CENTRIFUGAL COMPRESSION TURBO HEAT PUMP MADE BY ECOP

Bernhard Adler, CTO, ECOP Technologies, Vienna, Austria
Sebastian Riepl, CEO, ECOP Technologies, Vienna, Austria
Karl Ponweiser, Professor, Institute for Energy Systems and Thermodynamics, Vienna University of Technology, Vienna, Austria

Abstract: When realizing a heat pump system, the key to get high performance is to minimize exergy losses. In this paper the characteristics of today's state of the art vapor-compression heat pump will be shown by analyzing two different examples. The first one is a typical vapor-compression heat pump as it is quite common on the market. The other one a vapor/single phase compression heat pump, realized by compressor and a turbine. Furthermore another way of realizing a Joule compression cycle in a vapor zone will be introduced, combining the advantages of both characteristics, which results in a system promising higher COPs, more flexible usage, new temperature areas and working fluids without increased global warming potential.

Key Words: heat pump, centrifugal compression, efficiency

1 INTRODUCTION

In today's world heat pump systems are well established. A broad variety of different technologies make it possible to realize such a machine. The most common one is the compression heat pump using a thermodynamic cycle in the two phase region. There are different refrigerants to realize this. A common one in Europe is for example R134a. Basically those refrigerants work as already mentioned, at least partly, in the two phase region in order to be able to use the stored latent heat. Machines with cycles that work in gaseous region only are known as well but so far solely used in special applications.
The problem about such single phase machines is that the realizable Coefficient of Performance (COP) is rather low compared with a vapor-compression version.
When realizing a heat pump system, the key to get high performance is to minimize exergy losses. In this paper the characteristics of today's state of the art compression heat pumps will be shown by analyzing two different examples. Additionally an innovative way of realizing a heat pump cycle in a vapor zone will be introduced.

2 ANALYSIS OF CONVENTIONAL HEAT PUMP SYSTEMS

Exemplary specific cases shall be calculated to have a better understanding of different compression cycles. The temperature of the heat source is assumed at 273.15 K, while the temperature of the heat sink is 308.15 K. In this assumed case the vapor-compression heat pump has a compression temperature of 312.22K. This means that the temperature spread of the thermodynamic middle temperatures is a bit higher. In the following Figures calculated COPs are compared to the maximum reachable COP to show the qualitative image.
In a first step all processes will be calculated as ideal. Afterwards the calculations are corrected by a typical assumed efficiency of compression between 100% and 80% to show the effects on the different cycles. Exergy losses due to heat transfer (temperature spread in

Heat Exchangers) as well as fluid flow friction losses shall be ignored in this simple calculation.

2.1 Analysis of vapor-compression heat pump

The exemplary chosen refrigerant for this analysis shall be R134a, since it is a very common one in Europe.

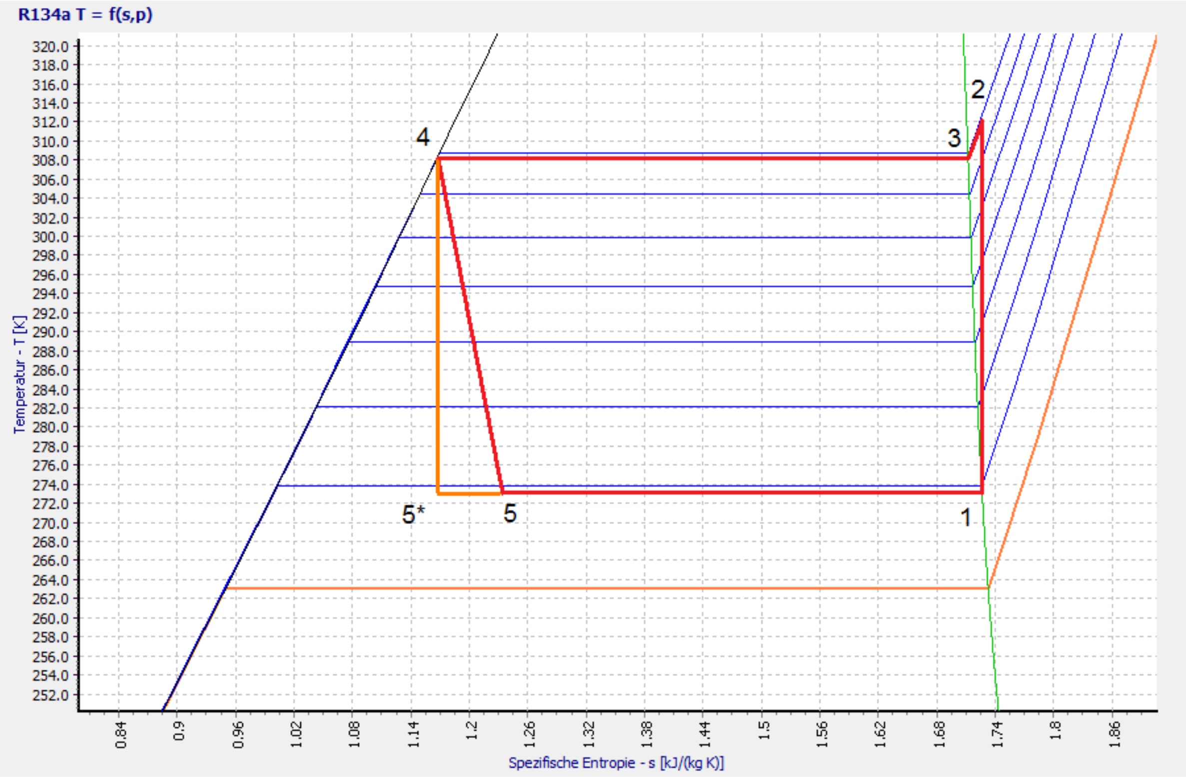

Figure 1: R134a heat pump cycle (Source ThermoFluid Springer V1.0)

Table 1: Conditions for the different points in Figure 1

R134a	Temperature [K]	Pressure [kPa]	Entropy [kJ/(kg K)]	Enthalpy [kJ/(kg)]
1	273.15	292.80	1.7271	398.60
2	312.22	886.98	1.7271	421.62
3	308.15	886.98	1.7128	417.19
4	308.15	886.98	1.1670	249.01
5	273.15	292.80	1.1794	249.01
5*	273.15	292.80	1.1670	245.61

The process basically consists of the steps shown above, consisting of isentropic compression (1-2), isobaric heat release (2-3), isotherm (and isobaric) heat release (3-4), isenthalpic expansion (4-5) and finally isothermal heat absorption to close the cycle. The COP would be ideal if the expansion after point 4 would be isentropic and go on to 5*.

The isenthalpic expansion in a choke (process 1-2-3-4-5) causes about 15% efficiency loss compared with the ideal calculated process (1-2-3-4-5*) using the isentropic expansion (and assuming that somehow the energy during expansion will be recovered).

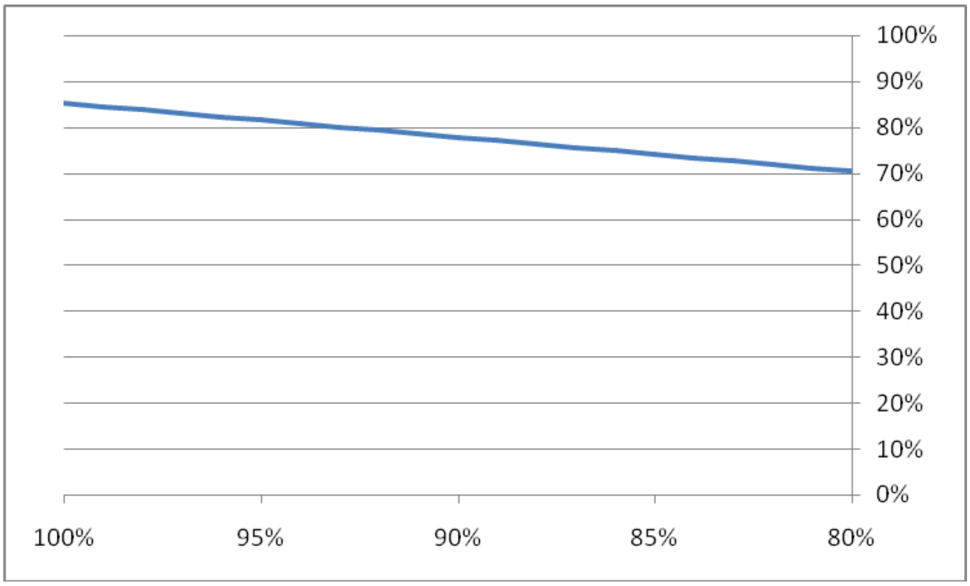

Figure 2: Influence of compression efficiency (x-Axis) on COP for mentioned example

The figure shows the influence of the compression efficiency for the relation reached COP to ideal calculated COP. As shown, the influence of the compression efficiency for a two phase region heat pump is rather small.

2.2 Analysis of single phase region heat pump – state of the art

The exemplary chosen working fluid for this analysis is nitrogen.

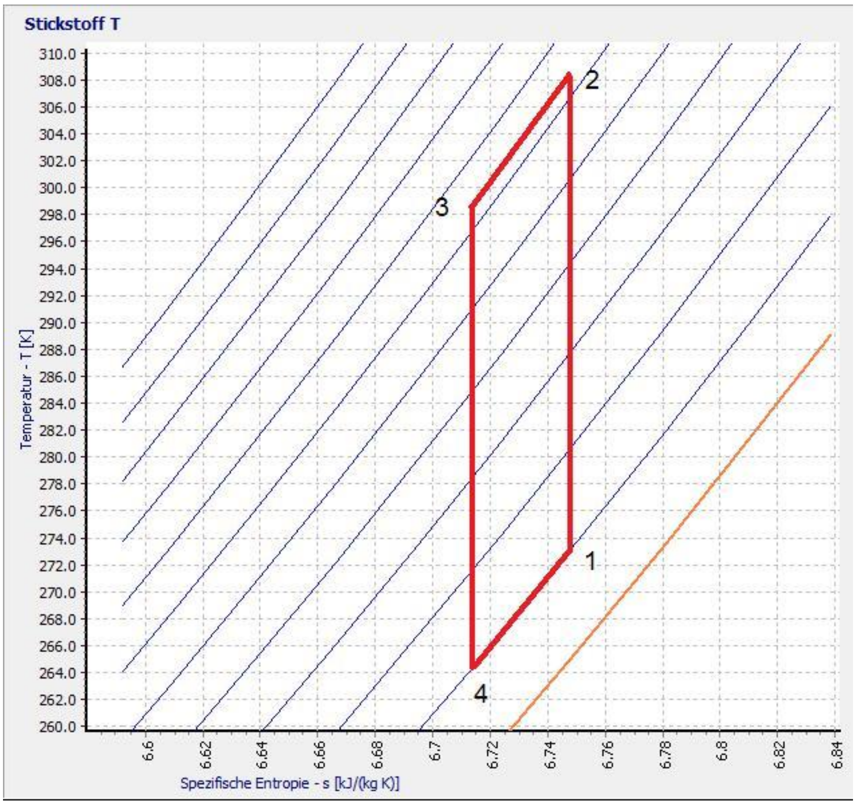

Figure 3: N_2 heat pump cycle, T-s (Source ThermoFluid Springer V1.0)

Table 2: conditions for the different points in Figure 3

N_2	Temperature [K]	Pressure [kPa]	Entropy [kJ/(kg K)]	Enthalpy [kJ/(kg)]
1	273.15	100.0	6.7480	283.24
2	308.15	152.5	6.7480	319.57
3	298.15	152.5	6.7136	309.15
4	264.27	100.0	6.7136	273.99

The process consists of isentropic compression (1-2), isobaric heat release (2-3), isentropic expansion (3-4) and isobaric heat absorption to close the cycle.
The difference to the vapor-compression heat pump is obvious. In a vapor zone an isotherm heat transfer is not possible. For this reason the thermodynamic boundaries for the compression temperature were fixed as shown. One advantage for a single phase heat pump cycle is that the pressure energy could high efficiently be recovered (e.g. in a turbine).

2.2.1 Ideal calculation of cycle for an example of a heat pump

$$h_{in} = h_2 - h_1 = 36.33 \text{ kJ/kg} \tag{1}$$

$$q_{out} = h_3 - h_2 = -10.42 \text{ kJ/kg} \tag{2}$$

$$h_{out} = h_4 - h_3 = -35.16 \text{ kJ/kg} \tag{3}$$

Equations (1) to (3) show exactly why efficiency of compression (and expansion) is of such big importance in a single phase region. The work needed for compression is relatively high compared to the transferred heat. Of course a large amount of this work can be recovered in the expansion section but state of the art compression devices will significantly lower the reachable efficiency of the whole process.

2.2.2 Influence by compression efficiency

The assumed efficiency of compression will raise the needed energy for the sketched example. In the best case the heat can be recovered without losses, raising the releasable heat. The recoverable work will be basically the same. The influence on the COP of the process is huge. With the assumed example the reachable COP drops to about 20% by just taking the compression efficiency of 80% into account.

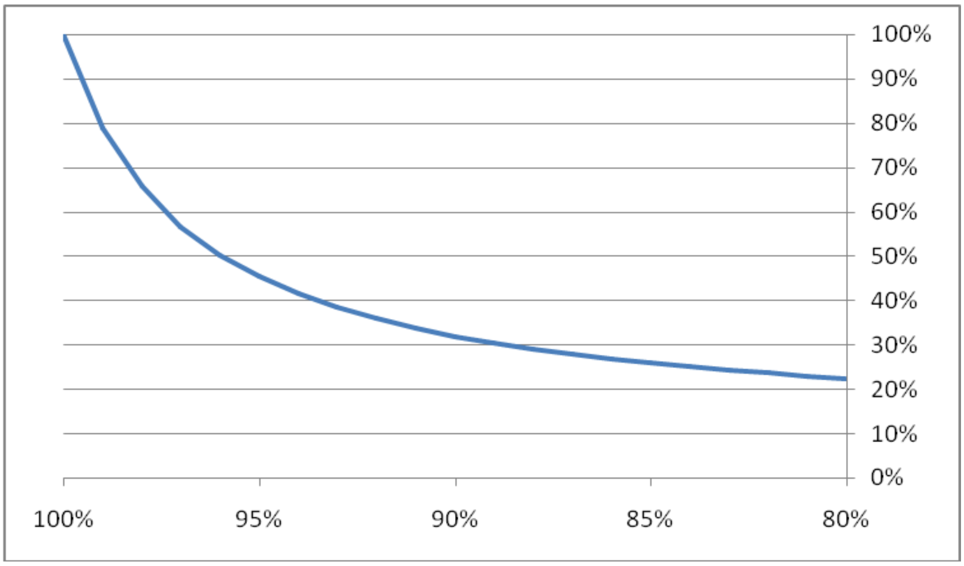

Figure 4: Influence of compression efficiency on reachable COP for mentioned example

2.2.3 Core Issue Efficiency Factor

Since the efficiency factor for a vapor operated heat pump is critical, a closer look on a compression device is needed. For the example of a radial compressor the pressure increase basically contains two components. The first is the increase because of centrifugal acceleration. So far mainly friction losses influence the efficiency of the compression, working with relatively high efficiency. The second component is the pressure increase caused by the conversion of kinetic energy into pressure. When the fluid leaves the rotating wheel with high velocity, it is pressed against the casing and the conversion takes place. The efficiency of this part is relatively low and the main influence factor for the overall efficiency.

3 CENTRIFUGAL COMPRESSION TURBO HEAT PUMP

To make a single phase vapor heat pump efficient, it would be necessary to improve the efficiency factor of the compression device. The most important step is to avoid the relatively inefficient conversion from kinetic energy into pressure.

3.1 Compression by acceleration field

The basic idea is simple and it can be observed every day. The earth possesses an acceleration field causing aerostatic or hydrostatic pressure increase. At the surface of the sea the pressure is around 100 kPa. For every meter below the surface the pressure is increasing. In 100 meter depth the pressure is around ten times as high as on the surface.

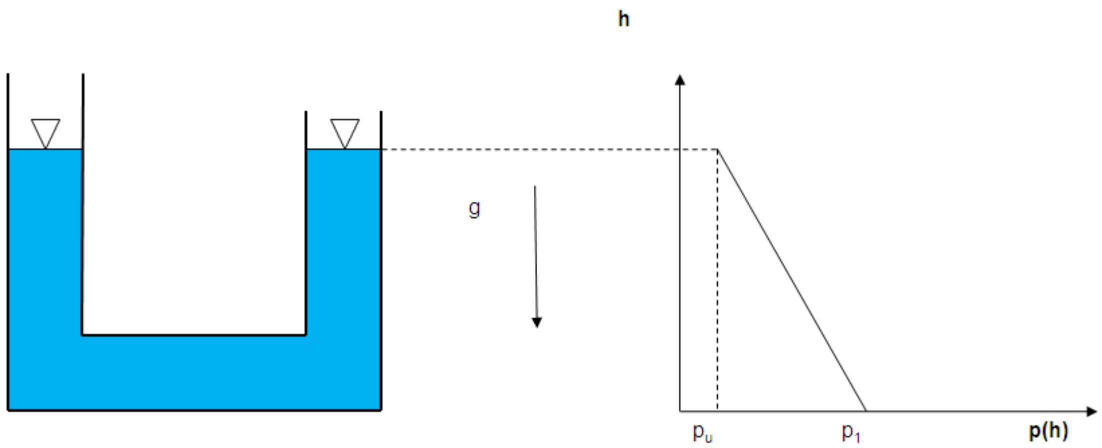

Figure 5: Pressure increase caused by gravitation

Of course this pressure increase also works if the gravitation (g) field is substituted by a centrifugal acceleration field, g*(r) caused by rotation. The centrifugal acceleration, which is depending on the radius and the angular velocity, causes a parabolic, instead of a linear, pressure increase.

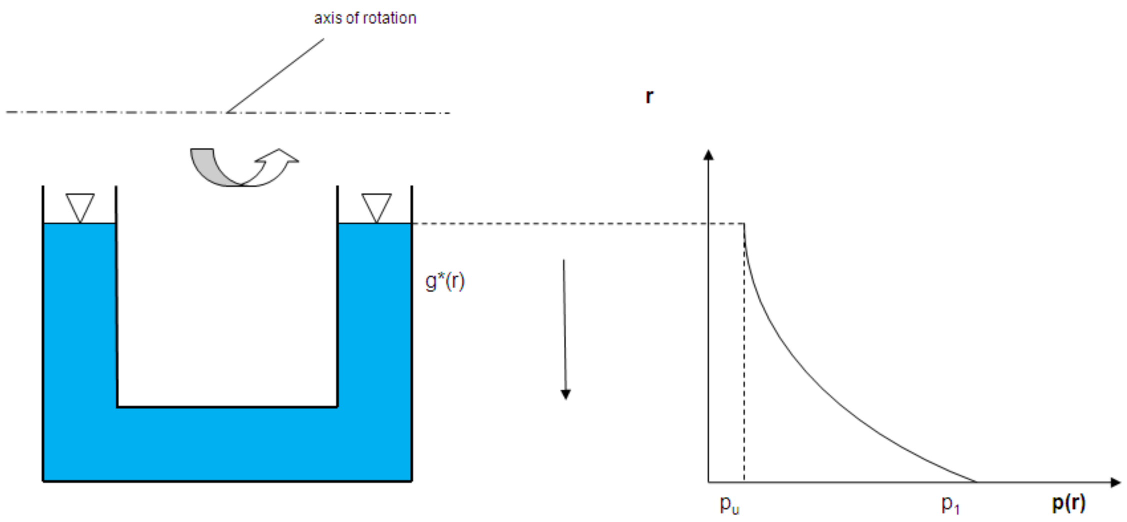

Figure 6: Pressure increase caused by rotation

3.2 Heat pump Cycle

When the U Pipe from Figure 6 is closed, a thermodynamic cycle can be realized in the following way.

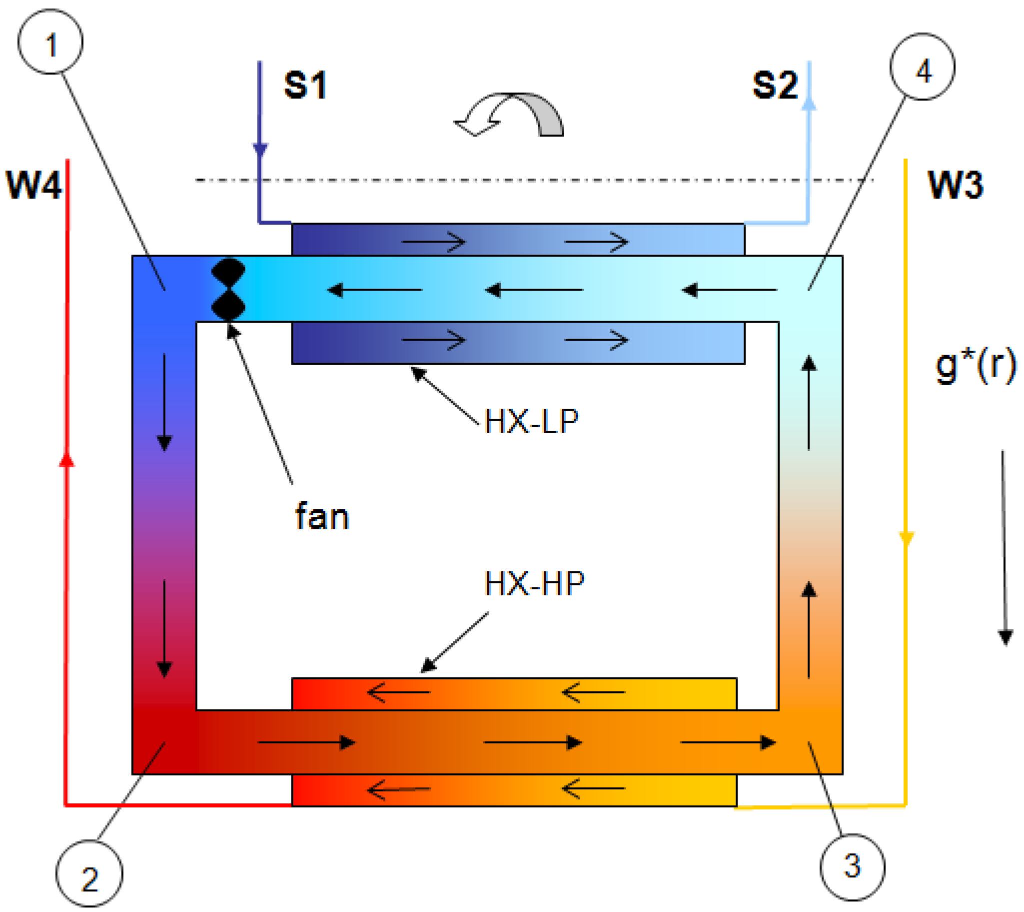

Figure 7: centrifugal compression heat pump cycle

Nearby the axis of rotation a Heat Exchanger for Low Pressure is situated (HX-LP). In the outer diameter a Heat Exchanger for High Pressure is realized (HX-HP). The thermodynamic cycle starts at point 1. The exemplary chosen working fluid is N_2. The points of the cycle are equivalent to the information given in 2.2.

The whole system rotates around the axis of rotation. The gaseous fluid is ventilated with very slow speed, below 7 meter per second, by the fan within the whole system. By compression, point 1 to 2, the temperature is rising from 273.15K to 308.15K. Because of the very slow speed of ventilation friction losses are very low. At the same time the higher pressure isn't realized by the rather inefficient conversion from kinetic energy but only by the centrifugal force itself. This means that exergy losses are rather low causing high efficient compression process.

From Point 2 to 3 the heat release takes place and the fluid is cooled down from 308.15K to 298.15K. The heat is removed through a heat exchanger (HP-HP) by a fluid, like water, which basically keeps its temperature even at higher pressures. It enters the rotating system at W3 and leaves it at W4.

At point 3 the expansion is starting. The temperature is lower and therefore the density at point 3 is slightly higher than before at point 2. Exergy is needed to transport the fluid from 3 to 4 supplied by the fan.

When the fluid has expanded the temperature has cooled down to a lower level at point 4 than on point 1. Because of the very low flow speed the friction losses are rather small again. Heat is needed to close the cycle. It is provided by a fluid entering the system at S1 and leaving at S2.

It can be shown by trivial transformation of the energy balance according to the system that the enthalpy difference from 1 to 2 is directly proportional to the difference of the peripheral speed of 2 to 1. These results can also be found in literature for turbo machines. The enthalpy difference itself is directly proportional to the isentropic temperature spread.

3.3 Analysis of single phase region heat pump – introduced cycle

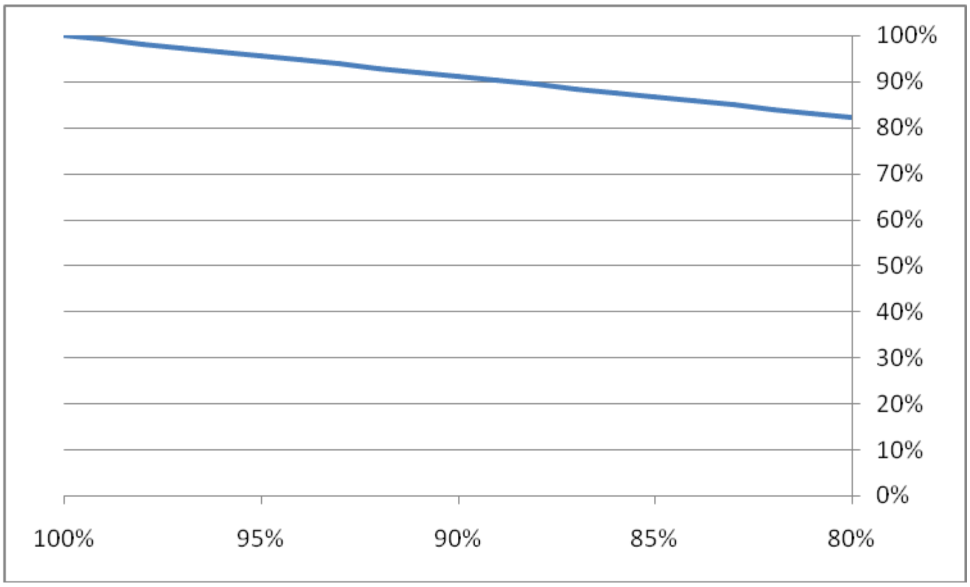

Figure 8: Influence of compression efficiency on reachable COP for mentioned example

Since compression and expansion work within the system high efficient, the main exergy losses will occur at the fan itself. For efficiency between 100% and 80% as well as assumed ideal expansion and compression the reachable COP drops as the figure shows at the mentioned circumstances.

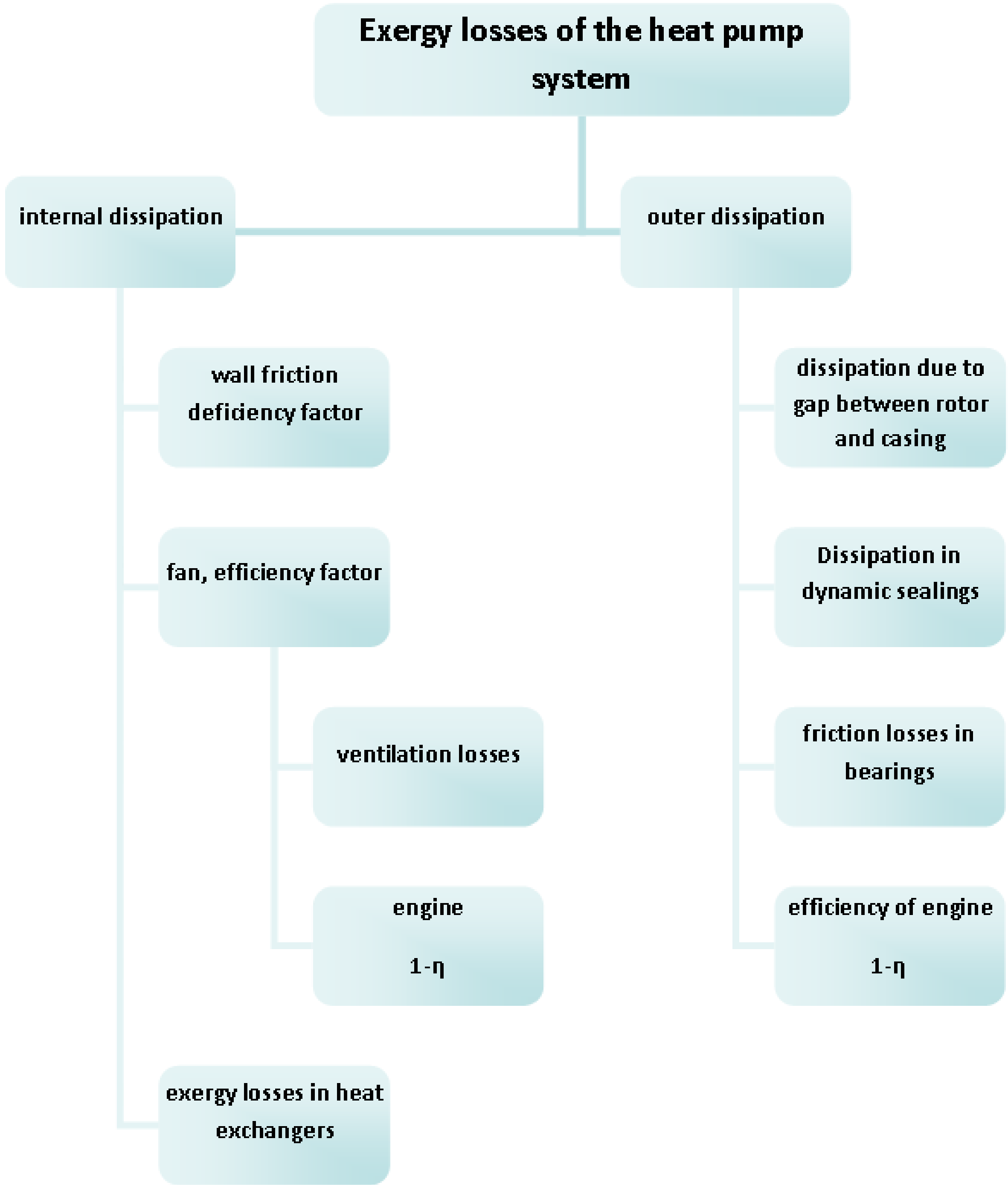

Figure 9: Exergy losses of the heat pump system

The figure above shows the exergy losses of the heat pump system. Of course it is not possible to reach the COP as shown in reality for those further losses will occur. Anyway a COP of around 7 for a 35K compared to a state of the art COP of around 5.5 for a vapor-compression or below 2 for a single phase system under similar circumstances sounds promising. First ideas to use the efficient way of compression by centrifugal forces go back several decades, like in Roebuck's device 1946. In order to judge if this system also looks promising after the mentioned losses, a set up of calculation models were developed.

The losses were identified by those calculation models. Due to engineering measures, all of them can be optimized. For example the dissipation due to the gap between the rotor and the casing can be minimized by evacuation. Anyway losses will occur, for those it is intended to "win them back" as heat within the system. The key is the internal dissipation since 80% will occur there, especially the implementation of the fan, with around 50% the biggest influence factor.

Afterwards those models were validated and if necessary corrected by real tests using a Laboratory Prototype, giving also detailed insight into the engineering problems of developing such a machine.

In the following figure the radius for the high pressure heat exchanger is about 0.6 m and 0.2 m for the inner one. To reach the given conditions for N_2 in 2.2 rotating speed of around 470 rad/s is needed. For the heat transfer it is of course beneficial if the starting pressure is higher, similar to CO_2 heat pumps.

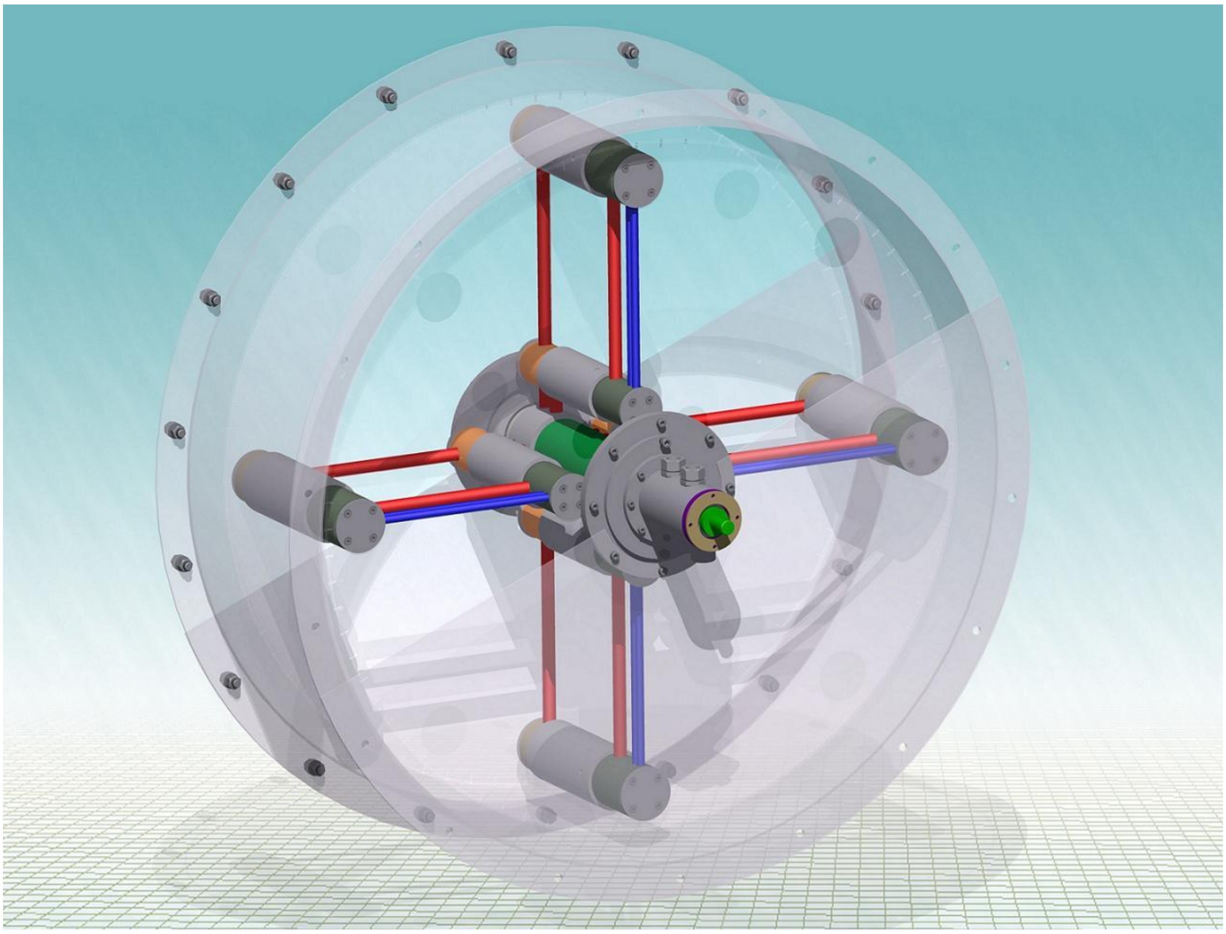

Figure 10: Laboratory Prototype of Heat Pump, diameter around 1.2 meter

The outcomes of those tests were more than promising, showing that the compression efficiency was indeed as high efficient as assumed. Moreover the amounts of losses and engineering challenges were identified. Analysis of this work validated the higher reachable COP at handleable development efforts.

4 CONCLUSION

The most significant difference of this technology is that it combines the advantages of both state of the art versions. The fan efficiency has rather small impact on the reachable COP as it is true for the two phase cycle while not having exergy losses because of an isenthalpic expansion. The system basically doesn't depend on the temperature level of the absorpt heat, as long as the working fluid is still in the vapor region. This has the important advantage that a machine is constructed for a temperature spread in the first place and not for specific absolute temperature levels. The reachable temperatures of this technology are mostly limited by materials, meaning that high temperatures can be realized like needed in industrial processes. The used working fluid is variable, which allows the usage of fluids without an increased global warming potential compared to CO_2. All in all the technology looks interesting, promising a final COP up to 6 for the example of a 100kW heat pump system with a temperature spread of 35K which is developed at the moment.

5 CREDITS

This paper gives a short introduction into the ideas and work done so far. Of course such a project is quite extensive and ECOP wouldn't have been able to realize it without the help of many different supporters and partners. Though it is not possible to name everybody since they are too numerous and growing from day to day we would like to mention at least the following.

We would like to express our thanks to AWS, FFG, INiTS and especially to ZIT (City of Vienna) for making the initiation of this project possible as well as to Linde AG for our great cooperation.

Thanks and regards to the RAG Rohöl-Aufsuchungs Aktiengesellschaft for believing in this project and the team, the co-financing of the work and the excellent supervision as well as our great way of working together to achieve our goals.

Thanks to Professor Willinger, from technical university Vienna, for his great support and time used to discuss our ideas.

Last but not least we want to express our special thanks to our coauthor Professor Ponweiser, from technical university Vienna, for his excellent supervision and support from the very beginning.

6 REFERENCES

NIST: http://webbook.nist.gov/chemistry, fluid database, Homepage of National Institute of Standards and Technology (NIST), 22.12.2010

BOHL W.1977. Strömungsmaschinen (Aufbau und Wirkungsweise), Vogel-Verlag, Würzburg