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Optimal Distribution of Condenser Area for Retrofits

A. Johansson
The Royal Institute of Technology

P. Lundqvist
The Royal Institute of Technology

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A. Johansson and P. Lundqvist
Dept. of Energy Technology
Div. of Applied Thermodynamics and Refrigeration
The Royal Institute of Technology, KTH
Stockholm, Sweden

Abstract

Test results from field retrofit situations, where R22 has been changed to Isceon 59[1] and R407C in two identical chillers, shows that both substitutes require more subcooling than R22 to give the system a satisfying performance. It was also noted that to achieve the same subcooling, one had to charge the system with a larger amount of refrigerant. To achieve the same efficiency, COP_2, with Isceon 59 as with R22, one had to add approximately 25% more refrigerant, than the system had with R22.

A computer model of a system consisting of coaxial heat exchangers, with refrigerant in the inner tubes, has been made. In the model five R22 substitutes (R134a, Propane, R404A, R407C and Isceon 59) and R22 were compared. The simulations showed a similar behaviour as was experienced in the field tests: To make the system perform its best with Isceon 59 the area used for subcooling had to be larger than when R22 were used. With other substitutes, like R134a, a smaller subcooling area gave the maximum performance of the system. R134a can be a very interesting R22 alternative if it is energy efficiency that is important, and not primarily capacity.

To find the correct amount of refrigerant charge for a specific system, it is necessary to look at the shape of the wet region when plotted in a property plot, but it is also important to look at the relative size of the heat exchanger area in both condenser and evaporator, for the specific refrigerant used.

Introduction

Legislation in many countries makes the study of replacing R22 in old heat pump and refrigeration facilities interesting. Usually the substitutes are chosen from refrigerants with look alike volumetric refrigeration capacity and vapour pressure. These two parameters has been dealt with extensively in the literature. This paper addresses another factor: System charge. To achieve the same, or at least satisfying performance with a new working media, it is often necessary to charge the system differently than was the case with R22[2]. The question is just how much refrigerant should one charge into the system when retrofitting, to get the maximum performance with the chosen R22 replacement? It is not possible to make a general study of the appropriate/optimum system charge from volumetric refrigeration capacity and vapour pressure alone. Parameters such as transport properties and other more fundamental thermodynamic properties has to be taken into account.

Comparing refrigerants in a system

When the performance of refrigerants in a certain facility are compared, it is common to use the carnot efficiency as a "benchmarking" tool. The carnot efficiency definition that is often used is the definition describing the efficiency for the refrigerant – the refrigerant cycle carnot efficiency, $\eta_{C,Cycle}$:

$$\eta_{C,Cycle} = \frac{COP_2}{COP_{2,C,Cycle}} \qquad \text{Where... } COP_2 = \frac{\dot{Q}_2}{\dot{E}} \qquad \text{...and } COP_{2,C,Cycle} = \frac{T_2}{T_1 - T_2}$$

[1] Isceon 59 is a zeotropic refrigerant mixture consisting of R134a (50 mass-%), R125 (46 mass-%) and R600a (4 mass-%).
[2] It is usually, more or less, impossible to achieve the same refrigeration capacity with the R22 substitutes presently available on the market, without altering the system design in a more thorough way than just changing the refrigerant, and possibly lubricant.

T_1 is the condensation temperature in Kelvin, T_2 is the evaporation temperature, $\dot{Q}_2$ is the refrigeration capacity ,and $\dot{E}$ is the power supplied to the compressor.

For zeotropic refrigerant mixtures the evaporation and condensation temperatures are defined as follows[3]:

$$T_1 = \frac{T_1' + T_1''}{2} \qquad\qquad T_2 = \frac{T_{2S} + T_2''}{2}$$

Where T_1' is the bubble temperature at condenser pressure, T_1'' and T_2'' are dew temperatures at condenser and evaporator pressure, and T_{2S} is the refrigerant temperature at the evaporator inlet.

These definitions give a somewhat misleading view of the system performance. In most systems you have both a secondary refrigerant, e.g. water or a mixture of water and propylene glycol, and a coolant for the condenser, e.g. water or air. Since the utilised temperatures are not the evaporation or condensation temperatures, but the brine and coolant temperatures, the latter ones gives a much better view of the system performance. This gives another definition of carnot efficiency – the system carnot efficiency, $\eta_{C,System}$:

$$\eta_{C,System} = \frac{COP_2}{COP_{2,C,System}} \qquad\qquad COP_{2,C,System} = \frac{\bar{T}_{brine}}{\bar{T}_{coolant} - \bar{T}_{brine}}$$

Where $\bar{T}_{brine}$ is the arithmetic mean temperature of the brine, and $\bar{T}_{coolant}$ is the arithmetic mean temperature of the coolant, in Kelvin[4]. The subcooling is calculated from the bubble point temperature at condensation pressure, T_1', and the superheat from the dew point temperature at evaporation pressure, T_2''.

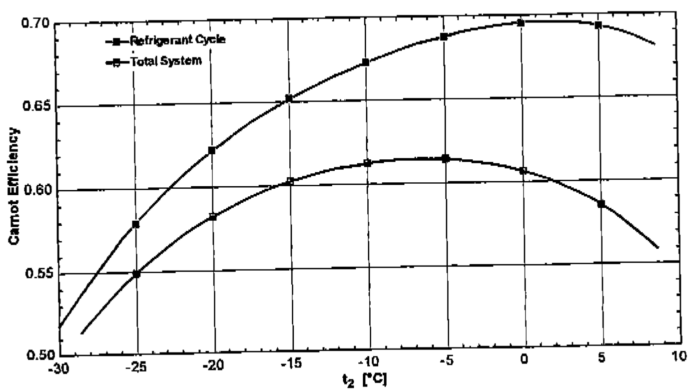

Figure 1 The two different definitions of carnot efficiency gives radically different results when plotted as functions of evaporation temperature. The diagram describes the principal behaviour of a system running with R22 as working media, and a condensation temperature of 40 °C. (The plot is taken from another study.)

Subcooling

What is hoped to be gained by subcooling is a decreased amount of vapour after the expansion device. The larger amount of refrigerant liquid, to be evaporated, gives a higher refrigeration capacity. How much of the refrigerant that remains in liquid phase depends on evaporation and condensation temperature, amount of subcooling, and the refrigerant used. If the temperatures, evaporation, condensation and subcooling, are set to fixed values, then all depends on the shape of the wet region[5]. The shape of the wet region differs between different refrigerants: A refrigerant with a wet region that is more tilted, e.g. R134a, needs a lot of subcooling

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[3] Refrigerant mixtures that under constant pressure evaporates under increasing temperature. For condensation the reverse is applied.
[4] For simplicity the arithmetic mean temperature is used instead of the logarithmic mean temperature. From a strict thermodynamic point of view the latter should be used. In these cases the difference between the two methods is so small that the first method is sufficient.
[5] The shape of the wet region depends on the molecular structure and molecular mass of the refrigerant[1].

to reduce the otherwise high vapour content after throttling, whereas a refrigerant with a more upright shape of the wet region, e.g. R22, gains less on increased subcooling.

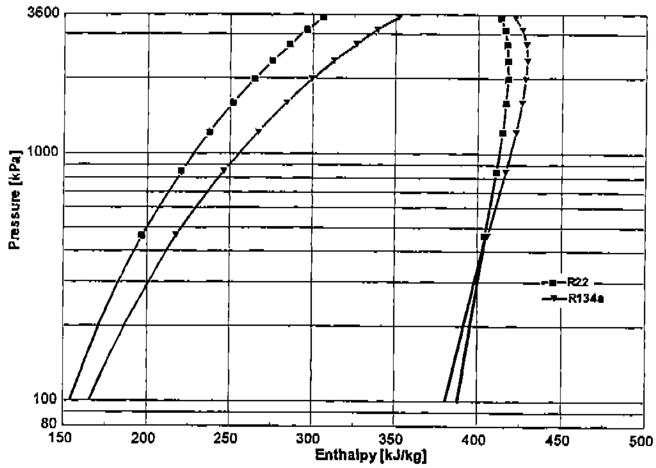

Figure 2 *As can be seen in the figure above, R22 has a much more upright shape of its wet region than R134a, when plotted in a enthalpy versus pressure diagram. This would imply that more subcooling would be beneficial to R134a's performance.*

It is thus obvious that the shape of the wet region for the chosen refrigerant has to be taken into account when recharging the system when retrofitting. To achieve the desired subcooling, the system has to be charged differently, so that the area used for subcooling in the condenser is large enough. This will however steal area from the condensation and de-superheating part. At a certain area used for subcooling, X_L, the COP and carnot efficiency will not increase anymore: It has reached its maximum for the specific facility, and further filling of refrigerant will lead to poorer performance. This can be seen in figures 5 and 6.

Model for computer simulations

To study the basic influence of how the system has been charged a computer model describing a system with coaxial heat exchangers, with refrigerant in the inner tubes, and a scroll compressor has been built, using *EES* and *RefProp*[6]. The boundary conditions for the model has been inlet temperature and mass flow of the coolant (water), 35 °C and 0.5 kg/s, and outlet temperature and mass flow of the brine (water), 6 °C and 0.5 kg/s. I.e. a simple chiller. COP, subcooling, condensation and evaporation temperature, system Carnot efficiency, and a few other parameters have been studied as functions of how much of the condenser are that has been used for subcooling, X_L; the relative length of the condenser used for subcooling[7].

The computer simulation model uses the *Dittus-Boelter* correlation to calculate the heat transfer coefficients for coolant, brine and refrigerant subcooling[8][2]. The *Pierre* correlation was used for evaporation heat transfer coefficients for the refrigerant[3]. To obtain heat transfer coefficients for the refrigerant condensation, the *Shao-Granryd* correlation for condensation in horizontal tubes has been used[4]. The compressor, a scroll compressor, is assumed to be adiabatic with a constant isentropic and volumetric efficiency.

Simulated refrigerants

When retrofitting from R22 and a minimum of system modifications is desired, the choice of refrigerant is restricted to a few alternatives. Refrigerants which will give a system pressure far higher than those experi-

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[6] Engineering Equation Solver by F-Chart Software. RefProp 6 by NIST.

[7] E.g. X_L=0.1 means that 10 % of the condenser length is used for subcooling and 90 % for condensation and de-superheating.

[8] The *Dittus-Boelter* correlation has been used in case of turbulent flow. This has always been the case in these simulations. If laminar flow had occurred the *Kays* correlation would have applied.

enced with R22, usually has to be put aside. E.g. R410A and R32. Both of these two alternatives would also require adjustments to compressor and compressor motor, otherwise the motor will quite likely be damaged. Therefore results from simulations with these refrigerants are not represented in this paper.

The hydrocarbon propane, R290, has been studied, even though it is flammable and not very likely to be used in a retrofit situation. Further refrigerants and refrigerant mixtures that have been put under test in these computer simulations are: R22, R134a, R407C, R404A and Isceon 59. R134a gives a significantly lower capacity than R22 when run in a facility originally designed for R22, but in cases where efficiency is more important than capacity, it is an interesting alternative.

Results from computer simulations

In all the following diagrams the different parameters are plotted as functions of the relative area, or length, of the condenser used for subcooling. Note that these values are to be studied ad hoc – they are only valid for this specific computer model.

Different refrigerants will give different values on subcooling, Δt_{sc}, at the same area used for subcooling, X_L. This is caused by many factors: E.g. if the refrigerant results in a significantly smaller heating capacity, as for e.g. R134a, the condenser will be relatively larger than for a refrigerant that gives a higher heating capacity, e.g. R404A, in the same system (originally run with R22). This will result in a smaller temperature difference between refrigerant and coolant for R134a. This will then lead to smaller subcooling, since the available temperature difference is smaller than for R404A. Other factors that influence the amount of subcooling achieved by a certain refrigerant at a certain X_L, are conductivity, viscosity and density of the refrigerant liquid. The latter two making the flow more or less turbulent, and thus effecting the heat transfer coefficient together with the conductivity for that specific refrigerant.

It should be noted that the definition for subcooling used in this paper, makes the temperature difference in the subcooling part a few degrees lower for zeotropic refrigerants than if the subcooling had been calculated from the condensation temperature. Since the subcooling is calculated from the bubble point temperature, which is a couple of degrees lower than the mean condensation temperature for zeotropes, the temperature difference between refrigerant liquid and coolant is smaller than for pure refrigerants.

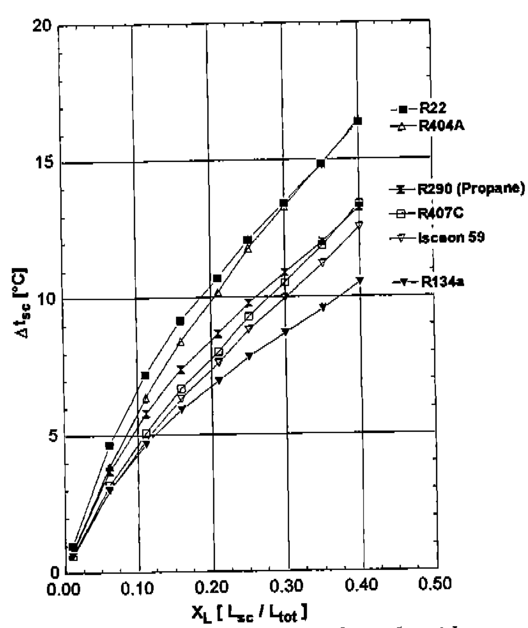

Figure 3 The subcooling achieved with different relative amounts of subcooling area, for R22 and five different R22 alternatives.

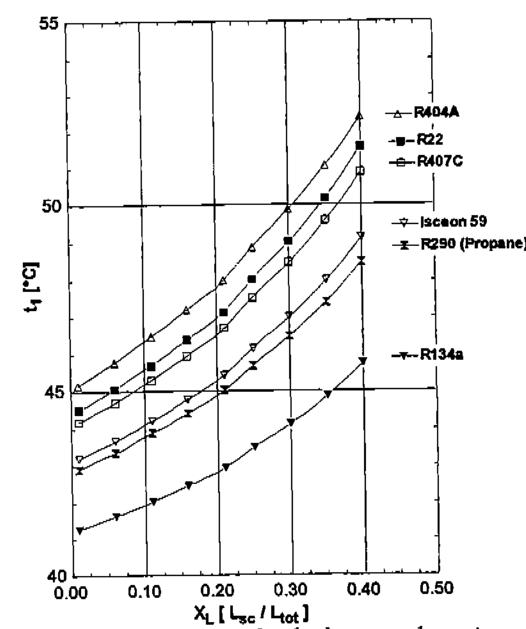

Figure 4 R404A gets the highest condensation temperature, t_l, and R134a the lowest. This is one of the reasons for R134a's smaller amount of subcooling.

R404A gives approximately the same amount of subcooling as R22 at the same X_L. Since the condenser area is relatively small when the system is run with R404A, it has a rather large temperature difference to utilise as a driving force, when all the refrigerant is in liquid phase. It is therefore possible for it to obtain a larger amount of subcooling than with e.g. R134a or Isceon 59, with the same system charge.

More condenser heat will result in larger temperature differences, and thus increasing the condensation temperature. This is beneficial for large amounts of subcooling, but decreases the carnot efficiency of the system. As can be seen in figures 5 and 6, that when retrofitting to e.g. R404A or Isceon 59, the system needs a larger refrigerant charge than it had with R22, to perform its best.

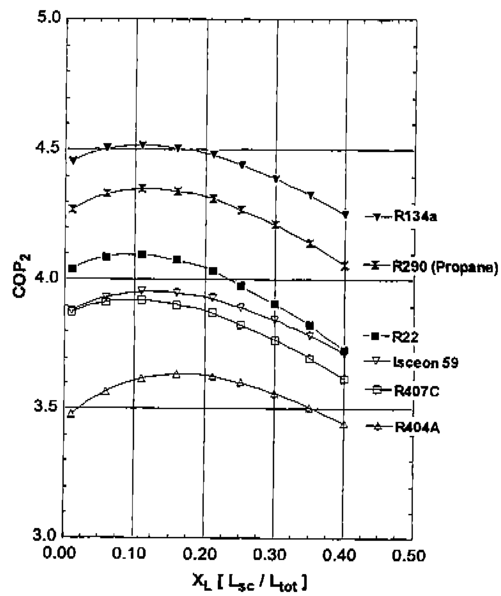

Figure 5 *COP_2 achieved with different charge (relative amounts of subcooling area).*

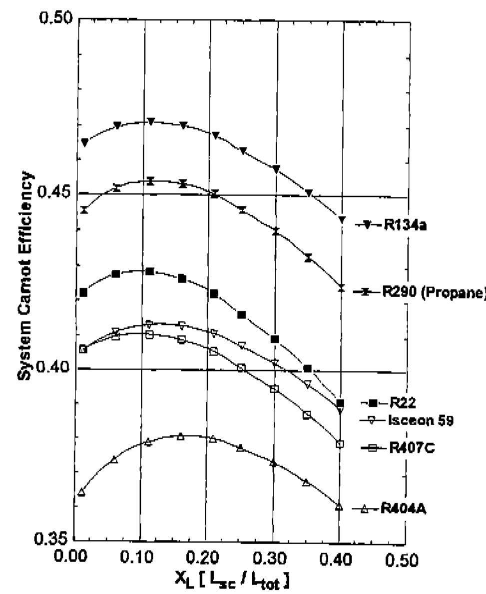

Figure 6 *Propane is benified by a larger amount of subcooling area, system charge, than e.g. R407C. Isceon 59 by an even larger charge.*

Comparisons between field retrofits and computer model

The performance of two R22 replacements, R407C and Isceon 59, have been studied in two identical chillers in a phone station outside Stockholm. The chillers uses two tube-and-shell heat exchangers, with refrigerant on the outside of the tubes, as condenser. The two heat exchangers are connected in series, and the first one is originally intended to be used as a heat recoverer/de-superheater. They have however never been used for this purpose, since it was discovered after installation that there in reality was no need for it(!). The evaporator is also a tube-and-shell heat exchanger but with the refrigerant on the inside of the tubes. Reciprocating compressors are used, and each chiller has two.

When the system used R22 as working media, the charge was 22 kg. To achieve approximately the same subcooling, 5 °C, with Isceon 59, the system had to be charged with 26 kg. And to achieve more or less the same COP_2 another 2 kg refrigerant had to be added. The system still delivered 20 % lower refrigeration capacity than it had done with R22, but the systems energy efficiency was the same. With R407C a similar behaviour could be noted. It was however not possible to achieve the same energy efficiency as with R22 in these chillers.

It needs to be stressed that the behaviour of subcooling is somewhat different in a horizontal tube with forced convection, as in the computer model, than it is in a case of more or less free convection outside tube banks, as in the case with tube-and-shell heat exchangers.

A study to verify the results from the computer model is currently done, on a system consisting of four plate heat exchangers, with different geometry but equal area, as condensers.

Conclusion

When a system is retrofitted from R22 to some R22 replacement, the system needs to be charged differently than it was originally. How the system should be charged to perform its best depends on the replacement used, and what is most important, refrigerating capacity or energy efficiency. The appropriate/optimum charge of a system depends on the overall system design, and:

- The relative refrigeration capacity (compared to R22) of the substitute.
- Whether the substitute is a refrigerant mixture with a considerable glide or not. The temperature glide will result in smaller temperature difference between refrigerant and coolant in the subcooling part of the condenser, and thus making the subcooling smaller than if the refrigerant had been a pure refrigerant or an azeotropic blend of refrigerants, with the same condensation temperature.
- Transport properties of the substituting refrigerant.
- The shape of the wet region in a property plot of the refrigerant.

If the heat exchanger area in the system is large, as seldom is the case, it is even more important to charge the system correct. A system with relatively small heat exchanger surfaces seems to need a relatively larger charge, more surface used for subcooling, than a system with relatively large heat exchanger surfaces, to perform its best.

In a system similar to the model in these computer simulations, with condensation inside horizontal tubes, or in a system with plate heat exchangers, it is possible to vary the area used for subcooling continuously. This is usually not possible in tube-and-shell heat exchangers, where the increase of subcooling area only can be varied in discrete steps. E.g. consider a tube-and-shell condenser, with condensation on the outside of the tubes, originally designed with a certain number of subcooling tubes. If the area used for subcooling is to be increased, the system needs to be charged with so much more refrigerant that it reaches the condenser tubes above. An increase of subcooling area would then take away a considerable amount of condenser area. The subcooling might increase dramatically, but so will the condensation temperature due to poorer heat transfer coefficients in the part used for condensation and de-superheating.

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Refrigerant data are taken from the RefProp 6 database for thermodynamic properties for refrigerants and refrigerant mixtures, by NIST.