Phase-out of refrigerant R22

Doctoral thesis by
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Stockholm, June 2003

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Trita REFR Report No. 03/39
ISSN 1102–0245
ISRN KTH/REFR/R–03/39–SE

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The current PDF-version has been revised and the changes from the errata sheet has amended.

Printed by Universitetsservice US AB
Stockholm, 2003

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av

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AKADEMISK AVHANDLING

som med tillstånd av Kungliga Tekniska Högskolan framläggs till offentlig granskning för avläggande av teknologie doktorsexamen i Energiteknik, fredagen den 6 juni 2003 klockan 10:00 i sal M2, Brinellvägen 64, Kungliga Tekniska Högskolan, Stockholm. Avhandlingen försvaras på engelska.
Abstract

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Phasing-out refrigerant R22 in existing machinery has proven to be complicated in other ways than phasing out CFCs like R12 once was. As the Swedish refill stop for HCFC-refrigerants came into force January 2002 the focus on replacing R22 in existing facilities and installations was strengthened. In the Swedish refrigeration community rumours started to flourish early on as zeotropic refrigerant mixtures was beginning to be used as substitutes for R22 –rumours concerning phenomena and behaviours previously unheard of in the business.

The current thesis questions how performance comparisons of various alternative refrigerants and refrigerant mixtures are conducted. It presents alternative systems extensions and model resolutions that better represent the issue of replacing refrigerants such as R22 in existing machinery; systems extensions and models somewhat different than the ones that gradually have become accepted within the international refrigeration and heat pump community. Results from predictions using a multitude of these models are presented.

If an analysis on why a particular unit using a zeotropic refrigerant mixture as working media behaves at it does is to be conducted – an analysis going further than considering momentary efficiency and capacity alone – the circulated composition has to be at hand. The current thesis presents one indirect method to use to get indications on that something has happened with the circulated composition, due to leakage etc. Another method for online measurement of the circulated composition without taking samples is also presented.

In the current thesis, explanations and explanatory models that answer many of the “strange behaviours”, reported already early on as zeotropic refrigerant mixtures were started to be used, are presented.

Keywords: Retrofit, Phase-out, CFC, HCFC, HFC, R22, R407C, R417A, R404A, R134a, Ozone depletion, Zeotropic refrigerant mixture, Circulated composition, Measurement, Energy, Energy efficiency, System, Systems analysis, Modelling, Simulation, Prediction
Retrofit

(1) To furnish (as a computer, aeroplane, or building) with new or modified parts or equipment not available or considered necessary at the time of manufacture. (2) To install (new or modified parts or equipment) in something previously manufactured or constructed.

"Encyclopaedia Britannica Online" (2000)
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The literature on R22 and R22-substitution is flourishing, but very little is aimed at retrofitting in the sense it is treated in the current thesis. Most experimental work treats heat transfer issues, materials compatibility, or system performance. The latter could have been relevant if the underlying models in theoretical evaluation or measurements had been reported; but they are generally not. Much of the literature is reporting the performance according to ASHRAE standard tests procedure (black box) or relatively simple cycle modelling exercises using standard software such as Refprop. The lack of model discussions in e.g. case records in the literature makes it very hard to draw conclusions that meet demands on generality. This is why there is no traditional literature review in the current thesis.

Other than technical aspects of refrigerant phase-outs and the causes there of are treated by e.g. Stern et al, Vedung and Klefbom, Nolin and in a series of reports from the Swedish Environmental Protection Agency (S-EPA). (Nolin 1995; S-EPA 1990; 1992; 1994; 1995a; b; 1996; 1997a; b; c; 2000; Stern et al. 1992; Vedung and Klefbom 2002) They however treat the issue either as (interesting) cases in political science, economy, policy making, applied philosophy, etc; and do not treat any technical problem aspects. These, other, aspects are important in there own right, but are not sufficient to handle the problem in technical practice: In e.g. Vedung and Klefbom (2002) the presence of any technical problems are not considered. It is important to have some understanding of these non-technical issues too, but they alone cannot solve the problem issues concerning the phase-out of refrigerant R22 either.

Stockholm, May 2003
The research preceding the fore lying thesis has been financed by the Swedish Energy Administration (Energimyndigheten) and to some extent by the Swedish Environmental Protection Agency (Naturvårdsverket).

I would like to thank every one who have worked at the Division of Applied Thermodynamics and Refrigeration at the department of Energy Technology, KTH, over the years I have been there; especially the following people:

First of all Per Lundqvist my supervisor and friend, for helping me on the way: Thank you for standing – barely though at times – my impatience and bad habit of loosing interest as soon as I feel that I have solved the tricky (and funny) part.

Fredrik Lagergren, my room mate, for all the hours of discussion and cooperative thinking. Sharing room with you for the past years has been like striding a constant intellectual uphill climb. You have had an enormous influence on this thesis – it’s your entire fault!

Hans Jonsson and Björn Palm, for, among many other things, giving me invaluable comments and questions on the manuscript of the fore lying thesis.

Martin Forsén, for giving me so much of his time and letting me use his software. Jaime Arias, for helping me out with the world of supermarket refrigeration. Joachim Claesson, for giving me tips on robust heat transfer correlations to use. Benny Sjöberg, our supreme lab technician. Peter Hill, for making such wonderful measurements for me, and letting me use your sampling apparatus for my strange test rig. Jerry Zetterqvist and Kenneth Weber for helping me ask relevant questions in my research.

Professor Sanford A. Klein, for providing me with the best software for quasi-static-thermodynamics computer-simulations and modelling, and for spreading Brineprop for me. Åke Melinder, for letting me use his correlations and parameters for thermo-physical properties of secondary refrigerants in Brineprop.

…but most of all, I would like to thank my wife, Jenny, for putting up with me.
If adequate methods do not exist, they have to be developed, invented and tested. A trial-and-error approach may be available theoretically, but, as a practical matter, it may be prohibitively costly or too risky.¹

– In 1996, the Electrolux home appliance corporation sponsored a masters of Science thesis on the development of a substitute for R12 in both domestic and commercial refrigerators and freezers. Naturally, the substitute had to fulfil the traditional safety demands for refrigerant fluids, such as non-toxicity and non-flammability, but also it had to be compatible with the old lubricants present in existing R12-units. Another aim was to obtain performance – cooling capacity and energy efficiency – similar to what had been the case when R12 still was in use in these particular units. To achieve this, a blend of fluids had to designed, and a set of performance and safety criterion was setup: The blend must be (sufficiently) non-flammable and non-toxic; The blend must have volumetric cooling capacity and energy efficiency in the same range as R12; The blend had to be compatible with mineral oils, and if a blend of HFCs were chosen it had to be complemented with, probably, a hydrocarbon (e.g. isobutane) to solve the lubricant in the refrigerant; The components of the blend all had to be compatible with (at least) Swedish legislation and phase out plans for ozone depleting substances. Hence, the blend could not include R22 as e.g. R401A does.

It was decided that the refrigerant mixture should contain R134a and R227ea as the main working fluids and a smaller amount of R600a (isobutane) should be added to maintain compatibility with the legacy lubricants.

¹ Miser and Quade (1985) p.14
The role of R227ea in the mixture was to provide flame-suppressing abilities to it even though R600a is highly flammable.

A series of tests and computer simulations were conducted. Tests to set ranges for composition to keep the blend non-flammable, simulations so set composition ranges to maintain desired cycle performance when operated in existing equipment, and finally tests of the fluid in a lab rig to estimate the relative performance when operated in legacy R12-units. The flammability tests (a miniature ignition barrel made of transparent acrylic) showed that the suggested composition was hard enough to ignite with an open flame to satisfy Electrolux, and the lab rig tests proved the computer simulations to be adequately correct – when used in existing machinery it rendered performance very close to what was achieved using R12.

Even though Electrolux were pleased with the results of the Master of Science thesis project, they still decided not to go forward in testing and implementation of the suggested refrigerant mixture, RetroFix-12. It was evident that to Electrolux (and perhaps even to their customers) it was better business to sell new units instead of conducting refrigerant retrofits with a homemade refrigerant mixture. Apparently, refrigerant phase-outs and refrigerant retrofitting is not only a question of thermodynamics and chemistry. Understanding something about business and how and what the circumstances and situation are for different applications, appears to be just as important.

Epistemological environment

Phasing out refrigerant R22 in existing machinery has proven to be a far more complicated business than phasing out the CFC’s, like R12 and R502, were. Not that the phase out of these were without problems – especially concerning lubricants and materials compatibility – they were very laborious indeed, but to the service personnel conducting the actual retrofits, things concerned changes in

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2 Johansson and Jansson (1997)
procedure not thermodynamics. The problems with lubricants and materials compatibility was a question primarily dealt with by equipment manufacturers – mainly multinational companies with a high degree of engineering knowledge, and a well developed ability to treat problems of this type. With these changes (mainly concerning evacuation of the circuit and how to handle the highly hygroscopic polyol-ester-oils etc), the phase out in existing machinery boiled down to basically just changes in procedures for the service technicians. When a piece of machinery had been retrofitted from R12 to R134a, it still behaved in the same manner as before the change of working media. Thus, the operation of the unit as a part of a larger system was not affected and there by the user and owner side did not notice any significant change in availability, capacity or efficiency.

To service technicians active in the refrigeration and heat pump business the refrigerant fluid, the working media, ought to be considered a machine component. Taking a mechanical point of view this is the obvious use of it. Still many, even with a mechanical background, would claim it is a chemical (which obviously also is true); when people are asked give to number of the necessary components it takes to build vapour compression heat pumping machine, most people answers “four”: Evaporator, compressor, condenser and an expansion devise. The need for a working media is neglected! I.e. the refrigerant fluid is used as a machine element, or component, but by many people in the field considered only a chemical – even rather mysterious by many.

The knowledge formation within the refrigeration service technician’s community is mainly based on personal experience and oral communication with colleagues. As the level of formal education generally is relatively low and deeper understanding of the thermodynamics of refrigerants and refrigeration equipment likewise, this group becomes dependent on recommendations from dis-

Further elaboration on the context and history of refrigerant phase outs is found in chapter 2.

**Overall aims and issues**

In the current thesis, and the preceding research done, the following issues are treated.

*Firstly: How will existing facilities perform after a refrigerant retrofit from R22 to any of the commercially available substitutes?*

The obvious answer is that “it depends”. In this thesis the focus is on how “it depends”, and why “it depends” on “what”. Reasons to how facilities will perform and as to why facilities will perform as they do after a refrigerant retrofit, is treated in chapters 3 and 6. Issues concerning finding the appropriate systems charge for the substitute refrigerant and the design of the software Desktop Retrofit is treated in chapter 4.

*Secondly: What can be done to improve the conditions for retrofitting, and what can be done to improve energy efficiency?*

How are retrofits to be made successful in a wider perspective than that “at least thing’s did not breakdown”? As a widened systems perspective, including the purpose of the facility (the application) increase the windows of opportunity, it is necessary to see

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3 Desktop Retrofit is software intended as an aid to obtain reasonable estimations of expected performance compared to what was the case when R22 was used, and is available in the internet on the web-page of KTH Energy Technology (www.energy.kth.se). Follow the link ‘Research’ and then ‘The phase-out of refrigerant R22 in a systems perspective’.
what can be done outside the liquid chiller or heat pump, or even the refrigerant circuit. This is treated in the end of chapter 3 and in chapter 6; somewhat indirectly in chapter 4.

*Thirdly: How are facilities in the field (real facilities) to be evaluated before and after a refrigerant retrofit? Is there need for new methods or techniques?*

There is a need for new techniques when it comes to analysis of the behaviour of units using zeotropic refrigerant mixtures. Not for the sake of estimating capacity of efficiency, but to understand some of the reported/experienced behaviours by service technicians and users. This is treated in chapter 5. An example of a set of explanatory descriptions for reported behaviour (why R407C seems to be running stable with only a very small degree of superheating) is presented in chapter 6.

**Methodology**

There are several ways of conducting research in this field. One way would of course be to conduct a (large) series of case studies in field and lab: Test retrofitting a number of different units in different applications to different substitute working media. Using this type of approach will result in (at least) one big problem: A problem of generality. Even though rather extensive knowledge on how refrigeration and heat pump equipment are designed, what their contents are; each individual unit as it is being measured have to be considered as a black box as it is subject to measurements after a refrigerant retrofit⁴. If research is to be conducted, and information is to be spread amongst academia, users and service technicians etc, a high degree of generality ought to be desired.

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⁴ Ashby (1964) pp. 86-88, 255
In the previously mentioned engineering licentiate thesis the problem of generality is emphasised by his study of oil contamination and machine failures after retrofitting from (primarily) CFC’s to HFC’s. One result of his study of this is that even though there were over one thousand samples analysed, one conclusion that however could be drawn was that if the retrofitting procedure in general was not good, there was a greater risk of machine failure. On basis of the oil sampling and analysis, one conclusion drawn was that at least ten times more samples of oil had to be taken to, maybe, be able to draw any strong conclusions concerning the metal- and acid contamination and how they might have appeared. (Herbe 1997) So, a high degree of generality concerning the results from the research done can only be obtained using a methodology based on the understanding (systemic, thermodynamic etc) and not purely on statistics.

To investigate issues concerning the replacement of refrigerant R22 in existing units and installations, and making statements on resulting coefficients of performance and capacity of individual units on which refrigerant retrofits has been conducted, contributes very little to increase the understanding of the underlying reasons for the behaviour as the actual unit and installation of which it is a part, essentially are black boxes. Using the case wise (naïve)\(^5\) inductive model to conduct research could probably work if, and only if, there was a high degree of coherence between the cases studied and very large number cases with sufficiently different pre-requisites were studied. Looking at studies done, there is very little coherence: They concern real plants in real operation, so the surrounding conditions are so different even for a single unit that the performance measurements before and after a retrofit are incomparable. So, using (naïve) inductive arguments to claim general results from unique cases does not work particularly well, for once

\(^5\) That is, naïve in the sense without prejudice, or unbiased.
one can always argue that things worked that way in those particular applications and units, but how will they work in other? I.e. how many is a large number?\(^6\)

Another way of approaching the subject is to consider the issue a developing subject as described in figure 1. In reality there are a number of issues and problems that arise due to concerns about ozone depletion and the possible consequences on life as we know it. This has led to the aspiration to reduce – and ultimately put a stop to – the use of ozone depleting substances, such as the CFC’s and HCFC’s. Thus, the Swedish parliament has passed laws on how this should be obtained. One of the HCFC’s that play a crucial role as working media in a great number of applications is R22. In 1998 a new installation stop came into force, and was in 2002 followed by a refill stop for existing facilities. As a consequence of this, a number of existing units and installations will be subject to retrofitting of some sort or another: In some cases refrigerant retrofitting without any further ambitions, in other more elaborate methods to maintain functionality of the installation or activity as a whole.

\(^6\) Cp. Herbe (1997), or the classical example within philosophy of science, where the assumption is that all swans are white since all swans seen by the observer at that time had been white. In Australia there are however black swans. I.e. The hypothesis claimed using naïve inductive arguments is easily falsified with further observation. So, how many is a large number? Even a large number of swans are not all swans, and a large number of oil samples analysed does not necessarily lead to increased understanding of the underlying phenomena of apparent oil induced machine failures.
Figure 1 The growth of a developing subject as described in Checkland (1999) p.8.

But how do you practically conduct an investigation this way? How do you find relevant questions to ask? One way of solving this methodological problem is naturally to conduct some test retrofits in cooperation with actors who have real problems and real
facilities, and real interest in solving their real problems. By letting them ask the questions they like to have answered, and develop (or integrate) theories to further elaborate on the problem formulation, and building models, aimed for simulation as well as explanation. The findings may then be compared to results in the case records from the test retrofits and other peoples work, and fed back to the source of the question: the service technician etc. This way of operating a study of this kind will likely lead to the fact that the problem formulation develops over time: New questions add to the older, as do the explanatory models – the subject develops and grows.

The evident outcome of research using this methodology, the results, will present itself in the form of rules of thumb and other tools for the man in the field as well as the development of new theories and explanations for academia, many of them a result of integration of established knowledge and models from a wider field. Results of the kind

.../changing from R22 to R134a, as working media in a unit, will lead to 40% reduction of cooling capacity/…

will be very sparse except in very specific cases, and as such used mainly for validation of simulation models etc, as absolute numbers are less interesting than the generality and qualitative relevance of the results with this approach. This method in many ways more resembles a classical hypothetic-deductive method but the falsification process is of course inductive in many ways, but not naively inductive. (Hempel 1969; Popper 1989) Then again, this is applied engineering science, not natural science.

Limits and extension

The questions treated in this thesis have all been raised in one form or another by service technicians or people in the “field” on
site, or at seminars and presentations, but elaborated and (hope-
fully) raised to an academic level by the author; Or, from labora-
tory measurements of which results have been made available in
either publications (journals), or in presentation by or communica-
tion with, colleagues at our or other research laboratories in (pri-
marily) Sweden. The focus of the work has been to build explana-
tory models and building understanding. To obtain this, one ambi-
tion has been to integrate models formed with perhaps more of a
reductive character: the sub-models used are more or less well es-
tablished within the paradigm, and sometimes they may even be
considered rough simplifications, but at the systems hierarchical
level explored they may be used within an explanatory model
since it is their behaviour that are interesting and not their very
nature. At some occasions it has however been necessary to inves-
tigate certain phenomena or components deeper. However, the
overarching approach has been to study the issues using a systems
approach, with modelling, simulation, and field and laboratory
measurements where appropriate.

The facilities of interest in the current thesis can be described as
mid-ranged capacity wise: from kW-size in cooling or heating ca-
pacity to, say, MW-size. I.e. not domestic appliances or industrial
sized facilities, but domestic heat pumps, air conditioning chillers,
split-type units, process cooling etc – applications where R22 has
been used extensively in Sweden.

The computer simulations conducted, have all been static. No dy-
namic computer simulations have been conducted what so ever.
Therefore no dynamic phenomena such as hysteresis behaviour of
the circulated compositions in units operating with zeotropic re-
frigerant mixtures have been simulated. However, all field meas-
urements are of course dynamic, as compared to most laboratory
measurements, where static operation is often sought. There is
however one study, where a sort of dynamic simulations has been
conducted: In chapter 6, the simulation of heat pump perform-
ance over an extended period of time, are essentially dynamic simulations of the building. As the heat pump’s time constants are much smaller than the building, they are simulated using a quasi static model based on experimentally measured data.

**How this thesis should, or could, be used**

This is not a reference book, nor a refrigerant retrofit handbook. Neither is it (probably) an ordinary engineering thesis. According to the author’s opinion, the results of the research conducted, and the ones also presented in this thesis, are the general conclusions and explanations that may have an impact on the outcome of a refrigerant retrofit in applications where R22 previously has been used as refrigerant.

In the research foregoing the current thesis, the focus has been to present methods, tools and conceptual models to improve the conditions for successful retrofits with high energy efficiency, such that the user and owner side of equipment and installations may have reasonable support to make sound decisions, and supply aid to service technicians so that they are able to conduct retrofits and reach these goals.

Different parts of the thesis may be of interest to different people, with different interests. In one of the later chapters, a methodology for decision making in retrofit situations is presented. This could very well be used (perhaps slightly adapted) as an aid to facility owners, service technicians and engineering consultants in the process of decision making in the case of retrofitting applications were R22 is used as refrigerant. The thesis can be said to be aimed for engineers at BSc or MSc level, concerning the discussions on perspective and the explanatory models. The question: ”Where is energy efficiency?” of chapter 3 is aimed at these practising engineers, maybe as engineering consultants. It is also aimed at researchers in the refrigeration field. For example in the respect
that it includes discussions on the importance to define the systems boundary of any system studied, and not take ones own paradigmatic extension for given: It is not! I do not think that a service technician, at any greater extent, has any direct use for this thesis. During the progress of the foregoing project this has been covered by the publications in the domestic non-academic refrigeration business publications, seminars and presentations.

**Publications**

This doctoral thesis is not consisting of previously published articles by the author. That does not mean that the author has not been published. Within the project foregoing the current thesis a number of articles and papers have been published – even more has been written but never actually submitted for publication – a summary of the publicised works follows below. Short descriptions of the contents are presented below each item (in italics).

**Conference papers**


Different refrigerant fluids have different throttling losses etc, thus they will be more or less benefited by subcooling. As many of the refrigerant mixtures apt for R22 substitution have more or less considerable temperature glide, the required use of subcooling-area in the condenser will be different…

The contents of this paper are essentially presented in chapter 5 of the current thesis.


As the title of the paper says, it treats the experiences from the Swedish phase-out of R22, especially in existing facilities.

Seminar papers

The computer model for the Purdue-paper above was completely rebuilt, and the following computer simulations were complemented by lab measurements.


This seminar contribution covers the contextual situation and the windows of opportunity for the facility owners to succeed in the phase-out. Focus is placed on one, the interdisciplinary form of the problem context and, two, how the problems context is different from the case of the R12 phase-out and, three, how the structure of the various organisations involved affects this – how companies within various fields and sizes handles this problem.

This paper covers essentially everything treated within the current thesis – somewhat condensed however, as it is only four pages long.

**Scientific journals**


The contents of this article are essentially presented in chapter 5 of the current thesis.

**Refrigeration industry publications (domestic)**

One way of providing feedback to the Swedish refrigeration and heat pump industry has been to publish findings from the research done in one of the most widely spread refrigeration business magazines. On top of this a number of presentations to various business organisations have been made over the years.


Would it be possible to phase out the HFC-media technically? What are the conditions for doing this with today’s technology?


The article treats the green house issues concerning the possible future phase out of HFC-media. Green house arguments have been used in the debate and these are questioned in this article.

As the use of zeotropic refrigerant mixtures caused, and still causes, puzzle-ment articles where published to straighten things out. As these articles only seemed to make things even more confusing and also contained faults, this article was published to set things straight without making them easier than they are, nor simplifying them as had been common.


This is a summary of the findings from the foregoing research project (see below). The article was aimed to provide especially service technicians with tools, and make them think about the installation instead of just the vapour compression unit. It also treated the “it depends” issue.

**Software**


Desktop Retrofit is computer software for making estimates of the performance of a unit as it is retrofitted from R22 to some substitute. It is written in, and compiled with, EES (Engineering Equation Solver). (Klein 1991-2003)


**Reports**


*This is the final report to the Swedish Energy Administration from the research program “Klimat 21” (Climate 21). It sums up and presents the conclusions from the preceding project.*


*A report written for the Swedish Environmental Protection Agency (S-EPA) as the refill stop for R22 were about to come into force: Aimed for the owners of equipment that could be subject to the phase-out of R22 in existing facilities.*


*Some of the experiences (also presented in the current thesis) lead to the S-EPA asking the author to apply for a research grant, to be used to investigate the possibilities of implementing some form of environmental and energy efficiency index of heat pump and refrigeration installations. In the report of the*
study both the “problem systems” as well as “problem solving system” are treated and elaborated upon.\(^7\)

**Publications which the author of the thesis has contributed to**


**The contents in brief**

**Chapter 2 – A history of phase outs**

Chapter 2 contains a resume on the historical and environmental context of this field of research. The last part of it may seem slightly short with source references, but is essentially based on the author’s personal experience from being a part of the process, taking part in meetings etc – at least in the periphery.

**Chapter 3 – Systems thinking for retrofitting**

Any analysis of energy efficiency or capacity requires a system and a systems boundary; heat and work only exists at a systems

\(^7\) Cp. both Checkland (1999) and Checkland’s thesis in Miser and Quade (1985), especially pp.153, 163.
boundary. Since most systems definitions are paradigmatic and never stated, confusion might arise when the models are scrutinised. This is the reason for including a chapter about the systems and models of concern. In chapter 3 the concept of parasitic and structural losses is also introduced, and some consequences thereof will be discussed.

**Chapter 4 – Experiences: field, lab and simulation**

Chapter 4 treats both measurements made in the laboratory at KTH Energy Technology and field measurements (measurements made on real plants with the service technicians standard equipment), as well as simulations and the conceptual models behind the computer simulation models. More focus is, as mentioned earlier, on the conceptual models than the actual numerical models. As measurements also are based on models, these models are also presented – briefly though.

In chapter 4 the questions that raised the need for a computer simulation model with relatively high conceptual resolution are treated, not only from a modelling point of view but also the outcome and verification of the simulation output data and results. The conceptual design of the simulation model is described and its generality and general usability to create explanatory models is also treated.

In the last part of chapter 4, the formation of predictive models is treated, both concerning conceptual models and the consequences of various systems extensions, as well as the conceptual formation of a tool for estimation of the performance of a vapour compression refrigeration unit after a refrigerant retrofit to any chosen substitute. The input data for the tool are the same measured data as the *ETM refrigerant computer* uses – a common tool amongst service technicians in Sweden. (A description of the function of the *ETM refrigerant computer* is also presented in chapter 4.)
Chapter 5 – Measurement and analysis

When test retrofits of refrigerating machinery to zeotropic refrigerant mixtures like R407C the composition may be different than the composition stated by the manufacturer and the ASHRAE classification. This means that conclusions cannot be drawn, if not these phenomena are not considered. This is something that is seldom done in case studies reported in the literature.

So, in chapter 5 consequences of component fractioning on measurement data are discussed. The issues are exemplified with both experiences – personal as well as questions asked by mainly service technicians, manufacturers etc – and examples found in the literature. Further, two methods to detect composition shifts, is presented also: One for online measurement and another indirect method.

Chapter 6 – Retrofitting R22-applications

In chapter 6 the aim is to provide tools for decision making. Reflections on the concept of energy efficiency are also presented.

Chapter 7 – Conclusions

In chapter 7 concluding remarks of the thesis can be found, as well as suggestions for future work. Thoughts on the future of mechanical refrigeration and the working media used, as well as the connections to global warming are also discussed.

Appendices

In Appendix A a nomenclature list ais found.

In Appendix B some of the correlations used in the computer simulation models are presented, as they are less important for the sake of argumentation but may be of interest to some readers.
References

At the end of the thesis two reference lists are found: Firstly a list of sources referred to within the thesis, and secondly a complete bibliography with all my sources – some not referred to within the thesis, but they have all contributed to the problem solving in some way or another.
A history of phase outs

The unreliability of scientific predictions was, in fact, a consequence of the growing awareness of the complexity of the issue. The ozone-destruction theory of Molina and Rowland however, remained intact.8

During the history of artificial refrigeration a number of refrigerant phase-outs have taken place. Thus the preferred working media in compressor driven refrigerating machines have changed over time. Usually because the “old” chemicals proved toxic, were flammable, caused nausea, or the fact that the “new” media were experienced as having better thermo-physical properties than the old. I.e. the aims of the earlier refrigerant phase-outs were either to improve working conditions for the operations personnel, reduce the risks connected with operation caused by the flammability of certain refrigerants, etc. In the end, most of the efforts were aimed at helping artificial refrigeration reach the home appliance market. In the late nineteen twenties this was a very big issue! General Motors was at this time an enormously large corporation with equally enormously large funds waiting to be invested. GM as well as others, argued that mechanical refrigeration would never reach the home appliance market if it wasn’t made safe. All the refrigerants used were either toxic or flammable, sometimes both, and this was pinpointed as the main hurdle to be cleared. In 1928 Thomas Midgley and a team within the Frigidaire division of General Motors were appointed to perform a screening of potential chemical compounds; compounds that could be used as re-

8 Peter Usher of the United Nation Environmental Program as quoted in Nolin (1995).
frigerants in domestic appliances. A number of targets were set up, of which low (or no) toxicity and normal boiling point between -40°C and 0°C, were two.

In 1930 Midgley held a famous demonstration to the American Chemical Society proving the harmlessness of the new refrigerant developed and patented: Dichlorodifluoro-methane or R12. This category of compounds, the CFCs, had actually been synthesised by Belgian chemist Swartz already at the turn of the last century. With the perspective of the 1930s, the CFCs seemed the ultimate refrigerants. They were non-toxic, non-flammable, functioned well with the available lubricants, showed good thermo-physical properties, and they were chemically stable. The stability of the CFCs was to become what caused the next major refrigerant phase out.

Compared to the old refrigerants (ammonia, sulphur dioxide, dimethyl-ether, carbon dioxide and various hydrocarbons) the CFCs, and later the HCFCs, seemed perfect. Other compounds were investigated, e.g. HFCs, but since the CFCs and HCFCs were considered supreme, no interest were shown to these alternatives. Not in the next fifty years...

The DuPont Corporation was invited to help commercialise the CFCs, initially mainly R12, as a shareowner of Frigidaire. The CFC refrigerants, and eventually the HCFC refrigerants, were branded Freon; a brand name that soon became the synonym for refrigerant. Later, General Motors left the partnership, and Frigidaire became a division of DuPont de Nemur Corporation. DuPont is still one of the largest manufacturers of halocarbon refrigerants.

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9 Professor Matts Bäckström even insisted on calling this group of fluids “Swartzians”, or “Swartzaner” in Swedish! E.g. Sw-12 (R12) and Sw-22 (R22).
During the 1960s, the development of supersonic high altitude civil aviation began on both sides of the iron curtain. Supersonic aviation was at that time thought to be the future way to travel long and medium distances. Environmentalists and other critics of the concept argued that extensive high altitude flight was likely to cause damage to the ozone layer, leading to serious consequences to life on earth. Eventually virtually all of the supersonic high altitude aviation projects were cancelled, but the issue of an easily damaged ozone layer was raised (Nolin 1995). In 1974 chemists Rowland and Molina showed how certain man made chlorine containing compounds could act as catalysts and break down the stratospheric ozone layer. The incoming ultra violet rays from the sun could break down the molecules of the chlorinated hydrocarbons; release the chlorine, which in turn would act as catalyst in breaking down the ozone (O₃) to ordinary oxygen gas (O₂)(Molina and Rowland 1974). Some of the man made chlorinated hydrocarbons of concern was the Freons. Still, at that time no one had shown how the heavy Freon molecules were actually able to reach the stratospheric ozone layer¹⁰. Critics argued that for example volcanoes would be a much larger source of atmospheric chlorine, and that no depletion had been detected. Soon however, Dutch meteorologist Paul Crutzen presented a model that could show how this was actually possible. Still, the critics argued, there was no sign of depletion of the stratospheric ozone layer, and governments ought to take caution before running of and legislating on the ban of the use of CFCs and HCFCs. The public debate started, affecting politicians and policy makers. The critics were troubled. In 1981, the president of the General Conference of the International Institute of Refrigeration (IIR), Professor Gustav Lorentzen, released an official statement of the IIR claiming that:

¹⁰ E.g. R12 has a molecular weight of 120 kg/kmol, whereas the average molecular weight of air is only 29 kg/kmol. According to the critics, the Freons would not be able to ascend through the atmosphere and reach the stratospheric ozone layer.
The International Institute of Refrigeration has been informed that some governments are considering introducing legislation to prohibit the use of chlorofluorocarbons (CFC) in all forms of refrigerating equipment. /…/ There are very considerable uncertainties in the calculations /and/ natural fluctuation of the Ozone concentration is far greater than the expected effect of CFC. /…/

CFC refrigerants are used extensively /and/ No satisfactory substitutes are available since all known alternatives are either toxic, inflammable or have other undesirable properties. A ban of CFC may well result in greater hazards than those associated with the ozone effect. A change will also involve enormous sums of money, indeed many billion dollars, in transforming and replacing existing equipment and installations. /…/

In consideration of the above, the International Institute of Refrigeration strongly advises governments not to introduce any drastic restrictions on the use of CFC refrigerants until the need for such measures has been definitely established. /…/ In the meantime measures can be taken to encourage the use of the potentially less harmful types of CFC, such as R22, and to reduce the leakage to the atmosphere of such media. (Lorentzen 1981)

This view was not shared by the Swedish government and a sharp letter and note was sent to the Swedish commissioner in the executive committee of the IIR, Eric Granryd (Hägerhäll 1981).11

Despite what was claimed in official statements, there was an available substitute for R12: R134a. It was known as a chemical compound since the thirties and its thermo-physical properties

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11 It should be noted that Professors Lorentzen and Granryd later became fore runners in the use of hydrocarbon and carbon dioxide as refrigerants.
surprisingly well described, but since the CFC’s were available, there was no use for it or other HFC’s. Therefore, there were no production of them either and therefore they were not commercially available. R134a has thermo-physical properties so close to those of R12 that you are able to change the working media of the refrigeration machine or heat pump and do some minor operations to make it perform as well as it had done with R12. The problems that did occur were mainly connected to the new type of lubricants needed and materials compatibility (Herbe 1997).

The British Halley Bay station in Antarctica observed a serious depletion of the ozone layer over the station, from 1982 and on. They were not entirely certain their measurements were correct, since their equipment was since 1957. Hence, they decided to upgrade it. NASA however took the Halley Bay report seriously, and both re-evaluated data from their Nimbus 7 satellite from earlier years and conducted new measurements. The amount of raw data sent from the satellite was large, and low values in the ozone readings had been assumed just poor measurements. After all, the Nimbus 7 was originally not designed to measure ozone levels. In 1986 a report on the measurements from NASA was published in Nature. The measurements showed on a depletion of the stratospheric ozone layer over Antarctica so far gone that a hole had actually formed. There were no longer any doubts about ozone depletion; the mechanisms behind atmospheric transport of heavy molecules such as CFCs, and their catalytic effect as ozone depleting substances seemed a fact, and the critics soon found them selves disregarded. (Nolin 1995)

In 1987, a number of the world’s countries signed the Montreal Protocol, suggesting a phase out of ozone depleting substances. Amendments were made to the Montreal Protocol in London, Copenhagen and in Vienna. Suggested phase out schedule was sped up with each amendment. In the London amendment, a
resolution on reduction on the much less harmful HCFC’s was included, with dates and quotas set in the Copenhagen.

<table>
<thead>
<tr>
<th>Type of refrigerant</th>
<th>Montreal 1987</th>
<th>London 1990</th>
<th>Copenhagen 1992</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFC</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>-20% 1993</td>
<td>-50% 1995</td>
<td>-75% 1994</td>
</tr>
<tr>
<td></td>
<td>-50% 1998</td>
<td>-85% 1997</td>
<td>-100% 1996</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HCFC</td>
<td></td>
<td></td>
<td>Resolution on reduction</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-35% 2000</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-65% 2010</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-90% 2015</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-99.5% 2020</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-100% 2030</td>
</tr>
</tbody>
</table>

Table 1 Dates for reduction in global production of CFCs and HCFCs according to the Montreal protocol and some of its amendments. (S-EPA 1994)

<table>
<thead>
<tr>
<th>ASHRAE number</th>
<th>Type of refrigerant</th>
<th>New install stop</th>
<th>Refill stop</th>
<th>Stop for use</th>
</tr>
</thead>
<tbody>
<tr>
<td>R22</td>
<td>HCFC</td>
<td>1998</td>
<td>2002</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 2 Phase out deadlines in Sweden. The stop for use is only valid for systems with charges larger than 900 g. (S-EPA 1995b)

The Swedish legislators set out a CFC phase-out schedule with a comparatively seen high pace. The schedule and legislation included not only new installation stops, but also refill stop for existing equipment and even a stop for use datum. Later a HCFC phase out schedule was set up, suggesting not only new installation stop but as with the CFC’s, a refill stop. The schedule for HCFC’s however does not contain any end of use date. This would imply that in Sweden existing machines would have to have
the present working media replaced with something else. I.e. existing installations would have to be retrofitted and use working media they were not originally designed for.

Phasing out R12 (and R500) first seemed very troublesome, since, as critics had argued, there was no commercially available substitute. Which of course was self-evident: Why would any one mass-produce a chemical no one asked for? However, there was a substitute available – not in commercial production at the time however: R134a, or 1,1,1,2-tetrafluoro-ethane. There were a range problems with R134a and the traditional materials and lubricants used in refrigeration equipment. R134a is a HFC-refrigerant, and thus not compatible with e.g. the mineral oils traditionally used with CFC-refrigerants. There were other problems concerning different polymers and rubbers used. The lubrication problem was solved (chemically) by changing the traditional mineral oils to ester oils. The use of polyol-ester oils (POE) in turn gave rise to other materials compatibility problems that had to be solved. In new designs and new equipment, the manufacturers of chillers, mobile air conditioning and domestic appliances could address this problem as any engineering problem. In existing installations, these could however cause problems. (Herbe 1997) I.e. refrigerant retrofitting existing plants from R12 to R134a could cause problems with extended ware on moving and non-moving parts etc. Thermo-physically, however, R134a is so similar to R12 (and R500) that a change of refrigerant can be conducted without the plant (or unit) changing capacity, efficiency or general operating behaviour, in any great extent. When the materials compatibility problems had been identified and measures had been taken, what remained to the service and maintenance personnel conducting the refrigerant retrofit was to adopt by applying new procedures. Procedures such as cleaning the system after removing the old re-

frigerant and lubricant, keeping cans with ester oils closed and not exposed to the moisture of the ambient air etc. (Herbe 1997) Evidently at least some service technicians underestimated the hygroscopic abilities of the POE-lubricants, since many did not use protective gloves initially had their hands exposed to it, with skin damage as a consequence. (Arfvidson 1999)

In other parts of the world were phasing out HCFC was still not an issue in a foreseeable future, various blends containing R22 was marketed for drop in use; blends such as R401A. These never gained any noticeable market share in Sweden.

R12 was not the only CFC with a considerable market share used in refrigeration. R502, a blend of R22 and R115, was a common working media especially in commercial freezing and similar applications. Thus R502 was used a lot in supermarket refrigeration: Systems with large system charges. Soon a substitute for R502 was marketed: R404A. As R134a compared to R12, R404A has thermo-physical properties so close to those of R502 that to service and maintenance personnel, replacing R502 with R404A in existing equipment was reduced to a question of retrofit procedures. (Herbe 1997) As was the case of R12 refrigerant retrofitting, there was essentially no need for them to increase their knowledge in thermodynamic behaviour of refrigeration working media. I.e. replacing CFCs in existing equipment was technically possible and could be conducted by the available personnel as soon as they had adapted to the new procedures needed.

What separated R502 from R12 was the fact that R12 was in Sweden, and in many other European countries, mainly used as working media in household appliances and similar short lived standardised products, or in very large industrial size plants such as district heating heat pumps. Phasing out R12 was thus in many respects a “new-design” problem. Problems were solved within mainly large multinational corporations, such as Electrolux,
Bosch, and Whirlpool. Automotive air conditioning units are commonly not manufactured by the individual car manufacturers, but by a few subcontractors, like Delphi in Luxemburg. These did supply the car-dealers service organisations with exact guidelines for how to conduct retrofits of every unique type of unit. Large district heat pumps on the other hand, were dealt with ad hoc solutions and adopted and retrofitted to R134a by highly qualified personnel, often from, or assigned by, the original plant manufacturer (large corporations such as ABB). I.e. household appliances, automotive air conditioning units and industrial size plants where dealt with by organisations and corporations with a high degree of engineering knowledge.

What separates household appliances from most commercial equipment is that they are sold to private persons who happen to also be the end users. Industrial sized plants as district heating heat pumps are sold to companies who might see themselves as the end users of the service supplied by the heat pump: heated water to be circulated in their district heating circuit. The products or services are quite clearly defined. Who the end user of commercial equipment is, is not always clear. Often it is a question of definition; i.e. how you define what the product or service is: suitable working conditions, fresh tomatoes in the grocery store, a piece of equipment such as a refrigerator or a liquid chiller. Further, phasing out CFCs in existing equipment was something going on in many countries all over the world.

Soon after the phase out of CFCs in Sweden the time had come for phasing out HCFCs, mainly R22, using basically the same plan as for CFCs. What separates the Swedish legislation from the phase out legislation in most other countries is that in Sweden, existing plants would have their working media changed. Most other countries choose another path, allowing R22 to be used in existing machines during their technical or economical life span. In many other countries with a HCFC phase out plan, you are allowed to
refill your existing R22 units with recycled R22; where as no new installations are allowed after a certain date. I.e. no new HCFC units, but you are allowed to re-charge your existing machines with used R22. According to Swedish legislation, there is a refill stop from January first in 2002, and the new installation stop has been effective since January first in 1998 (S-EPA 1995b). Effectively the refill stop is an end-of-use date for equipment which is leaking — *do something about the leaks, or change working media*. Further, in most countries, phasing out HCFCs is mainly new design problem. At a conference at the Royal Institute of Technology, a representative for EPEE\(^{13}\) even claimed that the R22 phase-out was finished — no *new* units with R22 was produced for the European market any more. The perspective is thus “a new design phase-out”. In Sweden it is in many respects a retrofit problem. This makes the Swedish situation slightly different from the situation in essentially the rest of the world. The present thesis to a great deal presents problems and issues unique for the Swedish situation.

<table>
<thead>
<tr>
<th>Type of refrigerant</th>
<th>Usage 1988 [metric tons]</th>
<th>Usage 1994 [metric tons]</th>
<th>Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFC</td>
<td>4620</td>
<td>280</td>
<td>-94%</td>
</tr>
<tr>
<td>HCFC</td>
<td>910</td>
<td>300</td>
<td>-67%</td>
</tr>
</tbody>
</table>

*Table 3* The usage (in any application) of HCFCs decreased considerably between 1988 and 1994. Until 1993 the usage of HCFCs, however, increased in plastics manufacturing. (S-EPA 1995b)

The phase out of the HCFCs may have seemed to start off long before the phase out schedule stated (S-EPA 1995a). If you only considered reported tonnage, that is. The number of units may have increased however — units with smaller charges than before, many falling outside compulsory report limit of a minimum of 10kg total systems charge in a unit.

\(^{13}\) European Partnership for Energy and the Environment, an interest organisation for a number of refrigeration equipment manufacturers.
As for the CFC phase out – mainly R12 and R502 – there had been available substitute refrigerants. Substitutes with thermophysical properties so similar to those of the original working media, that you could exchange the existing working media with something else without significantly loosing capacity, or efficiency. For the HCFC’s – mainly R22 – it might have seemed as if though there was a substitute at hand: R407C. R407C, a blend of refrigerants, presented volumetric cooling capacity and a vapour pressure curve close to that of R22. In theory, many argued that this would be a suitable substitute, and even to use as a drop-in in existing R22-machinery: The supporters argued that, as the case had been with e.g. R12 and R134a, you therefore should be able to use R407C in R22-equipment without loosing either capacity or efficiency significantly.

![Figure 2](image)

**Figure 2** Number of units with a larger total systems charge than 10kg’s reported to the Swedish Environmental Protection Agency (S-EPA). No data is available from years before 1995 in S-EPA’s statistics. (S-EPA 2001)
Figure 3 Total systems charge in units with a larger system charge than 10kg’s. When the refill-stop for CFCs came into force in 1995, there was a small increase in HCFC until 1997. (S-EPA 2001)

Figure 4 Service re-charging in units larger than 10kg’s of total systems charge. (S-EPA 2001)
Figure 5: The annual world production of refrigerants R134a, R12 and R22. During the 1960's the world production increase tremendously. (AFEAS 2002)

Figure 6: Estimated annual leakage from refrigeration and air conditioning equipment worldwide (after manufacturing year). (Calm 2002)

In the service and maintenance community, replacing R22 was not popular. Many in the service and maintenance, as well as facility owner, communities did not see a R22 phase-out as something important. There were also many rumours of problems and strange, unexplainable, behaviour and break down of units that
had been retrofitted. Still, they were mainly rumours since very few refrigerant retrofits had actually been conducted. In the autumn of 1997, there even was a hearing at the Swedish Environmental Protection Agency (S-EPA)\textsuperscript{14}, attended by the author of the present thesis, where fears and critique on the Swedish legislation with an upcoming new installation and refill stop, was ventilated. Arguments that were expressed concerned the “non existence” of available equipment in certain niches of applications\textsuperscript{15}. That some importers even would go bankrupt. Other arguments that were pronounced concerned the poor performance of the available substitutes, and even that “professors at the Royal Institute of Technology had said that R407C was a bad refrigerant and that phasing out R22 was down right stupid”. Some of the arguments were valid: Some at the time hard-to-explain behaviours were experienced (Herbe 1997). But since most refrigerant retrofits that had been conducted had been so in a laboratory environment or had been test retrofits made in connection to universities such as the Royal Institute of Technology in Stockholm or Chalmers University of Technology in Gothenburg, the experience within the refrigeration community – service and maintenance personnel and firms – itself was marginal.

Despite, or because, the issues mentioned above, R407C was marketed in Sweden by the large retailers as the R22 replacement that would work as well as R134a had worked in R12 units. –It did not. Certain peculiarities connected to the zeotropicity of R407C made it a less suitable choice for many types of applications and circuitry designs. Unfortunately, these designs and applications are not that uncommon in the Swedish refrigeration and heat pump

\textsuperscript{14} Naturvårdsverket

\textsuperscript{15} The Swedish market for e.g. split type air conditioners is extremely small to the large multinational manufacturers of this type of units. They would hardly care about a Swedish new installation stop.
fauna. At the time, about 1997, a large portion of the Swedish refrigeration community waited for the ultimate R22 substitute; the substitute that would work as well in existing R22 machinery as R134a had worked in R12 machines. I.e. they waited for the expertise at e.g. DuPont to solve their problem, as they believed “they had solved it the last time; they only had to come up with a substitute as similar to R22 as R134a had been to R12, and as simple to use”\textsuperscript{16}.

At about the same time another refrigerant blend was beginning to be tested in the United Kingdom by e.g. Star Refrigeration and in Sweden by e.g. the telecom company TELIA. The blend was branded Isceon59, and had been designed with a slightly different design approach than R407C. Substituting R22 with R407C in an exiting liquid chiller for example, would involve cleaning the system in the same way as was the case with R134a in R12 units. With Isceon59, the idea was that you could use it as an actual drop-in replacement. The existing lubricant worked with it, which implied that there was no need for cleaning the circuitry. Initially Isceon59 contained 4\%, by mass, of iso-butane. The purpose of the iso-butane was to reduce the viscosity of the oil, by dissolving itself in it and thereby help the exiting oil to follow the refrigerant flow around in the circuitry back to the compressor. At least that was the general idea. Later, the iso-butane was exchanged with n-butane, and the amount reduced to 3.4\%, by mass. The reason for this was troubles for the manufacturer with obtaining an A1 ASHRAE classification. Critics argued that by doing worst case

\textsuperscript{16} This Weltanschauung can be seen as an indication of the lack of engineering knowledge within certain parts of the Swedish refrigeration community; a very small amount of people actually had/have adequate knowledge concerning refrigerants and conceptual good knowledge about why refrigeration and heat pumping machines behave as they do. There is however a very large amount of tacit knowledge; and the dependency on tacit knowledge has actually had a hampering effect on the ability to deal with the problems that has occurred in actual retrofit situations. Cp. e.g. Checkland (1999), Henning (1998) and Östberg (1998), (2000).
fractioning studies, they could show that it in fact was inflammable, and by changing the composition slightly, the manufacturer managed to make it non-flammable\textsuperscript{17}. With the new composition Isceon59 obtained its ASHRAE classification and its ASHRAE number, R417A. Actually this process took about three years, and the formal classification was not ready until the autumn of the year 2000.

Once the ASHRAE 34 classification was through, R417A was accepted by some within the Swedish refrigeration community as the ultimate R22 substitute to be used in existing machinery. A few members however were concerned about oil return and similar questions. One interesting fact is that even though R417A also is a zeotropic refrigerant mixture, no one seemed to worry about that. The fact that R407C is a zeotropic blend was one of the main arguments used by some why they thought it should not be used. R417A being zeotropic was however not an issue as it seemed. Further, it got a rumour of running stable in existing R22 machinery with a very low superheat, just as R407C had a reputation of running stable with a seemingly small superheat but also for showing bubbles in the liquid line sight glass even though the subcooling seemed adequate. There is an answer to why it appears to be this way (as will be shown later in the fore lying thesis), but to the service technicians and maintenance personnel doing the actual retrofitting in the field these experiences and hearing the rumours zeotropic refrigerants were mysterious. They do/did not have the thermodynamic knowledge to be able to analyse their own as well as others (alleged) experiences etc. The business dependency on tacit knowledge – people “know” how things works, but they do not know “why” – becomes a problem when new refrigerants show other sorts of behaviour than the old ones did, for

\textsuperscript{17} An A1 ASHRAE-34 classification means that the fluid is non-toxic and non-flammable in a worst case fractioning scenario.
instance. If you do not understand how zeotropic refrigerant mix-
tures behave, it is of course very difficult to understand why a
plant behaves the way it does (why there are bubbles in the liquid
line sight glass even though there seems to be a fairly large
amount of sub-cooling, etc.) and how to deal with it. To many in
the refrigeration community, refrigerants and refrigerant mixtures
still is quite mysterious; making them ideal targets for marketing
of new blends with equally “bad” behaviour as other refuted ones.
This becomes particularly evident in times when things go wrong.

In a report from Swedish insurance company Folksam, a list of all
reported failures and damages on heat pumps in Sweden during
the years 1999 to 2001 were reported, covering all insurance com-
panies in the country. (Folksam 2003) For example the most fail-
ure inflicted make of air-to-air heat pumps, was imported from
the manufacturer as a R22-unit, and had its refrigerant substituted
for R407C in the importing firm’s warehouse. (R407C happened
to be the working media in all of the most failure inclined units,
except one where propane had been used.) The reaction from
many in the Swedish refrigeration and heat pump community has
been that R407C is a bad refrigerant. Obviously this is not the
case and a poor excuse. Looking at the design of the air-to-air unit
mentioned above, using R407C as working media was a bad
choice for that application (with high condensation and low
evaporation) and the components used within the unit: rotary
compressor, heat exchanger configuration etc.
.../there is a fundamental limitation of any modelling of a system, /.../ the system is always embedded in a larger system.\(^{18}\)

– One day during the late winter of 1999 Kenneth Weber, from ETM Kylteknik, called me on the phone. He had a question concerning the measurements he had made on a large heat pump (with uncommonly small heat exchangers for a heat pump) we had done a refrigerant retrofit from R22 to R134a on. As he was about to equip the machine with a frequency controller (as a part of his commitment to the research project), he wanted to understand why the heating capacity and efficiency (COP) of the machine had not gone down as much as one “would” expect. “Making predictions using property plots” he said, “one would expect the capacity reduction to be slightly larger than what I measured”.\(^{19}\)

Any analysis of energy efficiency or capacity requires a system and a systems boundary; heat and work only exists at a systems boundary. Since most systems definitions are paradigmatic and never stated, confusion might arise when the models are scrutinised. This is the reason for including a chapter about the systems and models of concern. In this chapter the concept of parasitic and structural losses is introduced, and some consequences there of will be discussed.

\(^{18}\) Churchman (1968) p.75

\(^{19}\) Weber (1997-2003)
Systems and models

There are several definitions as to what constitutes a system. One definition, by Kotas, is that…

A system is an identifiable collection of matter whose behaviour is the subject of study. For identification, the system is enclosed by a system boundary, which may be purely imaginary or may coincide with a real boundary.\(^20\)

Another definition from social science (operational and management research to be precise), by Churchman, is that…

…a system is a set of parts coordinated to accomplish a set of goals.\(^21\)

The entities in the surrounding may affect the units within the system, but the system itself cannot affect the surrounding.\(^22\) Churchman’s definition is focused on the purpose of the system: Maintaining agreeable climatic working conditions in an office, maintaining adequate temperatures in a display case in a supermarket so that the quality of the products is not compromised. Consequently, how these goals are achieved in detail, is not interesting: Cooling does not have to be achieved using mechanical refrigeration.

Kotas’ definition has its origin in thermodynamics, and according to his definition a system does not need a purpose. This may be exemplified by the general heat engine commonly used within the education of engineering thermodynamics. The general heat

\(^{20}\) Kotas (1995) p.1
\(^{21}\) Churchman (1968) p.29
\(^{22}\) Cp. von Bertalanffy (1968)
engine is a simple model of a machine working in a system producing work by the transfer of heat between two temperature levels. It takes heat from the high temperature reservoir and dumps it at the low temperature reservoir. During this operation work may be extracted (cp figure 7). How the heat engine actually works is not considered within this model. So, in this example the systems boundary is situated between the reservoirs and the “heat exchangers”. The heat transport from high to low does not affect the temperature levels of the reservoirs, but the temperature levels affect the operation of the general heat engine. The locus of the systems boundary will govern not only the spatial extension of the system, but also the temporal. As the systems boundary is subject to an increase in its spatial extension, so is the temporal extension. I.e. the required time frame is likely to become longer as the studied system becomes larger and perhaps more complex – other things becomes interesting (Churchman 1968).

Just as the extension of the systems boundary “decides” what is perceived as interesting; what is perceived as interesting will analogously decide the extension of the systems boundary of the modelled system. What is perceived as interesting depends on the background and aspirations, the Weltanschauung, of the viewer, and may partly be described as paradigmatic. What is considered to constitute the system is evidently paradigmatic and therefore it is an evident risk that actors, phenomena and issues of importance are omitted: What constitutes the parts of the system; what constitutes its environment; who is the actor who can influence the contents or operation of the system? The answers to these questions are not self evident. (Checkland 1999; Churchman 1968; Miser and Quade 1985)

So, various observers see different systems both concerning the extension and context as well as purpose of the systems of interests, and different systems approaches will lead to different conclusions concerning the “best choices and solutions”, different predictions of what performance to expect from a unit that is about to be subject to a refrigerant retrofit etc. In the following sections of this chapter these system definitions and the consequences on predictions will be discussed. The system extensions and models discussed are (1) the refrigerant cycle, (2) the five components of the cycle, (3) the integration of the five components and their function into a vapour compression machine, (4) the installation of which the heat pumping machine is a part, and finally, (5) reflections on the “system with a purpose” using something resembling Churchman’s definition.

What separates a model from reality is thus the fact that the model only describes certain features of the considered real object or phenomenon\(^{24}\). In the example above a lot of the phenomena and principles that actually make a heat engine operational are excluded from the model. The need for a working media, heat exchangers etc are neglected, and in reality it is quite likely that the heat reservoirs are affected by the heat engine operation: The high temperature reservoir will probably cool down and the low temperature reservoir will quite likely experience a temperature increase over time. Therefore, a model omits certain features of the reality of the modelled object or phenomena, but a model is also likely to describe phenomena not easily perceived in the real world.

---

\(^{24}\) A plastic kit model only focus on shape and perhaps colour of the aircraft, it does not take e.g. the avionics or dynamics of the engine into account, and neither does it take its role in the air force the original perhaps is a part of into account. In a sense this is actually the role of a photo model: he or she only represents a simplified description of a human being, mainly focusing on shape and perhaps colour.
The *systems* make out a certain class of models, and it is important to remember that “modelling is not systems analysis”\(^{25}\) – rather, systems analysis is one form of modelling.

**Example of a model common in thermodynamics**

One example of a model commonly used within the engineering thermodynamics is the general heat engine: In the 19\(^{th}\) century Sadi Carnot describes a general heat engine, cp figure 7. He concludes that maximum (theoretical) work that can be obtained from any heat engine is a heat engine where the working media is working between two constant temperatures, \(T_1\) and \(T_2\). Figure 7 shows an abstraction of a general heat engine working between to reservoirs with constant temperatures. As the heat, \(Q_1\), is allowed to transfer from the higher temperature, \(T_1\), to the reservoir with a lower temperature, \(T_2\), we are able to obtain work, \(E\). According to the first law of thermodynamics, heat and work are different types of energy, and since energy is indestructible heat and work are interchangeable. We conclude that:

\[
Q_1 - Q_2 = E
\]

Eq. 1

I.e. if we in the process of moving heat between to constant temperatures obtain the amount \(E\) of work, the heat delivered to the low temperature reservoir, \(Q_2\), will be reduced with \(E\) compared to the heat transferred from the high temperature reservoir, \(Q_1\).

Actually, to come to this conclusion another entity, not known to Sadi Carnot, had to be invented: Entropy, \(S\). In this case it works as proportionality factor between the temperature and the heat, but is an effect of the second law of thermodynamics.

\(^{25}\) Miser and Quade (1985) p.22
In the days of Sadi Carnot, the first law of thermodynamics still had to be defined. I.e. in those days one “knew” that work and heat were not interchangeable – they were two entirely different phenomena. Further, it was “common knowledge” that caloric was the media that carried heat. To meet the continuity equation he came to the conclusion that:

$$Q_1 - Q_2 = 0$$  \hspace{1cm} \text{Eq. 3}$$

…which we of course now know is wrong. He however, had to come to this conclusion, to fulfil the continuity equation – all caloric leaving the high temperature reservoir had to transfer to the low temperature reservoir. But, his model worked and fulfilled the continuity equation. Better ways to measure temperature and heat transfer developed in the 19th century brought us to equation 1. I.e. a model is not necessarily correct just because it seems to work.

---

Figure 7 A schematic of the ideal general heat engine. The systems boundary is shown as a dotted frame.$^{26}$

---

$^{26}$ It is from this model, the definitions in thermodynamics concerning the positive directions of work and heat have their origin: We are interested in producing work, i.e. work has its positive sign out from the system. In the steam engines, which originally
The refrigerant cycle

Another model commonly used is the vapour compression refrigeration cycle. The basic vapour compression heat pumping cycle, or reversed Rankine cycle, is often described as only the working media cycle: The other (at least) four components and supplementary piping are actually not included, only the effect of them. (Cp. figure 9) This systems description may be referred to as the vapour compression cycle, but the refrigerant cycle is a more descriptive name, as mentioned above, the interface components of the machine are not modelled particularly elaborately.

The system of concern is a closed system, according to classical thermodynamics, and is constituted by four separate open sub-systems: The subsystem where the refrigerant evaporation and heat absorption takes place, the evaporation system. The subsystem in which the refrigerant vapour has its pressure and temperature increased, the compression system; the subsystem in which the refrigerant vapour condensates and dissipates heat, the condensation sub-system; end finally the subsystem in which an adiabatic expansion takes place and the pressure of the refrigerant liquid is reduced, the expansion. In this systems model the cycle commonly operates between fixed temperatures – fixed evaporation and condensation temperatures – and it is usually the first description that students are approached with in engineering thermodynamics classes. So, what “decides” the evaporation and condensation temperatures, is not a part of the system but of the surrounding – what is not affected by the system is not a part of the system, this will be the consequence, how odd it ever may seem.

were the source of this idea, heat was supplied to produce work. Thus the positive direction for heat is in to the system.

Note that it says evaporation, compression and condensation, not evaporator, compressor and condenser.
The compression subsystem is modelled in a similar manner: The compression in itself is usually not subject to any elaborate modelling. Either it will be assumed to be ideal and free of losses or the losses will be described by an isentropic efficiency. I.e. the losses themselves are not considered, rather bundled and hidden in a constant (constant isentropic efficiency), \( \eta_{is} = \text{const} \). If the compressor behaviour is to be modelled in a more elaborate way, the compressor would be a part of the system, but not e.g. the evaporation temperature.

**Figure 8** The refrigerant cycle of the vapour compression heat pumping cycle: 2S – 2C, evaporation; 2C – 1C, compression; 1C – 1S, condensation; 1S – 2S, adiabatic expansion.
Consequences on predictions

As a refrigeration or heat pumping plant, or machine, is to be subject to a refrigerant retrofit, there are a number of plausible substitute working fluids commercially available. The first assigned R22-substitute to reach the market was R407C, but since then a number of alternatives has appeared. Before conducting a refrigerant retrofit, a decision has to be made on what substitute refrigerant to choose. To do be able to do this, it is of course desirable to have some estimation on expected capacity, $\dot{Q}_2$ or $\dot{Q}_1$, and efficiency, $COP_2$ or $COP_1$. When replacing R12 and R500 with R134a became an issue, comparisons of these parameters under constant evaporation and condensation temperatures was comme il faut, as it had been historically when refrigerant performance had been compared, and the information from these comparisons corresponded well with what was experienced in field and lab meas-

---

28 Initially R407 was available in a number of compositions, ranging from R407A to R407D. Commercially only R407C has survived to the present. Cp. ARTI (2001)
urements. This is however not a universally suitable method to use when estimating the performance of various substitutes in the same piece of machinery; machinery not designed for any of them.

In figure 11 a basic refrigerant cycle has been simulated. Each alternative has its performance compared to R22 in a total of nine cases. The condensation temperature has been set to 35, 45 and 55°C, and the evaporation temperature to -10, 0 and 10°C. When comparing estimates of the performance outcome of different substitute choices, it is convenient to compare the relative capacity versus the relative efficiency such that:

\[
\text{Rated capacity} = \frac{\dot{Q}_{2,\text{Alt.}}}{\dot{Q}_{2,R22}} \quad \text{Eq. 4}
\]

\[
\text{Rated efficiency} = \frac{\text{COP}_{2,\text{Alt.}}}{\text{COP}_{2,R22}} \quad \text{Eq. 5}
\]

![Graph showing rated cooling capacity versus relative efficiency, COP₂, if a refrigerant cycle model operating between the same temperatures is used. The symbols represent different operating points.](image)

**Figure 10** Rated cooling capacity versus relative efficiency, \(\text{COP}_2\), if a refrigerant cycle model operating between the same temperatures is used. The symbols represent different operating points.
The components of the circuitry

Each of the components constituting a vapour compression heat pumping machine is commonly described as open systems according to the general systems description of classical thermodynamics. When the system of the compressor, evaporator or condenser is described (often with paradigmatically set system boundaries) a mass flow ($\dot{m}$) or mass flux ($G$) flows in, and commonly (but not always) the same flow passes out over the systems boundary in the “other end” of the process(es) within the component. Over the systems boundary there is a flow of energy (mechanical, electrical or heat) as described in figure 9. Many of the descriptions and models at this level (where behaviour in and of individual components are studied) are reductionistic to their character.

Heat exchangers – evaporator and condenser

All heat exchanger descriptions may be referred to as systems. Depending on what level of phenomena that is studied, whether it is on micro or macro scale, it may actually be considered a quasi-static closed system. This would be the case of the study and analysis of e.g. bubble formation on the surfaces: that level has, on a systemic perspective, very little to do uniquely with the vapour compression cycle, maybe except concerning the choice of studied fluids and perhaps surface properties.

The heat exchangers have losses too: E.g. heat transfer loss, which is illustrated by the following example. A body (1) with the temperature $T$ comes into contact with another body (2) with the temperature $(T - \Delta T)$. The temperature different will lead to heat flux between the two bodies, $dq_r$. As the heat flux is the same...

$$-dq_r = T \cdot ds_1 \quad \text{and} \quad dq_r = (T - \Delta T) \cdot ds_2$$

Eq. 6
The changes in entropy in the two bodies may thus be expressed as

\[ T \cdot ds = dq \]  \hspace{1cm} \text{Eq. 7}

The total entropy production, \( ds_1 + ds_2 \), is thus

\[ ds_1 + ds_2 = \frac{-dq_r}{T} + \frac{dq_r}{T - \Delta T} = dq_r \left( \frac{1}{T - \Delta T} - \frac{1}{T} \right) \geq 0 \]  \hspace{1cm} \text{Eq. 8}

The losses from heat transfer are represented by rectangles when plotted in s-T charts (\( \Delta s \cdot \Delta T \)).

Evidently the entropy production is always equal to or larger than zero. In practice however it will always be larger than zero, since the temperature difference would have to be zero if the entropy production should be zero. Without temperature difference there would not be any heat transfer in the first place.

**Refrigerants – thermodynamic models**

There are a number of criterions to be set on refrigerant fluid. Some have to do with safety, others with limits to thermodynamic properties and practical issues, so that they are actually possible to
use. It is of course desirable that the working media does not occur in solid phase within the expected operation range; desirable, however not absolutely necessary, that the critical temperature of the fluid is above the highest operating temperature. A low critical temperature of the working media will render in low efficiency and capacity of the mechanical refrigeration cycle. It is desirable, however not absolutely necessary, that the vapour pressure of the fluid is higher than normal pressure at all operating temperatures; i.e. that the operating pressure is above approximately 100 kPa.

<table>
<thead>
<tr>
<th>Refrig.</th>
<th>(c_p) [J/mol.K]</th>
<th>(\kappa = c_p/c_v)</th>
<th>(t_{\text{crit.}}) [°C]</th>
<th>(p_{\text{crit.}}) [kPa]</th>
<th>(t_{\text{nbp.}}) [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R12</td>
<td>69</td>
<td>1.17</td>
<td>112.0</td>
<td>4 136</td>
<td>-29.8</td>
</tr>
<tr>
<td>R134a</td>
<td>81</td>
<td>1.15</td>
<td>101.1</td>
<td>4 059</td>
<td>-26.1</td>
</tr>
<tr>
<td>R600a</td>
<td>91</td>
<td>1.13</td>
<td>134.7</td>
<td>3 640</td>
<td>-11.6</td>
</tr>
<tr>
<td>R502</td>
<td>69</td>
<td>1.17</td>
<td>80.7</td>
<td>4 019</td>
<td>-45.3</td>
</tr>
<tr>
<td>R404A</td>
<td>76</td>
<td>1.17</td>
<td>72.1</td>
<td>3 735</td>
<td>-46.6</td>
</tr>
<tr>
<td>R507A</td>
<td>76</td>
<td>1.16</td>
<td>70.7</td>
<td>3 715</td>
<td>-47.1</td>
</tr>
<tr>
<td>R22</td>
<td>52</td>
<td>1.24</td>
<td>96.1</td>
<td>4 990</td>
<td>-40.8</td>
</tr>
<tr>
<td>R407C</td>
<td>67</td>
<td>1.19</td>
<td>86.0</td>
<td>4 634</td>
<td>-43.8</td>
</tr>
<tr>
<td>R417A</td>
<td>83</td>
<td>1.15</td>
<td>89.9</td>
<td>4 096</td>
<td>-38.0</td>
</tr>
<tr>
<td>R410A</td>
<td>58</td>
<td>1.24</td>
<td>70.2</td>
<td>4 770</td>
<td>-51.6</td>
</tr>
<tr>
<td>R32</td>
<td>46</td>
<td>1.33</td>
<td>78.1</td>
<td>5 782</td>
<td>-51.7</td>
</tr>
<tr>
<td>R717</td>
<td>39</td>
<td>1.35</td>
<td>132.3</td>
<td>11 333</td>
<td>-33.3</td>
</tr>
</tbody>
</table>

Table 4 The molar heat capacity \(c_p\) and isentropic coefficient \(\kappa\) of the saturated vapour at normal pressure (101.3kPa), and critical temperature \(t_{\text{crit.}}\), critical pressure \(p_{\text{crit.}}\) and normal boiling point \(t_{\text{nbp.}}\) of twelve refrigerants and refrigerant mixtures.\(^{29}\) (Klein 1991-2003; McLinden et al. 1998)

It is a not too uncommon misconception that complex molecules have lower normal boiling point, \(t_{\text{nbp.}}\), than molecules with a less complex structure, or that the same applies for light molecules; that light molecules would have lower normal boiling point than

\(^{29}\) For the refrigerant blends \(t_{\text{nbp.}}\) is given as the bubble point temperature at normal pressure.
heavy. As the figures below shows, there exists no such simple relation.

![Graph](image-url)

**Figure 12** The normal boiling point as a function of the mol masses of the refrigerant fluids of table 4.

The structures of the refrigerant molecules are important to consider. The effect of the shape and complexity of a refrigerant molecule may be described by the isentropic coefficient, $\kappa$. This will affect the performance of the working media less than the critical point but is still significant. The more complex a molecule is, the lower the isentropic coefficient and lower the molar heat capacity is. Now, assuming that the refrigerant vapour behaves like an ideal gas, and considering what would happen under an isentropic (reversible adiabatic) compression the relation between the temperatures before (1) and after (2) the compression may be described using the equation below.
Figure 13 Isentropic coefficient and discharge temperature according to the model described below for the refrigerant fluid of table 4. (Klein 1991-2003; McLinden et al. 1998)

\[
\frac{T_1}{T_2} = \left( \frac{p_1}{p_2} \right)^{\frac{\kappa-1}{\kappa}} \tag{Eq. 9}
\]

Where \( T \) is the absolute temperature and \( p \) is the absolute pressure. Studying the temperature of the refrigerant fluid after an isentropic compression (assuming ideal gas behaviour) of saturated vapour with a pressure corresponding to 0°C, to a pressure corresponding to a saturation temperature of 50°C, it becomes evident that a low isentropic coefficient renders in lower temperatures after the compression (cf. figure 14)\(^{30}\). This may be interpreted using a mechanical analogy using the concepts of inertia and mass-moment of inertia:

During the compression, work (energy) is applied to the refrigerant vapour to increase its pressure. The individual molecules may

\(^{30}\) For the refrigerant blends, the pressures corresponding to 0°C and 50°C are calculated as the dew point pressures at the given temperatures.
only store this energy in forms of motions – translation and rotation, and some vibration modes. Depending on the structure of the molecules energy may be stored different types of motion. A very compact and light molecule as ammonia, \( \text{NH}_3 \) (R717), may essentially just perform translation motions. Rotating a small compact structure will not store any significant amount of energy. A branched, but maybe still lightweight, structure as e.g. isobutane, \( \text{C}_4\text{H}_{10} \) (R600a), may store energy by translation movement \textit{but also} by rotating around its axes (axes constituted by its carbon skeleton). More energy will be stored in the rotational movements the heavier and longer or more branched the molecule is. Consequently, a relatively smaller amount of the energy will be stored through translation motion. As the amplitude of the translation motion is experienced as temperature, consequently the temperature increase will be smaller for molecules that are more complex. In practice this will be experienced as the discharge temperature from the compressor being lower for more complex molecules (e.g. R600a) than for smaller and more compact ones (e.g. R32).

So if low discharge temperatures are desirable since it reduces the stress on the used lubricants, one could assume that using molecules with low isentropic coefficient would be desirable. In the same manner, the molar heat capacity may be interpreted. I.e. how much translation motion does supplied energy turn into; translation motion – temperature. As above, if a significant amount of the energy may be stored in rotational movement less will become translation movement – or temperature. This may also be interpreted such that the slope of the isentropes in the superheated region is dependent on the structure of the fluid. The slope of the isentropes on the saturation (dew point) line in an \( h \log-p \) chart can be estimated with the following:

\[
T_{ds} = dh - vdp
\]

\text{Eq. 10}
Where: \( T = \) temperature \([K]\)
\( s = \) entropy \([J/kmol,K\) or \(J/kg,K]\)\(^{31}\)
\( h = \) enthalpy \([J/kmol\) or \(J/kg]\)
\( v = \) specific volume \([m^3/kmol\) or \(m^3/kg]\)
\( p = \) pressure \([Pa]\)

Constant entropy implies that \( ds = 0 \). Thus

\[
\left( \frac{\partial p}{\partial h} \right)_{dr=0} = \frac{1}{v} \quad \text{Eq. 11}
\]

I.e. at the saturation line the slope of the isentropes are proportional to the inverse of the specific volume. Assuming ideal gas behaviour of the saturated vapour

\[
p \cdot v = R_M \cdot T \quad \text{Eq. 12}
\]

Where \( R_M \) is the absolute gas constant = 8 314.3 \( J/(kmol,K)\)...

\[
\frac{1}{v} = \frac{p}{R_M \cdot T} \quad \text{Eq. 13}
\]

Thus equation 14 is obtained.

\[
\left( \frac{\partial p}{\partial h} \right)_{dr=0} = \frac{p}{R_M \cdot T} \quad \text{Eq. 14}
\]

This of course may be developed to:

\(^{31}\) In this section, unless anything other is stated; mol-base will be used.
\[
\frac{dp}{p} = \frac{dh}{R_M \cdot T}, \text{ where } dh = \epsilon_p \cdot dT \text{ and } \epsilon_p = R_M \cdot \frac{\kappa}{\kappa - 1} \quad \text{Eq. 15}
\]

After integration this turns into:

\[
\ln\left(\frac{p_2}{p_1}\right) = \left(\frac{\kappa}{\kappa - 1}\right) \cdot \ln\left(\frac{T_2}{T_1}\right), \Rightarrow \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa - 1}{\kappa}} \quad \text{Eq. 16}
\]

That is the same as equation 9.

Now, generally refrigerants in ordinary applications do not behave as ideal gases. The concept of ideal gases assumes that the average free distance between the molecules is so large that the individual molecules are considered as point masses and any forces between the molecules can be neglected. I.e. their physical extension is neglected\(^{32}\). Many gasses and gas mixtures, such as air, may be described as ideal gasses at low pressures and relatively high temperatures (e.g. air at normal atmospheric pressure and room temperature). The ideal gas law is an equation of state – relation between properties – that do not consider e.g. spatial extension of the molecules. But as long as the free average distance between the molecules is large enough, it yields quite good predictions. Close to the saturation temperature at a given pressure refrigerants deviates a lot from ideal gas behaviour, since they do have physical extension, a considerable mass and the free average distance between the individual molecules is too short (Boltzman 1964).

However, using Clapeyrons equation (equation 17), where

\[
(s'' - s')dT = (v'' - v')dp \quad \text{Eq. 17}
\]

\(^{32}\) Cp. Kennard (1938), Present (1958) and Boltzman (1964).
...and where \( s'' \) and \( v'' \) refers to the saturated vapour, and \( s' \) and \( v' \) to the saturated liquid. By defining \( r \) as the latent heat of vaporisation, such that

\[
r = (s'' - s') \cdot T \quad \text{Eq. 18}
\]

\[
r = T \cdot (v'' - v') \frac{dp}{dT} \quad \text{Eq. 19}
\]

\[
\frac{dT}{dp}
\]
is the slope of the vapour pressure curve.

The difference in \( v'' - v' \) between different refrigerant fluids (with very different \( \varphi \) and \( \kappa \)) is small. Thus the heat absorbed (cold generated) as a fluid evaporates will be approximately the same for a given swept volume at a given pressure. I.e. if the vapour pressure curves of two refrigerant fluids are “the same”, and their mol masses are “the same” as well, not only will the isentropes in the superheated region have about the same slope, but also the latent heat of vaporisation will be the same. This may be expressed with Troutons rule, 20 and 21.

\[
r = C_T \cdot T \quad \text{Eq. 20}
\]

\[
C_T = (v'' - v') \frac{dp}{dT} \approx 88 \text{ kJ/(kmol} \cdot \text{K)} \quad \text{Eq. 21}
\]

So, the latent molar heat is about the same for any fluid at normal pressure. I.e. when “designing” a substitute for a refrigerant to be phased out, one should look for a fluid or mixture of fluids with a

\[33 \text{Troutons constant}\]
(mean) molecular mass close to the one of the fluid that is to be replaced. Further, one should try to choose a fluid or mixture of fluids with a vapour pressure curve close to the one of the original fluid, and one would desire to have an ideal gas isentropic coefficient and a specific heat similar to the one of the original fluid. If this is achieved, using this molecular approach\textsuperscript{34}, a substitute refrigerant (mixture) that would show only very small deviations from the original would be obtained: A substitute that when used in existing machinery would render in “identical” performance of the heat pump or liquid chiller.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{Figure_14}
\caption{The value for Troutons constant for the refrigerant fluids in table 4, calculated using equation 21. (Klein 1991-2003; McLinden et al. 1998)}
\end{figure}

Zeotropic refrigerant mixtures are blends of two or more refrigerants that deviate from “perfect mixtures” (cp. azeotropic blends). That is, mixtures of refrigerants that do not behave as one pure fluid. One phenomena connected to this is the so called temperature glide. While evaporating at constant pressure the temperature of

\textsuperscript{34} Cp. McLinden and Didion (1988)
the boiling refrigerant mixture increases. In the case of condensation at constant pressure the temperature of the condensing fluid decrease. When using this type of refrigerant mixtures the analysis of the systems performance is complicated by the fact that the circulated refrigerant mixture composition is different from the one the system was, or was thought to be, charged with. To understand and correctly analyse the performance and behaviour of a refrigeration or heat pumping facility, one has to consider these composition shifts. Small deviations in composition may perhaps not affect the general analysis of capacity and efficiency, but can explain e.g. hunting or bubbles in the liquid line. Further elaboration on this subject can be found in chapter 5: Measurement and analysis.

**Compressors**

Compressors are modelled in a number of ways. When only the compressor as such – as an open system – is modelled by compressor manufacturers, not very many aspects specifically connected refrigeration is studied. Obviously refrigeration compressors are slightly different in their design than air compressors etc are. The differences are connected to mechanical differences due to different pressure ranges, different fluids etc. But from a thermodynamic point of view, there are no fundamental differences. (Cp. heat exchangers as described above) Looking at the losses, some of them evidently have mechanical origin – friction between moving parts etc – other’s due to thermo-physical behaviour of the compressed gas volume. In the following discussion a reciprocating open compressor will be used as the object of study.

When the piston of a reciprocating compressor is approaching its lower end position and the pressure within the cylinder has become lower than the suction line pressure, the vapour flows into the cylinder volume. This will continue as the piston passes the lower end position and starts compressing the vapour in the cyl-
inder space. As the pressure in the cylinder has increased to about suction line pressure, no more new vapour flows into the cylinder as the inlet valve closes. If we start off with the assumption that the compression is isentropic – i.e. without losses and adiabatic – the temperature of the compressed vapour increases. Cp. equation 9. Obviously, the compression is in fact not adiabatic: In contact with the hot gas is a cylinder wall. As described in the heat exchanger section above, heat will transfer from the warm gas to the relatively cooler cylinder wall – with heat transfer losses and all.

When the piston approaches it upper end position and the pressure of the vapour in the cylinder is slightly above the discharge line pressure (in a refrigeration application essentially decided by the condensation temperature) the hot compressed gas flows out through the outlet valve into the discharge line. This continues as the piston passes its upper end position and starts to expand the vapour still remaining in the cylinder. Nota bene, in this the first stage, the temperature of the high pressure vapour in the discharge line is lower than what it would have been if the compression had been truly isentropic – some heat has passed into the cylinder wall.

As soon as the pressure of the vapour in the cylinder reaches the discharge line pressure, the outlet valve closes and some, still high pressure, vapour remains in the cylinder volume and undergoes an expansion as the piston moves towards its lower end position. If we assume that the expansion is isentropic, the remaining fluid undergoes a temperature reduction as described by equation 9. Eventually the temperature of the remaining vapour becomes so low that the cylinder wall is actually warmer than the gas. I.e. the expansion cannot be adiabatic, but heat is re-transferred from the cylinder wall to the gas – with heat transfer losses and all.

When the second stage of this description approaches – the next piston stroke – and the pressure of the vapour in the cylinder gets
lower than the pressure in the suction line, and new gas flows in, the
temperature of the mixture of the remaining and new gas will
be higher than the temperature of the gas in the actual suction
line. As the second stage progresses undergoing the same steps as
described above, further heat will transfer to and from the cylin-
der wall, as the piston moves up and down, and the temperature
of the gas entering the discharge line will increase. To this, heat
generated through mechanical losses such as friction between pis-
ton and cylinder wall, losses due to friction in inlet and outlet
valves, blow-by etc, is added. This lead to the expression for the
technical work (\(\varepsilon_t\)) applied to the quasi static open system being
amended with loss components (\(\int db\)):

\[
\varepsilon_t = \frac{\kappa}{\kappa - 1} \cdot R_M \cdot T_1 \left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{\kappa - 1}{\kappa}}\right] + \int db \quad \text{Eq. 22}
\]

The losses may be bundled and attached to the isentropic coeffi-
cient so that it becomes something similar to a polytropic coeffi-
cient, \(n\), and thus leading to the transition of equation 9 to equa-
tion 23, and equation 22 to equation 2435:

\[
\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \quad \text{Eq. 23}
\]

\[
\varepsilon_t = \frac{n}{n - 1} \cdot R_M \cdot T_1 \left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{n - 1}{n}}\right] \quad \text{Eq. 24}
\]

35 This is not the common way of using the polytropic coefficient in thermodynamics.
Apparently not all refrigerant vapour residing in the compressor cylinder during the compression is exchanged at each piston stroke. Some remain due to the reasons described above, but this effect is amplified by the presence of a dead space volume at the top of the cylinder. This is placed there since it otherwise would be an imminent risk that the piston would actually hit the cylinder top, compromising the functionality of the compressor. The impact of this dead space ($V_0$) assuming isentropic behaviour of the compression, is a function of the total stroke volume of the compressor ($V_s$) and the pressure ratio of the compressor, expressed as volumetric efficiency ($\eta_{S, is}$).

\[
\eta_{S, is} = 1 - \frac{V_0}{V_s} \left( \frac{p_1}{p_2} \right)^{\frac{1}{\kappa}} - 1 \quad \text{Eq. 25}
\]

Analogously, the volumetric efficiency of a reciprocating compressor could be modelled similar to how the compression work can be modelled, e.g…

\[
\eta_s = 1 - C_{Vol} \left( \frac{p_1}{p_2} \right)^{\frac{1}{m}} - 1 \quad \text{Eq. 26}
\]

Where $C_{Vol}$ is a constant for the ratio between the stroke volume and the dead-space, and $m$ is a polytropic-exponent-like constant.

**Expansion devices**

There are two types of expansion devices used commonly today: Capillary tubing and thermostatic expansion valves. There are other types, but in the applications of concern in the current thesis these are the types predominantly used. The function of vari-
ous expansion devices is well described in the literature used in refrigeration technology, and will therefore not be elaborated on further in the current thesis.

The reason for using expansion devices in the mechanical refrigeration process is to maintain the pressure difference between the condenser and evaporator. In practice they have one more function to fill in practice, and that is to make sure that no refrigerant liquid is spilled over into the compressor inlet through the suction line – provide a certain degree of superheating to the refrigerant vapour. …but that is a practicality, not a necessity for the actual cycle.

The vapour compression machine
At this systems hierarchical level, the models of the individual components commonly utilise models derived from more reductionistic models, such as the ones described above. I.e. it is primarily the behaviour of the phenomena and component, not the nature of it that is interesting at this level of modelling. However, at a given systems hierarchical level, it is still possible to implement more elaborate models for the components, sub-systems. This is in the current thesis referred to as conceptual model resolution.

Compressors
Compressors can be modelled in various ways, depending on the scope of the modelling. If the aim of the model is to describe the systemic behaviour of the compressor as a circuit component and on a macroscopic level, it is often modelled with an equation describing an empirical relation of the consequences of the losses (isentropic efficiency) such as equation 27. Had the very nature of the compressor been in focus, relations like equation 22 or 24 been more appropriate, but considering the cycle as a whole and the “macroscopic” view of the components, an expression like the
one below, with perhaps very little direct physical coupling representing purely numerical adaptations of empirical data, is practical.\textsuperscript{36}

\[ \eta_{ic} = f\left(p_1, p_2, \ldots\right) \]  

\textbf{Eq. 27}

There are various ways of modelling the mass flow, or volume flow, through the compressor. It may be modelled with either, a constant stroke volume flow and volumetric efficiency, which in some cases may be set to a constant value (e.g. 1) or described with a mathematical function similar to that of the isentropic efficiency, or not modelled at all – which may be the case when only the refrigerant cycle performance is considered\textsuperscript{37}. There are a number of correlations for compressor behaviour on the form of equations 27, for both isentropic and volumetric efficiency. None will be presented in detail in the current work.

It is also possible, and sometimes relevant, to burden the compressor with what could be called an adiabatic efficiency of the compressor and its physical surroundings (\(\eta_{\text{adiab}}\)). “How much of the electric power applied to the compressor, is actually transferred to the working media?”

\[ \eta_{\text{adiab}} = f\left(T_{1C}, T_\infty, \text{geometry}, \ldots\right) \]  

\textbf{Eq. 28}

The value of the adiabatic efficiency is probably around 90-98\%. The adiabatic efficiency has to be used together with the isentropic efficiency. Such that:

\textsuperscript{36} Examples on correlations used is found in appendix b.

\textsuperscript{37} This is of course a type of model too.
\[ \dot{E}_{el} = \eta_{li} \cdot \eta_{\text{adiab}} \cdot \eta_S \cdot \frac{V_S}{v_{2C}} \cdot \varepsilon_{tr}, \text{ where } \varepsilon_{tr} = h_{1is} - h_{2C} \quad \text{Eq. 29} \]

**The expansion device**

The expansion device, commonly a thermostatic expansion valve, may be modelled in various ways. The probably most common way of modelling it is to set a superheat to a fixed value (in Kelvin). A value that is not affected by altered operating conditions. Another is of course to more use a model with a more detailed description of the function of a thermostatic expansion valve: In a thermostatic expansion valve the amount of superheat obtained depends on the saturation pressure difference induced by the force of the spring-and-screw arrangement. As this force does not vary with the operation pressure (or temperature) the resulting superheat will become larger as the operation pressure decreases. (Cp. vapour-pressure curves). If the superheat has been set under conditions different from the ones under which the unit actually operates, it is quite likely that the superheat during operation will be different. Likewise, if the circulated composition (if a zeotropic refrigerant mixture is used as working media) is not the one it is thought to be: the actual superheat will be quite different than the one thought to be set. It is also possible that e.g. a unit which previously used R22 but which now operates with another working media, still has its old R22-valve left. This may also be modelled. This expansions device model has occasionally been used within the current work.

**Integration of parts – conceptual model resolution – prediction**

As all five components making up the machine are integrated and allowed to interact with each other directly and indirectly, new phenomena and behaviour will emerge – effects not seen as each component is studied by itself. In the following scenarios, model resolutions, refrigeration machinery with stroke volume flows of
0.005 m²/s are studied. In all scenes the superheat has been set to 8 K and the unit is charged until 4 K subcooling is obtained. The following temperatures are thought to have been measured:

<table>
<thead>
<tr>
<th>t.water.in [°C]</th>
<th>t.brine.in [°C]</th>
<th>t₁ [°C]</th>
<th>t₂ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>10</td>
<td>63</td>
<td>2</td>
</tr>
<tr>
<td>45</td>
<td>0</td>
<td>53</td>
<td>-8</td>
</tr>
<tr>
<td>35</td>
<td>-10</td>
<td>43</td>
<td>-18</td>
</tr>
</tbody>
</table>

*Table 5* The temperatures measured on the three different chillers, which will be used as input data for the scenes below.

**Predictions based on a few measurement data**

*Scene I: Performance compared at the same evaporation and condensation temperatures as was measured with R22 are used. Isentropic and volumetric efficiencies are set to fixed values: \( \eta_{is} = 0.7 \) and \( \eta_S = 0.6 \).*

As the figure below shows, this model yields predictions that only to a very small extent deviates from when the pure refrigerant cycle was considered. The only conceptual difference is that the compression worked has been scaled with a factor \( 1/\eta_{is} \).

*Figure 15* Scene I.
Scene II: Performance compared at the same evaporation and condensation temperatures as was measured with R22 are used. Isentropic and volumetric efficiencies are calculated according to equation 27.

If functions for the compressor efficiencies are added – relations such that the efficiencies decrease as the pressure ratio increase etc; i.e. closer to the “real” behaviour of real compressors – some interesting phenomena can be seen. Alternatives with lower desuperheating loss and pressure ratios are benefited from this.

**Figure 16** Scene II.
Figure 17 As the compressor is modelled and the superheating is set, the functionality of these is integrated into the conceptual model.

Scene III: Performance compared at the same operating temperatures, $t_{\text{water,in}}$ and $t_{\text{brine,in}}$. Overall heat transfer coefficients ($UA$) are calculated from the evaporation and condensation temperatures for R22. These values are used for all four alternatives to calculate their evaporation and condensation temperatures. Isentropic and volumetric efficiencies are set to fixed values: $\eta_S = 0.7$ and $\eta_v = 0.6$. Driving temperature differences defining the overall heat transfer coefficients are the inlet temperature differences:

$$\Delta T_1 = T_1' - T_{\text{water,in}} \quad \Delta T_2 = T_{\text{brine,in}} - T_2''$$

Using this model to build predictions of expected behaviour shows that the heat exchangers have a very large impact on the outcome. Lower capacity reduces the load on the heat exchangers. This will make them appear larger for e.g. R134a and R417A than for R407C or R404A. – Or for R22 for that matter.
Scene IV: As scene III, but with the difference that the isentropic and volumetric efficiencies are calculated using relations as equation 27.

Figure 18 Scene III.

Figure 19 Scene IV.
Figure 20 The models used in scenes III and IV have included the overall behaviour of the heat exchangers.
Predictions based on phenomena – generality

Scene V: This scene differs from the ones above, as the model as described in chapter 4, in the section ‘Optimum systems charge’, is used. Thus heat transfer coefficient for each and every alternative (including R22) calculated, as is the surface distribution between subcooling and desuperheating-condensation, evaporation and superheating, using models described in chapter 4. The isentropic and volumetric efficiencies are calculated using a relation like equation 27. Three cases are studied:

Firstly, a unit with relatively small heat exchanger surfaces rendering in 12-14K inlet temperature difference as R22 was used.

Secondly, a unit with 7K inlet temperature difference in the condenser and 8K inlet temperature difference in the evaporator as R22 was used.

Thirdly, a unit with large heat exchanger surfaces, rendering in 8K inlet temperature difference in the evaporator, and 5K inlet temperature difference in the condenser as R22 was used.

The machines are equipped with plate heat exchangers, thermostatic expansion valves and open compressors.

This model does not necessarily lead to better predictions than e.g. the models of scenes III-IV, or even I-II, but it includes more models of more phenomena. Each and every one of the phenomena has been studied extensively in the literature, but their integrated behaviour is seldom taken under consideration in case of refrigerant retrofitting etc.

As an absolute quantitatively correct predictive model, this model obviously has its limitations. It describes unique units with open compressors and brazed plate heat exchangers (which in practice might seem a slightly odd combination). If units with tube-and-shell or fin-coil heat exchangers or other compressor designs and
principals are of interest, the code has to be adapted and perhaps other phenomena included.

Figure 21 Scene V-I: Small heat exchangers.

Figure 22 Scene V-II: Medium sized heat exchangers.
Figure 23 Scene V-III: Large heat exchangers.

Figure 24 In the model used for scene V, the heat exchanger models are refined – heat transfer phenomena are included, as is surface distribution between e.g. evaporation and superheating in the evaporator.
Predictions based on experiments – black box uniqueness

Scene VI: In some cases it may be possible to actually compare the performance of a specific (real) unit with various substitutes. As the system then is a black box (see chapter 4) we cannot know what goes on inside the unit – how the surfaces in the heat exchangers are used etc. All we can do is to assert that this particular unit worked this way with these refrigerant alternatives and under the tested conditions.

By doing a series of tests in their lab, heat pump manufacturer IVT has evaluated the performance of a make of a legacy heat pump unit. The purpose has of course been to find (1) what refrigerant substitute to use for R22, and (2) how much to put of it into this particular unit, so that they can inform their agents and their service personnel how retrofits should proceed come the situation that they have to conduct a change of working media. Further elaboration on the subject of systems charge may be found in chapter 4 – Experiences: field, lab and simulation. The tested domestic heat pump delivers approximately 9kW heating. The tests have been conducted at four different outlet temperatures of the radiator water, and four different inlet temperatures of the secondary refrigerant, brine.

\[ t_{\text{water, out}} = 35/45/50/55^\circ \text{C} \quad \text{and} \quad t_{\text{brine, in}} = -5/0/5/10^\circ \text{C}. \]
As the measurements are made, the heat pump is essentially a black box: We cannot exactly know what goes on inside, we only see the result.

Figure 26 Outcome of tests of a heat pump running at reality-like conditions; i.e. conditions representing common operating conditions. (Measurements by IVT AB)

The tests show that in this particular unit both capacity and efficiency will be lost what ever choice is made. At small temperature lifts the differences are small for FX90 and R407C, but the higher the temperature lift is the poorer will the unit perform. As it is a heat pump it is equipped with fairly large heat exchangers. See chapter 4 – Experiences: field, lab and simulation. As the chart above shows, R417A yields the highest rated efficiency. The margin is
however small. FX90 has two measurement readings that deviate a lot from the general trend. Those two measurements are hard to analyse since the measurements made are black box measurements.

**Which of the Models above yields the best predictions?**

What essentially has been done from scenes I and II, through scene V, is that the conceptual resolution of the model has increased. Still, they are just models, and as such white boxes that only try to replicate behaviour in the real world in more or less refined ways.

The best prediction of how a specific unit will perform after a refrigerant retrofit is of course to make tests on it under real or reality-like conditions. Evidently scene VI yields the best predictions on how that specific unit will perform with either of the tested substitute refrigerants. But scene VI does not really include the use of a predicting model, rather the use of the more expensive method of trial-and-error. And none of the phenomena causing the behaviour can be studied or evaluated. It gives no information on how a liquid chiller or split-type air conditioner will perform – something that (at least qualitatively) can be shown using models like the one in scene V.

As figures 22 through 24 indicates has the original design concerning heat exchanger area a tremendous effect on the output data for the predictions. This is something not seen using any other of the predicting models exemplified above than those similar to the one in scene V. It is also commonly neglected in the literature and retrofit manuals. The indirect consequences are of course there in scene VI, but we cannot see them directly. This will be somewhat elaborated on in chapter 4 – *Experiences: field, lab and simulation.*
Installation – general heat pumping machine

The next step of abstraction, or system hierarchical level, of this is the general heat pumping machine: A generalised heat engine working “backwards”. In this case what goes on inside the machine, and the nature of it, is totally uninteresting. The machine is just an abstract device lifting heat from a low temperature level to a high temperature level. To make this possible work (mechanical) is applied. No heed is paid to heat transfer characteristics etc.

There are various ways of modelling this machine: One is to give it a fixed coefficient of performance (COP) of, say three (3). Another is to calculate the Carnot-COP with the temperatures of the heat source and sink, $T_2$ and $T_1$, then burden it with a Carnot efficiency of the whole machine, $\eta_C$, (With a fixed value based on experience). Cp. equations 30 and 31. The capacity may be modelled in various ways. Either as a constant value – a given number of kilowatts is delivered no matter what the surrounding circumstances are – or the deliver capacity is modelled as a function of, say, the mean temperatures of the heat sink and source respectively (Cp. 32). I.e. it is only the behaviour of e.g. the liquid chiller, not the reason for the behaviour and nature of it that is modelled.

It also possible to, as often is the case when students are confronted with it in engineering thermodynamics classes, not model it at all: Delivered capacity is not considered interesting.

\[
\eta_C = \frac{\text{COP}_2}{\text{COP}_{2,\text{Carnot}}} \quad \text{Eq. 30}
\]

\[
\text{COP}_{2,\text{Carnot}} = \frac{T_2}{T_1 - T_2} \quad \text{Eq. 31}
\]

\[
\dot{Q}_2 = f(T_1, T_2) \quad \text{Eq. 32}
\]
Apart from introductory courses in engineering thermodynamics etc, the concept of the general heat pumping machine is commonly used when the installation as a whole is modelled or investigated. Chillers, heat pumps etc, are even symbolised with a schematic interpretation of a heat pumping machine in installation diagrams and construction drawings. At this level it is very apparent that the nature of phenomena within the actual unit is of very little interest. It is the behaviour of the unit in the installation-system, and how that system might be affected of this that is interesting. For this, the results from the heat pump tests in scene VI above can be used; thus the black box representing the heat pump performance measurements undergoes the transition to numerical relations in a white box\(^{38}\). Relations such as:

\[
\dot{Q}_1 = f_1(t_{\text{water, out}}, t_{\text{brine, in}}) \quad \text{and} \quad \dot{E}_{\text{el}} = f_2(t_{\text{water, out}}, t_{\text{brine, in}}) \quad \text{Eq. 33}
\]

\[\text{Figure 27} \quad \text{All components in this system are modelled as white boxes. Some of the models, as for the heat source for example, may have elaborate models with many types of phenomena included; whereas the heat pump uses numerical relations as described in the equations above.}\]

\(^{38}\) The concept of black and white boxes from cybernetics are explained and used further in chapter 4.
By using this model, it is possible to study how the heat pump (in this case) interacts with the house and its heat distribution system, and the energy source, the bore hole for instance. In the simulation model as described in figure 28, the studied time frame is one year (in the software used it is actually one year a number of years after the installation of the heat pump, so that the borehole temperature has stabilised), and the time step is one hour. The time constants for bore hole, the house and the heat distribution system are very much longer than the time constant for the heat pump, the heat pump is simulated with relations of the form described in equation 33. (Forsén 1997-2003; 2002)

Four cases are studied, and in all cases the heat pump has originally been chosen so that it (with R22) covered a certain amount of the energy need. The heating water circuit is a 55/45-circuit, as the studied building is built in the mid nineteen-sixties and as the building is situated in Stockholm the dimensioning outdoor temperature is set to be -20°C. I.e. high condensation temperatures are not uncommon. Thus, using the software Enlight, the borehole depth has been calculated. The energy consumption etc of the building-borehole-heat-pump-heat-distribution-system is studied when the heat pump has been designed to cover 70, 80, 90 and 99% of the annual energy need. Using the same building (with the same energy need – which of course are different in each of the four scenarios) and borehole the use of driving energy for the heat pump and auxiliary heating (direct electricity) is compared for all three alternatives in all four cases.
The figure above is designed in the same way as the previous comparisons. As so much information is superposed, it is very hard to actually draw any conclusions. In figure 29 it is very hard to draw out any significant information. At a first glance things might even seem contradictive to common sense. As so much information is superposed the conclusion becomes very blurred. However, relatively to R22 it covers the need less well as the original annual heating energy coverage increase. (To obtain the actual energy coverage for the alternative, the rated value should be multiplied to the value for R22. For example $0.9 \cdot 0.8 = 0.72$ for R417A in the case where the heat pump using R22 covered 80% of the annual energy need.)

The total energy coverage is still higher in the installation where the heat pump was originally designed to cover 90% than the installation where it was designed to cover only 70%. At 99% original annual energy coverage design, the actual operating hours of the heat pump are firstly significantly shorter than it even was in the 90% case. Secondly, in the 99% case, the need for auxiliary heating is minute whether R22 or any tested substitute is used, so the annual-COP (see below) is essentially only dependent on the momentary COP (effect-COP). In the other cases the heat pump not only delivers less heating capacity at a lower momentary
efficiency, the decrease in capacity leads to an increased need for auxiliary heating with electricity (at a corresponding momentary COP of 1.00).

\[ \text{COP}_{\text{Annual}} = \frac{Q_{\text{Need}}}{E_{\text{HP}} + E_{\text{Aux}}_{\text{Annual}}} \]  

Eq. 34

Figure 29 By multiplying the values with the price for electric energy the increase in actual heating cost are calculated.
The system with a purpose

The systems described thus far essentially concern thermodynamic systems, and are as such systems without purpose. The purpose is situated in a higher systems hierarchical level, what perhaps could be described as the contextual system – what includes the thermodynamic system into systems with a purpose.

From the user, owner and end user point of view, even the installation or liquid chiller unit is only a mean to achieve something else: Something from a systems engineering point of view more abstract perhaps. Adopting this perspective it is the desired function that is the issue. E.g. maintaining agreeable climatic conditions for ones employees; or maintaining a sufficiently low temperature in display cases in a supermarket, so that the quality of the sold products are not compromised. Whether this is achieved by a vapour compression unit or by for example district cooling, is in this respect perhaps less interesting.
In the heat pump example in the section above, the purpose of the heat pump and the installation is to provide heating at a low annual cost for the owner. Heating could be supplied using other means than a heat pump: Oil, gas or bio-fuel fired heating, district heating etc. If the house owner’s ambition is to heat the house at the lowest possible cost, and he or she is faced with the fact that the heat pump has to have its working media (R22) replaced with something else (e.g. R417A, R407C or FX90) with an increase in heating cost as a consequence. It might be interesting to consider alternatives outside the heat pump: Replace radiators for convec-
tors so that lower temperatures of the distribution water is ob-
tained and there by increase the energy efficiency of the heat
pump (probably an expensive alternative), or (which probably is
cheaper) if possible increase the insulation of the outer ceiling, and thereby reduce the annual heating need.

Hierarchical level and conceptual resolution

The systemic model levels that have been described in this chapter are representing different systems hierarchical levels. This may be compared to, and as analogously to, Boulding’s intuitive hierarchy of real-world complexity. The approach of this chapter however, may be seen as more of a systems engineering approach to it. Depending on the aim, scope and the situation, different assumptions as to what is included in the system described may be done. The locus of the systems boundary will be situated differently and with other extensions, spatial and temporal, in different modelling situations for example. In the case of the object systems of concern in a refrigerant phase out it is not self evident what constitutes the individual systems. It all depends on who is looking, and perhaps who it is that formulates what problem or question.

So, how should we conduct systems analysis in a retrofit situation? On what systems hierarchal level should the situation be investigated? There is no general answer as to on what level it should be investigated. Local factors will influence the conditions in such an extent, that the first thing to do is to evaluate is on what level(s) the individual situation should be investigated. Where ought one to make the effort, if the best economy, energy efficiency and maintained functionality are desired for the (end) user? In some cases a change of working media will result in the best result, in other a refrigerant retrofit with an adaptation of the circuitry to better fit the chosen substituting working media, in still other cases it might be best to buy a new unit, etc. (Johansson and Lundqvist 2001b)

![Diagram](image_url)

**Figure 31** A: Scene I-II. B: Scene II-III. C: Scene III-IV. As the heat exchangers are included in the model, R417A seems a better option than when they are not included. R404A is also benefited by the inclusion of rudimentary heat exchanger models, but is still hampered by high condensation temperatures. Using these models R407C is affected by changes in model resolution only to a comparatively very small extent.

---

40 A rather positivistic approach…
Using the different models in scenes I-IV different aspects and phenomena are included – even though rudimentary. In lab environment it is not uncommon that tests are conducted at the same evaporation and condensation temperatures, this strengthens the importance of two issues: Firstly that lab tests as they are conducted are only models of real facilities, and secondly that as long as the model used for predications is not stated it is not possible to use the predictions with any generality. The model used in a study (the extension of the systems boundary – spatial and temporal, the conceptual resolution etc) is only seldom described in the literature – independent of whether it is an experimental or theoretical study. I.e. what model that is used, is taken for granted even though it definitely is not clear and the model chosen has a tremendous impact on the outcome of the study.

**System models used within the paradigm**

It seems as if though the refrigeration community – academia as well as practitioners – has come to gradually accept the models described in scenes I and II. They have become characterising for the paradigm even, and people within the paradigm seem to let them prevail.

One peculiar effect of this is that as lab tests are conducted they are often done so at the same evaporation and condensation temperature. To achieve this, a lot of effort has to be put to actually maintain these temperatures constant. This emphasise the fact that laboratory measurements and how they are conducted, only represent models of real units in real facilities in real applications. And perhaps, when looking at new design situations it works well: If the unit is designed to operate between certain operative temperatures with economically sound sizing of heat exchangers etc, the same compressor will result in different capacity with different working fluids. (As engineering education traditionally is focused on new design and possibly operation this becomes the natural
approach to any person who has passed the educational system.) An existing unit designed for R22, will most certainly behave differently after a refrigerant retrofit. And, in retrofit situations the two models of scenes I and II work less well.

It is evidently hard to change (systems) perspective if you have used one particular systems extension (spatial and temporal), for example, long enough without considering that other perspectives are possible and even more suited for the analysis of a particular situation. (Churchman 1968; Kuhn 1996)

**Emergent phenomena**

A significant attribute of systems is emergence—emergent phenomena. I.e. the impact of a change on systemic level may be much larger than the apparent impact of a change on component level. In the field of interest of the current thesis, emergent phenomena are determining factors concerning the outcome of a R22 retrofit. Jakobsen points out that even though the losses in the compressor represent a relatively large part of the total losses in the vapour compression refrigerating cycle, reducing the heat transfer losses in the condenser and evaporator may not only reduce the heat transfer losses them selves, but also the losses in the compressor (Jakobsen 1995). Reducing the losses in the compressor reduces only the losses in the compressor (more or less) but has a very small impact on the overall losses in the vapour compression refrigeration system as a whole.

At a refrigerant retrofit situation the straight forward positivistic approach is to choose a substitute fluid that renders in the highest possible efficiency, while more or less maintaining the capacity of the retrofitted unit. Looking at a typical liquid chiller unit, choosing R417A not only reduces the capacity of 15 to 20%, but one may expect a reduction in coefficient of performance of a few percents too. R407C on the other side renders in about the same
capacity and only slightly lower COP. That is, if only the refrigerant cycle is considered. Looking at the unit in operation in an air conditioning installation, things become far more complex. If the unit is adapted to the new working media at the retrofit situation, by installing e.g. a liquid-suction heat exchanger, the loss in capacity may be reduced with R417A, but most of all, the COP increases significantly – possibly even to the same magnitude as the unit had using R22.

The impact of emergent effects is clearly visible in the transition from scene III to scene IV and even more clearly in the three cases in scene V. Obviously, as e.g. Jakobsen points out certain improvements of a system will render in larger impact on the system’s integrated behaviour. This may be referred to as gearing factors or gearing effects. I.e. heat exchanger improvements have higher gearing factors than compressor improvements. (Jakobsen 1995) No different from emergence – gearing effects – within a refrigeration or heat pumping machine, emergent effects occur as the systems spatial and temporal extension increase; as the system studied includes more components and longer time frames. If accomplishing the highest possible efficiency (effect efficiency) of a liquid chiller unit after a refrigerant retrofit is the purpose of the study, choosing e.g. R134a as substitute for R22 is a sound choice, as the heat exchanger surfaces becomes relatively larger etc. It is important to point out that the gearing effects, or emergence effects of a system’s componential integration, is something that continues throughout any extension of the systems boundary – as the system of (previous) interest is transferred to, and integrated into, a higher systems hierarchical level: Some improvements of individual components of the system will in fact have higher impact of the integrated system’s performance, or higher gearing factor. As is illustrated in scene VI, doing something to the actual heating need of the house has larger impact on the annual energy usage (energy efficiency in the system with a purpose – the system aimed at maintaining agreeable indoor climate for the building’s
residents) than choosing the substitute refrigerant with somewhat higher annual coefficient of performance than the others.
Experiences: field, lab and simulation

In this chapter of the current thesis not only will the experiences from the field and laboratory be elaborated, but also brief descriptions of the models used for field and laboratory measurements.

Model – measurement – simulation

All measurements are dependent on models. Models on which the theory behind how the measurement equipment has been designed, what is measured and how the measurements are conducted, and models for how to evaluate and analyse the measurement data. Not until the data has been analysed and explained has it actually turned into information. Whether they are results or not, depends on the objective of the study.

Doing measurements on units or components in a laboratory environment enables the use of sophisticated measurement equipment, as well as advanced and highly specialised computer software. Doing field measurements on units in operation in the field is something quite different. Even though the conditions for laboratory and field measurements are very different, they do share some fundamental (often unspoken) models and model assumptions. One such is the idea of measuring evaporation and condensation pressures and interpreting them into the corresponding temperatures. This has its epistemological explanation in the historical use of pure fluids and possibly azeotropic blends. The model used for transferring the pressure readings into corresponding temperatures is the vapour pressure curve, according to which pressure
measurement data is expressed as saturation temperatures according to the vapour pressure curve for the current refrigerant fluid. For pure refrigerants, this works quite well, and manometers are indexed with saturation temperature for a number of refrigerant fluids. These readings are then used in combination with saturation data tables and charts; usually enthalpy-pressure charts.

There are several ways of estimating the cooling and heating capacity of the unit. The perhaps simplest and most exact way is simply to use the bucket-and-watch-principle; possible to conduct with any significant accuracy only when using liquid secondary fluids, and when the inlet respectively outlet temperatures of this from e.g. the condenser is known. The method also needs reliable values on the specific heat capacity \( (c_p) \) of the secondary fluid of concern – such as a water-glycol solution to prevent freezing. (Johansson 1998; Melinder 1997; 1998; 1999) By multiplying the mass flow, specific heat capacity and the temperature change of the secondary fluid, e.g. the dissipated heat in the condenser may be calculated. Using this value and the calculated values of the enthalpies in and out from the condenser of the refrigerant fluid, it is then possible to calculate the mass flow of the refrigerant. In real world field measurement situations this is however hard to achieve. It is easier said than done to measure mass or volume flows of the secondary fluids in a plant operating in or under field like conditions!

In practice other approaches has to be conceived. E.g. by measuring the suction and discharge pressures and temperatures it is possible to estimate the isentropic efficiency of the compressor. Complementing these readings with the electric power supplied to the compressor motor \( (\dot{E}_{el}) \) it is possible to calculate the mass flow of the refrigerant fluid through the compressor \( (m_{\text{refrigerant}}) \), using an energy balance with an assumed (perhaps experienced based) value of the total efficiency of the compressor \( (\eta_{\text{total}}) \). The
mass flow is then used to scale the specific cooling and heating capacity to cooling and heating power.

\[ \dot{E}_{\text{el}} \cdot \eta_{\text{total}} = \dot{E}_{\text{compression}} \]
\[ \text{where } \dot{E}_{\text{compression}} = \dot{m}_{\text{refrigerant}} \left( h_{1\text{comp}} - h_{2\text{comp}} \right) \]

Eq. 35

Field measurements compared to lab tests

It is important to appreciate the conditions for field measurements; how much they deviate from the conditions for laboratory measurements, as the possibilities to conduct elaborate measurements in laboratory conditions are generally not at all present in the field. Doing field measurements on “real” plants, there is simply no possibility to measure an extensive amount of temperatures and pressures, or even doing so very accurately (as compared to what readily can be conducted in a laboratory environment). Simply because the service personnel often cannot handle the equipment necessary nor have the time, and the relevance of it is sometimes questionable. Even if the service personnel could handle the necessary equipment and conduct elaborate analysis work of the measured data, there are generally no means of taking e.g. accurate temperature measurements on “real” machines as there are no temperature pockets present etc. Concerning the easy to use equipment, there are fairly simple to use and commercially available devices to help the man in the field do measurements.

Within the project preceding the current thesis, one such device has been used for all field measurements, the **ETM2000 Refrigeration Computer**, a small device which has its input data supplied from two pressure transducers, a number of thermocouples and measurement of the electric power supply to the compressor.\(^{41}\) It is

\(^{41}\) 'Kyldatorm ETM 2000' by ETM Mätteknik AB in Solna, Sweden.
equipped with a small computer with relevant thermo-physical data available for a number of refrigerants and refrigerants mixtures. To obtain a simple analysis of the refrigerant cycle (e.g. cooling capacity and COP) three temperatures and two pressures has to be supplied (evaporation and condensation pressures, and liquid line, suction line and compressor outlet temperatures) as well as an estimation of the total compressor efficiency. By adding relevant secondary side temperatures, it is possible to calculate overall heat transfer values in condenser and evaporator. This method may not always generate “absolutely” accurate performance data, but it is sufficiently accurate for service personnel and most facility owners, and is the measurement equipment used for all field measurements in the research project preceding this thesis. Therefore, the ETM200 will be referred to here and there in the thesis, but should only be seen as exemplifying this type of equipment.

Doing field measurements on real facilities in real operation one problem is overwhelming: As these are real facilities in real operation, the operational conditions are not constant and it makes it impossible to compare alternatives under the same circumstances. In e.g. Herbe’s study of milk-tanks at dairy-farms it was possible to do adequate comparative measurements as the surrounding conditions were relatively constant and the units had but one purpose at all farms (Herbe 1997). In the current work this has generally not been the case. It would have been possible to include the charts etc from the field measurements; they would however not have contributed to anything valuable for the argument of the thesis as the conditions even for the individual facilities of different tests is so different from measurement to measurement. They have however provided invaluable sources of experience for the author, and sources of interesting questions to be elaborated upon.
The optimum systems charge

– On a warm summer morning in the beginning of June 1997, I was doing measurements after a retrofit of two 125 kW liquid chillers in a telephone station together with Jerry Zetterqvist, at that time a service technician at the telecom company TELIA. After half an hour in the noisy machine room, Jerry said: “But, how do I know how much to charge into the unit? I mean, putting in the same amount in kg’s of Isceon 59 as R22 can’t be right, can it? Should I charge until I get the same subcooling as I had with R22? How am I to know how much to put into each and every one of all the different types of units we have?”

In the event it is decided that an existing plant operating with R22 as working media is to be subject to a refrigerant retrofit, a number of problems may arise. Some are even inevitable! One issue of course pertains to the choice of substitute refrigerant fluid and other as to how the actual retrofit is going to be conducted when the decision of appropriate substitute fluid has been made. When conducting refrigerant field retrofits, one question is thus inevitable: How much of a substitute refrigerant should be charged into the unit? What are the means for the service personnel to establish the appropriate charge?

In early tryouts, comparisons of the substitute performance of three R22 alternatives (R407C, R404A and R417A) field tests were conducted in cooperation with the largest Swedish telecom company during the summer of 1997. The question of finding the optimum systems charge was raised imminently during the initial tests with R417A.

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42 At this time Isceon 59 had not yet obtained its ASHRAE-number, R417A.
43 Zetterqvist (1997-2001)
44 At the time, Isceon 59 had not yet been assigned an ASHRAE-number.
As different refrigerants have a different distribution of their flash and desuperheating losses (cp. chapter 2 and appendix A) they will react differently, cycle performance wise, on subcooling and superheating. Further, their different heat transfer abilities in single- and two-phase heat transfer complicate things further. One way of finding the optimum charge for a specific type (model) of unit, is naturally testing: Increasing the charge (stepwise) slowly and thereby see when the desired performance is obtained as a function of the circuit charge.

**Example 1: Optimum charge for a heat pump**

In some specific cases, it is seemingly easy for a manufacturer to tryout the optimum charge for a specific type of units, for after market care. A manufacturer of domestic heat pumps can be rather sure that a certain historical model of a heat pump is used in a heat pump application and that the operating conditions are predictable. Further, testing one unit and developing procedures for how to conduct refrigerant retrofits covers a large number of units.

![Figure 32](image-url) Heating capacity versus total systems charge. (Measurements made by IVT AB)
From the lab tests presented above, it is possible to set the optimum systems charge for which ever of the tested substitute that is chosen. Information to be spread to the manufacturers service representatives. Another thing that becomes evident is of course that as the COP does not vary that much as the systems charge varies.

**Example 2: Optimum charge for a chiller**

As the measurement readings from the telephone station showed such incoherence in surrounding conditions and generality was sought, a series of laboratory tests were conducted on a well equipped test rig. (The test rig itself was originally designed and built for an earlier research project.) The lab rig has one significant advantage over the real liquid chiller as it is equipped with brazed-plate-type-heat exchangers, whereas the real chiller has tube-and-shell-heat exchangers. In the real chiller, the impact of increased systems charge would come in jumps. In the lab rig changes would come continuously as the charge increased. This would benefit the generality of the results.
Figure 34 Liquid chiller in the lab being charged with R22. The arrow indicates where the condenser is starting to fill up with refrigerant liquid.

Figure 35 As the figure above but with R407C. In this case the inlet temperature difference is calculated from the average condensation temperature at the condensation pressure.
Modelling – computer simulation

What happens when the systems charge change? Looking at the refrigerant cycle alone, what it all boils down to concerns how much sub-cooling should the cycle run with. Depending on whether the throttling-loss of the refrigerant fluid is large or small when operating between two temperatures, it will, refrigerant cycle wise, benefit more or less from subcooling: As subcooling of the condensate is more benign to fluids with high throttling-loss\textsuperscript{45}, it is less beneficial to fluids with smaller throttling loss and thus larger desuperheating-loss\textsuperscript{46}.

In the following graphs with graphical descriptions of models, functions are represented by rectangles and data by parallelograms. Boundary conditions are presented as grey parallelograms and global output data as black. Processes and data exchanged within the models “interiors” are white.

The systems boundary – locus and surrounding

The type of plant modelled is a unitary machine, with compact brazed plate heat exchangers, an open (or semi-hermetic) reciprocating compressor and a thermostatic expansion valve. The unit may be used in any application, but in the simulations presented here, it has operated as a liquid chiller in an air conditioning installation. It has been “designed” to operate with R22 as working media, and in the current application, it delivers approximately 100 kW of cooling capacity. The heat from the condenser is rejected from the cooling water in liquid coolers (probably evaporative cooling towers situated on the roof of the building). The evaporator and secondary refrigerant circuit are configured such that the heat absorbed by the secondary refrigerant and trans-

\textsuperscript{45} E.g. R600a (iso-butane), R404A or R417A.

\textsuperscript{46} E.g. R717 (ammonia), or R32.
ferred to the refrigerant in the evaporator, passes through a circuit with fin-coil batteries distributed over the building.

It is assumed that the machine is under continuous operation and that everything is in equilibrium – a model constraint intrinsic in classical thermodynamics.

![Diagram of Compressor Model](image)

**Figure 36** A graphical description of the compressor function and its boundary conditions.

### Compressor model

There are several ways of modelling the compressor. In the model, it is assumed that the compressor has a constant *stroke volume flow*; i.e. it runs at constant speed. The *compressor function* calculates the volumetric efficiency and hence the *volume flow* and *mass flow* through the compressor as a function of *inlet state* of the refrigerant and the *condensing pressure*. The outlet state of the refrigerant vapour is calculated using a function for the isentropic efficiency of the compressor. Output data are *mass flow* and *state* of the refrigerant, as well as the *power consumption* of the compressor motor.
Condenser model

The condenser model is divided into two sections: A condensation and desuperheating section, and a subcooling section. Each of the two sections is subject to a heat balance restrained by heat transfer behaviour, and results in a balance of area distribution and load (heating capacity in each of the sections as well as on the whole area) between the two sections, as well as condensing pressure. The input data is coolant temperature, mass flow and heat exchanger geometry, as well as the inlet state and mass flow of the refrigerant vapour. Output data are the outlet state and mass flow of coolant and refrigerant liquid, as well as heating capacity.

Expansion valve and evaporator model

The evaporator and expansions device models are strongly interwined as they jointly constitutes an expansion-device-evaporator regulation and feedback system. The expansion device decides not only the inlet state of the refrigerant into the evaporator but it also has strong impact on the outlet state of the refrigerant vapour exiting the evaporator and the evaporation pressure. It is thus im-

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47 The condensing pressure is important input data for the compressor model.
important not to separate these two from each other numerically or conceptually.

The evaporator is, as the condenser divided into two sections: an evaporation section, and a superheating section. Via heat balance and heat transfer limitations, an area distribution between the two sections is calculated, as well as the evaporation pressure. Output data is the state of the refrigerant vapour after the evaporator as well as the state of the secondary refrigerant, brine, at given mass flows of both refrigerant and secondary media.

![Diagram of expansion valve and evaporator model](image)

**Figure 38** A graphical description of the expansion valve and evaporator model and its boundary conditions.

**Integration of sub models – a refrigeration plant**

All of the subsystems are thermodynamically open systems, but as they are integrated and connected to each other, they become part of a model of a thermodynamically closed system: A closed system where the number of boundary conditions is a lot smaller than sum of boundary conditions of the individual subsystems.

The boundary conditions and assumptions for the simulation are the following:
• the compressor is running at constant speed: constant stroke volume flow

• the geometry of the heat exchangers (condenser and evaporator)

• the inlet (or outlet) temperature and mass flow of a given type of coolant

• the outlet (or inlet) temperature and mass flow of a given type of brine (e.g. an ethanol-water mixture of a given composition)

• both evaporator and condenser are modelled such that in the subcooling section, there is only liquid, and no subcooling takes place in the condensing and desuperheating section; in the evaporator, it is assumed that no superheat takes place in the evaporation section and in the superheating section, there is only refrigerant vapour left.

• both heat exchangers operate in counter-flow configuration.

Thermo physical property data for the refrigerants are taken from Refprop 6.01 and property data for secondary refrigerants are taken from EES (pure water) and Brineprop (anti-freeze/water mixtures) (Johansson 1998; Klein 1991-2003; McLinden et al. 1998). For further elaboration concerning used correlations, see Appendix B.
Figure 39 A graphical description interpretation of the simulation model used to investigate the impact of varying systems charge, and its boundary conditions.
**Interpretation of simulation output data**

The data from the computer simulations indicates some interesting but not entirely unexpected behaviour. As the level of standing liquid increase (or amount of heat exchanger area used for subcooling), $X_c$, the subcooling increases. (Cp. figure 41) As R407C is afflicted with a considerable temperature glide, even at the condensing pressures of concern, the driving temperature difference needed to obtain subcooling is considerably smaller than is the case for R22 under the same operating conditions (i.e. boundary conditions in the cybernetic like interpretation above).

![Figure 40](image)

**Figure 40 R22:** A comparison of the resulting subcooling as a function of the level of standing liquid in the condenser, $X_c$. (Results form simulations.)

In figure 41 above, the same patterns as in the tests of the heat pump described earlier in this chapter can be seen in the small variations of momentary-COP. The lab tests showed how the subcooling increased until it “reached” the inlet temperature difference. After that they follow each other. The same pattern as was seen for R407C in the lab tests presented earlier in this chapter, can be seen in figure 42. As in the lab tests, the inlet temperature difference is calculated as the difference between the inlet temperature of the secondary media and the arithmetic mean
temperature of the saturation temperatures at the condensing pressure. The difference between the inlet temperature difference and the achieved subcooling at a given systems charge, represents approximately half the temperature glide of R407C at the condensing pressure.

![Figure 41 R407C: The coefficient of performance, COP, does not vary in particular as the condenser is filled up with liquid. (Results form simulations.)](image)

**Simulation model considerations**

There are obviously conceptual shortcomings in the computer simulation model described above. One such is the assumption that there is no subcooling taking place on the heat exchanging surfaces of the desuperheating and condensation sections of the condenser. Another, the assumption that there is no superheating of the refrigerant vapour taking place in the evaporation section of the evaporator – only in the superheating section. Etc. Apparently these assumptions does not inflict particularly on the qualitative results of the simulations as the comparison with the laboratory measurements show.
The behaviour of the surrounding systems, such as the installation of which the chiller in the simulations is but a part, is placed outside the systems boundary. (The simulated chiller does not affect the rest of the installation. The installation does however affect the chiller.) Quite likely, inlet cooling water to the condenser at least to some extent will be dependent on the heating capacity of the chiller with the considered working media. How the simulated chiller is matched in the rest of the installation capacity wise will of course affect the behaviour of the surrounding systems. However, the tests in the lab have been run in a way that they match the constant boundary conditions of the simulation model. I.e. the laboratory measurement itself is only a model of a “real” liquid chiller, as compared with measurements made on “real” units operating under “real” conditions in “real” installations.

In the simulated situation, no consideration to working media displacement between evaporator and condenser is taken. In two identical units, running under same surrounding conditions, except having different working media (R22 and R407C), the distribution of the refrigerant between evaporator and condenser (or even compressor crankcase) will likely not be the same.

**Recommendations**

The purpose of the studies described above, was of course to be able to give recommendations to the *man in the field* as the question was initially raised by a *man in the field*. recommendations on how to recharge a unit at a refrigerant retrofit situation, using indicators that the service personnel are actually able to measure and comprehend. The recommendation, given in a presentation and published in conjunction with a seminar at the department of Energy Technology at the Royal Institute of Technology, was that to obtain a reasonably “optimum charge” with a new refrigerant, the following procedure should be used:
1. Charge the circuit using standard procedures, until there are no bubbles visible in the liquid line sight glass and the charge is large enough to set the superheat and maintain stable operation (no hunting).

2. Wait a while before increasing the charge until the condensation temperature has increased 1 to 2°C.

As the pressure gauge dials are indexed with the corresponding saturation temperature and the service personnel are used to use the corresponding saturation temperature, and as it is the temperature that is interesting, it is better to use the temperature as indicator than pressure. As elaborated below, shifts in compositions occur due to a number of reasons. Shifts in composition would make temperature readings erroneous even though the pressure readings are correct, the temperature scale is what service personnel are used to look at and refer to. Further it should be pointed out that by looking solely in the sight glass, and not on pressure/temperature readings neither from gauges nor in property charts etc, changes in composition should not contribute to “information noise” to the service personnel when charging a unit. When a minute amount of subcooling has been obtained (no bubbles visible in the liquid line sight glass) and a stable superheat has been set, pressure/temperature readings are taken, and the charge is slowly increased until the corresponding condensation temperature has increased one to two degrees. This will result in other charge amounts for substitutes than for R22 in a given unit. E.g. for R417A it will most likely result in a significantly increased charge, 28 kg’s as compared to 23 kg’s when R22 was used in the Spånga installation. (Johansson and Lundqvist 1998)

As long as the unit of concern is equipped with plate heat exchangers or fin-coil heat exchangers this rule of thumb is safe and easy to use. However, if the unit is equipped with a tube-and-shell condenser with condensation on the outside of the tubes, the
subcooling will not increase continuously as in the lab or computer model, but rather stepwise as tube-row after tube-row is put under the liquid level. I.e. when charging units with tube-and-shell condensers with outside condensation, caution is needed.

**Conclusions – simulation model**

Ashby explains the phenomenon of black boxes\(^{48}\), and according to Ingelstam (Ingelstam 2002), Norbert Wiener defines to the idea of white boxes\(^{49}\). As we cannot know all about the contents of the black box, it is perhaps possible for us to detect its outputs and maybe inputs. (Ashby 1964) It is however possible to design a white box that behaves the same way and thus simulates the behaviour of the black box. The white box can by no means include the same things as the black box does (what ever that may be), but as long as it behaves in the same way we can be satisfied and use it as a model of the black box. This may be compared what is done with the correlations commonly used to calculate e.g. heat transfer coefficients etc. To claim that, for instance, the Dittus-Boelter correlation \((Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4})\) is the reality is not quite reasonable, as it obviously is a correlation that only replicates the behaviour of certain parts of reality, even though both the Reynolds- and Prandtl-number equations appear when writing the equation for conservation of energy and Navier-Stokes differential equation. (Holman 1992; Incropera and DeWitt 1996)(Naturally, both of these equations are only models describing the behaviour of certain domains of the reality – they are not the reality itself!) The Dittus-Boelter correlation as described above is said to be valid for turbulent fluid flow in pipes with a length of at least 60 times

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\(^{48}\) Cp. Ashby (1964) pp. 86-88, 255

the hydraulic diameter, Reynolds-numbers between 10,000 and 120,000 and Prandtl numbers between 0.7 and 120, and moderate temperature differences between the fluid and the pipe wall. Thereby it is said to be valid for a greater variation of pipes, cross sections and cases than made out the empirical material used to derive the correlation. I.e. the Dittus-Boelter correlation may be used for a larger domain of the reality than the empery likely covered, as long as the reality of which it may be used to predict the behaviour of does not deviate too much from the points in reality the empery actually covered.

The computer model yields data that are qualitatively coherent with what is experienced in the laboratory measurements. As the laboratory model, is essentially representative to how liquid chillers, heat pumps etc, are designed, it seems likely that it represents a viable description of how refrigeration and heat pump machinery behaves at least as long as their circuitry layout and operating conditions do not deviate from what could be considered to be normal operating conditions from experience. Thus, the simulation model (a white box) could be used to simulate the behaviour of refrigeration and heat pump machinery in other applications if modified slightly to represent actual dimensioning for the chosen application.
Figure 42 The reality of interest (R) is a liquid chiller unit. The unit is a part of an air conditioning installation, in a telecom station. Neither the nature of the telecom station nor the installation is of interest. Focus is maintained on the chiller unit, and the systems boundary of the measured system is placed around the liquid chiller. The surrounding is defined by the inlet temperature of the secondary media.

Figure 43 A service technician is about to conduct a series of refrigerant retrofits. The technician raise a, to him, critical question: “How much should I charge into the circuitry, to obtain best possible performance?” He conducts a range of measurements, while increasing the systems charge. When measurements are conducted, the chiller is a black box and all measured data are considered output data. Some of the readings are used as input when creating laboratory test rig (L) and a computer simulation model (S). Nota bene, compared to the reality (R) both the lab rig and the computer simulation model are white boxes.
While conducting measurements on the test rig, the computer simulation model is developed and evaluated. What actually is taking place in the computer simulation is parameter variation. The laboratory test rig is used to evaluate the output data from the computer simulation model. When doing this the test rig is a black box compared to the computer simulation model. After evaluation of the computer simulation and lab test data, feedback is provided to the service technician: He is supplied with a rule of thumb.

The evaluation of the computer simulation model shows that it represents the behaviour of the test rig well – at least qualitatively. As the test rig does not deviate in particular from most units and installations used in the applications that may be object of refrigerant retrofits, it is possible to use it for descriptions and qualitative predictions of other units in reality (R). Units in applications that never would be subject to the kind of interest in the current thesis are probably not well described with the current computer simulation model.
Building a predictive model

By refining the model described in scene III in chapter 3, *Systems thinking for retrofitting – The vapour compression machine*, slightly, it is possible to build a tool for prediction and estimation of what performance that may be expected of a unit after a refrigerant retrofit. In the current thesis the model will be referred to as the *desktop retrofit* model, since it is easily implemented in e.g. EES (Klein 1991-2003), and it thus enables the user to conduct “test retrofits” in a computer environment.

**Desktop retrofit operates in two steps:**

First it uses a “white box” model of the “Refrigerant computer” *ETM 2000*, to generate the same types of output data out of the same set of input data as the “black box” *ETM 2000*. These output data are *evaporation* and *condensation* temperatures, amount of *subcooling* and *superheat* (in K), compressor *isentropic efficiency* as well as *overall heat transfer coefficients* in the condenser and evaporator. The *stroke volume flow* through the compressor is also obtained as output data. (Cp. figure 47 below)

The output data from step one are then used as input parameters in step two (into another white box!) where the surrounding conditions are assumed to be conserved from step one: The same *stroke volume flow* and *isentropic efficiency*, inlet (or outlet) *temperatures of the secondary media* are maintained as well as the *overall heat transfer coefficients*. Now, it is possible to alter the amount of *superheat* and *subcooling*, but these are commonly set to the same values that were obtained from the first step. (Cp. figure 48 below)

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50 ‘Kyldatorn ETM 2000’ from ETM Mätteknik AB in Solna, Sweden

51 Cp. black boxes, and white boxes as models of black boxes, as suggested by Ashby (1964) and Wiener as described in Ingelstam (2002).
Within step two it is possible to insert heat transfer degradation factors, $\Omega_i$ (index $i$ indicates condenser (1) or evaporator (2)), such that:

$$UA_i = UA_{R22,i} \cdot (1 - \Omega_i)$$  \hspace{1cm} \text{Eq. 36}

The values of the heat transfer degradation factors may be based on experience – own and others\textsuperscript{52}.

\textbf{Figure 46} Step one in the "desktop retrofit" model: Generating key parameters from measured values.

Relative size of heat exchanger surfaces

A common concept in connection with heat transfer and refrigeration is surface load – the heat load per surface area of the heat exchanger, kW/m². This concept is an important factor of the heat transfer performance in nucleate boiling for example. In the connection to refrigerant retrofitting scenarios, another description of the heat exchanger prerequisites has a greater impact on the outcome of a change working media: The relative heat exchanger surface, $A/\dot{Q}$ [m²/kW]. Principally this is the inverse of...
the surface load, but in practice this is a much more pedagogic tool for understanding the possible outcome of refrigerant substitution.

What is the practical influence of the concept of relative heat exchanger area? The concept of relative heat exchanger surfaces is indirectly connected to another well established concept in practical refrigeration engineering: Economic temperature differences. The
The basic idea of the latter is that it is not economically sound to spend money on large heat exchangers if a unit is expected to have few annual operating hours. An application where this is the typical case in Sweden is air conditioning. Heat pumps on the other hand usually have a large number of annual operating hours. So in this case it is considered economically sound to actually spend a considerably amount of money on heat exchangers. The development within the Swedish heat pump business has over the years gone in this direction – as a consequence of market competition through claimed energy efficiency, but also as a consequence of various heat pump competitions organised by governmental organisation such as the Swedish Business Development Agency, NUTEK.

Changing from R22 to e.g. R134a in a typical liquid chiller in an air conditioning application is described by the move from operating in point A to point B. The same change of working media in a typical liquid-to-liquid heat pump is characterised by the move from operating in point B to point C.
This is can be explained by using the computer simulation model described earlier in this chapter. By decreasing the speed of the compressor, the heat flux in the heat exchangers decrease: the heat exchangers become relatively seen larger. Comparing R22 and a zeotropic refrigerant mixture such as R407C in the same “unit” and maintaining the same superheat (8K) and subcooling (5K), and reducing the speed of the simulated compressor, it becomes evident that the zeotrope requires a larger amount of the heat exchanger areas to maintain the same amount of subcooling and superheating. This behaviour is caused by the smaller available temperature difference between refrigerant and secondary media due to the temperature glide of the refrigerant mixture. As the heat transfer of the single phase refrigerant mixture’s being so small, a large area is needed to maintain the same amount of superheating and subcooling.

Figure 51 R22: As the heat exchangers become relatively larger, a larger amount of the heat exchanger area has to be used to maintain 8K superheating and 5K subcooling. The hatched line represents the area used for subcooling, and the solid line the area used for superheating:
Figure 52 R407C: Using a zeotropic refrigerant mixture, maintaining the same superheating and subcooling as in the chart above takes up more of the heat exchanger surfaces. The hatched line represents the area used for subcooling, and the solid line the area used for superheating:

(The absolute values of the surface distribution are of course only valid for the modelled unit.)

When the relative heat exchanger areas gradually become larger, the temperature difference between primary and secondary media will become smaller. Only to a certain extent though. Eventually the superheating will force the temperature difference to maintain more or less the same as the inlet temperature difference between the refrigerant and secondary media in a counter-flow heat exchanger cannot be smaller than the superheat.

In practice the amount of subcooling in degrees in itself is not interesting in many applications. The subcooling required should in many cases not be larger than that there are no bubbles visible in the liquid line sight glass and stable operation of the expansion device is obtained. Still, as systems charges continue to decrease and the condenser is used often used also as receiver to maintain a
refrigerant buffer in the circuit. A buffer needed to ensure stable operation in a wider range of operating conditions.

As the relative size of the heat exchanger surfaces increase, all substitutes with lower volumetric capacity and/or significant temperature glide lose relative efficiency. R404A however shows the reversed behaviour, and becomes a better choice as the relative heat exchanger surfaces becomes larger. R404A with its relatively low critical temperature is always hampered capacity wise by high condensation temperatures. This is shown in the figure below.

![Figure 53](image_url) By superposing the results from scene V, the following figure is obtained.
Measurement and analysis

– Jerry Zetterqvist, then a service technician at TELIA, called me on the phone one day in the late autumn of 1997. We had continued the refrigerant retrofit tests on the liquid chillers in the telecom station in Spånga, by retrofitting the second chiller to R407C. “Do you know why we got such strange readings during the measurements, last week?” he asked. “The evaporator is leaking on the upper side in one end by the flanges!” It made sense, and was what I already expected…

As phasing out HCFCs such as R22, in new and existing equipment became an issue, an extensive amount of measurements of the performance of various substitutes has been conducted and the results published in journals and on conferences. As many of the suggested and commercially available substitute working fluids are zeotropic refrigerant mixtures, the influence and behaviour of these fluids have been the main concern. The studies encompass both studies of isolated phenomena such as e.g. heat transfer characteristics, and e.g. measures to predict these, and overall system behaviour of units, as well as these behaviours in new as well in existing equipment.

53 Zetterqvist (1997-2001)
A brief note on zeotropic blends

Zeotropic refrigerant mixtures are blends of two or more refrigerants that deviate from perfect mixtures (cp. azeotropic blends). That is, mixtures of refrigerants that do not behave as one pure fluid. One phenomena connected to this is the so-called temperature glide. While evaporating at constant pressure the temperature of the boiling refrigerant mixture increases. In the case of condensation at constant pressure, the temperature of the condensing fluid decrease. Therefore, even the most apparent “peculiarities” of a zeotropic mixture, such as the temperature glide during phase change, present behaviour different from pure or azeotropic mixtures that will render in altered performance of a refrigeration, air conditioning or heat pump unit. This has been further elaborated on in an earlier chapter of the current thesis – Experiences: field, lab and simulation. When using this type of refrigerant mixtures the analysis of the systems performance is complicated by the fact that the circulated refrigerant mixture composition is different from the one the system was, or was thought to be, charged with. To understand and correctly analyse the performance and behaviour of a refrigeration or heat pumping facility, one has to consider these composition shifts. Small deviations in composition may perhaps not affect the general analysis of capacity and efficiency, but can explain e.g. hunting or bubbles in the liquid line, and even apparent heat transfer degradation.

There are a number of reasons as to why the circulated composition of a zeotropic working fluid in a vapour compression heat pumping unit. The obvious and initially probably most common cause to this was erroneous handling of the refrigerants either at the wholesalers and agents during recharging from bulk tanks to smaller service bottles, or at the circuit charging situation. Other possible causes are what could be described as selective solubility of the different constituting refrigerants in the lubricant, and
single phase leakages or deposits in locations of the refrigerants circuit where both liquid and vapour phase are present.

As the circulating composition of the refrigerant mixtures starts to deviate from the nominal composition — as constituted by the manufacturer and the ASHRAE-number designation — some of the thermo-physical data presented in saturation tables and property charts, analysis of the unit’s operational data becomes harder to deal with. Since pressure readings are used to estimate evaporation and condensation temperatures and the new circulated mixtures has another vapour pressure curve than the one of the nominal composition, these readings will be erroneously analysed; leading to further errors in the analysis of the unit’s operational data. Not only will the vapour pressure curve be different: The shape and width of the vapour dome will also change.

**Basic model for composition changes**

A vessel with the volume $V$ contains $n_z$ moles of a refrigerant mixture of $j$ components. The molar vapour quality is $Q$, and the specific volume is $v_z$ such that:

$$v_z = \frac{V}{n_z} \quad \text{Eq. 37}$$

The shares of the individual components sums up to unity; in general, $\psi_i$:

$$1 = \sum_{i=1}^{j} \psi_i \quad \text{Eq. 38}$$
\[
\begin{align*}
\text{Bulk: } & \quad 1 = \sum_{i=1}^{j} \zeta_i \\
\text{Vapour: } & \quad 1 = \sum_{i=1}^{j} y_i \quad \text{Eq. 39} \\
\text{Liquid: } & \quad 1 = \sum_{i=1}^{j} x_i 
\end{align*}
\]

…or specifically:

The mol fraction of the \(i\)th component in the bulk is described by equation 40, which also can be written as equation 41. The mol fraction of the \(i\)th component in liquid phase is calculated as the remainder of the \(i\)th component that is not in vapour phase: Equation 42, which is essentially the same as equation 40.\textsuperscript{54}

\[
\zeta_i = \frac{n_x x_i + n_y y_i}{n_x} \quad \text{Eq. 40}
\]

\[
z_i = x_i (1 - Q) + y_i Q \quad \text{Eq. 41}
\]

\[
x_i = \frac{n_x \zeta_i - n_y y_i}{n_x - n_y} \quad \text{Eq. 42}
\]

The leakage scenario is modelled as being a stepwise leak, such that during each time step \(\Delta n\) moles leaks out. As long as leaked out number of moles in each time step is much smaller than the total charge of the vessel, this approximation is sufficient. The share of the \(i\)th component in the bulk after a time step, \(\zeta'_i\), is described by equation 43 for a vapour leak and by equation 44 for a

\textsuperscript{54} The leakage model as described in this chapter is based on the model described in Kim and Didion (1995a).
liquid leak. It is also assumed that equilibrium and good mixing always appears.

\[
\zeta'_j = \frac{x_i n_x - y_i (n_y - \Delta n)}{n_z - \Delta n} \quad \text{Eq. 43}
\]

\[
\zeta'_j = \frac{x_i (n_x - \Delta n) - y_i n_x}{n_z - \Delta n} \quad \text{Eq. 44}
\]

As \( \Delta n \) moles leaks out from the vessel in each time step, the internal energy of the bulk of the refrigerant mixture, \( u'_\zeta (n_z - \Delta n) \), is described by equations 45 and 46 for vapour and liquid leaks respectively. During the leakage a certain amount of heat leaks in from the surrounding, \( \Delta q_{in} \).

\[
u'_\zeta (n_z - \Delta n) = u_x n_x + u_y n_y - h_x \Delta n + \Delta q_{in} \quad \text{Eq. 45}
\]

\[
u'_\zeta (n_z - \Delta n) = u_x n_x + u_y n_y - h_x \Delta n + \Delta q_{in} \quad \text{Eq. 46}
\]

\[
\Delta q_{in} = U A (T_\infty - T_\zeta) \quad \text{Eq. 47}
\]

Where \( T_\infty \) is the ambient temperature and \( T_\zeta \) the temperature of the refrigerant bulk, such that:

\[
T_\zeta = f(u_\zeta, v_\zeta, \zeta_1, ..., \zeta_j) \quad \text{Eq. 48}
\]
The values of the specific internal energy, \( u \), and enthalpy, \( h \), are calculated using some form of equation of state. Using off-the-shelf software such as \textit{Refprop}\textsuperscript{55}, one has to be aware of the fact that the default values of the mixing parameters may not be entirely correct for some of the component pairs of interest.\textsuperscript{56} Elaboration on this is however, far beyond the scope of the current thesis. The value of the overall heat transfer coefficient for the vessel depends on the geometry of the vessel, etc.

Specific leak scenarios may be modelled from these relations, by assuming certain relations between parameters or boundary conditions. Two fundamental scenarios are described below: Very slow leaks and fast leaks.

**Very slow leaks – isothermal leakage**

If the leakage process takes a significant amount of time, the heat leaking into the vessel, \( \Delta q_{in} \), will be in the same magnitude as the total enthalpy of the refrigerant leaking out each time step, \( h \Delta n \).

\[
h \Delta n \approx \Delta q_{in}
\]

Eq. 49

As the heat leaking into the vessel thus reheats the mixture, the driving temperature difference, \( T_\infty - T_z \), becomes smaller, reducing the heat leaking in until temperature balance is reached. Numerically this may be interpreted as the temperature of the vessel being constant. Consequently the last boundary condition be-

\textsuperscript{55} McLinden, et al. (1998)

\textsuperscript{56} If the studied mixture contains both hydrocarbons and halogenated-hydrocarbons such as HFCs, deviation of the values of the mixing parameters from the default values, \textit{may} affect the validity of the simulation.
comes equation 50, and equations 45 through 47 are not needed to numerically model an isothermal leak scenario.

\[ T'_{\zeta} = T_{\zeta} = \text{Const.} \quad \text{Eq. 50} \]

**Fast leaks – adiabatic leakage**

If it in the leak scenario is assumed that the leakage is fast, e.g. flashing, the total enthalpy of the leaked out refrigerant mixture, \( h\Delta n \), is much larger than the heat flowing into the vessel, \( \Delta q_{in} \). Thus, the fast leak becomes adiabatic, and equations 45 and 46 turns into equations 52 and 53. I.e. the last boundary condition turns into “internal energy of the bulk is preserved and becomes input data for the next time step”.

\[ b\Delta n \not\rightarrow \Delta q_{in} \Rightarrow b\Delta n - \Delta q_{in} = b\Delta n \quad \text{Eq. 51} \]

\[ \{\text{vapour leak}\} \Rightarrow u'_{\zeta} = \frac{u_{x}n_{x} + u_{y}n_{y} - b_{y}\Delta n}{n_{\zeta} - \Delta n} \quad \text{Eq. 52} \]

\[ \{\text{liquid leak}\} \Rightarrow u'_{\zeta} = \frac{u_{x}n_{x} + u_{y}n_{y} - b_{x}\Delta n}{n_{\zeta} - \Delta n} \quad \text{Eq. 53} \]

**Model considerations**

One obvious objection that could (and perhaps should) be raised to at least the adiabatic leakage model is the fact that it in the model is assumed that there is good mixing in the liquid and vapour and that there is thermodynamic equilibrium. This is probably not always the case. However, this type of models is described in the literature and used in off-the-shelf software such as Refleak.
Leak scenarios with R407C

R407C is a three component zeotropic refrigerant mixture that contains R134a, R32, and R125, with the nominal composition 0.52/0.23/0.25 in mass shares, or 0.4393/0.3811/0.1796 in mol shares. Using the leakage models described above, three leak scenario cases will be studied: A number of isothermal leaks, an adiabatic leak, and a case where a second vessel is charged with vapour from the first vessel, and then allowed to be process to another vapour leak.

In all cases refrigerant property data is taken from Refprop 6.01. (McLinden et al. 1998)

Case 1: A number of isothermal leaks

Refrigerant vapour or liquid leaks out from a vessel, containing a two-phase equilibrium of R407C, and the composition of the bulk changes. As it is a slow leak (isothermal) the assumption is made that the remaining vapour and liquid will assume new phase equilibrium. Three different temperatures have been tested (-10°C, 0°C and 10°C), and in each case both liquid and vapour leakages have been simulated. Figure 55 describes how the composition of the bulk, vapour and liquid varies during these different isothermal liquid and vapour leakages at various temperatures; a total number of over 600 composition points.

As the share of the third component (R125 for example) is a function of the two first components, it is possible to plot the share of R134a as a function of R32. Cp. figure 56.
Figure 54 The directions of change in composition of the bulk are indicated with arrows: Vapour leak – black; Liquid leak – white. The grey circle indicates the nominal composition of R407C.

Figure 55 The mol share of R134a as a function of the mol share of R32.
**Case 2: An adiabatic leak**

Figure 57 shows the occurring compositions of vapour and liquid phase during an adiabatic liquid leak starting at an equilibrium temperature of 0°C (i.e. approximately 500 kPa), and continuing until the pressure in the vessel is the same as the ambient pressure, 101.3 kPa. The compositions occurring in this case remains in the vicinity of the compositions occurring at the isothermal leakages mentioned earlier.

![Figure 56](image)

*Figure 56* The black diamonds shows the occurring compositions of the vapour and liquid phase at an adiabatic liquid leakage starting at 0°C. The grey arrow indicates the direction of change in composition of the liquid phase.

**Case 3: Two consecutive isothermal leaks**

In this case we can imagine a small refrigerant bottle being charged from the top (with vapour) from another larger bottle; all taking place at a constant temperature of 10°C. In other words, the leaked out refrigerant vapour is accumulated in a second vessel, that in turn may be subject to a vapour leak process. As figure 58 shows, not only the accumulated composition in the smaller vessel stays in the vicinity of the compositions occurring at the isother-
mal leaks, as shown in figure 56, but the compositions that occur in bulk, vapour and liquid as it is being subject to a vapour leak.

![Figure 57](image.png) Again, the mol share of R134a as a function of the mol share of R32. The grey diamonds indicates the occurring compositions of the bulk, vapour and liquid phase. The black square indicates the bulk composition of the accumulated refrigerant mixture. The black circle indicates the nominal composition of refrigerant R407C.

**Conclusion**

All the compositions that actually seem to occur in any leak scenario place them self on or in the near vicinity of a curve; as long as the original composition was the nominal composition of R407C. This line can be described by a simple polynomial where the mole fraction of R134a is a function of the mole fraction of R32. A curve fit gives the following relation:

$$\psi_{R134a} = -0.62 \cdot \psi_{R32}^3 + 0.88 \cdot \psi_{R32}^2 - 1.7 \cdot \psi_{R32}$$  \hspace{1cm} Eq. 54

In figure 56 it is shown that the compositions occurring in vapour and liquid phase and in the bulk for various leakage scenarios moves along a curve. The point of origin depends on phase and pressure. The direction of the composition shifts depends on...
whether it is the vapour phase or liquid phase, and whether it is a vapour or liquid leak. The compositions however seems to stay, more or less, on the line. Equation 54 describes this line transformed into a function of the mole fractions of R134a and R32, and is valid only for R407C.

To verify the presence of the relation described by equation 54, laboratory measurements were conducted on a specially designed test rig where it is possible to take samples of liquid circulating from and to a refrigerant bottle. The method used is well described and elaborated on by Hill, and Johansson and Lundqvist. The samples have been analysed using gas chromatography. (Hill 1997; Johansson and Lundqvist 1999; 2001a)

The tests have been conducted using a refrigerant bottle charged with refrigerant vapour from a heated bottle that in turn had been charged in an erroneous way by the refrigerant manufacturer’s
agent\textsuperscript{57}. As figure 59 indicates equation 54 still seems to be a reasonably good prediction of compositions that possibly can occur when R407C is subject to any kind of leak scenario, whatever has happened with the refrigerant since it was originally mixed by the manufacturer.

Since it in the leakage simulation models is assumed that there is always thermodynamic equilibrium, it would not have seemed too unlikely that the deviation from the predictions of equation 54 would have been larger – especially for the adiabatic scenario. The two measuring points from Hill’s study of 1997 with the highest concentrations of R32 in figure 59, were samples taken using liquid flashing into a sample bottle. (Hill 1997)

**Selective oil solubility**

Different refrigerants dissolve differently into the lubricant of the compressor. In some cases this is the intention of the design of the mixture, as in the case of e.g. R417A. R417A is a zeotropic refrigerant mixture of R134a, R125 and R600 (n-butane), with the nominal composition, expressed as mass shares, 50%, 46.6% and 3.4% respectively (Calm and Hourahan 2001). R134a and R125 do not dissolve into either of the lubricants traditionally used in combination with R22, so in this case the n-butane (R600) is intended to function as an oil carrier, enabling the continued use of the existing lubricant (mineral and/or alkyl-benzene oil) at a refrigerant retrofit situation. The n-butane dissolves into the existing lubricant and reduces the viscosity of it, enabling it to follow the refrigerant fluid around through the circuitry and returning it again to the compressor crank house. That’s general the idea anyhow.

\textsuperscript{57} Cp. Hill (1997)
Since the n-butane essentially disappears from the refrigerant blend it does not take part as working media in the vapour compression refrigeration cycle. I.e. the circulating composition is hence different than the nominal composition, the composition making out the mixture represented in the property tables and charts supplied by the manufacturer. The new circulating working media composition turns from R134a/R125/R600 (50/46.6/3.4)$^{58}$ into R134a/R125 (51.8/48.2)$^{59}$. This might seem a small difference but it is large enough change the shape of the vapour dome and its relation to isotherms etc.

### Apparent heat transfer

In this section an indirect method to indicate deviations in the circulated composition of a zeotropic refrigerant mixture$^{60}$. In this Gedankeexperiment R407C will be used as working fluid in a liquid chiller operating in an air conditioning installation. The Gedankeexperiment is conducted in three steps: In the first step the plausible compositions are generated; in the second they are used in the chiller computer simulation model, and measurement data are generated; and in the third step the measured data are analysed assuming that the circulated composition is the nominal composition of R407C.

As will be shown, the consequence of using the nominal formulation as circulated composition does not affect the values of capacity or COP. When trying to understand the behaviour of the unit, and trimming operation parameters such as superheating, things become more complicated, and perhaps even confusing. If capacity and COP are considered as effects of the first order (things the

$^{58}$ R417As mean molecular mass is 106.7 kg/kmol.

$^{59}$ R417A without R600 has a mean molecular mass of 110 kg/kmol.

$^{60}$ Cp. Lundqvist (1999)
owner of the machine may be interested of), superheating and heat transfer are effects of the second order (thing the service technician may be interested in for trimming and understanding).

**Step 1: Generating plausible erroneous compositions**

In this *Gedankenexperiment* the reasons for the particular shifts in compositions simply do not matter. It is the circulated compositions as such that are interesting. The computer simulations have been run with nine different compositions that could occur due to a number of reasons.

<table>
<thead>
<tr>
<th>Run. no</th>
<th>$\xi_{R134a}$ [kg/kg]</th>
<th>$\xi_{R32}$ [kg/kg]</th>
<th>$\xi_{R125}$ [kg/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.62</td>
<td>0.17</td>
<td>0.21</td>
</tr>
<tr>
<td>2</td>
<td>0.59</td>
<td>0.19</td>
<td>0.22</td>
</tr>
<tr>
<td>3</td>
<td>0.57</td>
<td>0.20</td>
<td>0.23</td>
</tr>
<tr>
<td>4</td>
<td>0.54</td>
<td>0.21</td>
<td>0.24</td>
</tr>
<tr>
<td>5</td>
<td>0.52</td>
<td>0.23</td>
<td>0.25</td>
</tr>
<tr>
<td>6</td>
<td>0.49</td>
<td>0.24</td>
<td>0.26</td>
</tr>
<tr>
<td>7</td>
<td>0.47</td>
<td>0.26</td>
<td>0.27</td>
</tr>
<tr>
<td>8</td>
<td>0.44</td>
<td>0.27</td>
<td>0.29</td>
</tr>
<tr>
<td>9</td>
<td>0.41</td>
<td>0.29</td>
<td>0.30</td>
</tr>
</tbody>
</table>

*Table 6* The nine different compositions that the simulated chiller has been operated with. These are shifts in composition that are not particularly extreme, and very well could appear in real situations. The nominal composition is the one used in run number 5. How these very compositions have been obtained is not interesting for the sake of the model. (Didion and Kim 1998)

In the first four runs the circulated composition is enriched in the least volatile component, R134a. In the last four runs it is enriched with the two most volatile components, R32 and R125. In the fifth run it operates with a composition identical with the nominal composition of R407C.

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61 Note that these compositions are presented in mass-fraction, as compared to the mol-fraction of e.g. figure 59 and equation 54.
Step 2: Simulation of operation and generating data

The model of the liquid chiller operating in the air conditioning installation is based on a real liquid chiller, and the real liquid chillers operating conditions are used\textsuperscript{62}. It is however downsized in capacity from approximately 125kW to approximately 8kW.

In the simulation the inlet temperature of the cooling water to the condenser is 40°C, and the inlet temperature of the secondary refrigerant (also water) is 8°C. It operates with constant superheating and subcooling of 7K respectively 4K. The stroke volume of the compressor is 10.6m³/h, the isentropic coefficient of the compressor is 70 % and the volumetric efficiency of the compressor is 100 %(!). As most liquid chiller in air conditioning applications in Sweden, it has relatively small heat exchanger surfaces. In the simulation model, it is assumed that the overall heat transfer coefficients are 1 kW/K in both the evaporator and condenser; referred to the inlet temperature differences.

\begin{tabular}{|c|c|c|c|c|c|c|}
\hline
Run. no & Electric power to comp. [kW] & $p_1$ [kPa] & $p_2$ [kPa] & $t_{2C}$ [$^\circ$C] & $t_{1C}$ [$^\circ$C] & $t_{1S}$ [$^\circ$C] \\
\hline
1 & 3.11 & 2052 & 424 & 7.4 & 88.3 & 46.5 \\
2 & 3.19 & 2107 & 433 & 7.3 & 89.2 & 46.7 \\
3 & 3.27 & 2162 & 442 & 7.2 & 90.2 & 46.9 \\
4 & 3.36 & 2218 & 451 & 7.1 & 91.1 & 47.0 \\
5 & 3.44 & 2275 & 461 & 7.0 & 92.0 & 47.2 \\
6 & 3.53 & 2333 & 472 & 6.9 & 92.9 & 47.4 \\
7 & 3.62 & 2393 & 482 & 6.7 & 93.8 & 47.7 \\
8 & 3.71 & 2453 & 493 & 6.6 & 94.7 & 47.9 \\
9 & 3.81 & 2515 & 505 & 6.5 & 95.6 & 48.1 \\
\hline
\end{tabular}

\textbf{Table 7} The measurement readings a service technician would obtain if measurements had been made on the liquid chiller with the nine different circulated compositions of the previous table. Temperature indexes represent the readings of suction gas temperature (2C), hot gas (1C) and the condensate (1S)\textsuperscript{63}.

\textsuperscript{62} Zetterqvist (1997-2001)

\textsuperscript{63} The simulated measurement data are generated using the computer software EES, Klein (1991-2003), with refrigerant property data obtained using Refprop, McLinden, et al. (1998). The compressor is assumed to have an adiabatic efficiency of 94%.
The data measured corresponds to the data a service technician could obtain using e.g. the ETM2000 Refrigerant Computer. See table above. The service technician has also measured the inlet temperatures of the secondary media.

**Step 3: Making the “erroneous” analysis**

Using equipment like ETM2000 Refrigerant Computer to analyse the performance of the unit, the nominal formulation of R407C is assumed to be the circulated composition. As it is not the analysis of cooling capacity and COP₂ may be slightly wrong.

![Figure 59](image)

**Figure 59** As the circulating composition is enriched in the least volatile component (R134a) the value of the capacity will seem higher than it actually is. For the cases when the circulating composition is enriched in the two most volatile components (R32 and R125) the opposite applies. The differences are however small.⁶⁴

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The values of COP₂ obtained from the ETM2000 Refrigerant Computer shows that it is higher than it actually is. ⁶⁵

Figure 60

The values of the apparent superheating and subcooling deviates significantly from the real values, 7 and 4 K. ⁶⁶

Figure 61

⁶⁵ Refrigerant property data are taken from McLinden, et al. (1998).
⁶⁶ Refrigerant property data are taken from Ibid.
When the service technician looks at the values of the superheating and subcooling, things start to seem perhaps somewhat strange. The compressor also seems to operate either poorly or splendidly, depending on what components the circulating composition is enriched in. One consequence of this is that had the actual subcooling been smaller than 4 K, it would still seem relatively large if the circulated composition is enriched with the two more volatile components. Especially when R407C was starting to being used in both new and existing machinery in Sweden, it was often reported, and rumours spread in the refrigeration and heat pump business, that you could still see bubbles in the sight glass in the liquid line even though the subcooling was considerable, and that that was quite normal and okay.

Figure 62 The values of the apparent isentropic efficiency of the compressor will vary as the composition varies. The real isentropic efficiency is actually 0.70 and in the simulation model independent of composition etc.\textsuperscript{67}

\textsuperscript{67} Refrigerant property data are taken from Ibid.
As the values of the evaporation and condensation temperatures, \( t_2 \) and \( t_1 \), obtained using this type of analysis, are calculated as functions of the measured corresponding saturation pressures, \( p_2 \) and \( p_1 \), deviations in the circulated will consequently lead to erroneous estimations of these temperatures.

![Figure 63](image)

**Figure 63** As the evaporation and condensation saturation pressures varies as the circulated composition decrease or increase compared as the composition shifts, the interpretation of them into the corresponding temperatures varies even more.\(^{68}\)

**Using the “wrong analysis” to make the right analysis**

It is understandable that all this may seem a little confusing to a service technician in the field. The natural assumption from their part is that even though the units perform as they would expect concerning the delivered capacity and efficiency, they are still very tricky to apply the correct amount of charge into, some times the superheating has to be set to be very large to obtain stable operation, bubbles in the sight glass even though the subcooling seems very large, etc.

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\(^{68}\) Refrigerant property data are taken from Ibid.
It is however possible to use the analysis data from the analysis made with the assumption that the circulated composition is the nominal composition of R407C as an indirect method to indicate changes in composition. That idea of the method is to actually use the errors in the “wrong” analysis to make the right. (Lundqvist 1999) By studying the overall heat transfer coefficients, and how they vary over time in a unit, it is possible to get at least an early warning that the unit is leaking and that the circulated composition is not the nominal, that there is some sort of problem in the process of charging units in the production line if the heat transfer coefficients seems to vary between identical units, etc. For example, if the UA-value of the condenser seems to be increasing over time in a unit, it is very possible that there is a liquid leak or accumulation in the evaporator – the circulating composition is enriched with R32 and R125.

**Figure 64** The apparent overall heat transfer coefficients as a function of the composition. The real UA-values are always 1 kW/K in the simulations of the liquid chiller.
**Reflection on the indirect method and the literature**

In the literature it is often mentioned that the heat transfer of zeotropic refrigerant mixtures in fact becomes considerably lower than what was experienced with the same equipment using e.g. R22\(^69\). Not only is this a question of lower volumetric capacity (as in the case of R417A) so that the pool-boiling component of the heat transfer might become lower but also a result of the fact that in some heat exchanger configurations there is an imminent risk that some part of the newly evaporated refrigerant re-condensates further down the evaporator, and consequently has to evaporate again. The presence of composition gradients in the in the liquid will also have a hampering effect on the heat (and mass) transfer. (Granryd 1991; Rohlin 1996)

Looking at figure 65 it is evident that the composition of the circulated refrigerant mixture does not have to deviate that much from the nominal – a few percents enrichment of the more volatile components only – for the condenser to seem to be operating a lot poorer than it actually is. If the unit of study is a machine with relatively large surfaces in the heat exchangers the temperature differences will be small to start with, say 4-5 K in inlet temperature difference. If the condensation temperature seems to be 1-2 K higher than it actually is, the heat transfer coefficient will appear to be, say 20-40 %, lower than it actually is. Surprisingly seldom it seems is any heed paid to possible composition shifts in the literature: Neither in field tests (or field like laboratory tests) or laboratory tests of unit performance as comparisons of e.g. different refrigerant choices, or even laboratory tests with the aim to

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evaluate the validity of generic heat transfer correlations using zeotropic refrigerant mixtures.\textsuperscript{70}

Many would argue that as long as the circuit is charged with liquid phase, using the liquid line connection on the source refrigerant bottle and there are no (obvious) hold-ups or leaks in the circuit, it should not be a problem. But as some equipment manufacturers have noticed in their operation, cracks in the liquid lines in refrigerant bottles is common. (Kronström 2002) When charging a circuit using the liquid line connection of the refrigerant bottle, vapour phase refrigerant from the top of the bottle is still charged into the circuit, even though the service technician may be doing everything by-the-book.

**Consequences of different secondary media**

In the general case the heat transfer degradation due to zeotropicity does effect the temperature difference between the primary and secondary refrigerant fluids even though the volumetric capacity at given temperatures etc. of the substitute refrigerant are very close to that of R22 – as is the case of R407C\textsuperscript{71}. Whether this heat transfer degradation in fact does affect the levels of evaporation and condensation temperatures significantly or not depends on the performance of the secondary side rather than that of the primary side of the heat exchanger.

In units with direct evaporation and condensation in fin-coil heat exchangers, the method may actually work reasonably well whereas it does not work at all in the case of indirect configura-

\textsuperscript{70} E.g. Aprea (2000), Boissieux, et al. (2000a), (2000b).

tions: Looking at the number of units, direct expansion machines and systems are perhaps the most common design. In this case the secondary fluid is air on both evaporator and condenser sides. This type of systems configuration is common in e.g. USA and Canada. Heat transfer coefficients on the airside are comparatively low, very low compared to the primary side. The magnitude of heat transfer coefficients on the primary side is, say, 1 000-2 000 W/(m², K), whereas the magnitude of the secondary side is in the range of 10-30 W/(m², K). Considering the fined area, and rating it down to the area of the inside of the tube, the magnitude reaches, say, 300 W/(m², K). Actually, a heat transfer degradation of 30% compared to R22 on the primary side becomes relatively insignificant, only 6-7% on the total overall heat transfer coefficient, since virtually all the thermal resistance is on the secondary fluid side. I.e. if the volumetric capacity is equivalent to the target (R22), even a large reduction in heat transfer coefficient on the primary side does not change the evaporation and/or condensation temperatures significantly, and to make predictions using the same evaporation and condensation temperatures will result in sufficiently good predictions. If, however, the secondary fluids have heat transfer coefficients in the same magnitude as the primary side, as is the case of water and other liquid secondary refrigerants and coolants, it does matter. Installations of this type are becoming more common since more systems are built as indirect systems, and are the most common systems design in e.g. Scandinavia. In these cases the evaporation and condensing temperatures of the substitute deviates (considerably) from that of R22, and using the values of the same temperatures yields results and predictions of no, or at least small, value.

72 Degradation of 30% on the primary side from 1500W/(m², K) with R22, and 300W/(m², K) on the secondary side (projected to the inside area of the tubing).
Estimating composition online

The traditional way of finding the composition of the circulated mixture has been taking samples, which then has been analysed using gas chromatography. As is shown by Hill (1997) doing this with a high degree of accuracy and repeatability is more complex than can be readily managed by the average service technician in the field. (Hill 1997)

A self sensing circuit for capacity control and correcting the superheating in multi-evaporator-direct-expansion air conditioning units using R407C was developed within a large Japanese manufacturing corporation in the mid to late nineteen-nineties. (Sumida et al. 1998) Their solution is an ad hoc solution to solve their particularly problem, but can be modified and has been developed to a general method. (Johansson and Lundqvist 1999; 2001a)

Online measurement of circulated composition

The enthalpy of the zeotropic refrigerant mixture may be calculated using temperature and pressure, together with the share of each component in the mixture, as input data. If the circulated composition is different from the nominal composition of e.g. R407C, and the nominal composition is used as input data in e.g. Refprop (McLinden et al. 1998) the calculated value of the enthalpy after the expansion device will be either larger or smaller than the calculated value of the enthalpy before the expansion. E.g. if the circulated composition is enriched with R134a, the calculated value of the enthalpy after the expansion will be larger than the calculated value of the enthalpy before the expansion device. The opposite applies for a case where the circulated composition is enriched in R32 and R125.

By measuring the temperature and pressure of the refrigerant before and after the expansion device – $t_1s$ and $p_1s$, and $t_2s$ and $p_2s$ respectively – it is possible to estimate the circulated composition
in a refrigeration or heat pump facility. This is done by the solution of an equation system as follows: R407C consists of three components (as does R417a and R404A), R134a, R32 and R125, consequently three equations are needed. The first equation states that the enthalpy of the refrigerant is the same before as after an adiabatic expansion – equation 55. The values of the enthalpies are obtained using Refprop (McLinden et al. 1998). The second equation is equation 56, and the third is equation 54 (shown again below).

\[
\psi_{R134a} = -0.62 \cdot \psi_{R32}^3 + 0.88 \cdot \psi_{R32}^2 - 1.7 \cdot \psi_{R32}
\quad \text{Eq. 54}
\]

\[
b_1 (p_1, \rho_1, \psi_{R134a}, \psi_{R32}, \psi_{R125}) = b_2 (p_2, \rho_2, \psi_{R134a}, \psi_{R32}, \psi_{R125})
\quad \text{Eq. 55}
\]

\[
\psi_{R125} = 1 - \psi_{R134a} - \psi_{R32}
\quad \text{Eq. 56}
\]

The equation system may now be solved, using for example the computer software EES together with Refprop (Klein 1991-2003; McLinden et al. 1998).
A test rig has been built to evaluate and verify the method described above. In the test facility the somewhat sub-cooled refrigerant liquid is expanded through a needle valve and capillary tubing; temperatures and pressures are measured before and after the expansion. The refrigerant vapour formed during the expansion is condensed in a small plate heat exchanger. The refrigerant liquid is sub-cooled further before reaching the pump. From the pump the refrigerant is lead back to the refrigerant vessel. The refrigerant vessel shell is heated by an electrical heater at half the vessel height to obtain a satisfying pressure on the high pressure side. The bottom of the refrigerant vessel is cooled to make the refrigerant liquid in the bottom somewhat sub-cooled before entering the expansion device. Cp. figure 66. With the six-way valve indicated in figure 66 it is possible to take small liquid samples of the circulated refrigerant liquid: As the fluid is pumped trough the
valve it is possible to turn the valve, and seal a small amount of refrigerant liquid in an evacuated capillary tube. This small amount of refrigerant liquid is flashed into an evacuated sample bottle. To be sure that no refrigerant mixture residues are left in the capillary tube, high pressure nitrogen gas is pushed through the tube into the sample bottle. The sample bottle (with the diluted refrigerant sample) is attached to the appropriate port on a gas chromatograph and the content is analysed according to standard procedures. (Hill 1997) In the same rig the online method described above is also applied, and takes measurements of temperatures and pressures, before and after the expansion device, on the same refrigerant mixture continuously. (Johansson and Lundqvist 1999; 2001a)

![Figure 66](image)

**Figure 66** The tests conducted to verify the reliability of the online method shows that the values obtained with it does not deviate from what is obtained through gas chromatography of samples more than 2-5% absolute of R134a – about half of that with R32 and R125. The same rig and refrigerant batch has been used to verify equation 54.

**Magnitude of the error-margin**

In the figure above ±5% absolute error lines are drawn. I.e. if the online method yields 50% R134a as result, the true composition is
somewhere between 45 and 55%. As the figure shows, the actual error margin is lower than ±5%.

**Concerning this method, oils and oil solubility**

As the different components of a multi-component mixture may have varying solubility in the used lubricants, the composition might change in ways not covered by equation 54. The online method described above would have to the complemented with some form of oil-concentration sensing device. There is a few such suggested working with interferometers or other indirect methods to estimate the oil concentration. (Lebreton and Vuillame 2000; Newell 1996) This is however something that has not been done within the research project foregoing the current thesis.

**Slow and fast leaks – reflection**

The leakage model as described in this chapter is based on a set of critical assumptions: One there is good mixing in each of the two phases, and two, the phases are always in thermodynamic equilibrium. This can of course be questioned, especially in the adiabatic leak scenario. However, practically, adiabatic leaks occurring in units under operation usually cause breakdown. Most leaks that can cause problems (instability, seemingly bad performance etc) are slow and take place under more or less isothermal conditions: In connection to evaporators or condensers. The causes may be leaking flange couplings, leaking gaskets in the gables of tube-and-shell heat exchangers etc. Under these conditions the mixing can be assumed to be (very) good.
Retrofitting R22-applications

This chapter is aimed at consolidating the findings presented previously in the thesis. The consolidation will be represented by three discussions: Firstly, a discussion about at what level the energy efficiency is found, and how the windows of opportunity increase as the perspective includes a wider and wider system. Secondly, a set of explanatory models to answer why R407C is reported to be running without hunting, with only a very small degree of superheating, are presented. And thirdly, a discussion on why things go wrong as reported in the report from Folksam (Folksam 2003).

A decision making aid

It is possible to come far in maintaining capacity and efficiency – even increasing the momentary-COP for the unit at the system hierarchical level in the first sub-section of the section below. As an aid for individual facility owners and service contractors etc, a decision tree can be used. A decision tree not aimed at giving absolute recommendations for how individual plants should be retrofitted, but to present a “thought style”.

Firstly, the application the unit is used for has to be considered. Is it an application where its part essentially concerns providing cooling or heating? I.e. is it a refrigeration or heat pump unit? Which one of these it is will practically govern the next level in many cases.
Figure 67 The decision tree: a tool to be used as a “thought style” when considering suitable substitutes for R22 in individual units and applications.
Secondly, the question of the unit’s capacity relative to the demand of the application has to be considered. Is the unit oversized as compared to the need, is it under sized capacity wise, or is the capacity the one absolutely necessary capacity to cover the applications need? In case of refrigeration and air conditioning equipment in Sweden the capacity is generally higher than the required capacity to cover the demand. Domestic heat pumps are, as is mentioned in the next section, usually under sized capacity wise, but during periods they have been designed to cover the maximum momentary-heating demand.

Thirdly, how are the heat exchangers (the evaporator and condenser) designed? Are they large or small? Are the temperature differences between the primary and secondary media small or large? This is an issue that is connected to Bäckström’s economical temperature differences: Units that are over dimensioned capacity wise have short annual operating hours. Thus, it is not economically sound to invest a lot of money in large heat exchangers for units like these.

How the “thought style aid” can be used will be exemplified with three cases integrated in the section below.

Where is energy efficiency?

As the systems perspective changes as the system boundary expands spatial and temporal; and as the systems boundary expands, the time frame required for evaluating the system increase. Thus, new things become interesting…

The unit – a common perspective

Looking at the unit alone, putting the systems boundary around the unit and perhaps include some rudimentary features of the secondary sides, the temporal extension is short. Classic quasi
static thermodynamic systems descriptions operating under steady state: the temporal extension is so short that the time derivatives of energy – heat and work – are considered. This definition of the coefficient of performance is not energy efficiency, rather effect efficiency. Assuming this type of perspective, the options at hand in a retrofit situation are limited to one, chose the substitute refrigerant that yields the highest capacity or highest COP, what ever is considered most important, or two, chose a substitute that may not be perfectly suited for the circuit design and alter the design. These alterations may consist of installing a liquid suction heat exchanger and chose R404A or R417A. As they are burdened by high throttling losses, the effect of this circuit change increases the cooling capacity without the loss of energy efficiency. See, we have already lifted our eyes from the refrigerant alone being the issue.

Figure 68 Systems hierarchical level: Unit

Case 1: An air conditioning liquid chiller

An air conditioning chiller with extensive over-capacity is to be retrofitted from R22 to some alternative refrigerant or refrigerant mixture. As it is over sized capacity wise and still sensibly designed internally, it is equipped with fairly small heat exchangers. The service technician will notice this when the temperature differences are measured. Since the heat exchangers are small, choosing a substitute with less volumetric cooling capacity is a sound choice
if the highest possible momentary-COP is sought. I.e. replacing R22 with R134a will result in a significantly higher momentary COP than what was the case when R22 was used. Further, as the momentary capacity is reduced the operating hours increase, leading to smaller annual integrated start-up losses.

The installation – components working together

The next systems hierarchical level includes the installation as a whole: A systems boundary extension (spatially) where the liquid chiller or heat pump unit is but a part of a larger system. This implies that the behaviour of the unit will both affect and be affected by the behaviour of the liquid coolers on the building roof etc. The temporal extension has also increased as the time constants of the installation are so much larger than the time constants of the vapour compression unit. It is no longer the time derivatives of the energy that are studied, but their integrated values – heat and energy.
Here it is important to point out a fact. In Sweden (at least) there is a fundamental difference between heat pumps and refrigerating equipment (including air conditioning chillers for instance). Domestic heat pump units are (generally) dimensioned to cover a certain percentage of the maximum heating need, rendering in only partial coverage of the annual heating need – for example 50% heating capacity coverage leads to approximately 80% heating energy coverage, and 30% heating power coverage to typically 70%.

There have over the years been different trends how large the “optimum” energy coverage rate should be (Forsén 1997-2003; Kronström 2002). So, over the years the “under size rate” of domestic heat pumps varies. But there is always an alternative to the heat “produced” by the heat pump: direct electric heating etc. Cooling equipment is as a rule over dimensioned capacity wise. There is of course a simple reason for this, to complement a heat pump’s insufficient capacity you can always add e.g. electric heating, whereas you cannot use any other mean to generate the necessary cooling. You can use free cooling in some applications, but never to handle peak load, only base load. A consequence of this is that heat pumps have longer annual operating hours and refrigeration and air conditioning equipment has shorter operating hours. To this it should be added, that much of the refrigeration equipment is even further oversized as contractors and engi-
neering consultants add capacity “just to make sure the installation can handle a larger future demand”. Refrigerating units having 200-400% capacity compared to the maximum need do occur (Weber 1997-2003; Zetterqvist 1997-2001).

Using this perspective the windows of opportunity multiply: Things may be solved outside the refrigerating unit. For example, even though a change from R22 to R417A in a unit reduces the capacity of it, this is sometimes something that can be handled as the original capacity was a lot higher than the maximum need. Further, reducing the temperature lift the unit has to operate with will increase the efficiency of it. This is something that can be obtained by reducing the return temperature of the cooling water to the condenser. Many units operate with fixed condensation temperature; this feature can be removed to let the condensation temperature float and allow reduce condensation temperatures at times when the cooling water has a lower temperature. Etc…

Case 2: A domestic heat pump

A domestic heat pump in a one-family house has to have its present refrigerant, R22, replaced with any of the commercially available substitutes. The house was retrofitted from oil-heating to a ground coupled heat pump in the late 1990’s, which was dimensioned to cover approximately 50% of the maximum heating demand, resulting in an annual heating energy coverage of 80%. The unit has long annual operating hours and partly because of this it is equipped with large heat exchangers. Choosing a substitute resulting in considerably smaller momentary-heating capacity will not lead to increased momentary-COP of the unit nor increased annual energy efficiency, as discussed in chapter 3. The most important issue is not to lose momentary-heating capacity. Any reduction in this will lead to increased need for auxiliary heating, usually with electric heaters that in turn results in lower annual energy efficiency.
Figure 71 Case 2: A typical domestic heat pump – large heat exchangers and under-capacity.

The application – the system with a purpose

If a higher system hierarchical level, the system with a purpose, is considered, the windows of opportunity multiply. Not only are the possible routes from lower hierarchical levels available, but a set of new paths for the individual facility owner to thread presents them selves. To “simply” change working media is but one opportunity available for the facility owner.

Some estate owners have chosen to replace all mechanical refrigeration units for district cooling where this is available. I.e. having a liquid chiller generating the necessary cooling capacity is no purpose in it self. It is the function, or effect, of it that is necessary.

Another way of handling this issue is of course to try to reduce the demand of space cooling in a building. This can be obtained by installing e.g. reflecting film on windows or some form of sun blinds to reduce the impact of solar irradiation; i.e. nothing new or even very radical.
Case 3: Another air conditioning liquid chiller

Even though most air conditioning units are over-dimensioned capacity wise the technical development has led to significantly increased cooling demand. E.g. most tele- and datacom-stations have historically been furnished with cooling capacity to handle approximately 500 W/m². An increase in data-communication services, both commercial and private, has caused the cooling need to increase considerably: As high values as 1 500 W/m² has been estimated in a near future if not the technical development in data-communications hardware reduces its energy need to a very great extent (Zetterqvist 1997-2001). This development may cause the liquid chiller in the current case to become optimised- or even under-dimensioned capacity wise.

There are at least two ways of handling the refrigerant substitution from R22 to any of the commercially available alternatives: Firstly,
it is probably not a bad idea to choose a substitute with significantly lower volumetric cooling capacity such as R134a. This would, as has been mentioned earlier, increase the momentary-COP of the unit. However, loosing say 30% cooling capacity would lead to an even larger under-dimensioning. This can be solved by installing yet another liquid chiller to compensate for this. Secondly, an economically sound solution can very well be to scrap the old R22-unit and simply by a new machine. …or, retro-fitting the installation to primarily cover its cooling demand with free-cooling using boreholes as they sometimes do in their new installations, or installing district cooling where available. The list of possible economically and environmentally sound possible decisions grows long…

No hunting with only a small superheat

It has been mentioned earlier in the thesis that R407C has been reported to be running without hunting with only a very small degree of superheat. This is something that has been reported from early on when R407C started being used and service technicians
got in contact with it. One explanation was presented quite soon. This is the second explanatory model that will be described in this section, and concerns issues not previously treated in the thesis.

1. Not nominal composition – not the real superheat

As is shown in chapter 5, the circulating composition does not have to be more enriched with the two more volatile components than a few percent to increase the systems pressure enough for the evaporation temperature to seem higher than it actually is. Hence, if the evaporation temperature seems to be higher than it actually is, the superheating will appear to be smaller than it actually is – if the superheat is calculated from evaporation pressure readings and suction gas temperature, and the pressure reading is translated to dew-point temperature using e.g. property tables or charts for R407C with the nominal composition. I.e. the real superheating is in fact not as small as it may seem.

2. Larger initial driving temperature difference

The second explanatory model has conceptually not been treated so far in the thesis, and covers phenomena connected to the temperature glide.

As R407C enters the evaporator it will have a lower temperature than R22 – in a given unit in a given application. This will lead to a larger temperature difference between the secondary media and the refrigerant fluid. (For this discussion we can assume that the heat transfer coefficient is the same for R407C as for R22 in the given unit in the given application.) The larger temperature difference will lead to a larger amount of the total heat transfer (larger surface load) taking place in the beginning of the evaporator, leaving less refrigerant to actually evaporate in the end. Consequently there will be less refrigerant liquid to create mist and droplets in the vapour flow. Fewer liquid droplets would then lead to a re-
duced need to superheat the vapour to get rid of said droplets and mist.

The argument against this would be that a larger amount of refrigerant liquid evaporating in the beginning of the evaporator would lead to smaller temperature differences in the end of the evaporator due to, one, the temperature glide makes the dew point temperature typically 3-5K higher than the inlet temperature and, two, less refrigerant liquid remains to evaporate. Thus a longer stretch of the evaporator would be handling less cooling capacity, which in turn would lead to lower surface load leading to poorer heat transfer etc. This would probably cause the boiling front to remain in more or less the same place. As has been shown in chapter 4, this would lead to a smaller superheat for R407C than for R22.

3. Minimum stabile signal – another way of seeing it

As the concept of the Minimum Stabile Signal, or MSS-line, is taught to students in refrigeration technology it is referred to as the minimum superheating in K that is required to maintain stable operation for a range of cooling capacities with a given expansion valve nozzle. I.e. it is referred to as a temperature difference. If the system thermostatic-expansion-valve-and-evaporator is considered as any regulatory system with a feedback loop, the explanation becomes different. In this case the system is inherently unstable. Cp. a circuit operating with a capillary tube or fixed orifice expansion device. The thermostatic expansion valve has a regulatory time constant that is significantly shorter than the evaporation system’s. If the system operates with a too small superheat the feedback loop will give positive feedback and thus amplify any cyclical boiling front movements; and vice versa when the superheat is large enough, the feedback is negative and as such suppresses any inherent cyclical boiling-front movements.
If instead the MSS-concept is considered as a regulation and feedback phenomena and that it is the locus of the boiling front it tries to regulate – it is the distance between the boiling front and the evaporator outlet that is interesting for suppressing mist flow and droplets. Studying figures 52 and 53 it becomes evident that to maintain the same degree of superheat at the same relative heat exchanger area, a larger amount of the area has to be used for superheating with R407C (or any other zeotrope) than for R22. In other words, if the MSS-line describes the necessary superheating distance between boiling front and evaporator outlet a refrigerant that yield a lower superheat when the same area is used for superheating, will of course run stable with smaller degree of superheating.

**Which one is it?**

It may be any one of the explanations above, or a combination of two or all three of them: The phenomena in explanation three, amplifies the phenomena in explanation two, so the risk of pinching in the beginning (from the refrigerant’s point of view) is smaller than for R22, whereas the possible problem with pinching in the end of the evaporator is a larger problem for R407C than for R22, as the temperature difference already is smaller for R407C without the phenomena in explanation one or three.

Circulating composition can be measured as has been discussed earlier in the thesis, but the two other explanations can not be proven with more than circumstantial evidence, unless more refined measurements are made. The question is whether there is any point in doing this (other than curiosity, which is a perfectly good reason in it self) as it really does not practically matter which ever it may be, as long as the service technician does not set the superheating higher than is necessary for stable operation.
**Why things go wrong**

In the report concerning actual domestic heat pump failures, no explanations are given: It consists of statistics only. (Folksam 2003) The current thesis covers discussions of systems extensions etc, and has a thermodynamic approach to the issue; it is aimed at explaining behaviour and present tools for prediction and decision. When refrigeration and heat pump equipment do break down, they generally do not do so because of thermodynamics. They do break down because of poor implementation and understanding leading to operation where the lubrication of the compressor work less well than is necessary, due to fabrication errors etc. So, if a type of split-type-reversible air conditioning unit breaks down, because of compressor failures, in great numbers after it being warehouse retrofitted from R22 to R407C, or a make of direct-expansion-ground-coupled heat pumps breaks down *en masse*, it is not the new refrigerant’s fault. Neither is it the compressor's or POE-oil’s fault. –Even if things worked fine when e.g. R22 was used.
Phase-out of refrigerant R22

Conclusions
When replacing CFCs like R12 in existing machines was an issue, a substitute (R134a) with thermo-physical properties so close to R12’s that a unit subject to a refrigerant retrofit to it presented essentially the same performance and behaviour as before the retrofit. As time has come to replace R22 in existing machinery things are complicated by the fact that none of the commercially available substitutes all have thermo-physical properties that in one way or another significantly deviates from R22’s. Thus, using any of them in existing facilities will in many cases render in not only changes in capacity and efficiency, but also new types of behaviour either not experienced with R22 or previously not considered significant. These issues raise a number of questions when a phase-out in existing facilities is implemented in the field.

Models and system extensions
As units are to be retrofitted from R22 to any of the commercially available substitute refrigerants, or with tailor made mixtures for unique installations, it is desired to obtain predictions on how the facility will perform after the retrofit. In this thesis, a presentation of the consequences of how various systems extensions – spatial and temporal – and conceptual model resolutions are decided, affects the predictions made, and that the perhaps most common predictive model (a model gradually accepted as natural within the
refrigeration paradigm) has very low general validity as predicting-tool for real facilities in real installations.

The models and system extensions that gradually have become accepted within the international refrigeration and heat pump community (they have become part of the paradigm) are not well fitted to handle refrigerant R22 retrofitting issues. Other more elaborate models have to be used if predictions and explanations with any generality are to be given.

As is shown in this thesis, many of the phenomena reported by service technicians and others can be explained. The computer simulation tool used for this present output data that (at least qualitatively) represents what is experienced in laboratory as well as field measurements.

**Circulated composition and measurement techniques**

If retrofitted units with zeotropic refrigerant mixtures in the field are to be analysed in a more elaborate way than just looking capacity and momentary-efficiency, and explanations to why the superheating behaves as it does, why the unit seems to be operating with a considerable subcooling when there are still bubbles in the liquid line sight glass, the actual circulated composition has to be at hand. In chapter 5 of this thesis are two methods for how to achieve estimations of the circulated composition presented: One indirect method and one online method.

Deviations, from the nominal, in circulated composition can explain a number of the phenomena and behaviours of new and retrofitted units that has been reported ever since zeotropic refrigerant mixtures as R407C and R417A started to be used. In chapter 4 one part of the explanation as to why units operating with e.g. R407C have been reported to be running without hunting even though the superheating is seemingly small. Another part of the
explanation is found in chapter 5, and the discussion concluding this is found as an example in chapter 6.

**Relative size of the heat exchanger surfaces**

In chapter 4 (some already in chapter 3) it is shown that one thing that has a tremendous impact on the actual outcome and the possibilities to affect the outcome of a refrigerant retrofit: the original dimensioning of the heat exchangers. In the thesis this is referred to as the relative size of the heat exchanger surfaces.

In the chapter 4 of the thesis it is shown that by using zeotropic refrigerant mixtures the same amount of area used for subcooling (in the condenser) and superheating (in the evaporator) will render in a smaller degree of subcooling and superheating. This is, as is mentioned above, one reason why units using zeotropic refrigerant mixtures can run without hunting with only a seemingly small degree of superheating, and is also discussed in chapter 6.

**Energy efficiency contra momentary-COP**

In chapters 3 and 6, the idea of energy efficiency is discussed. One conclusion is that the concept of Coefficient of Performance as it is commonly used within the refrigeration and heat pump community is not generally a good measure of the energy efficiency.

**Suggestions for future work**

It is not totally unlikely that the future will bring at least a new installation stop for HFC-refrigerants within the whole of the European Union. How are they to be replaced: By new refrigerants, new techniques or energy conservation? What kind of research can universities provide to answer these questions? It is critical that the development in the field of heat pumping technologies is followed as one discipline in itself. Various specialists
in lubrication and thermodynamics etc may not be able to see both engineering and science at the same time. Only refrigeration engineering scholars will be able to detect minute changes and thus sense the future.

Future development of working fluids will be determined by legislation and the development of refrigeration technology. Fundamental issues will be charge minimisation and circuit tightness; units with very little charge will justify the use of both natural and man-made refrigerants when ever applicable.

Refrigeration engineering is not thermodynamics or machine design, control theory, building physics, economy or management – apparently it is all of the above.

*I have yet to see any problem, however complicated, which, when /.../looked at /.../the right way, did not become even more complicated.*

---

73 Poul Anderson (Science-fiction writer), as quoted in Miser and Quade (1985) p.151.
## Nomenclature

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol(s)</th>
<th>Units</th>
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<tbody>
<tr>
<td>Temperature</td>
<td>$T, t$</td>
<td>K, °C</td>
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<tr>
<td>Temperature difference</td>
<td>$\vartheta, \theta$</td>
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<td>Entropy</td>
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<td>$u$ or $U$</td>
<td>kJ/kg, kJ/kmol or kJ</td>
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<td>Enthalpy</td>
<td>$h$ or $H$</td>
<td>kJ/kg, kJ/kmol or kJ</td>
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<td>Specific heat</td>
<td>$c_p$ or $c_v$</td>
<td>kJ/(kg, K) or kJ/(kmol, K)</td>
</tr>
<tr>
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<td>$\kappa$</td>
<td>$\kappa = c_p / c_v$</td>
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<tr>
<td>Mass flux</td>
<td>$G$</td>
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E.g. mechanical work, electric energy or power etc.
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<td>kmol/kmol (general)</td>
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<td>$y$</td>
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<td>$x$</td>
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<td>Usually &lt; 1</td>
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<tr>
<td>COP</td>
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<td>Coefficient of performance</td>
</tr>
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*75 Mass flux ≈ mass flow per square-meter cross section area.*
Fouling factor $\Omega$ Degradation factor $< 1$

### Subscripts, superscripts and other indexes

1. High temperature level or initial state.
2. Low temperature level or end state

$i$ General index

$\textit{is}$ Isentropic

$s, S$ Volumetric, stroke or throttling

$\textit{el}$ Electric

$C$ …relative to a Carnot cycle, or for compressor

$m$ Mean, such as in \textit{logarithmic mean temperature difference}

$x$ liquid

$y$ vapour

$z$ bulk

\textit{adiab.} Adiabatic

$\textit{is}$ Isentropic

$r$ Reversible

$\infty$ Ambient, physical surrounding

$\Delta$ Difference
After time step or saturated liquid

Saturated vapour

Time derivative, as in $\dot{E}_{el}$ for supplied electric power
Correlations used

In this thesis a number of computer simulation models are used. In the main body of the text, the numerical models used have not been presented as they would not contribute to the arguments of the thesis. The correlations for heat transfer coefficients etc. are however presented here. Significant for all correlations used, and why they have been chosen, are that they sufficiently fulfil two criterions:

1. They include relevant phenomena
2. They are numerically robust

The second criterion is important because the purpose of the modelling has been to establish conceptual understanding of a sub-problem, and not the actual process of coding itself. This is also why off-the-shelf software has been used as modelling environment – Engineering Equation Solver, EES – and source of property data – Refprop and EES. Some plug-ins has been developed, Brineprop, but only as a tool never as a purpose in it self. In this case off-the-shelf correlations has been used (Melinder 1997; 1998).

Compressor correlation

The correlations describing compressor behaviour are the ones suggested by Pierre, but they have been somewhat modified to also handle refrigerants not tested by Pierre, however following
the same pattern. The correlation covers large open reciprocating compressors. The first correlation concerns the volumetric efficiency of the compressor (Pierre 1979):

\[
\eta_s = k_1 \left( 1 + \frac{t_{2c} - 18}{100} \right) e^{k_2 \frac{p_1}{p_2}}
\]

Eq. 57

\[
\frac{\eta_s}{\eta_{ii}} = \left( 1 + k_s \cdot \frac{t_{2c} - 18}{100} \right) e^{a \left( \frac{T_1}{T_2} \right) + b}
\]

Eq. 58

The constants (the “ks”, “a” and “b”) are as presented by Pierre empirical values, but have been modified to represent a numerical function of the mol-mass of the refrigerants. E.g. high molecular mass increase the frictional losses in the compressor outlet valve.

**Heat transfer correlations**

**Two phase: Condensation**

The model for heat transfer of the refrigerant in the desuperheating and condenser section of the condenser, the model below has been used (Shao 1996).

\[
Nu = F(X_{pr}) \cdot C_1 \cdot \left( \frac{Pr}{Re^{\alpha}} \cdot \frac{L}{d_i} \right)^{0.2} \cdot Re^{\alpha/\nu} \]

Eq. 59

For \( Re^{\alpha/\nu} < 24000 \): \( \nu_l = 0.18 \); \( C_1 = 36.5 \) and \( F(X_{pr}) = 0.846 \) and
\[ Re_{r(x=1)} > 24000 \cdot n l = 0.80; C_1 = 0.0675 \]

and \[ F\left( X_{tr} \right) = 0.37 + 0.476 \cdot \frac{24000}{Re_{r(x=1)}} \]

\[ Re_{lo} = \frac{4 \cdot \dot{m}}{\mu_l \cdot \pi \cdot d_i} \quad \text{and} \quad Re_{r(x=1)} = \frac{4 \cdot \dot{m}}{\mu_l \cdot \pi \cdot d_i} \cdot \left( \frac{\rho_l}{\rho_v} \right)^{\frac{1}{2}} \quad \text{Eq. 60} \]

Where:

- \( Pr_l \): Prandtl number for liquid
- \( Re_{lo} \): Reynolds number assuming all mass flowing as liquid
- \( Re_{r(x=1)} \): Reynolds number assuming all mass flowing as saturated vapour
- \( \mu_l \): Dynamic viscosity of the liquid \( [\text{Pa} \cdot \text{s}] \)
- \( \rho_l \): Liquid density (sat. liq. at cond. temp.) \( [\text{kg/m}^3] \)
- \( \rho_v \): Vapour density (sat. vap. at cond. temp.) \( [\text{kg/m}^3] \)
- \( L \): Length of channel \( [\text{m}] \)
- \( d_i \): Hydraulic diameter \( [\text{m}] \)

**Two phase: Evaporation**

The correlation used is a correlation for plate heat exchangers suggested in *Heat Exchanger Design Handbook* (Kumar 1998), but modified according to Claesson’s recommendations (Claesson and Palm 1999).

\[ h_{evap} = 1.5 \cdot 55 \cdot p_r^{0.12} \left[ -0.4343 \cdot \ln \left( \frac{Pr}{Pr} \right) \right]^{-0.55} \cdot M^{-0.5} \cdot q_{evap}^{0.67} \quad \text{Eq. 61} \]
\[ q_{\text{evap}} = \frac{\dot{Q}_{\text{evap}}}{A_{\text{evap}} \cdot (1 - X_{\text{evap}})} \]  

Eq. 62

Where:

- \( b_{\text{evap}} \) Heat transfer coefficient for the evaporation \([\text{W/(K \cdot m^2)}]\)
- \( p_r \) Reduced pressure, \( p_r = \frac{p_{\text{evap}}}{p_{\text{crit}}} \)
- \( M \) Molmass \([\text{kg/kmol}]\)
- \( \dot{Q}_{\text{evap}} \) Heat absorbed by the refrigerant in the evaporation section of the heat exchanger \([\text{W}]\)
- \( A_{\text{evap}} \) Total area of the evaporator \([\text{m}^2]\)
- \( X_{\text{evap}} \) The share of the evaporator area actually for evaporation.

**Single phase**

The correlation used for single phase heat transfer in plate heat exchangers is also from *Heat Exchanger Design Handbook* (Kumar 1998).

\[ Nu = 0.2 \cdot Re^{0.67} \cdot Pr^{0.4} \left( \frac{\nu}{\nu_w} \right)^{0.1} \]  

Eq. 63

Where:

- \( \nu \) is the dynamic viscosity in the bulk \([\text{kg/(s \cdot m)}]\)
- \( \nu_w \) is the dynamic viscosity at the wall \([\text{kg/(s \cdot m)}]\)
It is assumed that there is no difference between the viscosity in the bulk and at the wall. The equation used thus looks as follows:

\[ Nu = 0.2 \cdot Re^{0.67} \cdot Pr^{0.4} \quad \text{Eq. 64} \]

This is the single phase correlation used in both condenser and evaporator, for both single phase refrigerant and secondary media.
Sources

In this section the sources referred to within the current thesis. In the following section a complete bibliography may be found.


Lebreton, J.-M. and Vuillame, L. (2000). Real-time measurement of the oil concentration in liquid refrigerant flowing inside a refrigeration machine. The 15:th International Compressor Conference at Purdue University, West Lafayette, Purdue University.


Bibliography

In this bibliography a complete list of literature is presented. Many of the references have not been referred to within the current work explicitly.

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Lebreton, J.-M. and Vuillame, L. (2000). Real-time measurement of the oil concentration in liquid refrigerant flowing inside a refrigeration machine. The 15:th International Compressor Conference at Purdue University, West Lafayette, Purdue University.


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Software


**Online sources**


**Personal communication, letters etc**


