Increasing the evaporation temperature with the help of an internal heat exchanger

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1. ABSTRACT

Increasing the evaporation temperature by one degree lowers the power consumption of a refrigerator by about 3 percent. A method will be presented, how in many applications the evaporation temperature can be increased by about 3 to 5 K with the help of an internal heat exchanger.
This arrangement is advantageous for all refrigerants, even for ammonia, where normally internal heat exchangers are detrimental.

2. MOTIVATION

It is well known, that the power consumption of refrigerators is reduced by about 2-3 %, when the evaporation temperature is increased by 1 K. Therefore it is worth while to look for simple means to increase the evaporation temperature, if possible without changing the design of the evaporator or the type of the expansion valve.
In normal "dry" evaporators, as shown in figure 1, the heat transfer area on the refrigerant side is often only partially used effectively, because the last about 15 % are needed for the superheating of the refrigerant. The superheating is needed for two reasons: One needs a clear measurement for the control action of the thermostatic expansion valve and one wants to protect the compressor from liquid in the suction stream, which may lead to mechanical damage.

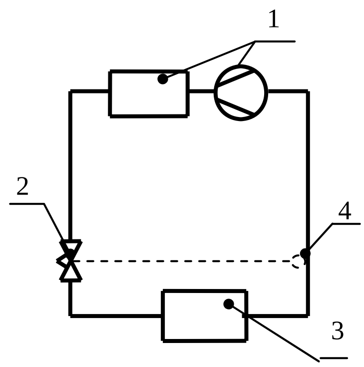

Figure 1 Simple refrigerating plant

The Motivation of this investigation was a non complete using of the evaporator area for the evaporation of a refrigerating medium. The figure 1 shows the foundation of the engineering [1]. The refrigerating plant consists of a compressor with condenser 1, an expansion valve 2 and an evaporator 3. The expansion valve regulates with help of his heat sensor the mass flow of a refrigerating medium to the evaporator in dependence from the refrigerating medium superheat after the evaporator.
It is well known, that the normal thermostatic expansion valve – due to a need of at least 6 K superheat – reduces the liquid content in the evaporator below the optimum level. Manufacturers of electronic expansion valves point out, that with their products a superheat of only about 2 K can be realized [2]. But if one looks closer into the dependence of the heat transfer coefficient on the liquid content and on superheating, one realizes, that it would be best, if there would be still 2 to 5% liquid at the outlet of the evaporator.

The superheat in this cycle is realized in the evaporator. The heat-transfer coefficient in this superheated area of evaporator is very low. For achievement of necessary superheat there is a need to work by low evaporating temperature. The low evaporating temperature leads to produce quick frost and to frequent thaw requirement.
The purpose of this investigation was to ~~re~~move the superheating out of the evaporator.

3. REALIZATION

If the superheating does not take place in the evaporator, then it must occur in an other heat exchanger. One possibility is to install an internal heat exchanger (IHX). Figure 2 shows an IHX as a counter-current exchanger 5. The intention is that in this IHX the residual liquid refrigerant is evaporated and then the vapour flow is superheated.

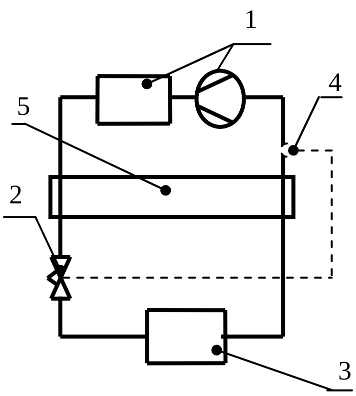

Figure 2 Realization

The sensor 4 of the expansion valve 2 must be placed between the IHX 5 and the compressor 1. This arrangement allows to use all of the evaporator area for the evaporation of the liquid refrigerant with a very good heat-transfer coefficient. Therefore it is possible to work with a higher evaporating temperature. The energy consumption is reduced and the volumetric refrigeration capacity is increased. In addition in air coolers a more homogeneous air temperature at the evaporator exit is obtained and the air temperature can be controlled in a tighter regime.
But unfortunately it has turned out in reality, that the cycle shown in figure 2 can not be used, because the control system is not stable. The fundamental problem of the flow diagram in figure 2 is that the deviation of the coming liquid from the evaporator to the IHX reaches the acting valve faster than the sensor of the controller. To avoid this, one has to look for an alternate flow diagram.

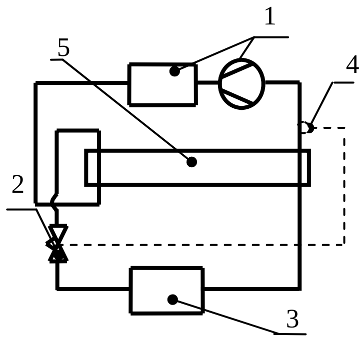

Figure 3 Alternative refrigerating plant

Figure 3 shows a flow diagram, where the IHX is arranged as a co-current heat exchanger. In this arrangement the disturbance by the change of liquid content on the evaporator reaches the sensor of the expansion valve sooner than the expansion valve itself. So the expansion valve has enough time to close the expansion valve to reduce the mass flow of the condensate before the better subcooled condensate reaches the valve.
This refrigerating scheme was investigated theoretically and experimentally.
The design of the IHX has an important influence. The used IHX consists of a shell and 5 internal pipes with longitudinal fins. The condensate flows inside the pipes and the low pressure flow runs on the shell side. It was theoretically and practically established that there is a maximum liquid content in the inlet of the low pressure flow, under which the control system can not work stable. The maximum for this design of IHX is about 8% liquid in the refrigerant flow rate.

4. THEORETICAL RESEARCHES

The theoretical researches have been done with refrigerating mediums R507 and ammonia. The refrigerating capacity was constant 10 kW and the cooling of the air through the evaporator was set from 8 to 4 °C for all investigations. In the case of the refrigerating cycle without IHX the superheat was chosen to 6K. The calculations were carried for different condensing temperatures in the range between 25 and 45°C. The superheating in the IHX was varied from 6 to 16K. In the case of ammonia the superheat was chosen to 6 and 8K.

To achieve the best results the IHX area was optimized for each superheating case. The higher the condensing temperature is, the more heat must be transferred in the IHX. For a constant control of the superheat under such conditions the IHX area was defined, so that the liquid content at the inlet to the IHX was never more then 8% for the highest condensing temperature.

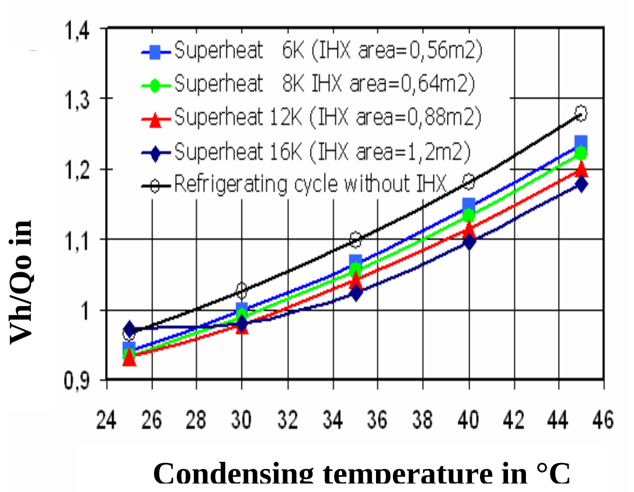

Figure 4 Required compressor volume (Refrigerating medium R507).

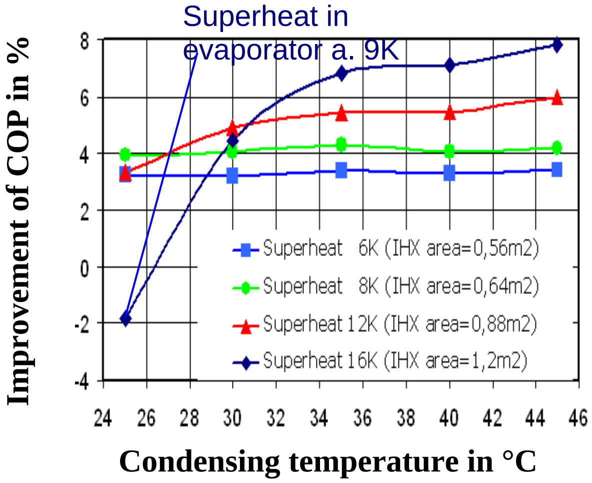

Figure 5 Improvement of COP (Refrigerating medium R507)

Figures 4 and 5 show for R507 the change of the required compressor suction volume and the change of the COP for different condensing temperatures depending on the IHX area. One realizes that the increase of the IHX area leads to a decrease of the required compressor suction volume. In addition, it reduces the energy consumption and improves the COP. But it also shows that one should not try too much. If one chooses a too large internal heat exchanger and aims for a too high superheat, this leads to a deterioration of the system at lower condensing temperatures. The reason is, that under these conditions, the evaporation end moves back into the main evaporator and one comes back to the operation without internal heat exchanger.

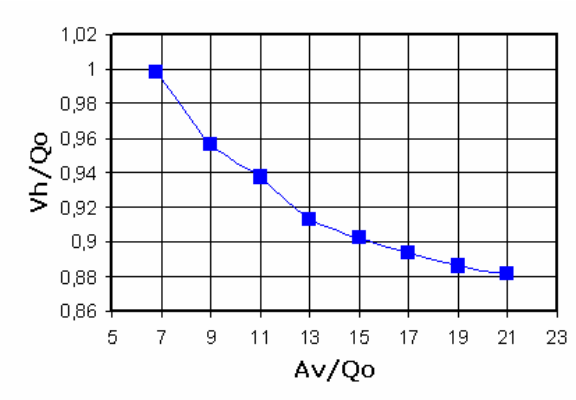

Figure 6 Required compressor volume (Refrigerating medium R507)

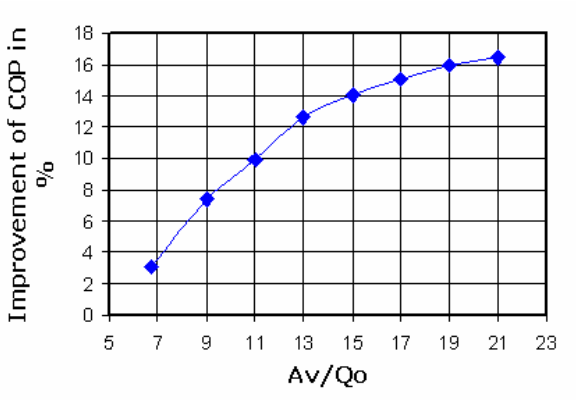

Figure 7 Improvement of COP (Refrigerating medium R507)

Figures 6 and 7 show a further decrease of the compressor suction volume and an improvement of the COP by use of one oversized evaporator. The calculations were done with a condensing temperature of 30°C and a superheating of 6K downstream of the internal heat exchanger. The results of the calculation show that with this flow diagram it pays to use an oversized evaporator. It can bring a remarkable increase of the COP. Of course there will a limit of improvement taking into account pressure drop and cost.

Similar investigations were done for a refrigerating plant with ammonia (NH_3) as refrigerant. Because of the special properties of HN_3 (small molecule weight) a use of an IHX is normally not recommended for HN_3-refrigerators, because the COP would decrease. But our calculations show that in our arrangement the effect of a co-current internal heat exchanger is positive both for the volumetric capacity as for the COP.
Figures 8 and 9 show the change of the required compressor suction volume and the COP for different condensing temperatures dependent on the IHX area. The calculations were done for the two superheats 6 and 8K. As can be seen the improvement of the COP with 8K superheating is smaller than with 6K. This is the case because of the negative influence of the superheating on the compression work with NH_3.
Similar to the same effect with R507, the COP can be further increased using an oversized evaporator.

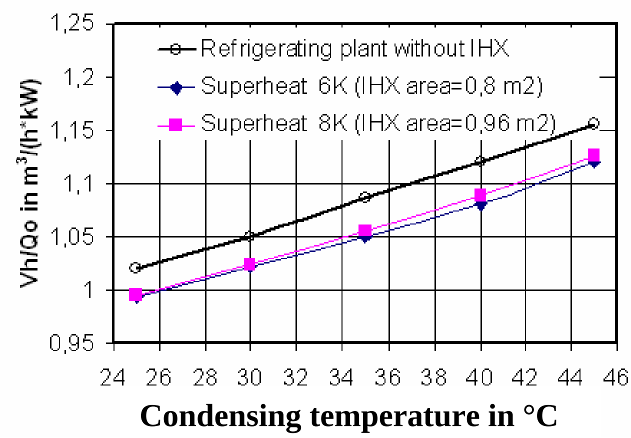

Figure 8 Required compressor volume (NH_3)

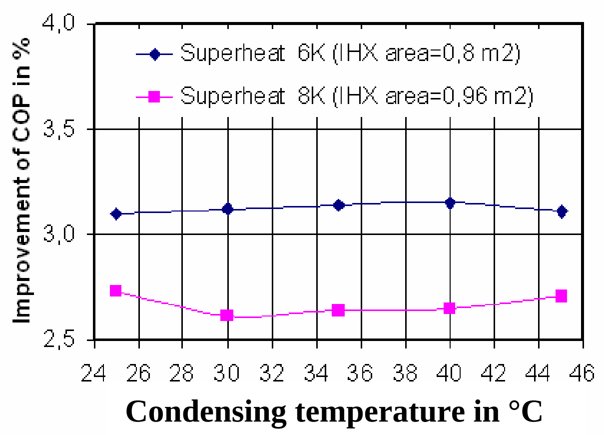

Figure 9 Improvement of COP (NH_3)

5. EXPERIMENTS

For a confirmation of the theory the experiments were conducted with the refrigerant R507. The task was to cool an air stream from 7 to 4°C. First the plant was operated with the conventional cycle (Figure 1) without IHX with 6 K superheat. Then with the cycle with IHX, the superheating was varied between 10 and 16K. The condensing temperature was varied between 30 and 39°C.
Then the results of the two experiments were compared.
Figure 10 shows an increase of the evaporating temperature of more than 2 K due to the IHX with unchanged air temperatures. As all experiments were conducted with the same compressor, the increase in evaporation temperature allowed an increase in refrigeration capacity. The large increase in refrigeration capacity with only a small increase in compressor power lead to a corresponding increase of the COP (Figure 11)

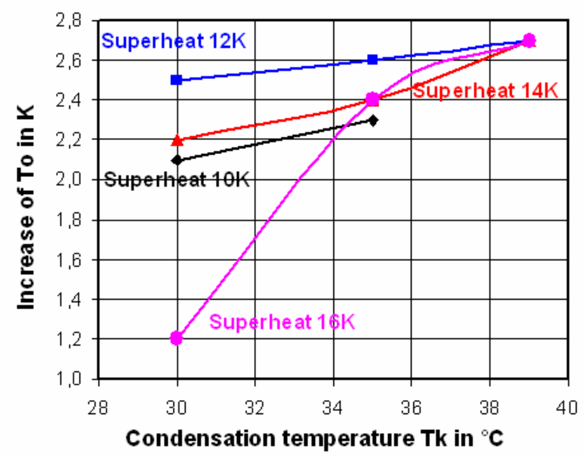

Figure 10 Increase of To

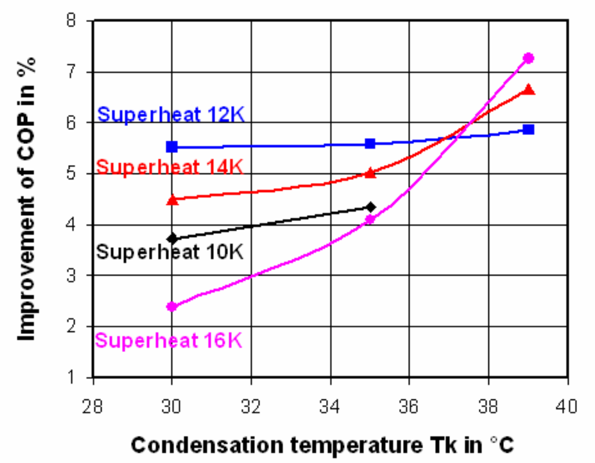

Figure 11 Improvement of COP

6. CONTROL SYSTEM

For a test of the control system the evaporator load was decreased rapidly from 100 to 70%. Figure 12 shows the temperatures at the inlet and the outlet of the evaporator and the change of the controlled superheat in function of the time. Figure 16 shows that after

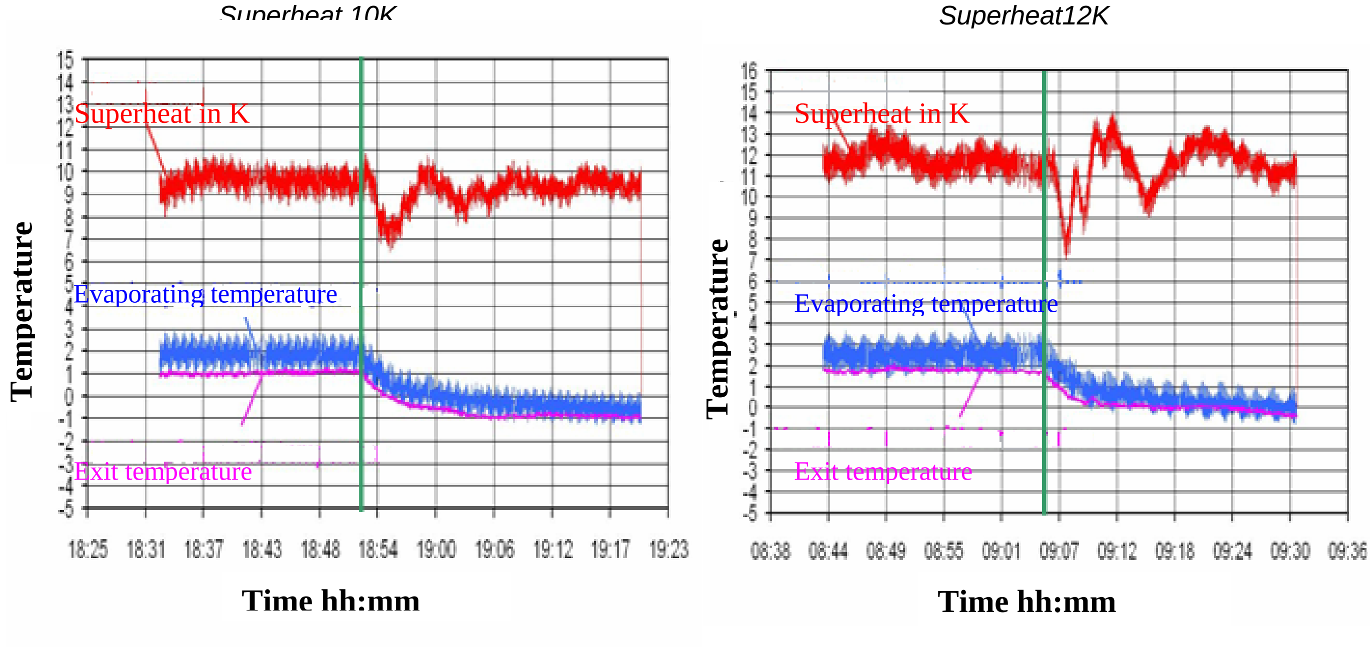

Figure 12 Dynamics of the control system (condensation temperature: 35°C)

the decrease of the evaporator load the superheat goes down in both cases till about 7K and then the control system finds the stable working point again. In the case of the 10K superheat, the liquid content oat the IHX inlet was about 5% and in the case of 12K superheat about. 3%.

Figure 13 shows what happened when the liquid content was about. 7 - 8% (higher condensation temperature). In this case, the superheating decreased from 12 to 2 K for a short time and then the control system found the working point again.

For these experiments a pulsating electric expansion valve was used.

Experiments with a simple thermostatic expansion valve are planned.

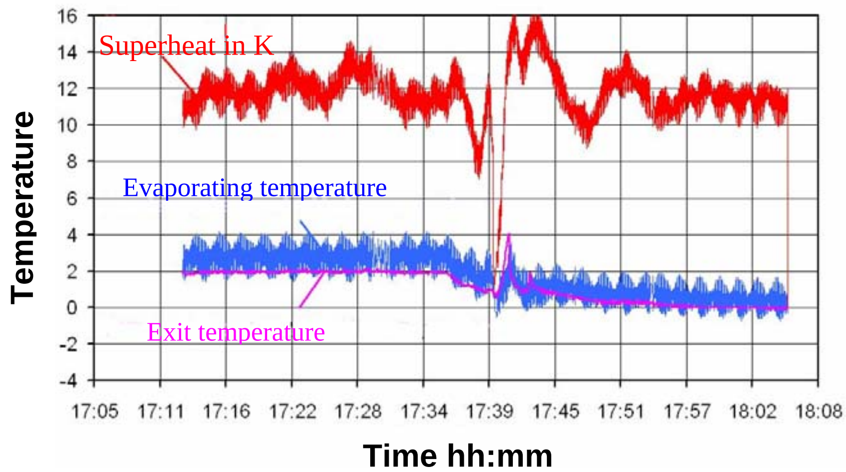

Figure 13 Dynamics of the control system (Superheat: 12K, condensation temperature: 39°C)

7. SUMMARY

By using the co-current IHX the evaporation temperature can be increased by more than 2K. It leads to a reduced of the energy consumption and an increase of the volumetric refrigeration capacity. The refrigeration cycle works stable even under large and rapid changes of the evaporator load.

The increase of the evaporation temperature leads to additional benefits:

There is less frost formation on the air side and therefore less need for defrosting.

Owing to the use of all evaporator area only for the evaporation of the refrigerant and no superheating, the minimal difference between air and the refrigerant temperature in the evaporator can be reached. It enables to control the air temperature e. g. in a cold store rather accurately.

The choice of an oversized evaporator in the refrigerating cycle would bring a further increase of the COP.

8. REFERENCES

[1] Mickan, P.; Fischer, H.; Handbuch Grundlagen der Kältetechnik

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