Heat pump COP from definition to sales argument

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Introduction

The oil crisis of the early 1970's initiated energy cost-related interest in heat pumps. The low energy prices and the abolition of subsidies in the 1980's saw the heat pump sink into oblivion. Large amounts of fossil fuels are burned to generate electricity, heat and power. The CO_2 that this releases has an enormous influence on the greenhouse effect and ultimately on our climate. The latter fact prompted the EU to sharpen its policy and demand that the contribution of sustainable energy to energy consumption be doubled by 2010. The significant part heat pumps play in this policy has caused a resurrection; we could easily call it hype.

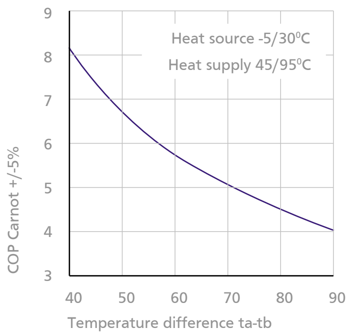
One negative side effect of hype is that the market is swamped with insufficient information for the purpose of higher sales figures. The efficiency of a heat pump, wrapped up in the abbreviation COP (coefficient of performance), changes from definition to sales pitch. There is nothing wrong with that in principle, but there is something wrong if it is used whether it is relevant or not, without any relation to the usage objective, the application and means of business operation. The aim of this article is to separate the proverbial apples and oranges in COP communication.

Just as water does not flow uphill, heat will also not flow automatically from a lower temperature Tb(K) (b for heat source) to a higher Ta (K) (a for heat emission). You need energy for that. The COP that is theoretically achievable, called the Carnot efficiency, related to the source and emission is thus

$$COPcarnot_{source+emission} = \frac{Ta}{(Ta-Tb}$$

Graph 1 shows the theoretical COP for most applications.
The Carnot efficiency decreases just as fast as the temperature difference to be conquered increases!

Figure 1



The actual COP is quite a bit lower. The relationship between the actual and theoretical COP is called the total efficiency.

$$COP_{actual} = \eta_{total} \times COPcarnot_{source+emission}$$

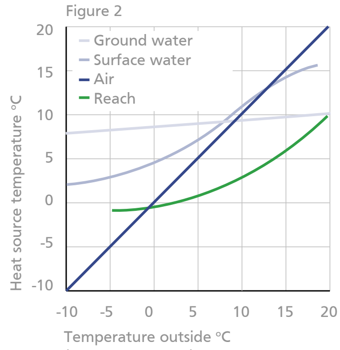
As an indication we used the following maximum values for the current heat pumps available on the market, including peripheral equipment with variable temperature of heat source and heat emission over the entire year.

Heat source	ηtotal
Ambient air	0.40
Terrestrial heat (probes)	0.45
Ground, river and lake water	0.50
**Ausbildungsmodul Geothermei Waermepumpen Dr.Martin Zogg*	

The following matters are taken into account in this total efficiency:

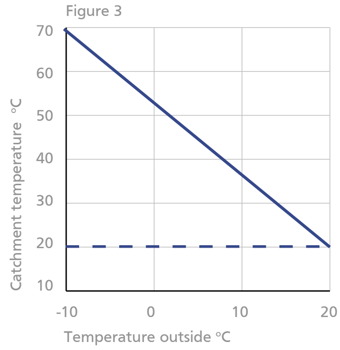
1. That the temperature of the heat source is not usually constant, determined by the influence of the seasons but also by heat being withdrawn or supplied. See graph 2.
2. That the heat emission temperature is not usually constant. See graph 3.
3. That peripheral equipment such as pumps and other auxiliary equipment is needed to transport and distribute the heat.
4. That the heat has to be transferred via condensers. For the highest possible COP the temperature difference has to be as small as possible between heat source-return temperature and evaporation temperature or heat emission temperature and condensation temperature.
   a. A temperature difference of <3K is feasible commercially and technically in the condenser.
   b. With the evaporator, it depends on the method of evaporation. If a bath or spray evaporator (FX) is used, the lower limit is about 1K. With full load, the design usually aims at a temperature difference of <3K. For an evaporator with refrigerant quantity control based on the exit superheating (DX), the

Figure 2



(Source; VDI-2067)

Figure 3



temperature difference will have to at least correspond with the minimum stable superheating. See also graph 7. Generally about 7 to 8K, which is why DX always has a worse COP (4 to 9%) than FX.

c. The chosen refrigerant influences the heat transfer enormously. Ammonia (R717) transfers heat much better in condensing and evaporating than the synthetic refrigerants. That can easily make a difference of 1 degree with an equal heat transfer surface.

The Carnot efficiency for the heat pump with the heat transfer temperature differences calculated as well is:

$$COPcarnot_{heat\ pump} = \frac{Tc}{(Tc-To)}$$

Tc and To are the condensation and evaporation temperatures respectively.

5. The efficiency of the heat pump itself naturally has the greatest influence. That is reason enough to devote the rest of this article to it.

There are various technical models of heat pumps. The most advanced technical and commercial designs are the compression and absorption heat pumps. The compression heat pumps have a COP many times higher than that of the absorption versions. The presence of sufficient waste heat of high level definitely makes the use of absorption pumps attractive. This article is about COP as a sales argument, so we will examine the compression heat pump as highest achievable in more detail.

If we look at the theoretical process, see graph 4, we see that:

- from 1 to 2 the isentropic compression P,
- from 2 to 3 the isobar and partially isothermal heat emission in the condenser Qc,
- from 3 to 4 the isenthalpic throttling,
- From 4 to 1 the isobar and partially isothermal heat absorption Qo, takes place.

The ratio of the theoretical condenser heat divided by the theoretical work of compression is the theoretical COP of the heat pump:

Figure 4

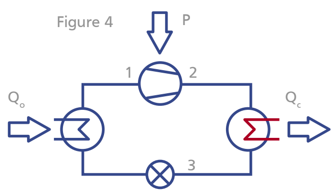
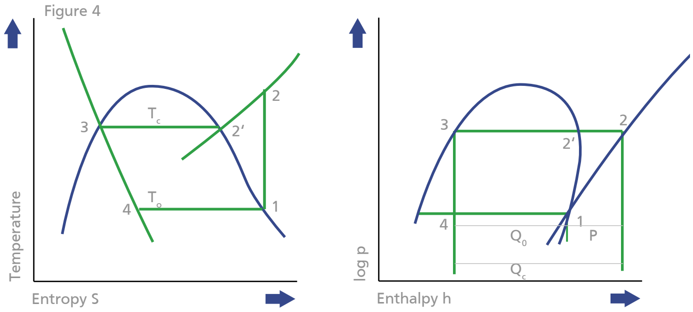


Figure 4



$$COPth_{heat\ pump} = \frac{Qc}{P}$$

Expressed in enthalpy:

$$COPth_{heat\ pump} = \frac{(h2-h3)}{(h2-h1)}$$

For the condition to=0°C superheating=15K and tc=55°C the efficiencies for a number of known industrial refrigerants turn out like this:

*	COPth	kJ/m³	NKP	t25	t50	tcrit	ODP	GWP
R507	3.95	3553	-47.3	54.1	68	72.8	0	3900
R404A	4.01	3506	-46.7	55.1	69.4	72	0	3800
R410A	4.17	5150	-51.7	42.9	67.4	69.6	0	2000
R407C	4.45	3410	-43.9	57.1	82.9	86	0	1700
R134A	4.6	2265	-26.2	79.3	98.6	101	0	1300
R22	4.62	3677	-40.9	63.1	93.4	96	0.05	1700
R717	4.82	4241	-33.5	59.6	89.7	132.3	0	0
R600	4.89	898	-0.5	129.1	148.7	152	0	20

* *Calculated with ASEREP32*

kJ/m³ *condenser capacity per m³ displaced swept volume*

NKP *boiling point at atmospheric pressure*

t25, t50 *condensation temperature at 25 and 50 bar operating pressure respectively*

tcrit *critical temperature of the refrigerant.*

The distance from the working point to the critical temperature influences the COP significantly, see graph 5, as well as the volumetric condenser capacity, see graph 6.

Figure 5

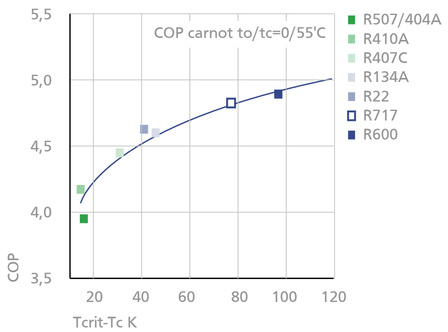


Figure 6

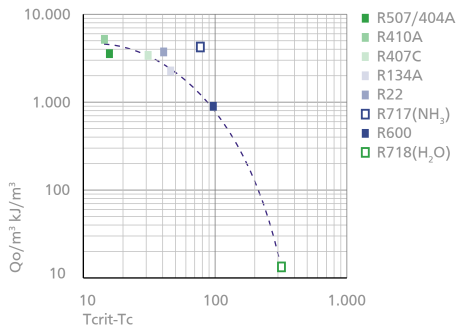
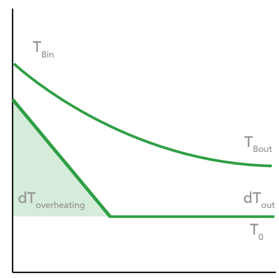


Figure 7



The higher the critical temperature, the closer $COPth_{heat\ pump}$ comes to $COPcarnot_{heat\ pump}$ but the less condenser capacity is supplied at equal swept volume. Ammonia differs from this in a very positive way!

What means do we now have to crank up the efficiency further? Because the condenser heat is the numerator and the work of compression is the denominator in the Carnot efficiency formula, the answer to this question is simple:

- increase the condenser heat and/or
- decrease the work of compression

Reducing the work of compression will decrease the condenser heat, but the COP will still increase. Check it out, on the basis of a COP of 4 for instance: a 25% improvement in the work of compression will increase the COP to (4-0.25)/(1-0.25)=5!

1. Increasing the condenser heat is possible by:

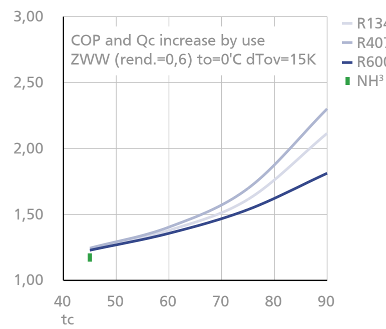
a. Having the heat carrier supercool the condensation on the emission side. The optimum is therefore achieved with an infinitely large supercooler and low return temperature of the medium to be heated. That means that a large temperature difference of the medium to be heated is advantageous.

Using other possibilities for supercooling the condensation is an often-made error, prompted by thinking in terms of refrigeration. Consider it: the compressor keeps moving the same amount of gas; after all, the pressure ratio and suction condition do not change (h1 stays constant). The enthalpy of the discharge gas does not increase either (h2 stays constant). So nothing changes.

b. increasing the discharge gas temperature by:

- letting the useful superheating in the evaporator increase. This is a contradiction, as the higher the superheating at constant supply temperature of the heat source, the lower the evaporation temperature has to be. The latter will also entail a decrease in the COP; see graph 7.
- using a condenser that supercools the condensation by heating the suction gas. This reduces the density of the suction gas, and through that the quantity of circulating refrigerant and therefore the capacity. If we take a suction gas condenser (ZWW) with an efficiency of 0.6, the COP and therefore the condensation heat increase considerably with increasing condensation temperature. See graph 8. The result is also positive for ammonia (+17% at tc=45C) but an excessively high discharge gas temperature obstructs its practical use.
- Increasing the condenser mass output by using a two-stage compression with refrigerant injection to the interstage pressure results in the COP increasing marginally for ammonia and decreasing for the synthetic refrigerants. If we use the interstage injection to supercool the condensation, the result is negative for all refrigerants (R717 <10%, R134A and R407C >>10%).

Figure 8



2. Reducing the work of compression by:

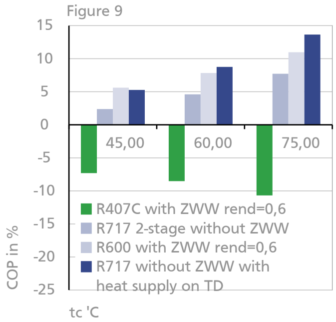
a. multi-stage compression does not achieve anything with systems with interstage injection.

b. Heat discharge during the compression is not an option for piston compressors; there is simply not enough time and surface to discharge enough heat. With screws, the oil used for the seals is used as refrigerant.

c. If the return temperature of the water to be heated is sufficiently low, there is an option of drawing heat from the discharge gas for ammonia on the interstage. This extra heat thus increases the COP.

If we plot out all these effects in a graph and compare it with R134A, then $T_0 = 0^oC$ dT_{ov}=15K. See graph 9.

Figure 9



Conclusion

Refrigerants with a high critical temperature give the best COP; the natural refrigerants are superior in this respect. To make water of 70' C, R134A needs almost twice as much swept volume as ammonia (HP and LP) and about 1.5 times as much as R407C.

With ammonia, the high pressure cylinders will have to be able to resist a higher pressure than the usual 25 bar(e). For R407C the same applies for the entire discharge part. The Grasso 5-HP can resist a discharge pressure of 50 bar(e). The highly flammable butane (R600) needs about 2.5 times more swept volume than R134A, so 5 times that of ammonia and is therefore not interesting commercially.

What is the current situation with efficiency achievable in practice?

As we are looking for the highest achievable COP, we will ignore the screw compressors that do not match up to the piston compressors energy-wise. The energy savings to be gained are apparently so high that a colleague manufacturer devoted an entire issue of their HVAC&R ENGINEERING update to it. For the sake of completeness we will quote the conclusion of that report literally: "For air-conditioning or heat-pump duty, the energy-cost savings of reciprocating chillers can exceed the maintenance-cost savings of screw chillers by 100 to 300%!"

The density of the gas influences the compressor efficiency greatly. A heavy gas has a positive effect on the compressor cooling and results in higher isentropic efficiency. The heavier the gas, the higher the pressure losses, which keeps the swept volume down. The extent of this influence is measured daily in the compressor laboratories of the various compressor manufacturers. We will field-test this with a Grasso 10 compressor, tested with the refrigerants R717, R22, R134A, R507, R404A and R407C. For our starting point we will take:

- a suction saturation temperature of 0°C,
- effective superheating of synthetic refrigerants of 15K (ammonia=0K)
- single-stage compression for the synthetic refrigerants
- for ammonia, 2-stage compression if the maximum discharge gas temperature is exceeded.
- with the synthetic refrigerants, a ZWW with an efficiency of 0.6% (efficiency of 1 means that the suction gas comes out at the same temperature as the incoming condensation).
- with ammonia, also heat emission on the interstage pressure
- a temperature difference between the evaporation and source return temperatures of 3K,
- a temperature difference in heat source medium over the evaporator of 5K, so the annual average source temperature becomes 8°C
- 3 different condensation temperatures 45, 60 and 75°C
- a temperature difference between the condensation temperature and annual average emission temperature of 3K, the annual average emission temperatures become 42, 57 and 72°C
- temperature difference of heat emission medium 20K
- use of low heat emission temperature for supercooling. Temperature difference of exiting condensation and return temperature is 3K, that makes the extra supercooling 20K

The maximum heat pump full load COP's achievable then become:

tc/KM	Carnot*	R717	R134A	R407C
45	9.27	5.52	4.66	5.02
60	6.74	4.40	3.68	3.88
75	5.39	3.68	3.07	3.11

* at source emission

If the cooling cannot be used effectively, the cooling capacity is added to the condenser capacity in the numerator, and we get the figures below.

tc/KM	R717	R134A	R407C
45	10.04	8.32	9.04
60	7.8	6.36	6.76
75	6.36	5.14	5.22

These values even exceed the Carnot efficiency of the source emission. **So it is all a question of definition!!!**

If you would like to react to this article or one of the previous articles, or have questions, you can contact: Grasso Products b.v. jphabraken@grasso.nl.