Isentropic-Isothermal Efficiency for Optimized Compressor Rating

Robin Langebach[1]*, Craig Bradshaw[2], Jonas Schmitt[1], Amjid Khan[2]

[1]University of Applied Sciences, The Schaufler Foundation Professorship for Compressor Technology, Institute for Refrigeration, Air-Conditioning and Environmental Technology
76133 Karlsruhe, Germany
Phone: +49 721 925 1853, Email: robin.langebach@h-ka.de.de

[2]Center for Integrated Building Systems, Oklahoma State University, Stillwater, OK

* Corresponding Author

ABSTRACT

Millions of compressors are used in refrigeration, air conditioning and heat pump systems. Their energy consumption is the most significant contributor to the efficiency of an entire HVAC system. The component specific isentropic efficiency is used for the energetic evaluation of compressors in accordance with globally standardized test procedures. The isentropic efficiency represents a relationship between the total power input and the friction-free, adiabatic power needed when compressing a refrigerant at a defined operating point. However, this basis of comparison neglects to account for a scenario where significant compressor heat transfer is achieved, so a new basis of comparison is proposed. For a Carnot cycle, the compressor outlet point is on the dew line, so that isobaric condensation can take place in the condenser. Depending on the fluid properties of the refrigerant used, a significant proportion of the ideal compression (isothermal proportion) may lie outside the liquid-vapor region. However, it would be advantageous to take this proportion into account in the energy efficiency to emphasize the refrigerant-specific influences of compression on the cycle and the system more clearly. Presented in this work, a new proposal for the isentropic-isothermal efficiency is to be introduced and compared with typical efficiencies discussed in the literature for the energetic evaluation of compressors. The main result is that the use of isentropic-isothermal efficiency can provide a better insight into the evaluation of the compressor. The user can directly see how well the compressor works with the refrigerant used at the operating point.

1. INTRODUCTION

Every year, around 486 million compressors are placed on the market worldwide [1]. The vast majority of these, around 58%, are used in refrigeration technology, around 41% in air conditioning and one percent in heat pumps. These impressive figures result in a global electrical energy consumption of approx. 20 % for the refrigeration, air conditioning and heat pump sector with an assumed 5 billion appliances on the market worldwide [2]. The vapor compression cycle is by far the most frequently used cycle. Estimates for commercial packaged products, and chiller systems assume that compressors consume a 60-70% share of electrical energy consumption in the system [3]. In refrigeration systems themselves, compressors consume around 70% [4]. These rather conservative consumption numbers for the compressor can rise to almost 100%, especially for smaller household refrigeration appliances without any other significant additional electrical consumers. It can therefore be concluded without any doubt that the compressor plays a far-reaching role in terms of energy consumption and therefore also system efficiency.

The performance evaluation of compressors is carried out according to internationally recognized standards [5, 6, 7], in particular for data sheet specifications. The compressors are tested at specific operating points under defined conditions and their performance data, in particular mass flow and electrical or mechanical power consumption, are determined. Polynomials can be used to determine the characteristics of each machine. Manufacturers usually do not explicitly state the energy efficiency of individual machines in data sheets. Often only a coefficient of performance (COP) is given for an associated, idealized comparative refrigeration cycle, but with the restriction of a specific temperature reference. Calculating the energy efficiency of the machine based on the provided performance data is possible, but it is typically only done for academic purposes. However, it is important to note that there is little or no distinction between the subsequent use of the compressor in specific applications such as heat pumps, refrigeration systems, or air conditioning systems. The paper addresses which efficiency data would be particularly advantageous for the user when evaluating compressors. The authors investigate the shortcomings of the current calculation and specification of the isentropic efficiency and propose an extension.

2. DERIVING THE OPTIMAL COMPRESSION

The Carnot cycle is scientifically accepted as the theoretical upper limit of cycle efficiency for refrigeration systems and heat pumps. It prescribes the optimal compression process, which comprises of two distinct stages: isentropic compression (1_c-2_c) and isothermal compression (2_c-3), as illustrated in **Figure 1**. The vapor compression process utilizes the fact that isothermal compression in the two-phase region can be interpreted as isobaric heat rejection and liquefaction simultaneously.

Carnot-like isentropic compression (1_c-2_c), as shown in **Figure 1**, presents a technical challenge because the suction condition (1_c) is in the vapor-liquid region. The vapor quality varies depending on the refrigerant used. However, this challenge can be overcome by shifting the isentropic compression with its starting point at or slightly above the dew line (point 1 in **Figures 1 and 2)**. The position of point 2 (refer to **Figure 2**) is affected by the course of the dew line specific to the refrigerant used.

Compression of refrigerants with an anterograde shape of the dew line, such as for R744, R717, or R718, yields discharge conditions in the superheated vapor and results in a significant increase in the discharge temperature and a deviation from the ideal Carnot cycle, recognizable by the isentropic peak, highlighted with hatches in **Figure 1**. Conversely, compression of refrigerants with retrograde course of the dew line, such as R600a or other long chain hydrocarbons, results in either a return to the wet vapor region or termination near the dew line. The compressor discharge temperature (state 2) of these refrigerants is typically close to the hot temperature, either at ambient or condensation level. An isentropic peak is not typically present or is only minimally present. It is important to note that these observations are supported by empirical data and should be considered when designing compression systems.

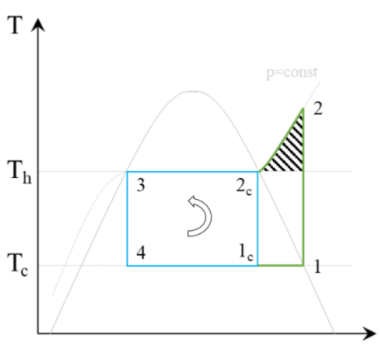

Figure 1: Carnot-cycle using wet compression (blue) and shifting of compression out of the vapor-liquid-region to superheated vapor (green).

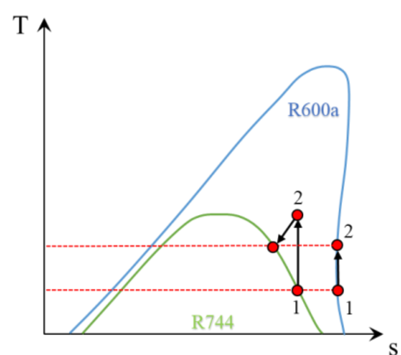

Figure 2: Compression processes for refrigerants with anterograde (green) and retrograde (blue) course of the dew line.

When assessing energy efficiency, it is important to consider both the proportion of isentropic compression and the proportion of isothermal compression. Isothermal compression is mainly handled by the condenser and is not typically the main focus of the compressor as a component. From the perspective of the Carnot process, it is crucial that the compression process in the compressor is only considered 'technically' complete when the dew line is reached. Therefore, the energy assessment of the compressor must take into account the loss of efficiency caused by isobaric discharge gas cooling.

It is important to note that the ideal vapor compression cycle deviates from the Carnot cycle due to the isenthalpic expansion. Depending on the refrigerant, the isenthalps may run into the vapor-liquid region, resulting in an increase in specific entropy. Therefore, subcooling and superheating may be advantageous for cycle efficiency in the ideal vapor compression process, which can be achieved e.g. through the use of an internal heat exchanger. However, we will not delve further into these considerations. The isenthalpic expansion deviates from the Carnot cycle and is therefore a non-ideal process. Instead, from a thermodynamic perspective, efforts should be made towards at least partially recovering work during expansion.

4. CURRENTLY USED ENERGY EFFICIENCY VALUES FOR COMPRESSOR EVALUATION

Energy efficiency is an important comparison criterion that is regularly advertised by manufacturers.
The coefficient of performance (COP) is an underlying key figure for comparing machines:

$$COP = \frac{\dot{Q}_o}{P_{tot}} \tag{1}$$

The COP represents the ratio of the cooling capacity $\dot{Q}_o$ to the mechanical power input (at the shaft) or electrical power (at the terminals) P_{tot} under specific circuit conditions. These conditions are standardized, e.g. according to DIN EN 12900 [5] and AHRI STD 540 [6] or ASHRAE STD 23 [7]. The COP allows the relative comparison of two compressors for the same refrigerant under the same test conditions. It also gives the user an indication of the expected coefficient of performance in the application with a similar cycle design. However, the COP does not indicate the extent to which the compressor is optimized for use with precisely this refrigerant at the operating point. Similarly, the COP is not a possible comparative figure for heat pump processes.

Another underlying key figure for comparing machines is the isentropic efficiency η_{is}:

$$\eta_{is} = \frac{P_{is}}{P_{tot}} = \eta_c \cdot \eta_{mec} \cdot (\eta_{el}) \tag{2}$$

This is the ratio of the power for an isentropic compression process to the actual measured power applied to the shaft or motor terminals. The isentropic efficiency η_{is} can be expressed as a chain of the actual compression efficiency η_c, the mechanical efficiency η_{mec} and the electrical efficiency of the motor η_{el}, if present. The isentropic efficiency suggests a value between 0...1 or 0...100%. This is the case for most compressors in the field. However, for machines with sufficient cooling of the compression process, the isentropic efficiency can initially take on implausible values above 100%. However, a closer look reveals a significant disadvantage of isentropic efficiency, which we have already discussed in section 2. Depending on the refrigerant under consideration, the underlying assumption of isentropic compression is not the energetic optimum with respect to the Carnot process. It is mandatory to consider the isentropic and, if necessary, the isothermal part of the compression.

It should be noted that cooling the compression is an increasingly used method to improve the energy efficiency of machines. Effective approaches are active cooling of the compressor with another medium, e.g. water [8]. Similar studies were conducted for effective cooling of the compression process in order to achieve near isothermal compression [9],[10]. Vapor injection of refrigerant (e.g. in scroll compressors) is another technique used for effective improvement in compression process to reduce the discharge temperature [11]. Similar results were found, when experimental investigation with vapor injection was carried out [12], [13], [14]. Lastly, another effective cooling of compression process is performed with oil flooding into the compression process [15].

5. EXTENSION TO ISENTROPIC-ISOTHERMAL EFFICIENCY

The isentropic efficiency is now to be modified. **Figure 3** shows two different cases to be distinguished.

- Case 1: The left-hand side shows a section of the dew line for an anterograde refrigerant - e.g. for R744. Here, isentropic compression would result in an isentropic peak between the low- and high-pressure levels. In the condenser, isobaric recooling of the refrigerant to the dew line would be necessary.
- Case 2: The right-hand side shows a section of the dew line for a retrograde refrigerant - e.g. for R600a. Here, isentropic compression between the low- and high-pressure levels results in an end point in the vapor-liquid region. To return to the dew line, the refrigerant would have to be vaporized again by adding heat.

For case 1, an extension of the isentropic efficiency is useful to include the minimum work of compression for the isobaric gas cooling part. Starting in suction state 1, isentropic compression should now only take place up to an intermediate pressure that belongs to the condensation temperature T_h. This intermediate state 2_{it} now serves as the starting point for isothermal compression. This takes place up to the point $2_{is,it}$ on the dew line. The specific heat q_{rej} released corresponds to the minimum specific compression work for that step.

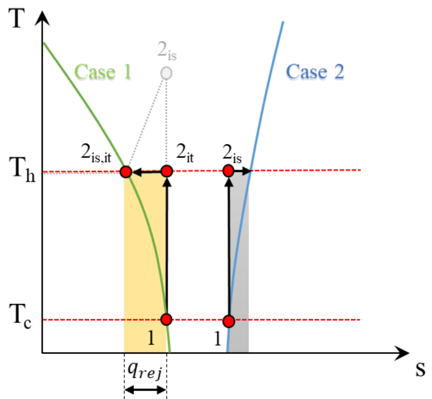

Figure 3: Compression process for anterograde course of the dew line (green, case 1) dividing compression into isentropic and isothermal proportion, and retrograde course (blue, case 2) of the dew line with compression end point within the vapor liquid region.

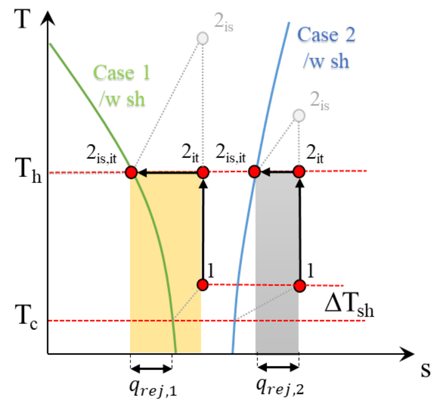

Figure 4: Compression process for anterograde course of the dew line (green) and retrograde course (blue) of the dew line including a superheating.

For case 2, an extension of the isentropic efficiency is not necessary. The isentropic compression leads to the high pressure corresponding to the desired condensation temperature T_h. The total pressure ratio is already fully achieved. The specific heat to be added at a later time would be positive and does not need to be considered further. This results in the following calculation definition with case differentiation for the extended isentropic-isothermal efficiency $\eta_{is,it}$:

$$\eta_{is,it} = \begin{cases} \dfrac{\dot{m} \cdot \left(w_{is,it} + |q_{rej}|\right)}{P_{tot}}, & q_{rej} < 0 \\ \eta_{is} = \dfrac{P_{is}}{P_{tot}} = \dfrac{\dot{m} \cdot w_{is}}{P_{tot}}, & q_{rej} \geq 0 \end{cases} \tag{3}$$

The definition is subject to the convention that specific energies supplied are regarded as positive. It should further be noted that $w_{is,it}$ and w_{is} are not identical. $w_{is,it}$ only corresponds to the proportion of the specific isentropic compression work up to an intermediate pressure at T_h.

In addition, the method for determining the isentropic-isothermal efficiency can also be applied to standardized points from standards with a defined superheat. **Figure 4** shows the two cases again with a defined superheat at state point 1. There is also a proportion of isothermal compression for case 2 due to the additional superheat. This means that the isentropic-isothermal efficiency can also be used directly when analyzing compressors according to standard measurements - even for refrigerants that have a retrograde course of the dew line. For R744 and other refrigerants where the test conditions require supercritical operation, the point at the gas cooler outlet can be defined.

6. EXAMPLE CALCULATION AND COMPARISON

This section will utilize the new efficiency definition and apply it to two theoretical compressor applications with a variety of refrigerants, both retrograde and anterograde thermodynamic behavior. Two operating points from the AHRI 540 standard are used for calculations and given below:

Test Point 1 – Freezer:

- Cold temperature / evaporation temperature $T_c = -25°F = -31.6°C$
- hot Temperature / condensation temperature $T_h = 105°F = 40.5°C$
- Superheating $\Delta T_{sh} = 22°R = 11.1K$

Test Point 2 – Heat Pump:

- Cold temperature / evaporation temperature $T_c = 45°F = 7.2°C$
- Hot temperature / condensation temperature $T_h = 130°F = 54.4°C$
- Superheating $\Delta T_{sh} = 22°R = 11.1K$

The calculations are to be carried out with five different refrigerants. These are ammonia (R717), the refrigerant mixture R410A, isobutane (R600a), carbon dioxide (R744) and propane (R290). The selection includes 4 natural refrigerants as candidates that are currently used in a wide variety of applications and are future-proof. In addition, R410A was integrated as a typical representative of refrigerants for heat pumps.

The authors are aware that some of the selected operating points cannot be achieved using single-stage compression. However, as the test points are specified in the standards, no changes should be made to the typical operating limits of compressors available on the market.

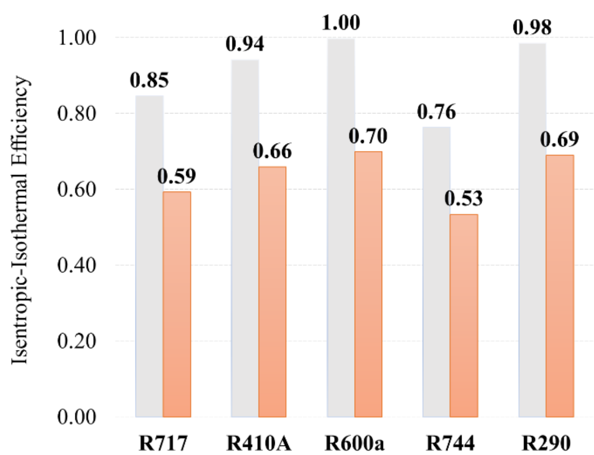

Figure 5: Isentropic-isothermal efficiencies for test point 1 for $\eta_{is} = 1$ (grey bars) and $\eta_{is} = 0.7$ (orange bars).

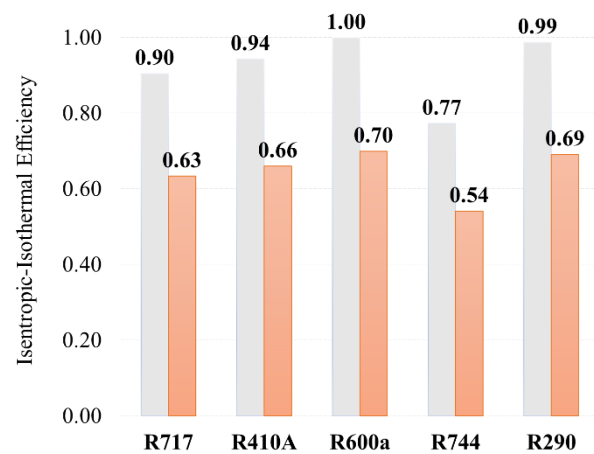

Figure 6: Isentropic-isothermal efficiencies for test point 2 for $\eta_{is} = 1$ (grey bars) and $\eta_{is} = 0.7$ (orange bars).

In a first calculation test point 1 shall be analyzed. It is assumed that all compressors for each refrigerant operate isentropically with $\eta_{is} = 1$. The results of the isentropic-isothermal efficiency are shown in **Figure 5** as gray bars. The gray bars show that in particular R600a and also R290 have an isentropic-isothermal efficiency of almost 1. This means that both compression processes are thermodynamically very good, almost optimal, in the sense of an associated Carnot cycle. In contrast, R717 and R744 (here as a supercritical cycle) in particular, with isentropic-isothermal efficiencies of around 85% and 76% respectively, still have potential for optimization. With regard to compressor development, this means that with R600a and also R290, attention must be paid exclusively to compression, mechanics and electric motor that are as frictionless and loss-free as possible. For R717 and R744, on the other hand, additional cooling measures should be taken during compression.

This becomes even clearer if an isentropic efficiency of $\eta_{is} = 0.7$ is assumed for all compressors. This value can certainly deviate for real machines depending on the manufacturer, but should be chosen constant here for the sake of clarity. The results of the isentropic-isothermal efficiency are shown in **Figure 6** as orange bars. The general trend can be confirmed here in the same way as for isentropic compression. For R600a and also R290, the isentropic efficiency is almost the same. For the other refrigerants, the imperfection with regard to the cooling of the compression process is now also taken into account in the efficiency.

In a second calculation test point 2 shall be analyzed. It is again assumed that all compressors for each refrigerant operate isentropically with $\eta_{is} = 1$. The results are this time shown in **Figure 6** as gray bars. It can be seen that for this test point 2, R600a and R290 can also be mapped very well using pure isentropic compression. Compared to test point 1, it can be seen that the need for heat release during compression is slightly lower for R717. For the compressor design, this would mean that additional cooling would be more effective for test point 1 than for test point 2. It is, of course, beneficial in both cases. This general trend can also be confirmed by again assuming an isentropic efficiency of $\eta_{is} = 0.7$ for all compressors. The results are this time shown in **Figure 6** as orange bars.

7. CONCLUSIONS

The compressor is the component with the highest electrical energy consumption in refrigeration and heat pump cycles. It is therefore necessary to design compressors with the lowest possible energy consumption and high energy efficiency. The energy efficiency of refrigerant compressors is typically specified on the basis of isentropic efficiency. This is the ratio of the power for an isentropic compression process to the actual measured power applied to the shaft or motor terminals. With respect to the Carnot process, it is crucial that the compression process in the compressor is only considered 'technically' complete when the dew line is reached. Therefore, the energy assessment of the compressor must consider the compression itself, but also the loss of efficiency caused by isobaric discharge gas cooling.

However, isentropic efficiency has decisive disadvantages. It is only suitable to a limited extent for externally cooled compression, which has already been technically implemented. Furthermore, it cannot be used to determine the extent to which a compressor is adapted to the thermodynamic properties of the refrigerant. As a consequence of the current definition of isentropic efficiency, two identical compressors, e.g. for R600a and R717, would have exactly the same efficiency.

To address these limitations, the use of an extended, isentropic-isothermal efficiency is proposed. It divides the compression into an isentropic and an isothermal proportion, the sizes of which can vary depending on the suction condition, operating point and refrigerant. In particular, it directly penalizes unused potential in the cooling of the compression process and reveals this in the efficiency.

For example, the refrigerant-specific consideration for the operating point ($T_c = -31.6°C$; $T_h = 40.5°C$) in the case of R717 leads to a reduction from 100% to 85% of the purely isentropic to isentropic-isothermal efficiency. In contrast, for R600a this consideration leads to no reduction. The isentropic-isothermal efficiency remains at 100%, identically to the isentropic efficiency. This makes it easier for the user to see how well a compressor is adapted to the operating point and the refrigerant used.

The isentropic-isothermal efficiency can be defined for common standard measuring points from standards, as well as for transcritical compression processes.

NOMENCLATURE

COP	Coefficient of performance	(…)
$\dot{m}$	Mass flow	(kg/s)
P_{tot}	Total power input	(W)
$\dot{Q}_o$	Cooling power	(W)
q	specific heat	(J/kg)
s	Specific entropy	(J/kg-K)
T	Temperature	(K, °C, °F)
ΔT	Temperature difference, for superheating	(K)
w	Specific work	(J/kg)
η	Efficiency	(…)

Subscript

c	cold temperature level
h	hot temperature level
sh	superheating
rej	rejected/released
is	isentropic
is,it	isentropic-isothermal
c	for compression
mec	mechanical
el	electrical

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