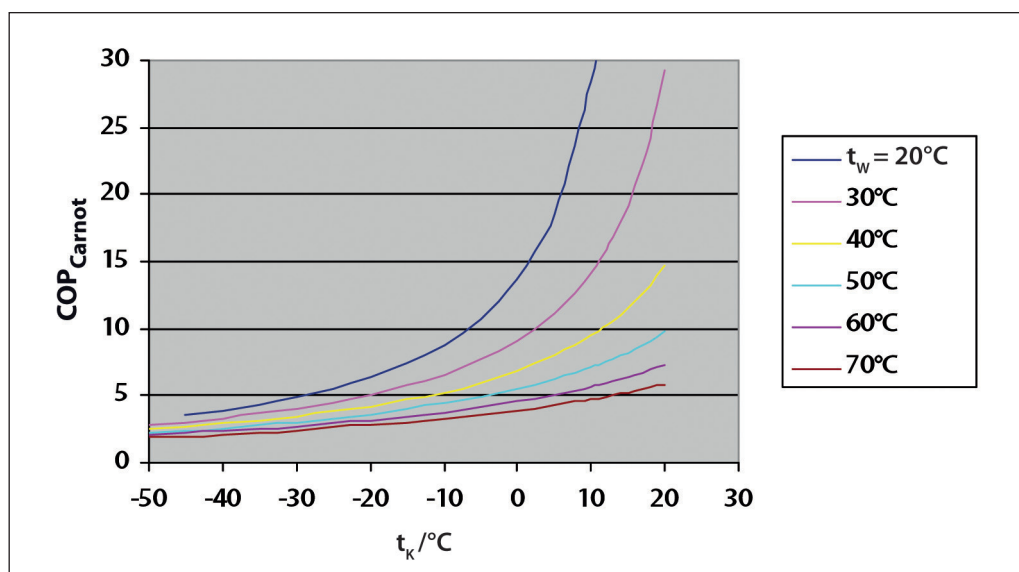


Figure 2: COP of Carnot cycle

As the reversed Carnot cycle assumes ideal processes, it is not applicable to real working systems.

Theoretical refrigeration cycle

A system designed to model the ideal Carnot cycle is shown in Figure 3.

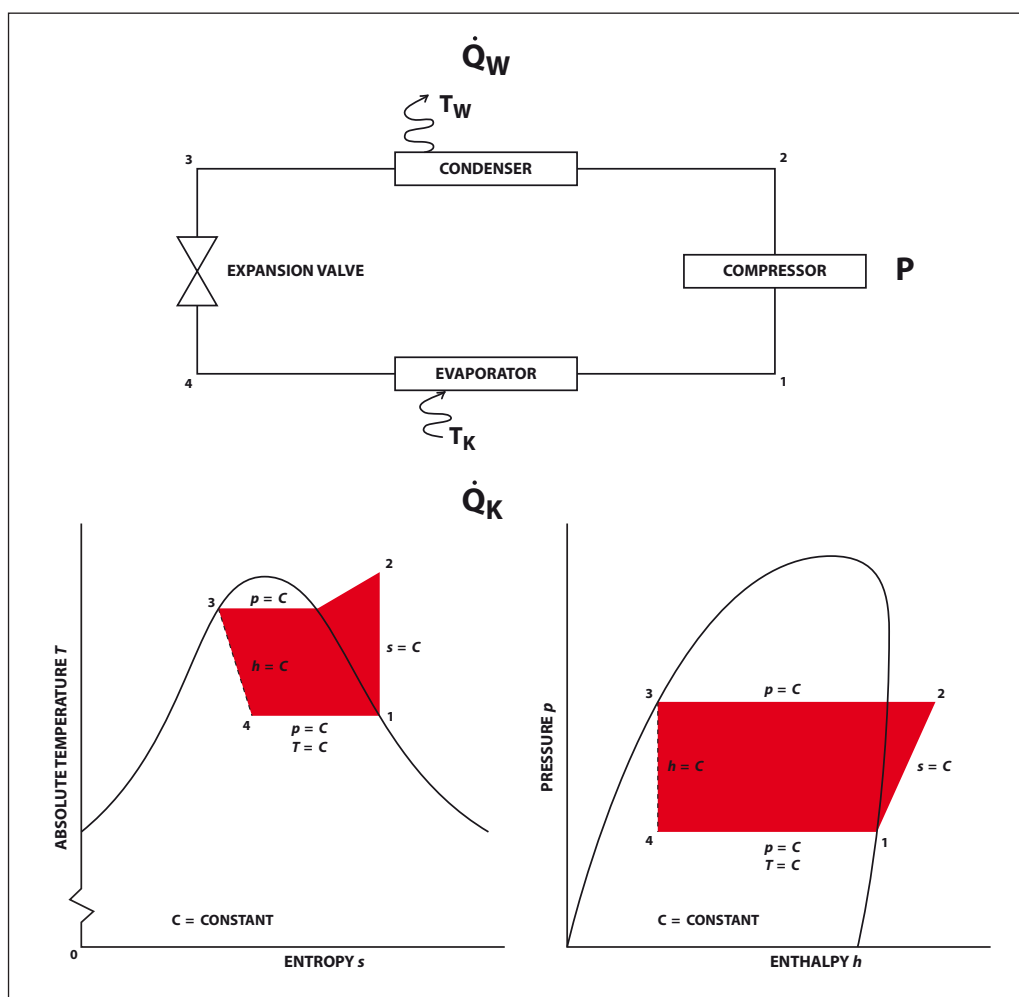


Figure 3: Theoretical single-stage vapour compression refrigeration cycle.

In order to achieve heat addition and rejection at constant temperatures, as defined by the Carnot cycle, the process is placed in the two-phase region of the refrigerant. Here, the refrigerant can maintain a constant temperature during phase change by maintaining a constant pressure.

There are two major differences with this model compared to the reversed Carnot cycle:

- Due to high initial costs and increased maintenance requirements, the machine used for the isentropic expansion of the refrigerant is replaced by a simpler expansion device that throttles the refrigerant isenthalpically from high to low pressure. Because of the use of a simpler expansion device, such as a thermostatic or electronic expansion valve, a capillary tube to throttle the refrigerant, energy is wasted that could be recovered by a more complicated expansion machine. Current expansion machines are economical in large capacity refrigeration systems only.

- For increased reliability, the compression process should take place outside the two-phase region. Therefore, the temperature at point 2 in Figure 3 is normally higher than T_w . Part of the condenser has to be used for de-superheating and the heat rejection process at point 2 and 3 does not occur at a constant temperature anymore.

Due to these two changes, the theoretical cycle is no longer reversible.

As the corner points of the theoretical cycle depend on the refrigerant properties, the COP_t differs from refrigerant to refrigerant.

The Carnot factor of the theoretical cycle is a measure of how close the cycle is, compared to the ideal reversible Carnot process.

$$\eta_t = \frac{COP_t}{COP_{Carnot}} = \frac{COP_t}{\frac{T_0}{T_c - T_0}}$$

Figure 4 illustrates the Carnot factor of the theoretical cycle for different refrigerants.

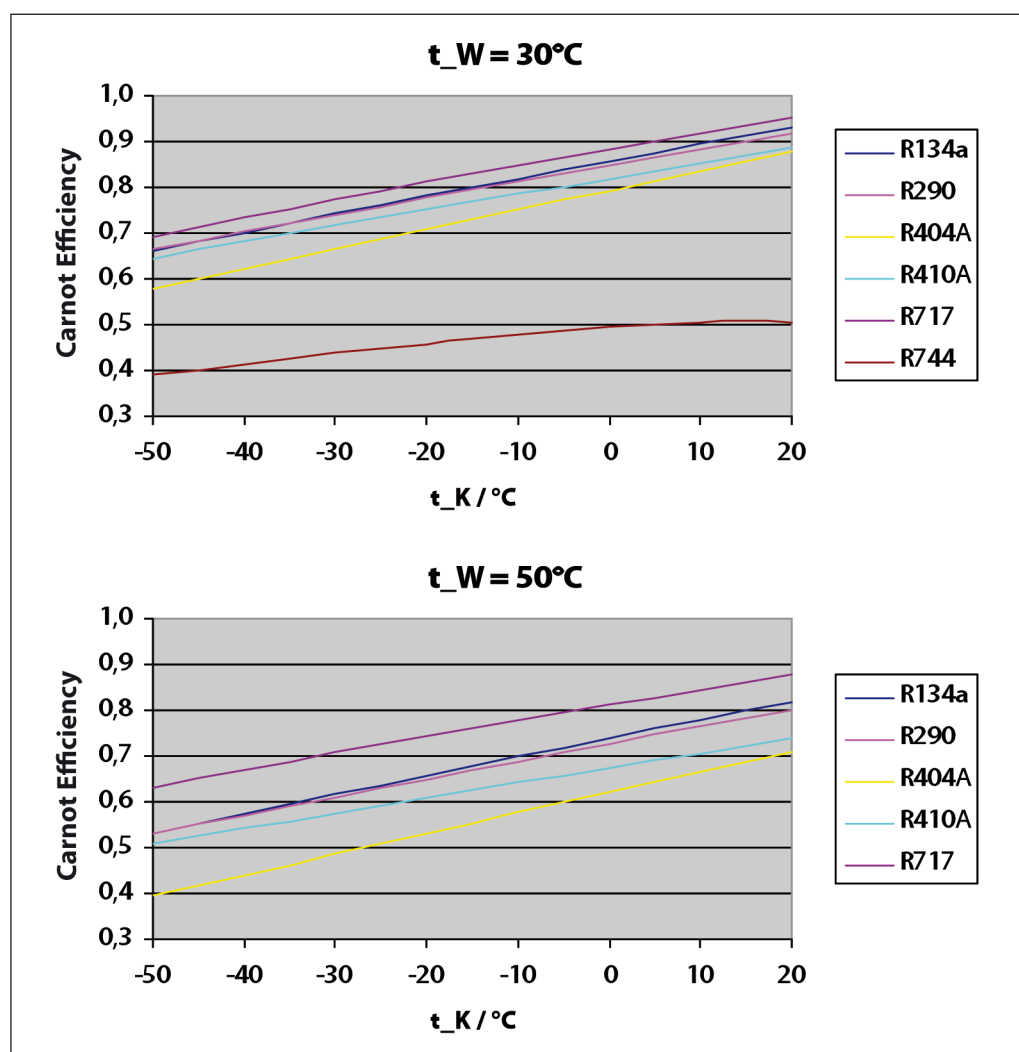
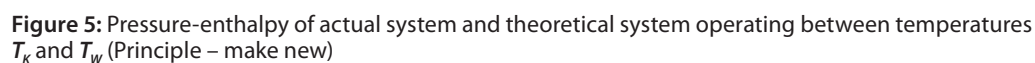


Figure 4: Carnot efficiency of the theoretical cycle with different refrigerants

Real refrigeration cycle

For energetic optimisation of the whole system, it is useful to understand how these irreversibilities are distributed throughout the system and which components should be replaced or redesigned to improve performance.



Compared to the Carnot refrigeration cycle, the real refrigeration cycle has the following losses:

a. Compression

In most cases, the component with the largest loss is the compressor. This loss is due to motor inefficiency, friction losses and irreversibilities caused by pressure drops, mixing, and heat transfer between the compressor and the surroundings.

The efficiency of the real compression process compared to the ideal process is rated by the isentropic compression efficiency η_{is} . Typical values for smaller hermetic compressors are 0.5, for larger semi-hermetic types 0.6, and for higher capacity screw or centrifugal compressors, 0.7. The maximum values can be higher than stated here, because compressors do not always operate under optimal conditions.

Efforts to increase isentropic efficiency focus on improving the efficiency of the electrical motor. This is done not only for efficiency at the nominal rating point, but for efficiency over the entire operating range of the motor. Further improvements can be realised by, for example, minimising the pressure loss through the suction and discharge valve, or reducing mechanical losses in the compressor's drive mechanism.

b. Heat Exchangers

In heat exchangers, a temperature difference is required to transfer heat into the evaporator and out of the condenser. This lowers the evaporation temperature T_e below the ideal value T_K , and increases the condensation temperature T_c above the minimal value T_w , thereby lowering the COP of the real cycle. The temperature difference on both sides can be reduced by a

larger or more efficient heat exchanger, providing this is efficient in terms of both cost and energy.

But even with an optimal evaporator, a certain temperature difference is required as some degree of refrigerant superheating at the evaporator outlet is required. In several applications, such as residential air-conditioning units or heat pumps, superheating limits the evaporation temperature. The superheat should therefore be as small and stable as possible.

c. Pressure losses

When a fluid runs through a pipe it loses pressure, which has to be compensated for by a compressor. Larger pipes would reduce the pressure loss, but might reduce the oil return inside the system. Again, the ideal level must take into account energy and economic factors, and also oil return issues.

Besides minimising the above losses, the type of refrigerant can have a significant impact on the COP of the system.

Although this article only examines the influence of the thermodynamic cycle on refrigeration systems, other components also affect the total energy consumption, such as fans, pumps and defrost heaters.

Efficiency values of existing systems

In an attempt to document the efficiency of existing systems on the market, representative data for four applications (Table 1) has been gathered. Please note that there are more and, of course, less efficient systems available. The COP values are compared to the COP of the ideal Carnot cycle, as well as the COP of the theoretical cycle. In all cases, the

data is valid for operation at a nominal operating point, while in reality other operating conditions may apply.

Table 1: The COP and Carnot factor of the ideal Carnot cycle, the theoretical cycle and available refrigeration systems in different applications

Application	Refrigerant	t_K	t_W	COP_{Carnot}	COP_{theor}	η_{theor}	COP_{real}	η_{real}
Air-water heat pump (1)	R290	2.0	35.0	9.34	7.61	0.81	3.2	0.34
Residential air Conditioning	R410A	26.7	35.0	35.98	31.58	0.88	3.01	0.08
Cold room refrigeration	R134a	5.0	32.0	10.30	8.91	0.87	1.85	0.18
Cold room freezer	R404A	-18.0	32.0	5.10	3.58	0.70	1.12	0.22

(1) For the heat pump, the COP definitions for heating systems are applied:

$$COP = \dot{Q}_c / P \text{ and } COP_{Carnot} = \frac{T_w}{T_w - T_K} = \frac{T_K}{T_w - T_K} + 1$$

The data in Table 1 shows that the COP of the theoretical cycle is still quite close to the ideal COP of the Carnot cycle. However, the COP drops rapidly with increasing temperature differences between the cold and the warm side due to the losses discussed earlier. For some applications, the Carnot factor of the real systems η_{real} is significantly below the limit of the theoretical cycle η_{theor} . The high Carnot factor of the heat pump η_{real} shows that this application has been improved significantly in order to achieve a higher heating efficiency. For a fair evaluation it should be noted that the values of the real systems include the energy consumption of auxiliaries, in particular fans. Also, in reality, lower supply temperatures than those given in the table are required.

An interesting, simple evaluation proposed in VDMA Einheitsblatt 24247 [1] can be used to compare different systems. Using four efficiency factors, it can support the designer in the evaluation of the total efficiency of the system and the selection of components.

Seasonal efficiencies

The above considerations regarding the efficiency of a refrigeration system are based on the process, the components and the refrigerant, assuming stationary operation under fixed conditions. At present, in most cases refrigeration systems are selected and compared at rated conditions. These design conditions meet peak demand, while in reality most systems operate at lower loads. Although it's inefficient, the conventional way to deal with varying refrigeration loads is on-off operation. Variable capacity systems, using continuous or step control, can more closely match varying loads and improve system efficiency.

Fluctuating operating conditions over a period of time, for example a whole year, can be assessed more accurately by comparing equipment at representative reduced capacities, and by determining a seasonal efficiency. For a refrigeration system, this may lead to a design in which the highest efficiency has to be achieved in a certain range of part load conditions, while efficiency at full load – due to low operational time – is less important.

Due to a large number of external influences, evaluating the seasonal efficiency of installed refrigeration, heat pump or air conditioning systems can be a very complex process. For residential heat pumps for example, the evaporating conditions and the desired heating capacity depend on the ambient temperature, which in turn affects the condensation temperature. And depending on the geographical location and season, the heat pump will operate at higher, medium or lower ambient temperatures at different periods of time.

To compare system efficiency, standards have been developed that represent operating conditions for various climatic regions over a year. One example, the AHRI 210/240 [2] standard, defines a Seasonal Energy Efficiency Ratio (SEER) and a Heating Seasonal Performance Factor (HSPF) for unitary air-conditioning and heat pump equipment. Both SEER and HSPF are based on the efficiencies and capacities measured in up to five test conditions, each using a typical load profile.

Another example is SEER for cooling, and the Seasonal Coefficient of Performance (SCOP) for heating defined in prEN14825 [3] for air-conditioners, liquid chillers and heat pumps. Both calculations are based on four efficiencies in different operating conditions. These efficiencies are weighted by four factors that represent an average European climate profile and an average building load.

Literature

- [1] VDMA Einheitsblatt 24247-2: Energieeffizienz von Kälteanlagen. Teil 2: Anforderungen an das Anlagenkonzept und die Komponenten (Entwurf). Verband Deutscher Maschinen- und Anlagenbau, Frankfurt, Juli (2010)
- [2] AHRI Standard 210/240: 2008 Standard for Performance Rating of Unitary Air-Conditioning & Air Source Heat Pump Equipment. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, USA (2008)
- [3] prEN 14825:2010: Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling – Testing and rating at part load conditions (2010)

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