FILE COPY
DO NOT
REMOVE

STUDY OF WORKING FLUID MIXTURES AND HIGH TEMPERATURE WORKING FLUIDS FOR COMPRESSOR DRIVEN SYSTEMS

Final Report

Part A

Prepared by:

Thore Berntsson, Chalmers University of Technology, Gothenburg, Sweden

Hans Schnitzer, Graz University of Technology, Austria

September 1985

This report is the Step 2 product of the project "Study of working fluid mixtures and high temperature working fluids for compressor-driven systems", which is Annex VI of the International Energy Agency working group Advanced Heat Pumps. Operating Agent for the annex is the Swedish Council For Building Research.

TABLE OF CONTENTS

SUMMARY 1

General 1

Part A 3

1. INTRODUCTION 10

The organization of the Step 2 work 11

2. HEAT TRANSFER ASPECTS OF MIXTURES 14

Heat transfer in evaporation 14

Experimental results 14

Empirical equations 16

Discussion 17

Heat transfer in condensation 19

References 20

3. SYSTEM ASPECTS OF HEAT PUMPS WITH NONAZEOTROPIC MIXTURES 23

Conclusions 32

4. RECENT DEVELOPMENTS IN THE FIELD OF HIGH TEMPERATURE HEAT PUMPS 33

References 37

5. TECHNICAL SENSITIVITY ANALYSIS 38

5.1 Introduction 38

5.2 Computer programs 39

Computer programs with infinite heat exchanger areas 39

Computer program with finite heat exchanger areas, pressure drops, and realistic compressor efficiencies 40

Physical properties 43

5.3 General parameter study 45

Influence of the type of mixture and of the composition 47

The space heating case 49

5.4 Parameter study with finite heat exchanger areas and realistic compressor efficiencies 50

Industrial case 51

Space heating case 53

5.5 Discussion and conclusions 55

5.6 Nomenclature 57

5.7 References 58

6. TECHNICAL-ECONOMICAL EVALUATIONS ON COMPRESSION HEAT PUMPS WORKING WITH NONAZEOTROPIC MIXTURES OF FLUIDS 79

6.1 Introduction 79

6.2 Mathematical model 81

6.2.1 General arrangement 81

6.2.2 The process-simulation program 82

6.2.3 The apparatus design program 83

6.2.4 The economy program 86

6.2.5 The maximization algorithm 88

6.3 The economical comparison of pure- and two component compression heat pumps for one particular application 88

6.3.1 Requirements on the system, boundary conditions and general definitions 88

6.3.2 Results 94

6.3.3 Critical reflections on the results 105

SUMMARY

General

This report is the result of the Step 2 work in the Annex VI of the International Energy Agency working group Advanced Heat Pumps.

According to the Annex text the objectives, scope and means of the project are

Objectives

(a) Objectives. The objectives of this study are to gather theoretical and experimental information for heat pump systems operating with new working fluids or mixtures with special emphasis on nonazeotropic mixtures, and to evaluate their potential in comparison with other heat pump systems.

(b) Scope. Working fluids and systems to be included in this Task are:

(1) Pure fluids of any chemical composition capable of working with condensing temperatures above 100 ^{o}C;

(2) Non-azeotropic mixtures for any condensing temperature in heat pump applications (with special emphasis on fluorohalocarbons);

(3) All types of compressor-driven heat pumps with closed circuits, i.e. for space heating and cooling and for industrial purposes (refrigeration applications will not be included), using the above working fluids;

(4) All kinds of heat sources and heat sinks.

Means

The work under this Task is carried out on a jointly funded basis in two sequential Steps:

Step 1: Preparation of a report on the state-of-the-art on working fluid mixtures and high temperature working fluids for compressor-driven systems, a general estimate of the technical and economic performance of working fluids compared with conventional systems and elaboration in more detail of the work to be undertaken in Step 2.

Step 2: Preparation of a report on one hardware project, heat pump design, or on an experimental activity on state or transport properties of working fluids from each Participant and a technical and economic evaluation of working fluids and heat pump systems described in paragraph 1 above.

The State-of-the-art report, the result from the Step 1 work, has been available since April 1984.

The official Operating Agent is the Swedish Council for Building Research. The Operating Agent work has been carried out as a cooperation between the Chalmers (Göteborg, Sweden) and the Graz (Austria) Technical Universitites. Seven countries have participated in the Step 1 work: Austria (University of Graz, Prof. Moser), Denmark (Technical University of Denmark, Prof. Worsoe-Schmidt), Finland (Tampere University, Prof. Aittomäki), Japan (Tokyo Sanyo Electric and Daikin Industries, Osaka), Sweden (Chalmers University of Technolcgy, Prof. Berntsson and Grén), United States (Oak Ridge National Laboratory, B. Levins, University of Illinois, Prof. Stoecker, and

National Bureau of Standards), and West Germany (University of Hannover, Prof. Kruse). Each country has appointed a National Team for this work.

The report is divided into two parts, one A and one B part. The A part contains the results from the Operating Agent work regarding technical and economical evaluation of working fluids and heat pump systems. The B part contains the reports from the National Teams' projects. The two parts are available in two separate volumes.

Part A

Below a chapter-by-chapter summary for Chapters 2-7 is given. Chalmers University of Technology has been responsible for Chapters 1, 2, 5, and 7 and the Technical University of Graz has had the responsibility for Chapters 3, 4, 6, and 7.

Chapter 2: Heat transfer aspects of mixtures

Heat transfer in evaporation

During recent years two experimental projects on evaporative heat transfer in horizontal tubes and one project on falling film evaporation have been reported.

From the experimental results for horizontal tubes it can be concluded that the mass flux is an important parameter when differences in the heat transfer coefficient between pure working fluids and mixtures are discussed. This difference decreases with increasing mass flux. At high heat fluxes, i.e. fully developed annular flow, typical reductions of the heat transfer coefficient of up to 30 % can be expected when mixtures are used instead of pure working fluids.

In falling films the turbulence seems to be so good that the reduction of the heat transfer coefficient for mixtures is smaller, typically 5-15 %.

Heat transfer in condensation

One project on condensation heat transfer coefficients for mixtures in horizontal tubes has been reported. Also here the heat transfer coefficient decreases when mixtures are used. The maximal decrease for R12/R114 mixtures was approximately 30 % compared with pure R12, i.e. the same order of magnitude as for evaporating conditions.

Chapter 3: System aspects of heat pumps with nonazeotropic mixtures

With the aid of a flow sheeting program experimental results of whole cycle measurements from the Universities of Denmark, Illinois, Tampere, and Graz have been theoretically evaluated and compared. From this work it can be concluded that the experiments did not show as high an improvement of the COP with mixtures, as could be predicted. The simulations showed that the mixture behaved as predicted, but the equipment did not. Especially the compressor behaviour was changed. Obviously it seems not possible to improve the COP of existing heat pumps with mixtures, but with a tailor-made heat pump for mixtures it should be possible to achieve performances as theoretically predicted.

Chapter 4: Recent developments in the field of high temperature heat pumps

During recent years some papers have been published in this field, especially in the seminar "Energy Conservation in Industry", Düsseldorf 1984 and the conference "The

large scale application of heat pumps", York 1984. Results with R114 as well as "unconventional" substances (such as perfluoro-n-hexane) have been reported. No real installation of a closed cycle compression heat pump condensing above 100 ^{o}C has been made since 1983 in the participating countries as far as the author knows.

Chapter 5: Technical sensitivity analysis

The influence on the COP, the heating capacity, and the condensing pressure of various technical parameters are presented for two cases, one space heating case and one industrial application.

For the calculations two computer programs have been used. The first one simulates the heat pump process with infinite heat exchanger areas. With this program the following parameters have been studied: Type of mixture, composition, temperature gradients, degree of subcooling, internal superheating, and compressor efficiency. With the second program a more realistic behaviour of a heat pump in a given situation has been simulated. The performance of the heat exchangers, the compressor, influence of the heat exchanger sizes, pressure drops, etc. have been taken into account.

From the calculations it can be concluded that the possible improvements with mixtures are decreased considerably when real behaviour of the components and finite heat exchanger sizes are assumed instead of infinite heat exchanger sizes. For the industrial case, having large temperature gradients on the heat sink and the heat source, improvements of the COP of 10-15 % and of the capacity of more than 80 % seem to be possible in reality. For the space heating case, with small temperature gradients,

mixtures do not give higher COP:s but the capacity can be increased by at least 20 %.

Chapter 6: Technical-economical evaluations on compression heat pumps working with nonazeotropic mixtures of fluids

Technical-economical optimizations of the various components in a heat pump for mixtures and pure fluids have been made with the aid of a computer program for one of the cases mentioned in Chapter 5, the industrial one.

Considering certain nonazeotropic mixtures with which temperature adaptations to the cooling of the heat source and the heating of the heat sink are possible, economical improvements can be achieved. E.g. by mixing 40 % R12 (by mole) to R114 the payback period of a 5 MW-unit decreases 6 % compared with R114, or by adding 20 % R12 to R11 the reduction of the payback period is 9.5 % compared with R11 (Table 6.5). It turns out, that from an economical point of view, mixtures that allow optimal temperature adaptations are not doing as well as it was expected. The reason for that are the minimum temperature differences in the condenser and evaporator (ΔT_c, ΔT_e) as well as the pressure drop in the evaporator (Δp_e) that have to be increased in order to avoid too high investment costs. Therefore the increase of the coefficient of performance, gained by the nonazeotropic mixture, is not so high that a drastical reduction of the payback period can be achieved.

In some cases it is possible to reduce even the investment costs of the heat pump due to the increase of the volumetric efficiency.

Comparing all layouts of large heat pumps the one with pure R21 represents the optimal economical solution, not considering systems with mixtures of pentane-butane or

butane-hexane. If one cannot apply R21 (e.g. because of its high discharge temperature at an uncooled compressor) and the nonazeotropic mixtures shall be compared with pure R114, up to 10 % reduction in the payback periods can be achieved.

As a matter of fact all conditions might differ on other places of application. The question which working fluid represents the most economic solution in heat pump systems for particular applications, can be answered only by comparisons of economically optimized designs with the different substances. No calculations have been made on small scale units.

Chapter 7: Discussion and conclusions

Nonazeotropic mixtures

In the preceding chapters it has been shown that the comparison between pure fluids and mixtures is very complex and that general statements about advantages and disadvantages with mixtures can be made only to a very limited extent. The benefits with mixtures must be considered from application to application. As was shown in Chapter 5, it is very important to compare pure fluids with mixtures at given outer conditions and to take all influences in the heat exchangers, compressor, etc. into account.

The results and conclusions from the technical and economical calculations are presented in the summaries of Chapters 5 and 6 above.

The perhaps most important further work needed at this state is to use the present knowledge about mixtures,

presented in this report, in technical-economical studies on interesting applications for heat pumps with mixtures in addition to those included in this report. Other areas, where further work is needed, are:

- measurements and development of estimation methods for state and transport properties

- measurements of compressor efficiencies for various types of compressors

- measurements of heat transfer coefficients in the condenser and the evaporator

- problems related to lubricating oils, especially chemical stability at high temperatures and solubility

- system simulations, containing appropriate types of mixtures, compositions, and component designs.

Of these areas, the one about state and transport properties has been considered by the Working Group to be appropriate for further international collaboration. Therefore, the group has suggested an extension of this Annex, which has been submitted to the Executive Committee of Advanced Heat Pumps. The extension is entitled "State and Transport Properties of High Temperature Working Fluids and Nonazeotropic Mixtures".

High-temperature working fluids

Not much has been reported on high temperature working fluids, through Step Two of the Annex. Some work has been done, by various research groups on the use of water and alcohols with oil-free compressors.

The search for new fluids has brought up some interesting substances, but no final choice has been made.

1. INTRODUCTION

This final report is the main result from the Step 2 work of Annex VI of the IEA group Advanced Heat Pumps. The title of the annex is "Study of Working Fluid Mixtures and High Temperature Working Fluids for Compressor-driven Systems". According to the annex text the objectives, scope, and means of the project are:

Objectives

(a) Objectives. The objectives of this study are to gather theoretical and experimental information for heat pump systems operating with new working fluids or mixtures with special emphasis on nonazeotropic mixtures, and to evaluate their potential in comparison with other heat pump systems.

(b) Scope. Working fluids and systems to be included in this Task are:

(1) Pure fluids of any chemical composition capable of working with condensing temperatures above 100 ^{o}C;

(2) Nonazeotropic mixtures for any condensing temperature in heat pump applications (with special emphasis on fluorohalocarbons);

(3) All types of compressor-driven heat pumps with closed circuits, i.e. for space heating and cooling and for industrial purposes (refrigeration applications will not be included), using the above working fluids;

(4) All kinds of heat sources and heat sinks.

Means

The work under this Task is carried out on a jointly funded basis in two sequential Steps:

Step 1: Preparation of a report on the state-of-the-art on working fluid mixtures and high temperature working fluids for compressor-driven systems, a general estimate of the technical and economic performance of working fluids compared with conventional systems and elaboration in more detail of the work to be undertaken in Step 2;

Step 2: Preparation of a report on one hardware project, heat pump design, or on an experimental activity on state or transport properties of working fluids from each Participant and a technical and economic evaluation of working fluids and heat pump systems defined in the Objectives above.

The organization of the Step 2 work

The State-of-the-art report, the result from the Step 1 work, has been available since April 1984. The Step 2 work has been carried out by two Operating Agents and one National Team from each of the seven participating countries. The following organizations have participated in this project:

Operating Agents:

Austria: Institut für Grundlagen der Verfahrenstechnik (Prof. Moser), Technical University of Graz

Sweden: Dept of Heat and Power Technology (Prof. Berntsson), Chalmers University of Technology, Göteborg

National teams:

Austria: Institut für Grundlagen der Verfahrenstechnik (Prof. Moser), Technical University of Graz

Denmark: Dept of Refrigeration (Prof. Worsoe-Schmidt), Technical University of Denmark, Lyngby

Finland: Tampere University of Technology (Prof. Aittomäki), Tampere

Germany: Institut für Kolbenmaschinen, Abteilung Kältetechnik (Prof. Kruse), University of Hannover

Japan: Daikin Industries (K. Fushimi), Osaka.
Tokyo Sanyo Electric (H. Watanabe), Tokyo.

Sweden: Dept of Heat and Power Technology, (Prof. Berntsson), Chalmers University of Technology, Göteborg.
Dept of Chemical Engineering (Prof. Grén), Chalmers University of Technology, Göteborg.

USA: Oak Ridge National Laboratories (B. Levins), Tennessee.
Dept of Mechanical Engineering (Prof. Stoecker), University of Illinois.
National Bureau of Standards (D. Didion), Washington D.C.

In the Step 2 work three working meetings have been held; according to the following:

1983, August 29-30 in Paris

1984, February 2-3 in Graz

1984, September 28 in Copenhagen

The report is divided into two parts, one A and one B part. The A part contains the results from the Operating Agent work regarding technical and economic evaluation of working fluids and heat pump systems. The B part contains the reports from the National Teams' projects. The two parts are available in two separate volumes.

For the different chapters in the A part, the following persons have been responsible:

Technical University of Graz:

Chapter 3: K. Harler, H. Schnitzer
" 4: H. Schnitzer
" 6: H. Ofner, H. Schnitzer, F. Moser
" 7: H. Schnitzer

Chalmers University of Technology:

Chapter 1: T. Berntsson
" 2: K. Munch Berntsson, T. Berntsson
" 5: " , "
" 7: T. Berntsson

2. HEAT TRANSFER ASPECTS OF MIXTURES

Aspects of heat transfer coefficients for both high-temperature fluids and nonazeotropic mixtures have been discussed in detail in the State-of-the-art report, pp. A44-A54, A73-A74, and B48-B56, respectively. Since that report was printed, new information in this area about mixtures has become available in the literature and in the NT's work in Annex VI. In this chapter the knowledge about heat transfer for mixtures is updated and an evaluation of the NT's work as well as the new results in the literature is made.

Heat transfer in evaporation

Experimental results

Since the State-of-the-art report was printed, three projects dealing with heat transfer for mixtures in evaporation have been presented. These are:

Radermacher et al. /2.2/
Pedersen /2.1/
Berntsson et al. (NT work in this report, see the B part).

Pedersen /2.1/ has made heat transfer studies in a horizontal tube-and-shell evaporator consisting of 30 tubes in parallel. In each tube a surface extension device was placed. The tubes had a length of 2.63 m and on the brine side 39 baffles were placed. Measurements were made with mixtures of R12/R114. Some typical results are shown in Fig. 2.1. As can be seen the decrease of the heat transfer coefficient when a mixture is used is considerable. The slopes for the pure components follow approximately the expression:

$$\alpha = k \cdot \dot{G}^{0.8}$$

i.e. independent on the heat flux, which should be expected in a situation with surface evaporation. This indicates that there is no nucleate boiling at the heat fluxes used in the experiments. The different slope for the mixture indicates that the differences between the heat transfer coefficients for mixtures and for pure refrigerants decreases with increasing mass flux. The mass fluxes used are low and in the region where annular flow is not fully developed.

Radermacher et al. /2.2/ have reported results for the mixture R152a/R13B1. The experiments were carried out in one horizontal tube which was heated electrically by using the tube as one of the electrodes in a DC system. The inlet quality was in all cases 0 % and the outlet one varied with the conditions. The flow pattern was in most cases annular flow. The main results are shown in Fig. 2.2. As can be seen, pure R13B1 gives somewhat higher heat transfer coefficients than R152a at given flow rates in moles/hr, except at very high flow rates, where they are equal. The mixtures gave considerably lower values, approx. 30 % lower than the corresponding ones for the pure working fluids. It is not clear if the boiling was evaporative or nucleate or a combination at the heat fluxes used in the experiments.

Berntsson et al. /this report/ have reported values for R12/R114 in a falling-film evaporator. The falling film is formed inside a 3 m long tube, which is heated from the outside by liquid R11, flowing upwards in an annular space. They report both mean values and local heat transfer coefficients along the tube for both surface evaporation and nucleate boiling conditions. As can be seen in the figures in the paper, see the B part of this report, the

decrease of the heat transfer coefficient for the mixtures compared with the corresponding ones for the pure fluids is small, 5-15 % compared with pure R12. In fact, the mixture values fall between those for R12 and R114. Hence, these results are quite contradictory to the ones for horizontal tubes. Possible reasons for this will be discussed later in this chapter.

Empirical equations

Pedersen compared his results with several empirical equations, those of Pierre /A74 in the State-of-the-art report/, Chaddock-Noerager /A40/, Lavin-Young /A141/, and Varma, Sharma and Mishra /B105/. For the pure working fluids the measured values are higher than the corresponding ones for all the equations. For the mixtures the measured values were found to vary with the magnitude of $|x - y|$. Pedersen proposes a new equation:

$$\alpha_{mix} = \alpha_{id}(\xi) \cdot c \cdot F$$

where

$$c = \left(1 + \frac{a}{Re_{Lo} + b}\right) |x - y|)^{-1}$$

F is a correction factor between the experiments and the used empirical equation for pure working fluids. With the Chaddock-Noerager equation as a base, the constants have the following values:

$F = 1.257$
$a = 3425$
$b = 59.5$
(Re_{Lo} is the liquid Re number)

Based on their experimental results, Radermacher et al. proposed an equation of the form:

$$\bar{h}_{mix} = \bar{h}_{lv}(1 - \overline{|y - x|})$$

where

$\bar{h}_{lv}$ is the average heat transfer coefficient of the less volatile component at the same mole flow rate and heat flux.

They compared this equation with the Happel and Stephan one /B33/:

$$\bar{h}_{mix} = \bar{h}_{id}(1 - K|\bar{y} - \bar{x}|^{n})$$

where h_{id} is the average heat transfer coefficient, based on a mole fraction weighting of the pure refrigerants, and K and n are empirically determined constants. It was shown that their own equation could predict their experimental results better than the Happel and Stephan one. The equation could predict the results within ±15 % for 90 % of the tests.

Discussion

From the experimental results for horizontal heat transfer presented above it can be concluded that the mass flux is an important parameter when differences in the heat transfer coefficient between pure working fluids and mixtures are discussed. According to Pedersen the difference between mixtures and pure working fluids decreases with increasing mass flux. This seems logical and is most probably due to the increased turbulence when the mass flux is increased. The results from Radermacher at al. at

annular flow conditions also show a mass flux dependence, although much lower than for the low flow rates in Pedersen´s experiments. Also the general level of the reduction in heat transfer coefficient for mixtures is smaller. Both these results are probably due to the high turbulence at annular flow conditions.

The influence of the heat flux in horizontal tubes has not yet been studied in such a detail that any general conclusions can be drawn.

The choice of mass flux in horizontal tubes is a question of optimization between the desire for a high heat transfer coefficient and a small pressure drop. Calculations made by the Operating Agent have shown that optimized mass fluxes for mixtures are in the well developed annular flow regime, which is also the normal regime used for pure working fluids. According to the experimental results in this regime, typical reductions of the heat transfer coefficient of up to 30 % can be expected when mixtures are used instead of pure working fluids.

In falling films the turbulence seems to be so good that the reduction of the heat transfer coefficient for mixtures is smaller, typically 5-15 %. This is valid both for film evaporation and nucleate boiling conditions. Another advantage with falling films is the negligible pressure drop.

When comparing the various equations suggested for the heat transfer coefficient of mixtures in horizontal tubes, it can be noticed that none of them, Pedersen /2.1/, Radermacher et al. /2.2/, and Varma and Sharma /B105/ takes the absolute pressure into account. For pool boiling conditions, on the other hand, this parameter seems to have a significant influence, see page B51 in the State-

of-the-art report and reference B110 in that report. This indicates that experiences and empirical correlations for pool boiling cannot be used for flow boiling conditions.

Heat transfer in condensation

The knowledge about heat transfer coefficients for condensation of mixtures is today very poor. The only experimental results reported in the literature are the ones made by Stoecker et al. /2.3/. These measurements have been made in a heat pump test rig with two condensers in series, the first one having a slightly higher temperature level than the second one. The condensation took place inside horizontal tubes with an inner diameter of 12.7 mm. Some typical results from these measurements are shown in Fig. 2.3. These results have been obtained by keeping the cooling capacity of the heat pump constant. This means that the refrigerant flow rate increased slightly with the content of R114, with less than 15 % difference between pure R12 and the mixture 70/30 % R114/R12.

As can be seen in Fig. 2.3, the heat transfer coefficient decreases when mixtures are used, in spite of the slightly higher flow rate (in general condensing heat transfer coefficients are much less dependent on the flow rate than evaporating ones). The maximal decrease, compared with pure R12 is approximately 30 %, i.e. the same order of magnitude as for evaporating heat transfer coefficients.

Before any general conclusions about condensing heat transfer coefficients for mixtures can be drawn, much more information about the influence of the type of mixture, the heat flux, the flow rate, the temperature and pressure levels, etc. must be known. Also the effect of the superheating and subcooling parts of a condenser must be taken into account.

References

2.1 Pedersen, P.H., "Heat transfer in evaporation of nonazeotropic mixtures". (In Danish). EFP 81, Projekt 75, Department of Refrigeration Technology, Denmark University of Technology, 1983.

2.2 Radermacher, R. et al., "Experimental determination of forced convection evaporative heat transfer coefficients for nonazeotropic refrigerant mixtures". Trans. of ASME, Nov. 1983.

2.3 Stoecker, W.F., and McCarthy, C.J., "The simulation and performance of a system using an R12/R114 refrigerant mixture". Department of Mechanical and Industrial Engineering, University of Illinois. Report No. ORNL/Sub/81-7762/3&01, prepared for Oak Ridge National Laboratory.

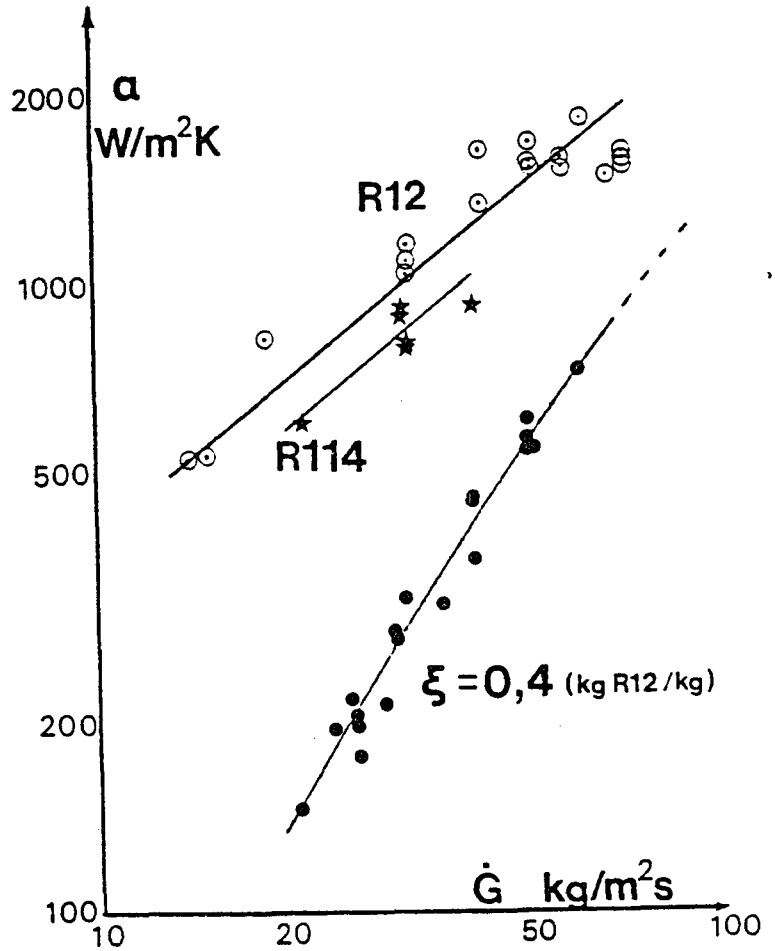

Fig. 2.1. Heat transfer coefficient in evaporation versus mass flux according to Pedersen /2.1/.

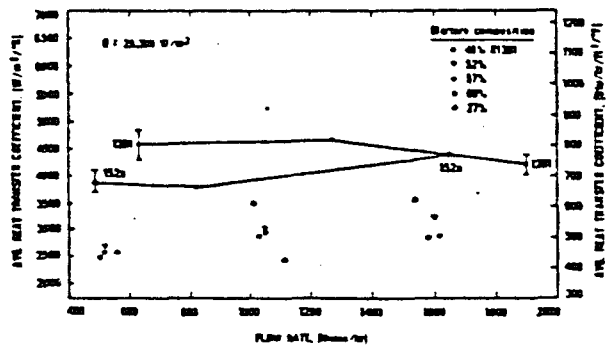

Fig. 2.2 Heat transfer coefficient for mixtures of R13B1/R152a in evaporation according to Radermacher et al. /2.2/. (I will try to get a better figure from the authors to the final report).

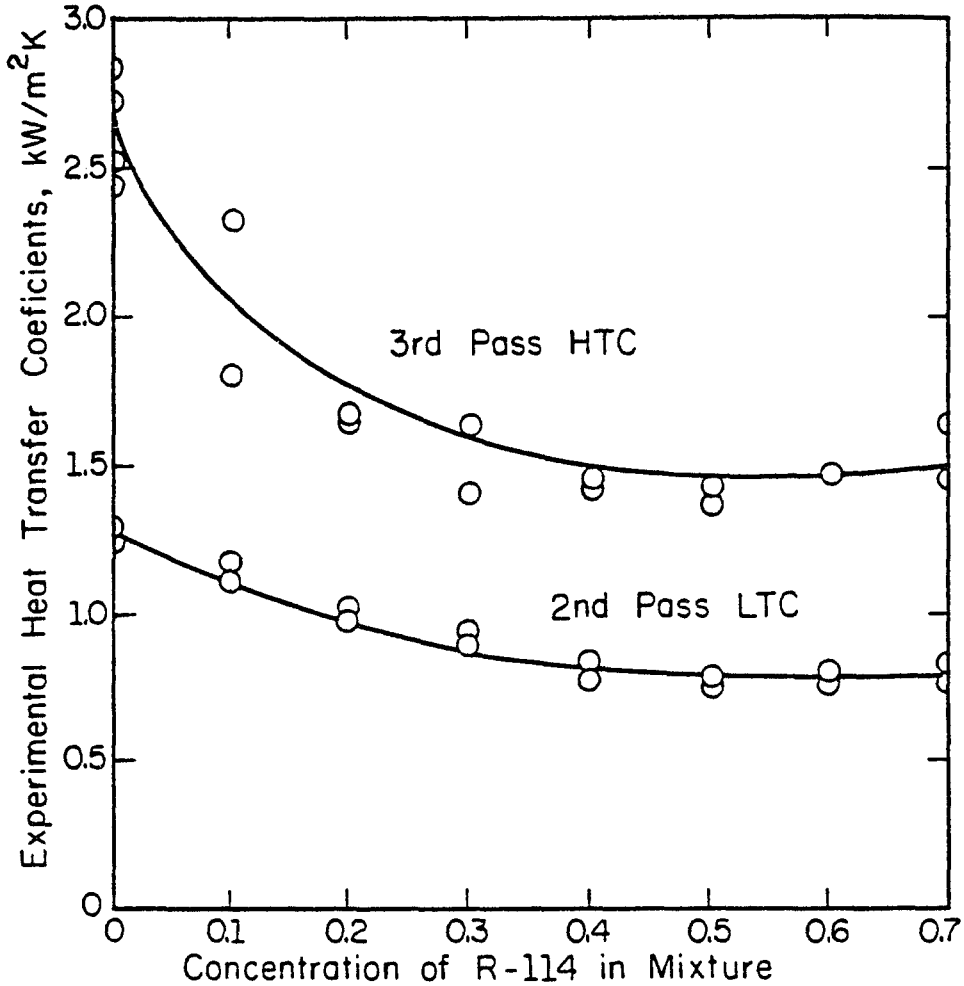

Fig. 2.3. Heat transfer coefficients of R12/R114-mixtures in condensation according to Stoecker et al. /2.3/.

3. SYSTEM ASPECTS OF HEAT PUMPS WITH NONAZEOTROPIC MIXTURES

Experiments with heat pump systems using nonazeotropic mixtures have been carried out by various national teams before and during work of Annex VI of the IEA Implementing Agreement on Advanced Heat Pumps. The aim of this chapter is to compare the experimental results with the simulations.

For the theoretical simulation of the heat pump circuit modified flow sheeting programs have been used. These programs are arranged like a modular system, which means, that single parts can be changed or added easily. In this way compression heat pumps as well as absorption heat pumps can be simulated and a number of different apparatus types can be designed.

For the comparison of theoretical and experimental data four different sets of measurements were available.

1) At the refrigeration laboratory of the Technical University of Denmark (Lyngby) H.J. Høgard Knudsen made cycle measurements with a R12/R114 mixture. A brine/water compression heat pump was used with a nominal heat output of 45 kW at 0/55 ^{o} and R12 as the refrigerant. The heat pump was driven by a gas motor and its capacity could be controlled by unloading cylinders and/or reducing the speed. Measuring results are presented for R12, R114 and some mixtures with different compositions.

2) At the department of mechanical and industrial engineering of the University of Illinois, W.F. Stoecker also tested the mixture R12/R114. He wanted to test especially the heat transfer in the condenser. An electric driven

brine/water heat pump system is used. Measured data are presented for an R12/R114 (70 %/30 % by weight) mixture. In his publication "The simulation and performance of a system using an R12/R114 refrigerant mixture" written for U.S. Department of Energy, he describes the test plant and also gives results of his computer simulation of the heat pump system.

3) Antero Aittomäki from the Tampere University of Technology (Finland) studied the plant conditions of a brine/water and an air/water compression heat pump with R22 and mixtures of R22/R114. He compared the heat pump cycles with a brine evaporator and an air evaporator.

4) At our department in Graz we installed a compression heat pump designed for the refrigerant R12 including a simulation plant for the heat sink and the heat source. With the simulation plant we controlled the inlet and outlet temperatures and the mass flow of the heat sink and the heat source. The heat source is simulated by electric heating rods, the heat sink with a plate heat exchanger. We investigated the refrigerant R12, the refrigerant R114 and two mixtures of them. For the pure refrigerants we used thermostatic controlled valves. These valves did not give satisfying results for the mixtures. Therefore a manually operated one was installed. An influence of the valve position on the COP of the heat pump was observed. The COP for an R12/R114 (48.5 %/51.5 % by weight) mixture changed from 2.5 to 3.6 being at the optimal working range of the compressor.

In all experiments mentioned above the same tendencies can be seen. If the tests are made with a heat pump designed for a pure refrigerant, optimal results for a mixture are not possible to obtain. In every case the

value for the COP decreases if a refrigerant mixture is used. Some of the reasons are the following facts:

- the heat transfer coefficients decrease because evaporating and condensing of a mixture is a combined problem of heat and mass transfer

- the heat transfer area is too large or too small for the mixture because the heat pumps have been designed for pure refrigerants

- the velocities in the compressor changes and the efficiency is reduced.

These facts had to be taken in consideration, as we fixed the parameters for the estimation of the heat pump cycles with our computer program. The results of the first calculations deviated rather much from the experimental data, because our program estimates optimized heat pump circuits. If one changes some input data like the compressor- and motor efficiency, a satisfactory agreement between experimental and theoretical results can be observed.

The following tables show the comparison of experimental and theoretical results.

Beside other effects the modification of the isentropic efficiency influenced the heat pump circuit most. As the flowsheet program only calculates an adiabatic gas-compression the simulation of the compressor was done in that way that the heat that in reality is removed by cooling during compression, is removed isobaric after the adiabatic compression.

Experiment Nr.: MIX 3 (AUS.)

Refrigerant mixture: R12.../ R114..

Composition: weight percent 48,50 / 51,50
mole percent 57,16 / 42,48

COP: exp.: 3,55 / calc.: 3,55

Internal heat exchanger	exp.	calc.	dev.
t_{in}	42,9	42,9	-
t_{out}	48,9	48,9	-
p_{in}	4,69	4,94	+5,3
p_{out}	4,69	4,91	+4,7
Q	-	0,43	-

compressor Q_{off} = 1,38 kW	exp.	calc.	dev.
	48,9	48,9	-
	90,6	90,6	-
	4,69	4,91	+4,7
	14,37	14,37	-
	3,28	3,28	-

condenser	exp.	calc.	dev.
	90,6	90,6	-
	69,8	69,8	-
	14,37	14,37	-
	13,73	13,73	-
	11,64	11,64	-

Internal heat exchanger	exp.	calc.	dev.
t_{in}	69,8	69,8	-
t_{out}	64,9	65,75	+1,3
p_{in}	13,73	13,71	-
p_{out}	13,73	13,73	-
Q	-	0,43	-

valve	exp.	calc.	dev.
	64,9	65,75	+1,3
	31,8	31,8	-
	13,73	13,73	-
	5,07	5,32	+4,9

evaporator	exp.	calc.	dev.
	31,8	31,8	-
	42,9	42,9	-
	5,08	5,32	+4,9
	4,69	4,94	+5,3
	9,79	9,74	-0,5

Experiment Nr.: U.S.

Refrigerant mixture: R.12 / R.114

Composition: weight percent 70 / 30

mole percent 76,7 / 23,3

COP: exp.: 2,47 / calc.: 2,47

Internal heat exchanger					compressor Q_{off} = 0,58 kW				condenser			
	exp.	calc.	dev.		exp.	calc.	dev.		exp.	calc.	dev.	
t_{in}	-11,0	-11,0	-		5,9	5,9	-		99,1	99,1	-	
t_{out}	5,9	5,9	-		99,1	99,1	-		35,1	35,1	35,1	
p_{in}	0,835	0,846	+1,32		0,735	0,746	+1,49		9,0	9,0	-	
p_{out}	0,735	0,746	+1,49		9,0	9,0	-		8,95	8,95	-	
Q	-	0,40	-		2,64	2,64	-		6,51	6,51	-	
Internal heat exchanger Q_{off}=0,92					valve				evaporator			
	exp.	calc.	dev.		exp.	calc.	dev.		exp.	calc.	dev.	
t_{in}	35,1	35,1	-		-1,0	-1,0	-		-25,0	-25,0	-	
t_{out}	-1,0	-1,0	-		-25,0	-25,0	-		-11,0	-11,0	-	
p_{in}	8,95	8,95	-		8,90	8,95	+0,56		1,0	1,01	+1,0	
p_{out}	8,90	8,95	+0,56		1,0	1,01	+1,0		0,835	0,846	+1,32	
Q	-	0,4	-						3,83	5,38	+40,5	

Experiment Nr.: B45 (Dän.)

Refrigerant mixture: R.12../ R114..

Composition: weight percent .33,7 / .66,3

mole percent ..42../.58...

COP: exp.: 2,755 / calc.: 2,755

Internal heat exchanger					compressor Q_{off} = 1,27 kW				condenser			
	exp.	calc.	dev.		exp.	calc.	dev.		exp.	calc.	dev.	
t_{in}	3,9	3,9	-		15,0	15,0	-		84,7	84,7	-	
t_{out}	15,0	15,0	-		84,7	84,7	-		43,2	42,63	-1,3	
p_{in}	1,11	1,14	+2,7		1,08	1,11	+2,8		6,5	6,5	-	
p_{out}	1,08	1,11	+2,8		6,5	6,5	-		6,41	6,41	-	
Q	0,69	0,81	17,4		5,97	5,97	-		16,45	16,45	-	

Internal heat exchanger					valve				evaporator			
	exp.	calc.	dev.		exp.	calc.	dev.		exp.	calc.	dev.	
t_{in}	43,2	42,6	-1,3		35,8	35,3	-1,4		-8,1	-8,1	-	
t_{out}	35,8	35,3	-1,4		-8,1	-8,1	-		3,9	3,9	-	
p_{in}	-	6,41	-		6,41	6,41	-		-	1,23	-	
p_{out}	-	6,41	-		-	1,23	-		1,11	1,14	+2,7	
Q	0,69	0,81	17,4						12,2	11,75	-3,7	

Experiment Nr.: W10/W 35 (AUS.)

Refrigerant mixture: R. 12 / R. -

Composition: weight percent 100 / -

mole percent 100 / -

COP: exp.: 3,8 / calc.: 3,8

Internal heat exchanger					compressor Q_{off} = 0,54 kW					condenser				
	exp.	calc.	dev.			exp.	calc.	dev.			exp.	calc.	dev.	
t_{in}	9,6	9,6	-			16,0	16,0	-			76,4	76,4	-	
t_{out}	16,0	16,0	-			76,4	76,4	-			34,4	34,4	-	
p_{in}	-	3,34	-			3,15	3,33	+5,7			9,05	9,05	-	
p_{out}	3,15	3,33	+5,7			9,05	9,05	-			8,79	8,79	-	
Q		0,24	-			2,47	2,47	-			9,31	9,31	-	
Internal heat exchanger					valve					evaporator				
	exp.	calc.	dev.			exp.	calc.	dev.			exp.	calc.	dev.	
t_{in}	34,4	34,4	-			-	30,3	-			3,1	3,1	-	
t_{out}	-	30,3	-			3,1	3,1	-			9,6	9,6	-	
p_{in}	8,79	8,79	-			8,79	8,79	-			3,22	3,41	+5,9	
p_{out}	-	8,79	-			3,22	3,41	+5,9			3,15	3,34	+5,7	
Q	-	0,24	-								6,84	7,38	+7,9	

Experiment Nr.: 72 (Fin.)

Refrigerant mixture: R.22../ R.-...

COP: exp.:..2,57 / calc.:.2,57.

Composition: weight percent ..100./..-...
mole percent ..100./..-...

Internal heat exchanger	exp.	calc.	dev.
t_{in}	2,49	2,49	-
t_{out}	22,28	22,28	-
p_{in}	4,3	4,4	+2,3
p_{out}	4,0	4,1	+2,4
Q	-	0,23	

compressor Q_{off} = 0,49 kW	exp.	calc.	dev.
	22,8	22,8	-
	118,25	118,25	-
	4,0	4,1	+2,4
	18,2	18,2	-
	1,43	1,44	+0,7

condenser	exp.	calc.	dev.
	118,25	118,25	-
	44,3	44,3	-
	18,2	18,2	-
	18,1	18,1	-
	3,67	3,67	-

Internal heat exchanger	exp.	calc.	dev.
t_{in}	44,3	44,3	-
t_{out}	33,85	33,24	-1,8
p_{in}	18,1	18,1	-
p_{out}	18,1	18,1	-
Q			

valve	exp.	calc.	dev.
	33,85	33,24	-1,8
	-3,0	-3,0	-
	18,1	18,1	-
	4,4	4,5	+2,2

evaporator	exp.	calc.	dev.
	-3,0	-3,0	-
	2,49	2,49	-
	4,4	4,5	+2,2
	2,5	2,72	+8,8

Experiment Nr.: 164 (Fin.)

Refrigerant mixture: R.22../ R..114

COP: exp.: 2,62 / calc.: 2,62

Composition: weight percent 60 / 40
mole percent 74,68 / 25,32

Internal heat exchanger					compressor Q_{off} = 0,16 kW					condenser				
	exp.	calc.	dev.			exp.	calc.	dev.			exp.	calc.	dev.	
t_{in}	4,66	4,66	-			8,31	8,31	-			106,58	106,58	-	
t_{out}	8,31	8,31	-			106,58	106,58	-			41,89	41,89	-	
p_{in}	3,0	3,17	+5,7			2,71	2,88	+6,3			13,8	13,8	-	
p_{out}	2,71	2,88	+6,3			13,8	13,8	-			13,8	13,72	0,6	
Q		0,04	-			1,07	1,07	-			2,8	2,8	-	
Internal heat exchanger					**valve**					**evaporator**				
	exp.	calc.	dev.			exp.	calc.	dev.			exp.	calc.	dev.	
t_{in}	41,89	41,49	-			41,16	39,25	-			-6,06	-6,06	-	
t_{out}	41,16	39,35	-4,4			-6,06	-6,06	-			4,66	4,66	-	
p_{in}	13,8	13,72	-0,6			13,8	13,72	-0,6			3,1	3,27	+5,5	
p_{out}	13,8	13,72	-0,6			3,1	3,27	+5,5			3,0	3,17	+5,7	
Q		0,04	-								1,5	1,89	+26,0	

Conclusions

The experiments did not show as high an improvement of the COP as the theory predicted, if any at all. Simulation of the experiments with a computer program showed, that the refrigerant mixture behaves as predicted, but the equipment does not. Especially the compressor, which works at a substancially changed load and suction volume, drops significantly in its efficiency. Furthermore, it was necessary to simulate the heat losses of the system. Also temperatures and flow rates in the heat exchangers changed in a way, that performance was reduced.

The work showed, that it is obviously not possible to improve the COP of existing heat pumps by the use of refrigerant mixtures, but at the same time it became clear, that the design of a tailor-made mixture heat pump must be possible and that it will show performance as predicted.

4. RECENT DEVELOPMENTS IN THE FIELD OF HIGH TEMPERATURE HEAT PUMPS

In the State-of-the-art report of the IEA-Agreement on Advanced Heat Pumps, Annex VI, the status of the research and development on high temperature compression heat pumps has been overviewed up to midth 1983. Since this time some further papers have been published. They will be discussed briefly here.

Some interesting papers on high temperature heat pumps of various types - Brayton-cycle, compression and absorption - have been presented at the international seminar "Energy Conservation in Industry" of the Commission of the European Communities /4.1/. Beside others Eustace reported on experiments on a steam generating heat pump. A detailed description of this unit is given already in the State-of-the-art report of Annex VI. Table 4.1 summarizes experimental results. Results are reported of the part load running tests, using a cylinder by-pass arrangement on the reciprocating compressor (Figure 4.1). The refrigerant used is R114.

Table 4.1: Experimental results of the IRD high temperature heat pump /4.1/.

Test	Fuel Input kW	Total Heat Output kW	Evap-orating Temp °C	Cond-ensing Temp °C	Engine Speed RPM	Compr-essor Load %
Off-Design Point	180	228	44	118	1100	100
	196	284	43	119	1100	100
	211	311	55	119	1100	100
Design Point	218	327	60	120	1100	100
Part Load	201	299	60	118	1100	80
	165	223	61	116	1100	50
Application Simulation	201	292	60	120	1000	100
	287	400	60	122	1350	100

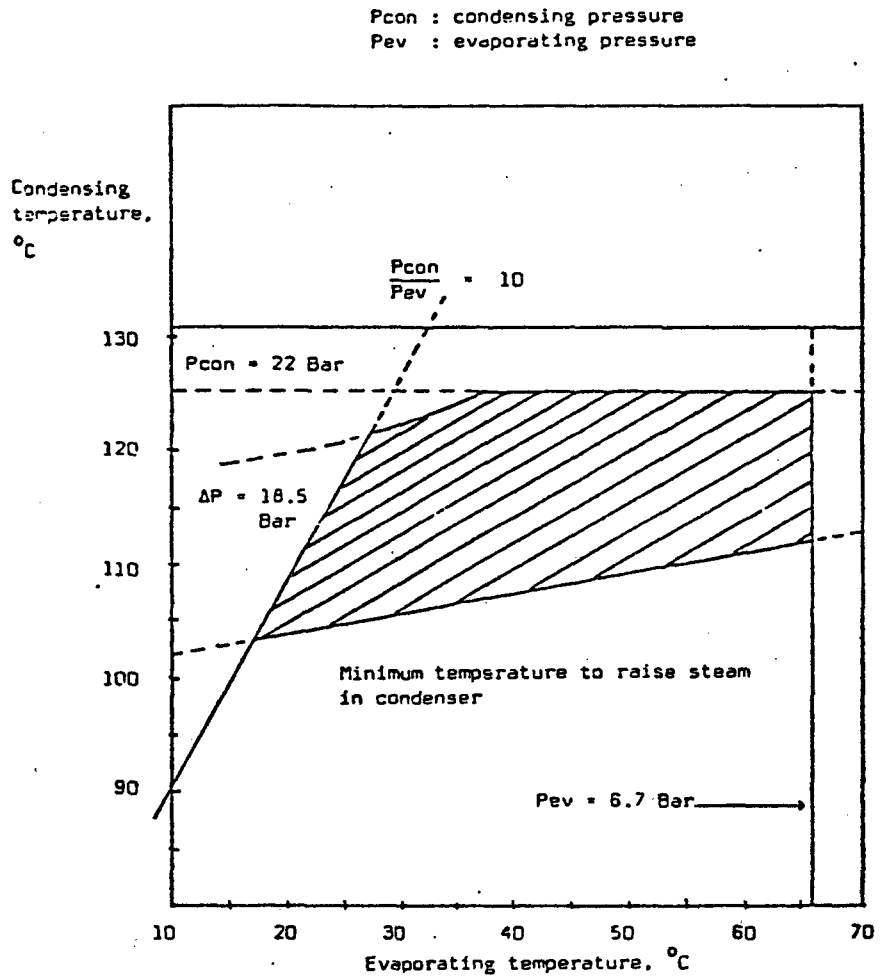

Figure 4.1 Limits of operation of IRD heat pump system imposed by compressor and heat transfer characteristics.

One other theoretical and experimental work has been presented by Taylor and Bertinat at the 2nd BHRA International Symposium on the Large Scale Application of Heat Pumps at York in September 1984 /4.3/.

Their survey of the physical and thermodynamic properties of working fluids for use at temperatures up to 150 °C ruled out the conventional halocarbon refrigerants. Operating within the constraints on non-flammability, stability,

non-toxicity and moderate working pressures a fully fluorinated heat transfer fluid (perfluoro-n-hexane) was chosen for further study. The solubility of this working fluid in various compressor lubricants and its effect on their viscosities and chemical and thermal stabilities were measured. A synthetic hydrocarbon lubricant was chosen. A test rig was designed and constructed using a 5 kW semihermetic compressor and coaxial finned tube heat exchangers with pressurised water as the heat source or the heat sink (Figure 4.2). Tests were carried out at evaporating temperatures of 75-105 ^{o}C and condensing temperatures of 130-155 ^{o}C. Depending on the temperature range, coefficients of performance between 2 and 4 were obtained (Figures 4.3 and 4.4), somewhat less than anticipated. It was found that heat losses from the compressor body accounted for a substantial part of the inefficiencies. Satisfactory operation for several thousand hours was carried out and examination of components after these tests showed no major wear or stability problems.

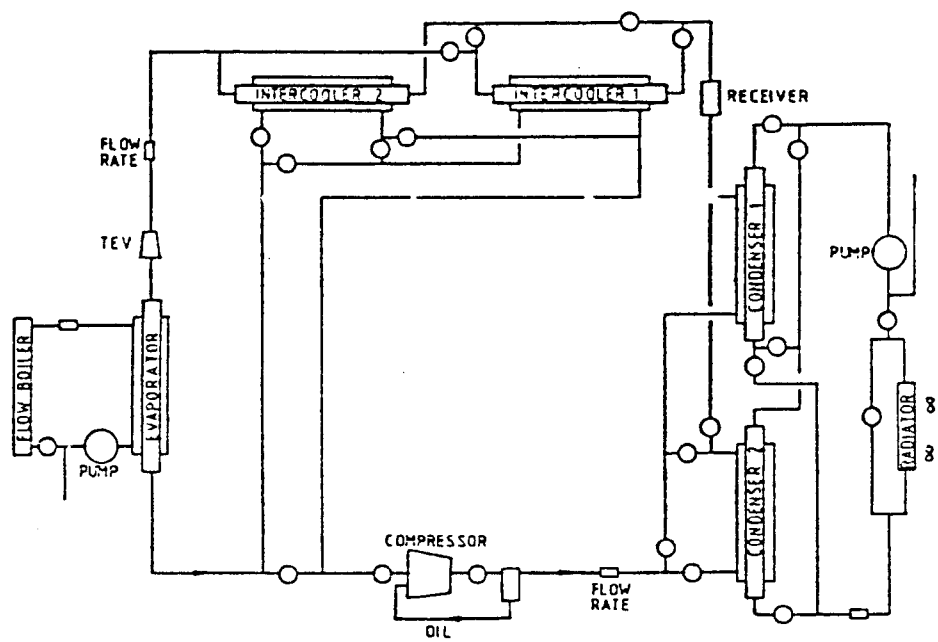

Figure 4.2 Schematic of test rig.

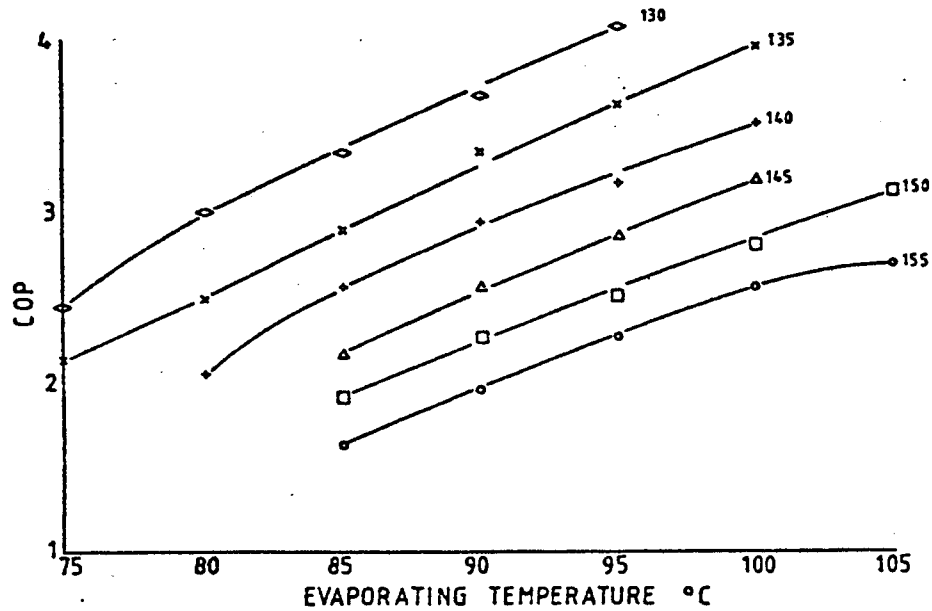

Figure 4.4 COP against evaporating temperature for all condensing temperatures.

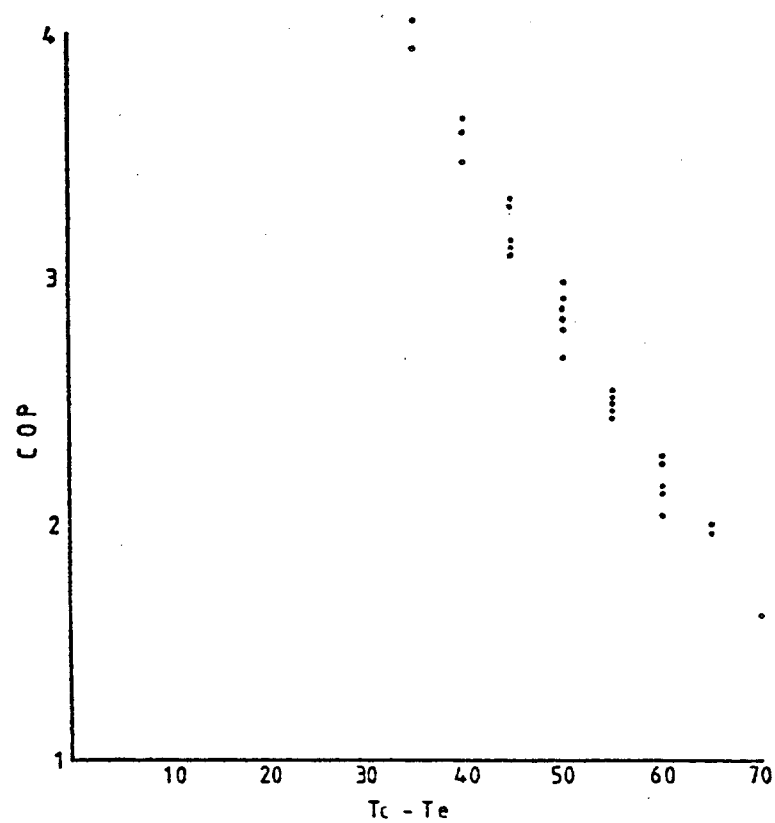

Figure 4.4 Coefficient of performance against temperature difference $T_c - T_e$.

Not too many activities have been published in Europe in this period. Also no National Team of this working group has brought in any report on high temperature working fluids or heat pump systems. No real installation of a closed cycle compression heat pump condensing at a temperature level above 100 ^{o}C has been done in the participating countries, as far as the author of this paper knows.

References

/4.1/ Commission of the European Communities: Energy Conservation in Industry, Düsseldorf, Feb. 1984.

/4.2/ Commission of the European Communities: The Community´s second energy R & D programme (1979-1983), edited by P. Zegers and P.A. Pilowachi 1983.

/4.3/ BHRA: The large scale application of heat pumps, International symposium York, England, Sep. 1984.

5. TECHNICAL SENSITIVITY ANALYSIS

5.1 Introduction

In this chapter various technical aspects of using non-azeotropic mixtures in heat pumps are discussed. The various possibilities for improving the performance with the aid of mixtures have already been discussed in the State-of-the-art report of this annex, see pp. B8-B12.

In the following the influence on the COP and the heating capacity of various parameters are presented. From these results general conclusions about several of the other aspects mentioned above can also be drawn. All results are shown for two cases, one space heating application and one industrial application.

The calculations have been made with two computer programs. The first program simulates the heat pump process with infinite heat exchanger areas. The second, considerably more complex, program simulates the process with given conditions on the heat sink and the heat source sides. With the first program the general influence on the COP and the heating capacity of the following parameters have been studied:

Type of mixture

Composition

Temperature gradients

Degree of subcooling

Internal superheating

Compressor efficiency

With the second program a more realistic behaviour of a heat pump in a given situation has been simulated. The performance of the heat exchangers, the compressor, influence of the size of the heat exchangers, pressure drops, etc. have been taken into account.

5.2 Computer programs

In this section the computer programs used for the parameter study in this chapter is described. The thermodynamical and physical properties and the technical components used in the calculations are also described.

The calculations have been made with two different computer programs. The first program simulates the heat pump with infinite heat exchanger areas. The second program is more complex with given heat exchanger areas etc.

The programs are made in a flexible way with relatively short main programs and a large number of subroutines. All the technical information as compressor efficiencies, heat transfer equations, etc., are situated in the subprograms. The main program handles the input and output of data and some of the iterations. Thermodynamical and physical data for the pure fluids and their mixtures are taken from a library program /5.4/.

Both programs have been made as purely technical ones.

Computer programs with infinite heat exchanger areas

The reason for evaluating and using this program is to study the general influence of different pure working fluids and their binary mixtures in a one stage heat pump cycle. This heat pump cycle includes possibilities of subcooling

on the condenser side and superheating by internal heat exchanging, see Figure 5.2.1. As the heat exchanger areas are infinite no heat transfer relations are included. The compressor efficiencies are given as input data to the program. The program calculates COP, capacity, discharge pressure, etc., from given outer conditions as heat sink and heat source temperatures.

Computer program with finite heat exchanger areas, pressure drops, and realistic compressor efficiencies

The aim of this program is to simulate the components in the heat pump as realistically as possible. The program simulates a single stage heat pump with possibilities of subcooling on the condenser side and superheating by internal heat exchanging as shown in Figure 5.2.1. From given outer conditions as heat source and heat sink temperatures, heat demand and given heat exchanger areas the program simulates the heat pump process. The heat exchanger processes are simulated in the program by stepwise calculations along the heat exchanger areas. This gives the temperature profiles in the heat exchanger as output data. Other output data from the program are: COP, capacity, discharge pressure, heat transfer coefficients, pressure drops, and compressor efficiencies. The technical data of the condenser, evaporator, and compressor used in this program are described below.

Thermodynamical and physical properties

This section describes the methods that have been used for estimating thermodynamical and physical property data of mixtures in this project. The only mixtures considered here are binary nonazeotropic mixtures of halogenated hydrocarbons.

Thermodynamical properties

There is today too little experimental data and knowledge about properties of mixtures to give the complete set of data needed for simulating heat pump cycles. Due to this, it has been decided in this project to use the rules for ideal solutions /5.13/ combined with the properties of pure fluids. The rules for ideal solutions give an accurate estimation of the mixture properties if the mixture consists of similar fluids. Data of the pure working fluids have been taken from a library program constructed by Johansson /5.4/. In this program the thermodynamical data are calculated by the following four equations:

- State equation
- Equation of saturated vapour
- Equation for density of saturated liquid
- Equation for heat capacity of ideal vapour

The state equation used is the one by Martin-Hou and the rest have been developed by Martin. The equation constants are from Downing /5.2/ and some equations limits are from Ekroth /5.3/.

Bubble and dew point data

As the mixture is looked upon as en ideal solution, the calculation of bubble and dew points has been made by using Raoult´s and Dalton´s laws.

Specific enthalpies

The enthalpies of the mixtures have been calculated by linear weighing of the enthalpies of the pure components

$$h_{12}(T, P) = \xi\, h_1(T, P) + (1 - \xi) h_2(T, P)$$

When estimating vapour enthalpies near the dew point of a mixture the state of the less volatile pure component often lies within the saturated area. This means that the pure fluid at this temperature and pressure in reality is partly condensed. In these cases the vapour enthalpy of the less volatile component is estimated by extrapolating from the state of saturated vapour along an isobar to the temperature of the mixture. This state is a pseudostate. The method of using pseudostates for calculating properties is commonly used within chemical engineering.

Specific volumes

The specific volumes of the mixtures have been calculated with the same method as the enthalpies.

$$v_{12}(T, P) = \xi\, v_1(T, P) + (1 - \xi) v_2(T, P)$$

When estimating specific volumes of vapours near the dew point the same situation as in the case of enthalpy estimation can appear. In these cases the specific volume of the less volatile component is calculated by using the compressibility factor and extrapolating along the actual isobar.

Specific entropies

The specific entropies of the mixtures have been calculated as follows:

$$s_{12}(T, P) = \xi\, s_1(T, P) + (1 - \xi) s_2(T, P) + \Delta s_{mix}$$

$$\Delta s_{mix} = -R \cdot (z \cdot \ln(z)) + (1 - z) \ln(1 - z)$$

In the case of estimating vapour entropies near the dew point the same method of extrapolating as for the enthalpies have been used.

Physical properties

When estimating physical properties for the mixtures empirical mixing rules from Reid et al. /5.8/ have been used together with data for the pure working fluids. The data for the pure working fluids have been taken from ASHRAE /5.1/.

The mixing rule used for estimating heat conductivity is:

$$\lambda_{12} = \left[\xi \, \lambda_1^{-2} + (1 - \xi) \lambda_2^{-2} \right]^{-\frac{1}{2}}$$

The liquid viscosity of the mixture is given by:

$$\mu_{12} = \left[x \cdot \mu_1^{\frac{1}{3}} + (1 - x) \cdot \mu_2^{\frac{1}{3}} \right]^3$$

Evaporator

The evaporator is assumed to be a shell and tube heat exchanger with horisontal tubes. It is a water-refrigerant heat exchanger with the working fluid evaporating inside the tubes. The estimation of heat transfer coefficients and pressure drops has been done using equations by Pierre /5.10/. These equations are valid for evaporating of pure working fluids inside smooth horisontal tubes. To take into account the extra heat transfer resistance caused by the mass transport resistance when evaporating a mixture, the following correlation recommended by Varma and Sharma /5.12/ has been used.

$$h_{mix} = h_{ideal} \cdot C$$

where $C = (1 + |x - y|)^n$.

Varma and Sharma have suggested a value of n between -0.90 and -0.94, which was based on their own measurements on R12/R22 mixtures. Comparisons with other measurements published recently, above all Rademacher et al. /5.7/ and Pedersen /5.6/, have shown that a slightly higher value of n seems to give a better agreement between the equation and experimental data. In the calculations n = -1.1 has been used. However, before more accurate values for n can be determined, more experimental data must become available.

The heat transfer coefficient on the water side has been kept at a constant value of 4000 W/m^2, K. The reason for keeping it constant is a desire to be able to study the influence of the heat transfer on the working fluid side in a more distinct way than otherwise would be possible.

For a given evaporator size the number of paralell tubes and hence the tube length can be varied. For all evaporator sizes in all the different cases this number has been technically optimized to give maximum COP when the variation in the pressure drop and the heat transfer coefficient is taken into account.

Condenser

The condenser type chosen for the simulation program is a tube and shell heat exchanger with horisontal finned tubes. It is a water-refrigerant heat exchanger with the refrigerant condensing outside the finned tubes and water inside the tubes. It has been assumed that it is possible to receive counter flow in this type of condenser by using a baffle arrangement on the working fluid side. The heat transfer coefficients for the condensing fluid has been estimated by the Nusselt equation modified by Katz and Beatty /5.5/. There is very little knowledge about how the heat transfer coefficient is influenced by the additional mass transfer resistance when condensing a mixture. Due to this no such

decrease of the heat transfer coefficient has been included in the calculations. However, as the parameter study in section 5.4 includes a study of the influence of the condenser size it should be possible to estimate the influence of such a decrease. This is discussed in more detail in section 5.5.

Compressor

The type of compressor chosen for the simulation is a large twin screw compressor. The data used for describing the compressor behaviour are experimental ones received from Swedish Rotor Machines /5.11/. When estimating the volumetric and isentropic efficiencies the following variables are taken into account: type of pure fluid, pressure ratio, discharge pressure, and condensing temperature. Figure 5.2.2 shows the principal variation of the isentropic and the volumetric efficiencies with the pressure ratio for a large twin screw compressor. The efficiencies when using mixtures have been estimated by linear weighing of the efficiencies of the pure fluids. Recent investigations of compressing, refrigerant mixtures made by Swedish Rotor Machines indicate that this is a reasonable approximation.

5.3 General parameter study

As already mentioned, two examples of applications have been used in this study. These are:

	Heat source	Heat sink
Space heating case	+8 - +4 °C	+62 - +72 °C
Industrial case	+60 - +45 °C	+70 - +90 °C

These examples correspond to case studies example 2, application No. 4, and industrial case 2, respectively, in the State-of-the-art report. In the case of space heating, a higher gradient of the heat source compared with the case study has been used, from 12 ^{o}C to 4 ^{o}C instead of from 8 ^{o}C to 4 ^{o}C. The reason is that this higher gradient is as common in real plants as the other one and that it is more interesting for mixtures.

The results shown below have been calculated with the program for infinite heat exchanger areas, described above. This means that the evaporating temperature for the pure fluids is equal to the outgoing heat source temperature and for the mixtures the evaporating temperature is equal to the incoming or outgoing heat source temperature, depending on the difference in temperature gradient of the heat source and the mixture. For the calculation of the condensing temperature the amount of superheat after the compressor has been taken into account. This means that the condensing temperature for the pure fluids and for the mixtures with smaller temperature gradients than the heat sink one is somewhat lower than the outgoing heat sink temperature.

The maximum allowed condensing temperature for a mixture should always be higher than the corresponding one for the pure more volatile component. The reason is that the mixture has both a lower pressure and, normally, a lower discharge temperature than the pure more volatile component at the same condensing temperature. The knowledge about possible highest condensing temperatures for mixtures is today poor. A reasonable assumption is that condensing pressures up to approx. 23 bar, the pressure limit for much of the standard equipment used today, can be allowed. Therefore, results for systems giving condensing pressures above this value are shown with dotted lines in the diagrams. Results at conditions, where the evaporating pressure is below 1 bar, are also shown with dotted lines.

Influence of the type of mixture and of the composition

For this case three types of mixtures have been used in the calculations, R12/R114, R12/R11, and R22/R114. The results for the COP, the relative capacity (Cap), and the condenser pressure are shown in Figures 5.3.1, 5.3.2, and 5.3.3. The isentropic efficiency in the compressor is in all cases 0.80, the superheating before the compressor is 10 ^{o}C (obtained by internal subcooling), and the external subcooling is 0 ^{o}C.

As can be seen in the figures the mixtures give higher COP:s than the pure working fluids in all three cases. The composition giving the highest COP is always the one having temperature gradients as close as possible to the corresponding ones of the heat sink and the heat source, respectively. The highest COP:s are obtained for the R12/R11 and the R22/R114 mixtures. The reason for the lower COP:s of the R12/R114 mixture is that the highest possible gradients for this one is only about 10 ^{o}C, compared with about 22 ^{o}C for the other ones.

The capacity of course increases with increasing amount of the more volatile component. It can be seen that the capacity can be increased considerably with a mixture compared with the pure working fluids R11 and R114. (Pure R12 is not possible to use in this case due to too high pressure and discharge temperature.) It is normally not possible to achieve high COP:s and high capacities at the same time. However, in the case of e.g. R12/R11 the composition 70 % R12 by weight gives a COP close to the highest one and the capacity is close to that of pure R12. At the same time the condensing pressure is reasonable, 15 bar.

An example of the actual temperature gradients achieved on the condenser and evaporator sides is shown in Figure 5.3.4. It is noticeable that the ratio between the temperature

gradient on the evaporator side and that on the condenser side increases with increasing content of R12 and is in the end close to 1. The reason is that the temperature gradient is almost negligible when the quality is increased from 0 % to about 30 % for a mixture with small amount of the less volatile component.

Influence of external subcooling

For two of the mixtures, R12/R114 and R12/R11, calculations with the maximum possible degree of external subcooling have been made. This means that the condensate is subcooled down to the incoming water temperature of the heat sink. The results are shown in Figures 5.3.5 and 5.3.6. As can be seen the improvement of the COP with the subcooling is considerable for the pure working fluids but is only small for the mixtures. This is of course due to the fact that the temperature gradient of the mixtures decreases the available amount of subcooling. In practice one should of course use external subcooling as much as possible. This is therefore a more correct way of doing the comparisons than the one done in Figures 5.3.1, 5.3.2, and 5.3.3.

Influence of the temperature gradients

For two mixtures, R12/R114 and R12/R11, calculations have been made with a smaller temperature gradient in the heat sink, from 80 ^{o}C to 90 ^{o}C, and in the heat source, from 55 ^{o}C to 45 ^{o}C. The other conditions are the same as the original ones. The results are shown in Figures 5.3.7 and 5.3.8. For R12/R11 there is a considerable decrease of the COP:s for the mixtures, whereas for R12/R114 the difference is small. The reason is that in this case it is not possible to make benefit of the high gradients. These comparisons show the high influence on the heat pump performance of the temperature gradients as well as on the relation between these gradients and the choice of mixture.

Influence of the compressor efficiency

For one mixture, R12/R114, calculations have been made with a lower isentropic efficiency in the compressor, 0.6 instead of 0.8. The result is shown in Figure 5.3.9. Compared with Figure 5.3.1 the decrease of the COP is, of course, considerable. The decrease is approximately the same for the pure working fluids and the mixtures.

Influence of the degree of superheating

By increasing the internal heat exchanger between the condensate and the suction gas, the performance can be improved. Results from calculations with a superheating of 20 ^{o}C instead of 10 ^{o}C have been made for R12/R114. The result is shown in Figure 5.3.10. Compared with Figure 5.3.1 the COP is increased, but only marginally. The improvement is slightly higher for mixtures than for pure working fluids due to the fact that the internal subcooling results in a higher temperature gradient in the evaporator in the mixture case. However, when also external subcooling is used, this difference between mixtures and pure fluids would decrease.

The small improvement achieved with increased superheating indicates that this is not a very important parameter.

The space heating case

Influence of the type of mixture and of the composition

In these calculations the same conditions as in the industrial case have been used, i.e. isentropic efficiency 0.8, superheating of the suction gas 10 ^{o}C and no external subcooling. The results are shown in Figures 5.3.11, 5.3.12, and 5.3.13. As can be seen the possible improvement with mixtures is smaller in this case than for the industrial one. The reason is of course the smaller temperature gradients. For the same

reason the improvement with R12/R114 is of the same order as with the other mixtures. Besides these differences, the general conclusions are the same as for the industrial case.

Influence of external subcooling, temperature gradients, and compressor efficiency

In Figure 5.3.14 the results for R12/R114 with maximal external subcooling are shown. As can be seen, the improvement on the COP is small for the pure fluids and almost negligible for the mixtures. The reason is that the possible degree of subcooling in this case is small, especially for the mixtures.

The influence of the temperature gradients is shown in Figure 5.3.15 for R12/R114. In this case the temperature gradients of the heat sink and the heat source are +66 ^{o}C to +70 ^{o}C and +8 ^{o}C to +4 ^{o}C, respectively. The possible increase of the COP when mixtures are used is considerably smaller than with the original gradients. It is noticeable that the maximum COP:s are achieved for mixtures with compositions 10%-20% or 80%-90% of R12 in this case. The reason is that these compositions give gradients of the same order as the heat sink and the heat source ones. In Figure 5.3.16 the influence of a lower compressor efficiency, 0.6, is shown for R12/R114. The whole level of the COP is decreased by approximately 20 % for both the mixtures and the pure fluids.

5.4 Parameter study with finite heat exchanger areas and realistic compressor efficiencies

The most accurate way of comparing heat pumps with different pure working fluids and their mixtures is to compare them with economically optimized components for each case. The

results from such a comparison are shown in Chapter 6. In this section the influence on the COP and the capacity etc. of the heat exchanger areas and real compressor behaviour are shown.

The two examples of applications that have been studied, one industrial and one space heating system, are the same as mentioned in the former section. The study of the influence of the heat exchanger areas has been done relative reference sizes of the condenser and evaporator. The reference sizes have been chosen to give reasonable minimum temperature differences for the pure working fluid.

Industrial case

Three mixtures have been used in the calculations, R12/R114, R12/R11, and R22/R114. The reference sizes of the condenser and evaporator have been chosen to give minimum temperature differences of 3.0 ^{o}C and 3.5 ^{o}C, respectively, for pure R114.

Figures 5.4.1, 5.4.2, and 5.4.3 show the COP, the capacity, and the condensing pressure versus composition for the three mixtures. In all cases the heat exchanger areas are equal to the reference sizes. The superheating before the compressor is 10 ^{o}C and it is obtained by internal subcooling. The external subcooling is always chosen to be as high as possible. A minimum temperature difference in this heat exchanger of 3 ^{o}C has been assumed.

The results show an increase of COP for the mixtures compared to the pure fluids. A comparison of these results with those from the study with infinite heat exchanger areas shows that the level of COP is much smaller in these cases. This is due to the lower compressor efficiencies and the temperature differences in the heat exchangers. The maximum increase of the COP is also smaller in this case. For R12/R114

this maximum increase is 12 %. For R12/R11 and R22/R114 the maximum increases are 13 % and 9 %, respectively. The COP curve for R22/R114 is decreasing rapidly at large amounts of R22 due to large pressure ratios, and hence low compressor efficiencies for R22 near its critical point. Taking the condensing pressure and the capacities into account, R12/R114 and R12/R11 seem to be the most interesting mixtures for the industrial case.

For the industrial application all the investigated mixtures give a remarkable increase of the capacity. In the calculation of the capacities shown in Figures 5.4.1, 5.4.2, and 5.4.3 the volumetric efficiency has been taken into account. The difference in pressure ratio and working fluid does not influence the isentropic efficiency very much in this application as the pressure ratios are small in all cases except those with larger amounts of R22.

Influence of the degree of subcooling

In Figures 5.4.4 and 5.4.5 the influence on the COP for the case of no subcooling is shown for R12/R114 and R12/R11, respectively. Compared with the maximum possible degree of subcooling, the COP of course decreases. It is noticeable that this decrease is higher for the pure fluids than for the mixtures. The reason for this is that the possible degree of subcooling is always smaller for mixtures than for the pure fluids due to the temperature gradients.

Influence of the heat exchanger sizes

When increasing the heat exchanger areas for a heat pump with pure fluids, the resulting minimum temperature differences will at large sizes be close to a case with infinite areas. Due to the more even temperature differences in the condenser and the evaporator when using a mixture an increase

of the areas would be expected to give a larger increase in COP compared to the corresponding ones for pure working fluids.

Figures 5.4.6 and 5.4.7 show the influence on the COP by varying the evaporator area for R12/R114 compared to pure R114 and for R12/R11 compared to pure R11. All other variables have been kept constant. Figure 5.4.8 and 5.4.9 show the corresponding results from variation of the condenser area. The results in Figures 5.4.10 and 5.4.11 show the influence on the COP by varying both the evaporator and condenser area.

The results show that the COP for the mixtures is larger for all area sizes investigated here. Figure 5.4.6 and 5.4.7 show that the influence of the evaporator area is larger for the mixture cases than for the pure fluid, which could be expected.

The results from varying the condenser area in Figures 5.4.8 and 5.4.9 show a somewhat larger influence on the COP. The influence is also here larger for the mixtures compared to the pure fluids. This effect is more pronounced in the results from varying both the condenser and evaporator areas in Figures 5.4.10 and 5.4.11. The conclusion of this study is that the heat exchanger sizes have a large influence on the improvement on the COP when mixtures are used. It is also possible to achieve considerable increases compared with pure fluids, if the heat exchanger sizes are increased when mixtures are used.

Space heating case

The investigated mixtures are the same as in the former case. The reference sizes of the condenser and the evaporator have been chosen to give minimum temperature differences of 3 ^{o}C and 4 ^{o}C, respectively, for pure R12.

Figures 5.4.12, 5.4.13, and 5.4.14 show the variation of the COP, the capacity, and the condenser pressure versus the composition. The heat exchanger areas are in all cases equal to the reference size. The superheating before the compressor is 10 ^{o}C and the external subcooling is the maximum possible.

For this case the COP is smaller for most of the mixture compositions, compared to the pure fluids. Only for the cases with a large amount of the most volatile component a little higher COP can be noticed for the mixtures compared to the pure most volatile working fluid. The capacity decreases when adding less volatile working fluid and the condensing pressure decreases.

Comparing these results with those obtained for infinite heat exchanger areas and constant compressor efficiency the most pronounced difference is the very low COP when using large amounts of the less volatile components. The big decrease of the COP in these cases is due to the high pressure ratios resulting in low compressor efficiencies and the high pressure losses in the evaporators (in spite of optimized number of tubes) caused by large specific volumes of the vapours.

The results in Figures 5.4.12, 5.4.13, and 5.4.14 show that the only mixtures of interest of those investigated for this case are those with large amounts of the more volatile component.

Influence of the compressor efficiency

Figure 5.4.15 shows the variation of the COP with the composition while keeping the isentropic efficiency constant and equal to the value received for pure R12. The other conditions are kept equal to those for the former results. The COP achieved here is considerably larger for the mix-

tures and the pure R114 compared to the results from calculations with varying efficiency. This shows that the increase of pressure ratio while increasing the amount of less volatile component has a large influence on the COP.

Influence of the heat exchanger sizes

The influence on the COP by varying the evaporator size for R12/R114 is shown in Figure 5.4.16. Corresponding results received by varying the condenser size and both the heat exchanger sizes are shown in Figures 5.4.17 and 5.4.18, respectively.

The results show that the COP for the mixtures is more sensitive to a variation in heat exchanger size compared to the pure fluids.

It can also be seen that in the cases of medium or large amounts of the less volatile component it is not possible to achieve higher COP:s than for the pure fluids, not even with large extensions of the heat exchanger sizes.

5.5 Discussion and conclusions

From the discussions above it can be concluded, that it is very important to include the real behaviour of the heat exchangers and the compressor, when mixtures are compared with pure fluids. Also when these aspects are taken into account, mixtures can give considerable increases of the COP for cases where the temperature gradients of the heat sink and the heat source are high.

In space heating systems the temperature gradients are in most cases small, as has been shown in the State-of-the-art report of this Annex. One example where the gradients can

be high on both sides is domestic tap water heating by using e.g. ambient or exhaust air as heat source. For this application R13B1/R152A could be an appropriate mixture, as it gives high gradients and at the same time a high capacity. Any calculations of this application have not been made in this work, as data for this mixture have not been available in the computer program used. The mixtures used in this work are most probably not appropriate for this application due to the low capacities and the low evaporator pressure levels that would be achieved.

There are, however, other reasons for using mixtures in space heating applications. This is discussed in Chapter 7 , General Conclusions.

In many industrial applications of heat pumps, the temperature gradients of the heat sink and the heat source are high and hence mixtures could be appropriate. Increases of the COP by at least 15 % should be possible to achieve. However, a common application for future industrial heat pumps would be the production of low pressure steam. In such cases the increase with mixtures would of course be smaller, as there is no temperature gradient of the heat sink.

For the industrial case investigated the capacity can be increased considerably with the aid of a mixture. For the space heating case the capacity can only be improved marginally, when the mixtures R22/R114 is used. For the other mixtures pure R12 always gives the highest capacity. However, for high-temperature space heating systems, where the condensing temperatures are above approximately 80 ^{o}C, and hence pure R12 cannot be used, the improvement could be considerable compared to the possible pure working fluids, e.g. R114 or R11.

For the calculations with finite heat exchangers and realistic compressor efficiencies the most uncertain parameter is the heat transfer coefficient for mixtures in the condenser. Due to the lack of data no additional mass transfer resistance has been assumed, as has already been discussed. However, there is in reality most probably such a resistance. If this resistance decreases the heat transfer coefficient of, say, 30 %, the overall heat transfer coefficient would be decreased by approximately 20 %. This would have approximately the same effect as a decrease in the heat transfer area of the same magnitude. As can be seen in the figures presented, a decrease of the condenser size of 20 % means a decrease of the COP of typically less than 2 %. Hence, the error made by disregarding the mass transfer resistance should be small. A change of the evaporator area of 20 % also means a change of the COP of less than 2 %. Hence, the mass transfer resistance does not have a considerable influence on the COP when mixtures are used. The most important reason for the decrease of the improvement in COP when finite areas are used instead of infinite ones is the influence on the mean temperature difference in the heat exchangers when temperature gradients on the working fluid side are introduced.

5.6 Nomenclature

COP	coefficient of performance	-
n	specific enthalpy	$kJ\ kg^{-1}$
p	pressure	bar
s	specific entropy (mass or molar)	$kJ\ kg^{-1}\ k^{-1}$
v	specific volume	$m^3\ kg^{-1}$
x	molar liquid composition	-

y	molar vapour composition	-
z	molar composition (general)	-

Greek letters:

α	heat transfer coefficient	$W/m^{-2}\ K$
λ	thermal conductivity	$W\ m^{-1}\ K^{-1}$
μ	dynamic viscosity	$kg\ m^{-1}\ s^{-1}$
ξ	weight fraction	-

Subscripts:

1	component 1
2	component 2
12	mixture of comp. 1 and 2

5.7 References

5.1 ASHRAE, Handbook of fundamentals, 1981.

5.2 Downing, R.C.: Refrigerant Equations. Du Pont Company No. 2313.

5.3 Ekroth, I.A.: Termodynamiska egenskaper för köldmedier. Diss. Stockholm 1979.

5.4 Johansson, T., Department of Heat and Power Technology, Chalmers University of Technology, Sweden.

5.5 Katz, D.L., Beatty, K.O.: Condensation of vapours on outside of finned tubes. Chemical Engineering Progress, Vol. 44, No. 1, 1948.

5.6 Pedersen, P.H.: Varmeovergang ved fordampning av blandede kölemidler. Laboratoriet för Köleteknik, DTH, Denmark, Report No. 7, 1983.

5.7 Radermacher, R., Ross, H., Didion, D.: Experimental determination of forced convection evaporative heat transfer coefficients for non-azeotropic refrigerant mixtures. Trans. of ASME, Nov. 1983.

5.8 Reid, Prausnitz, Sherwood: The properties of gases and liquids. McGraw Hill.

5.9 Pierre, B.: Strömningsmotstånd vid kokande köldmedier. Kyltekn. Tidskrf., No. 6, Dec. 1957.

5.10 Pierre, B.: Värmeöverföring vid kokande köldmedier i horisontella rör. Kyltekn. Tidskr., No. 5, Nov. 1969.

5.11 Swedish Rotor Machines (SRM), Sweden. Personal communication.

5.12 Varma, H.K., Sharma, C.P.: Heat transfer coefficients during forced convection evaporation of R-12 and R-22 mixtures in annular flow regime. XVIth Int. Congress of Refrigeration (Venice 1979), Proceedings Vol. II.

5.13 van Wylen, G.J., Sonntag, R.E.: Fundamentals of classical thermodynamics. 2^{e}, John Wiley & Sons.

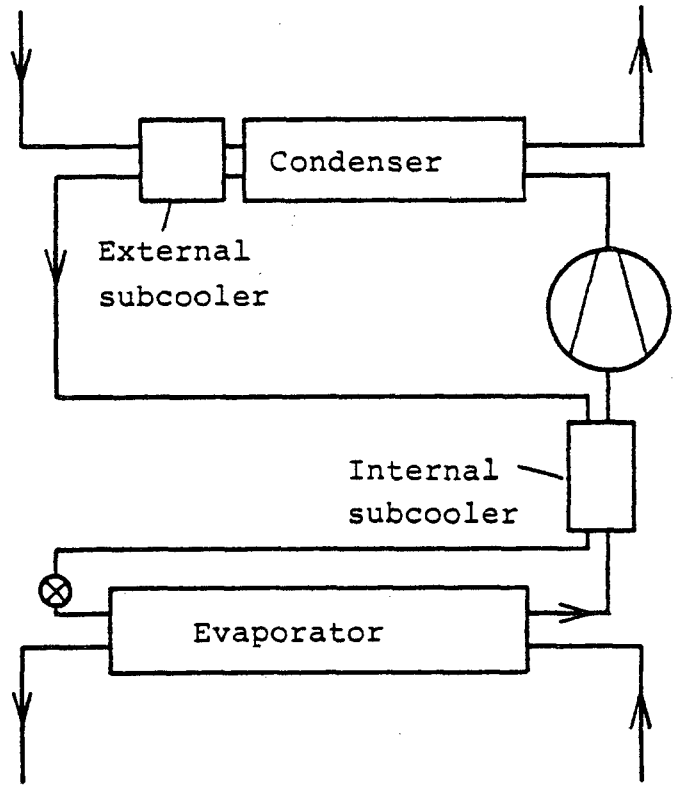

Figure 5.2.1 Single stage cycle.

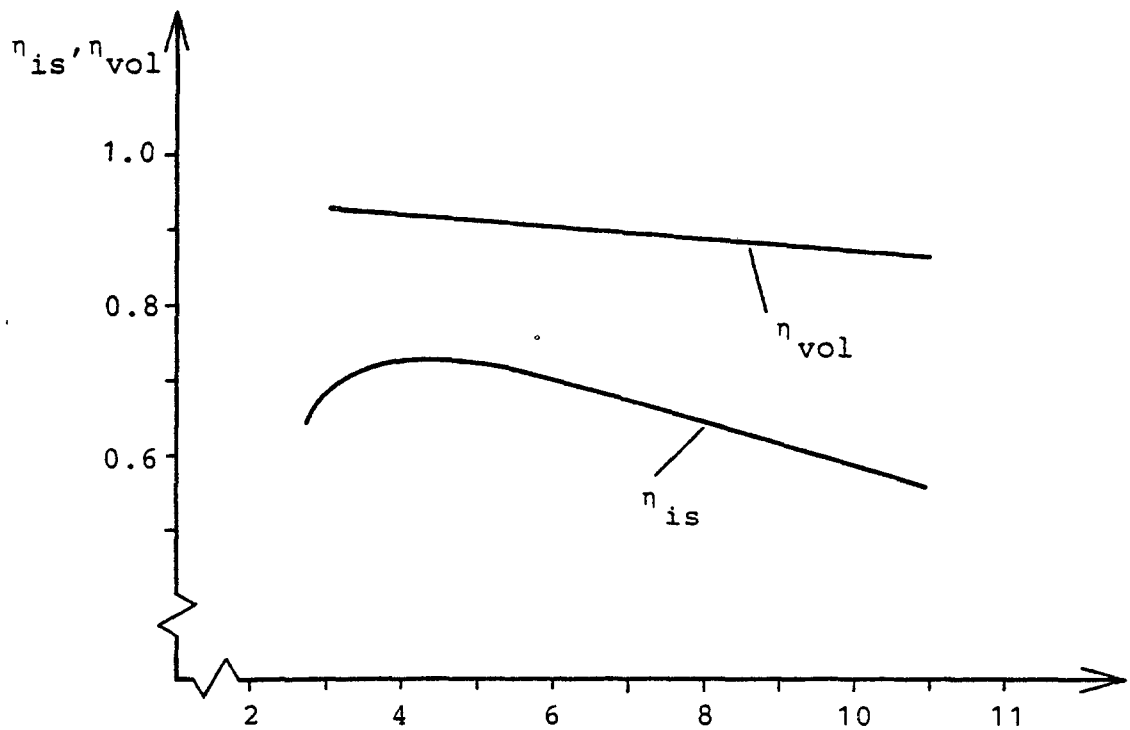

Figure 5.2.2 Principal variation of isentropic and volymetric efficiencies with pressure ratio for a large screw compressor.

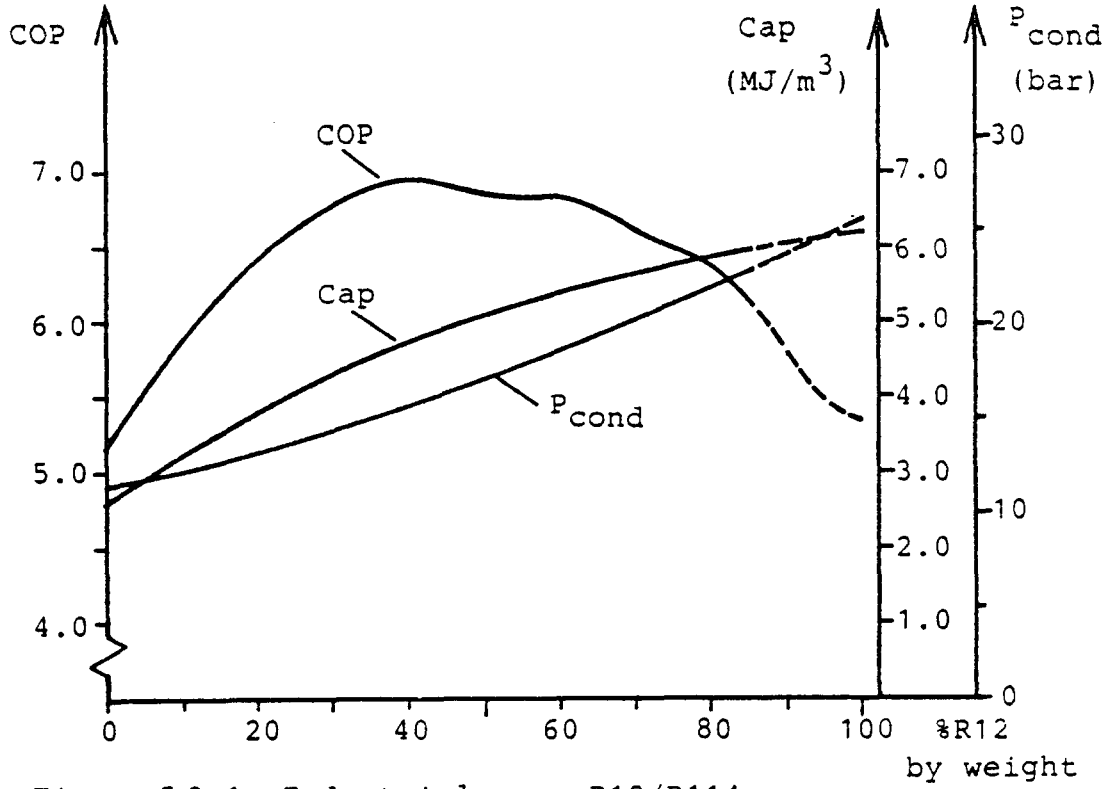

Figure 5.3.1 Industrial case R12/R114.

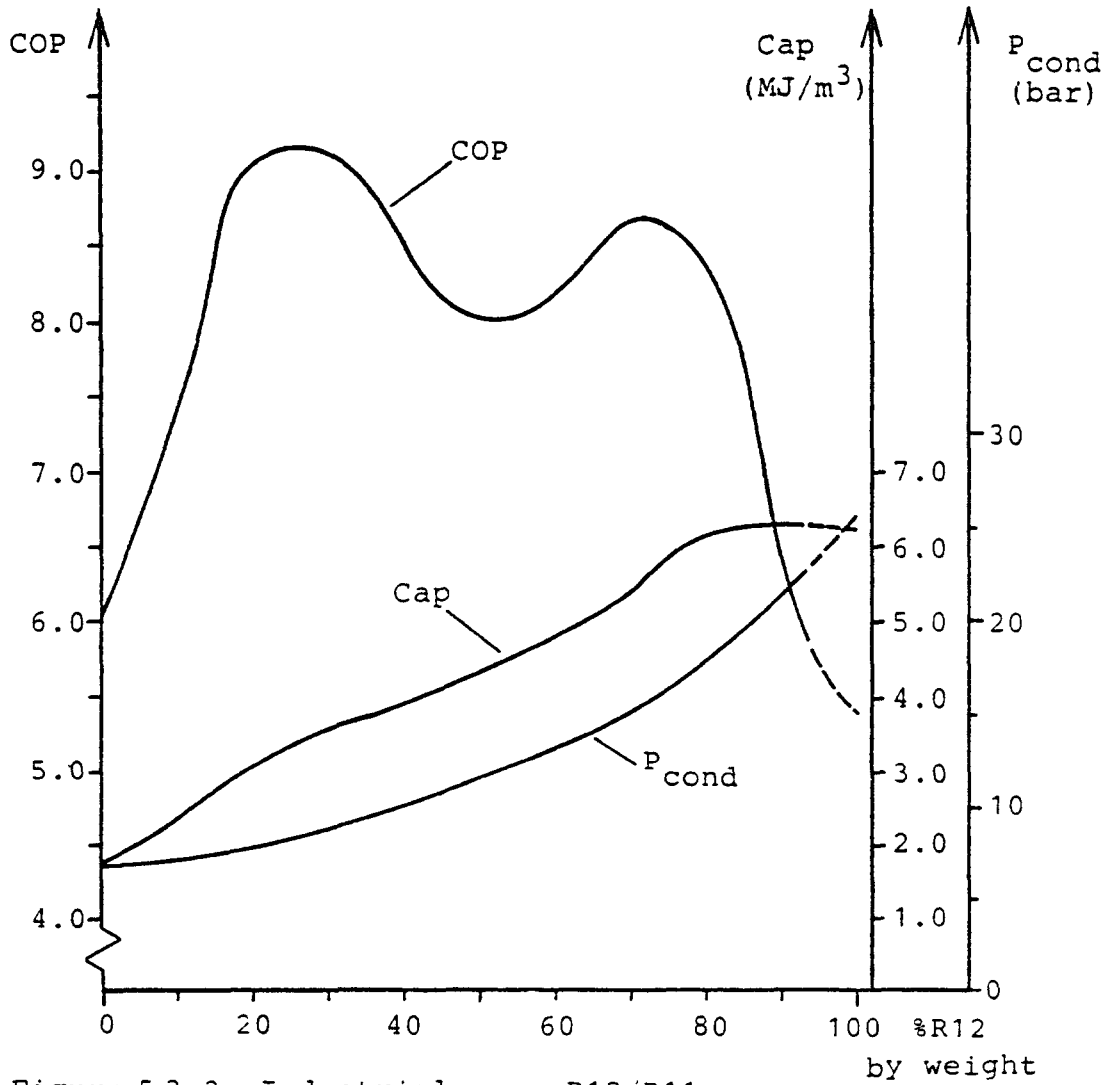

Figure 5.3.2 Industrial case R12/R11.

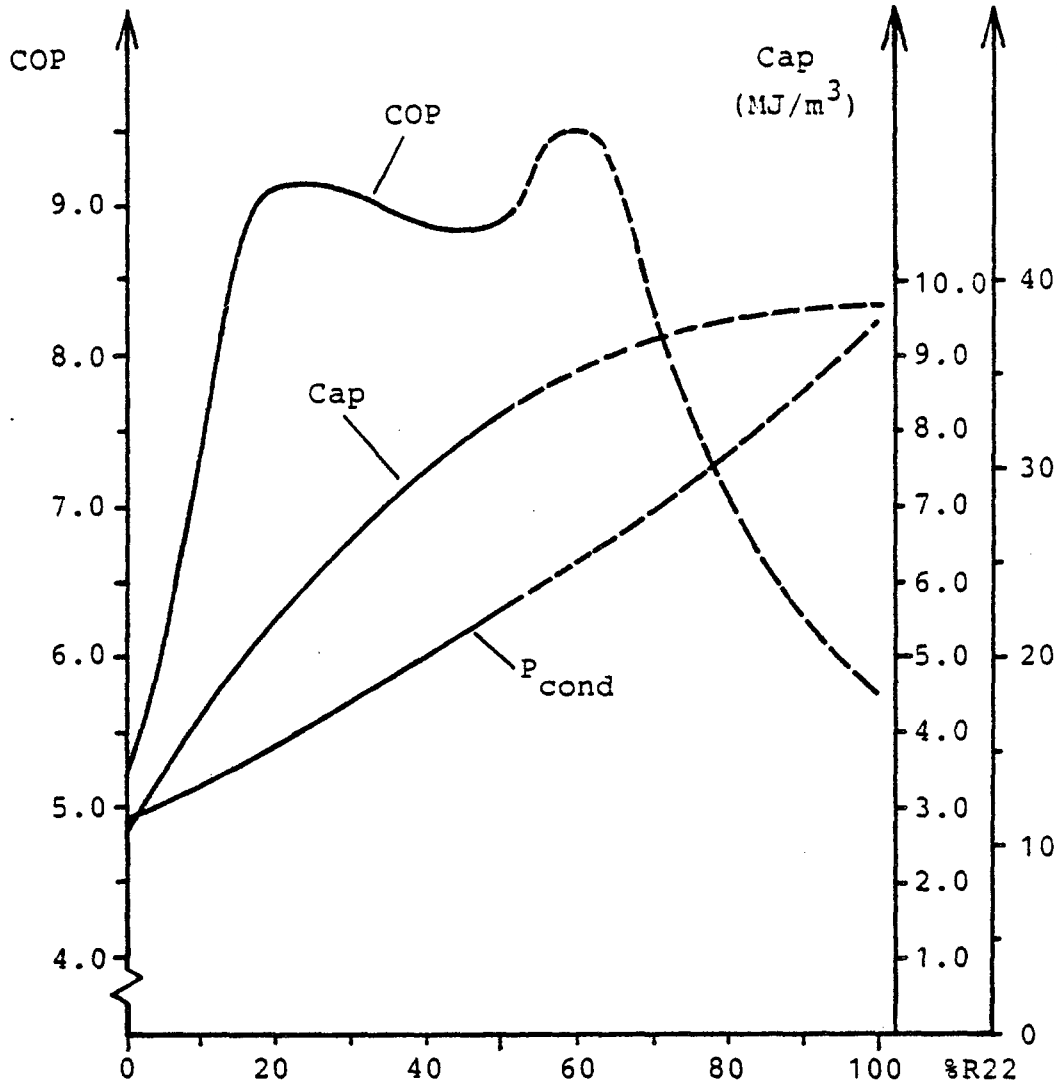

Figure 5.3.3 Industrial case R22/R114.

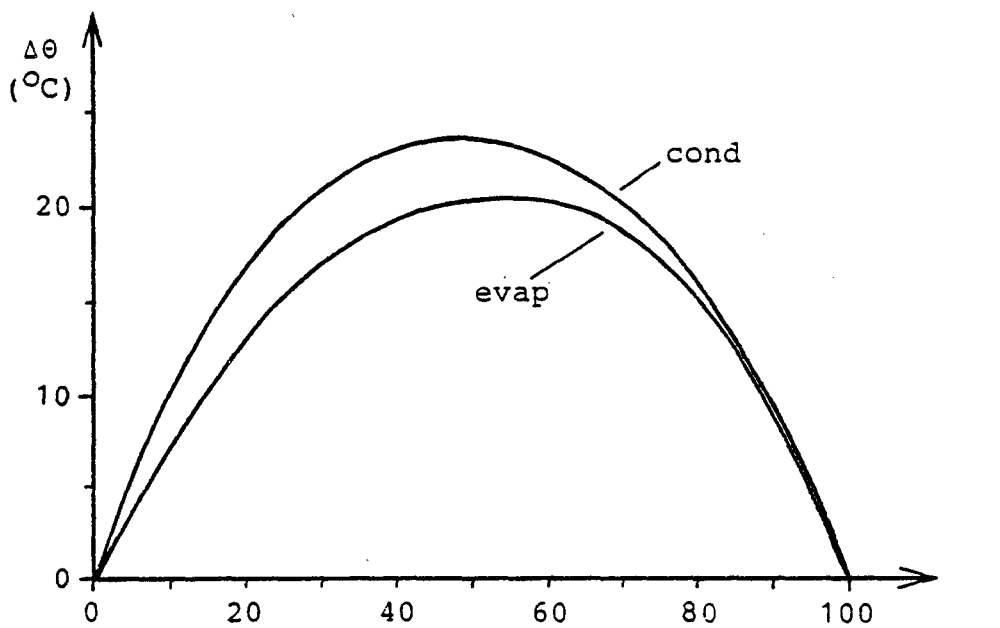

Figure 5.3.4 Temperature profiles in evaporator and condenser versus total composition. R12/R11.

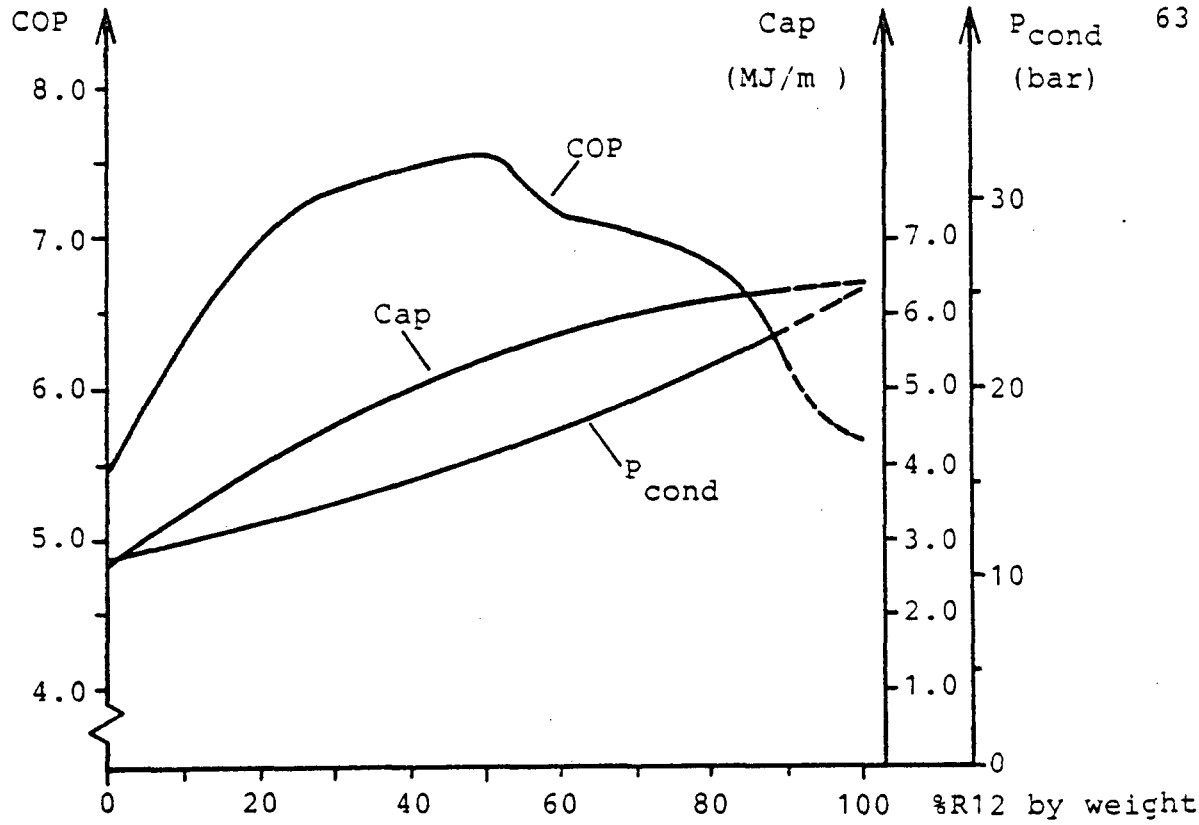

Figure 5.3.5 Industrial case, R12/R114, maximal external subcooling.

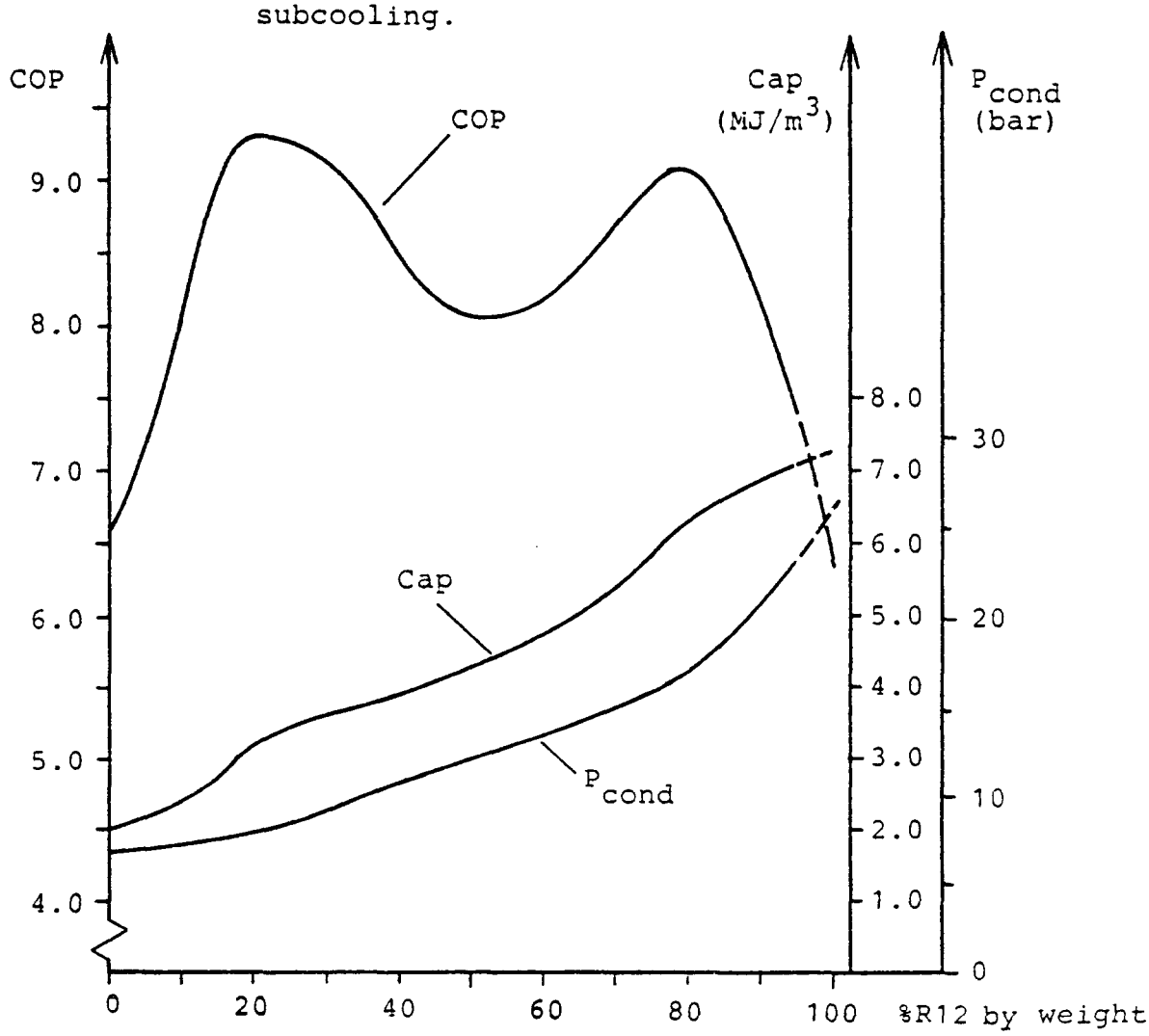

Figure 5.3.6 Industrial case, R12/R11 maximal external subcooling.

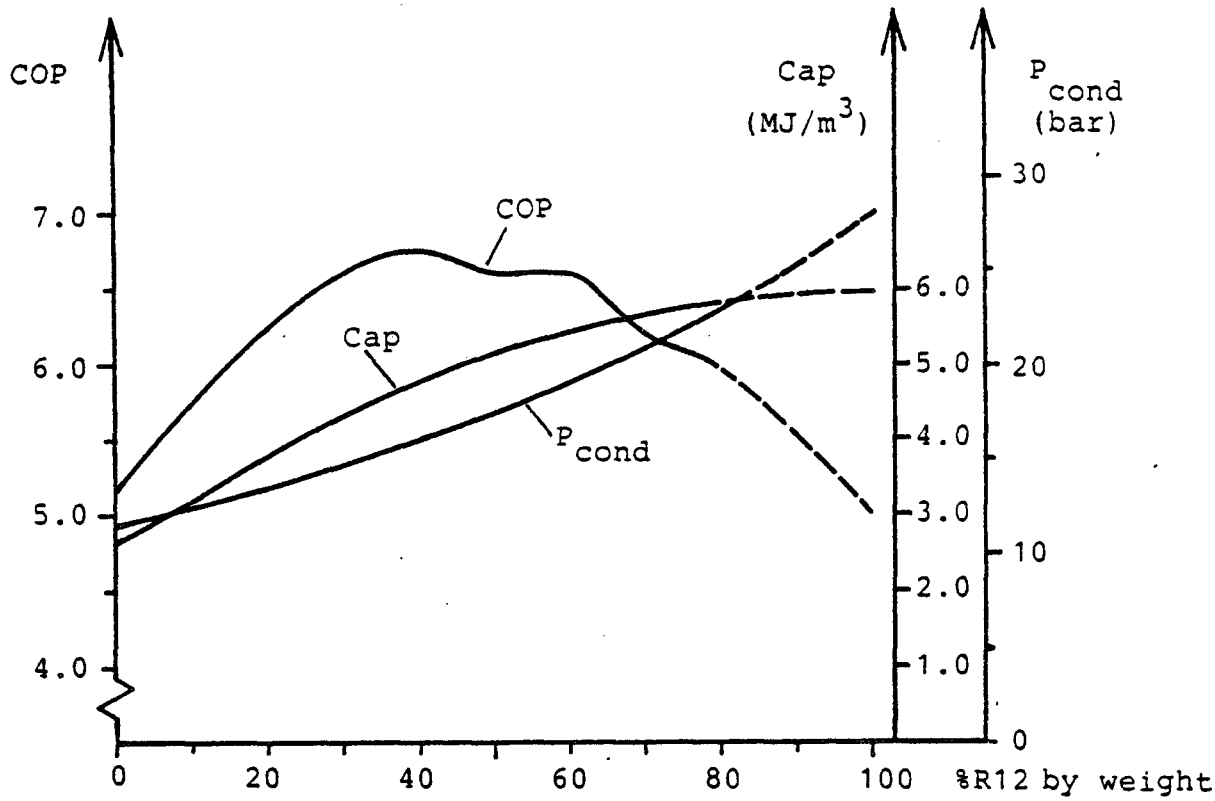

Figure 5.3.7 Industrial case, R12/R114, smaller temperature gradients on heat source and heat sink.

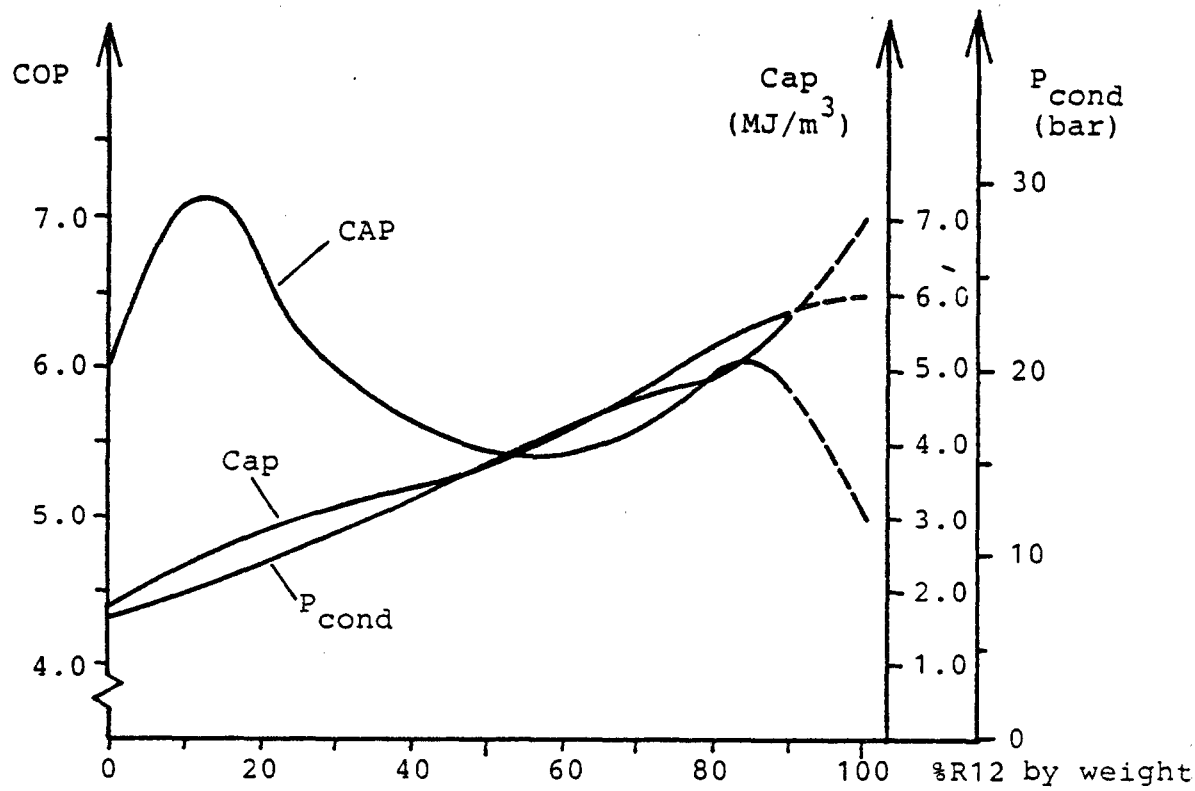

Figure 5.3.8 Industrial case, R12/R11, smaller temperature gradients on heat source and heat sink.

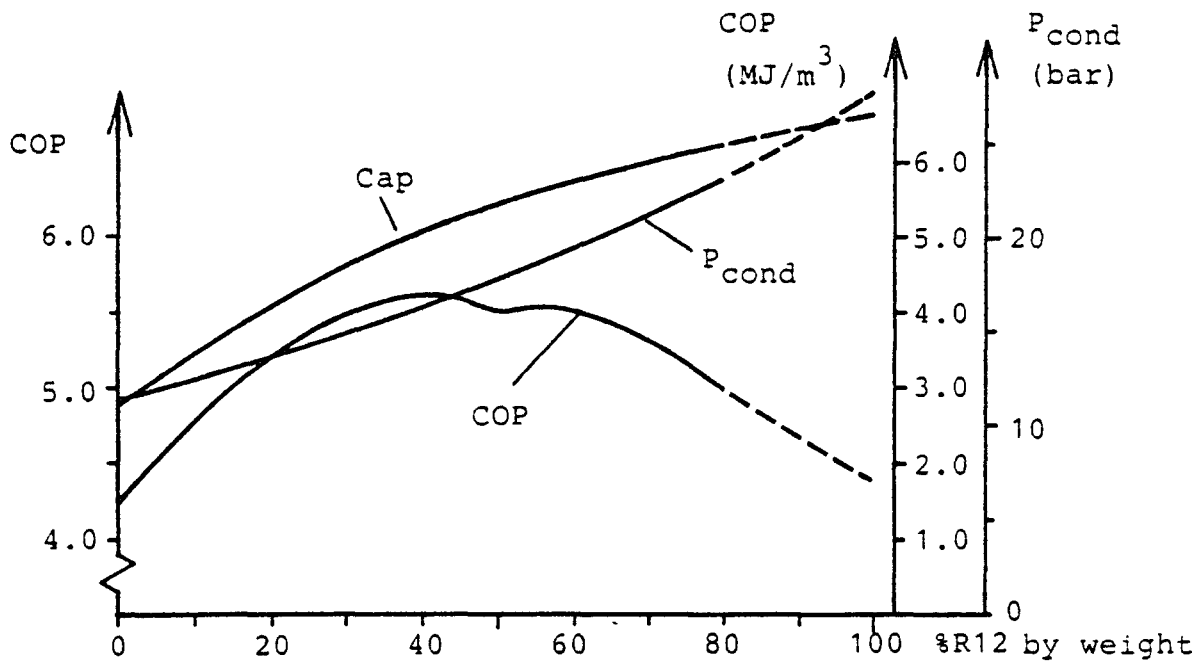

Figure 5.3.9 Industrial case, R12/R114 compressor efficiency equal 0.6 .

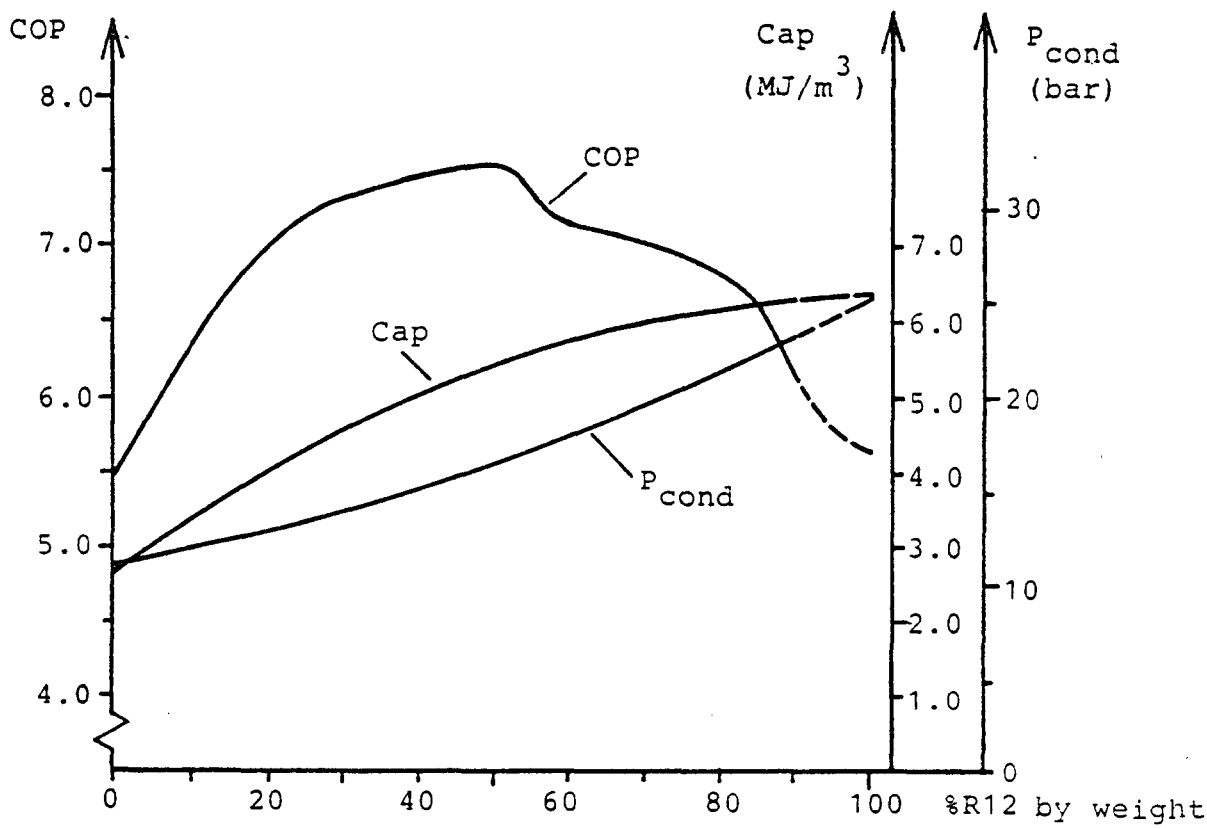

Figure 5.3.10 Industrial case, R12/R114, superheating equal 20 °C.

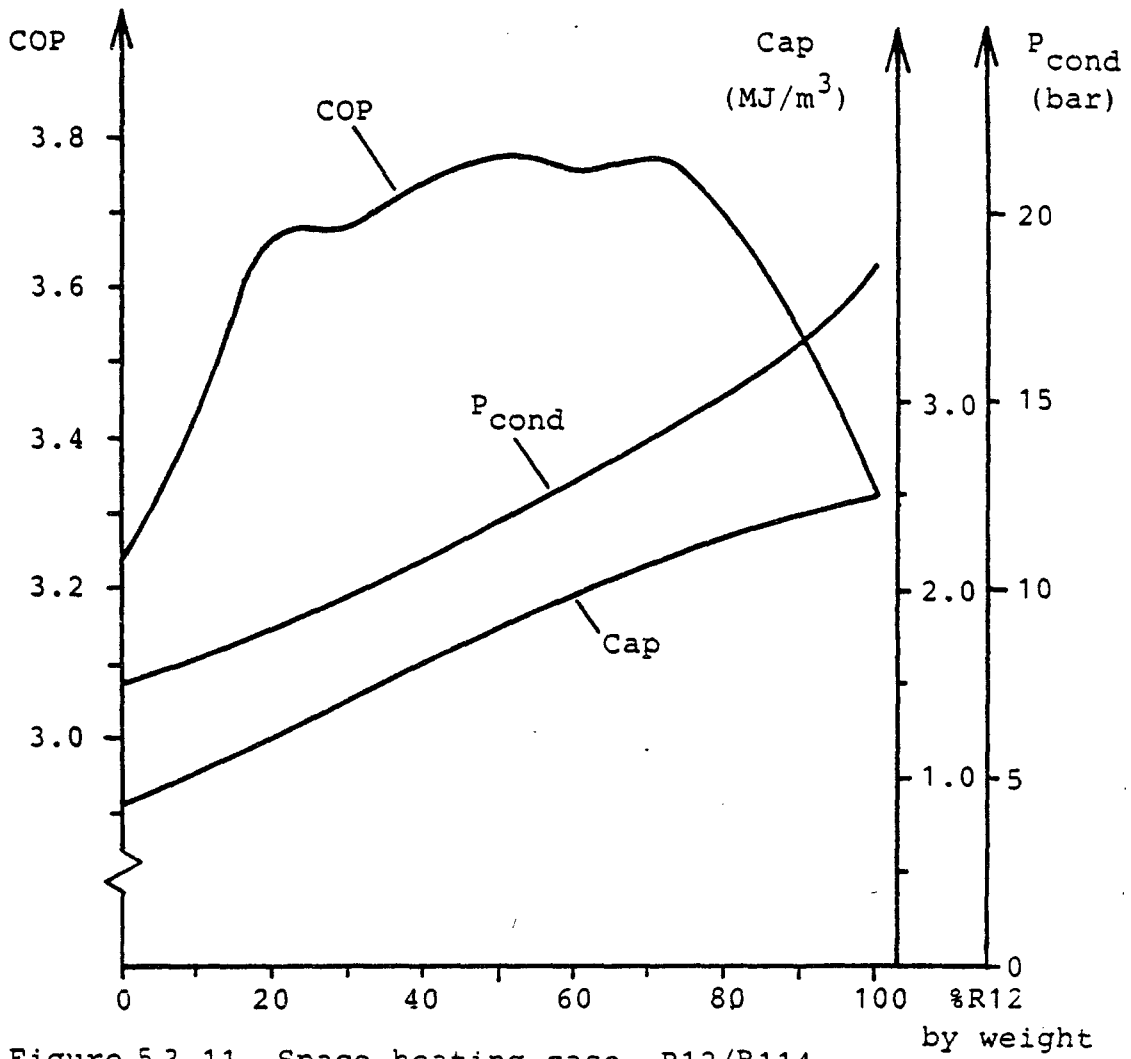

Figure 5.3.11 Space heating case, R12/R114.

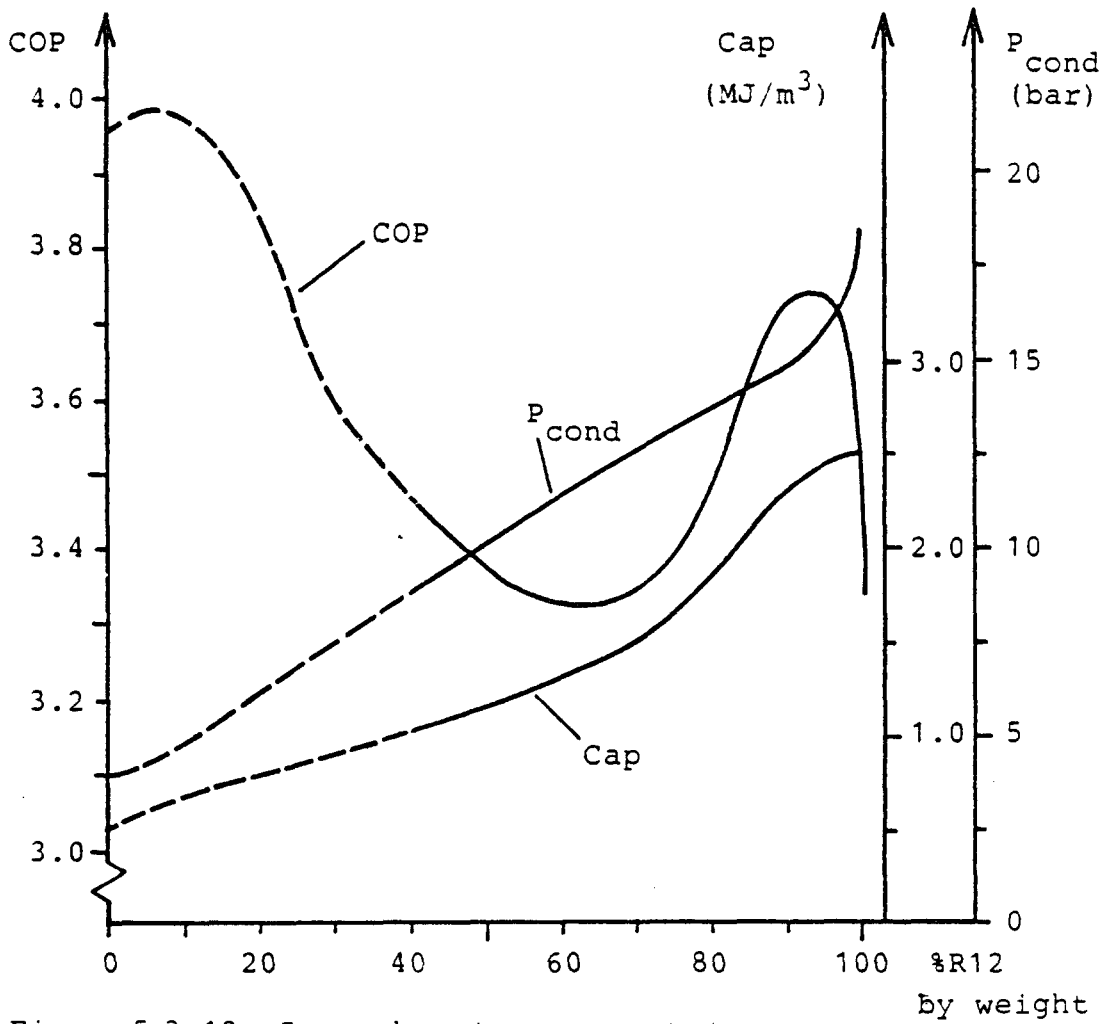

Figure 5.3.12 Space heating case, R12/R11.

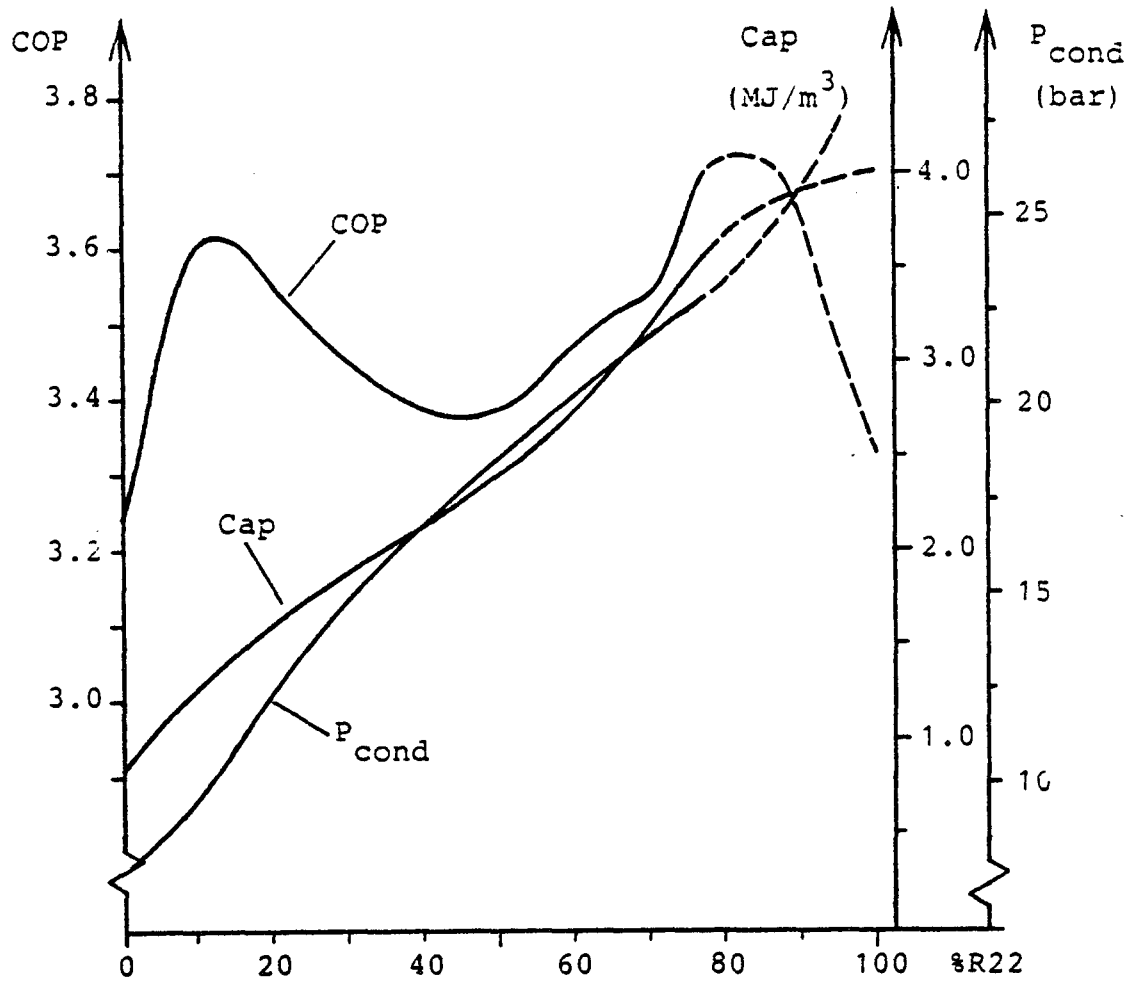

Figure 5.3.13 Space heating case, R22/R114.

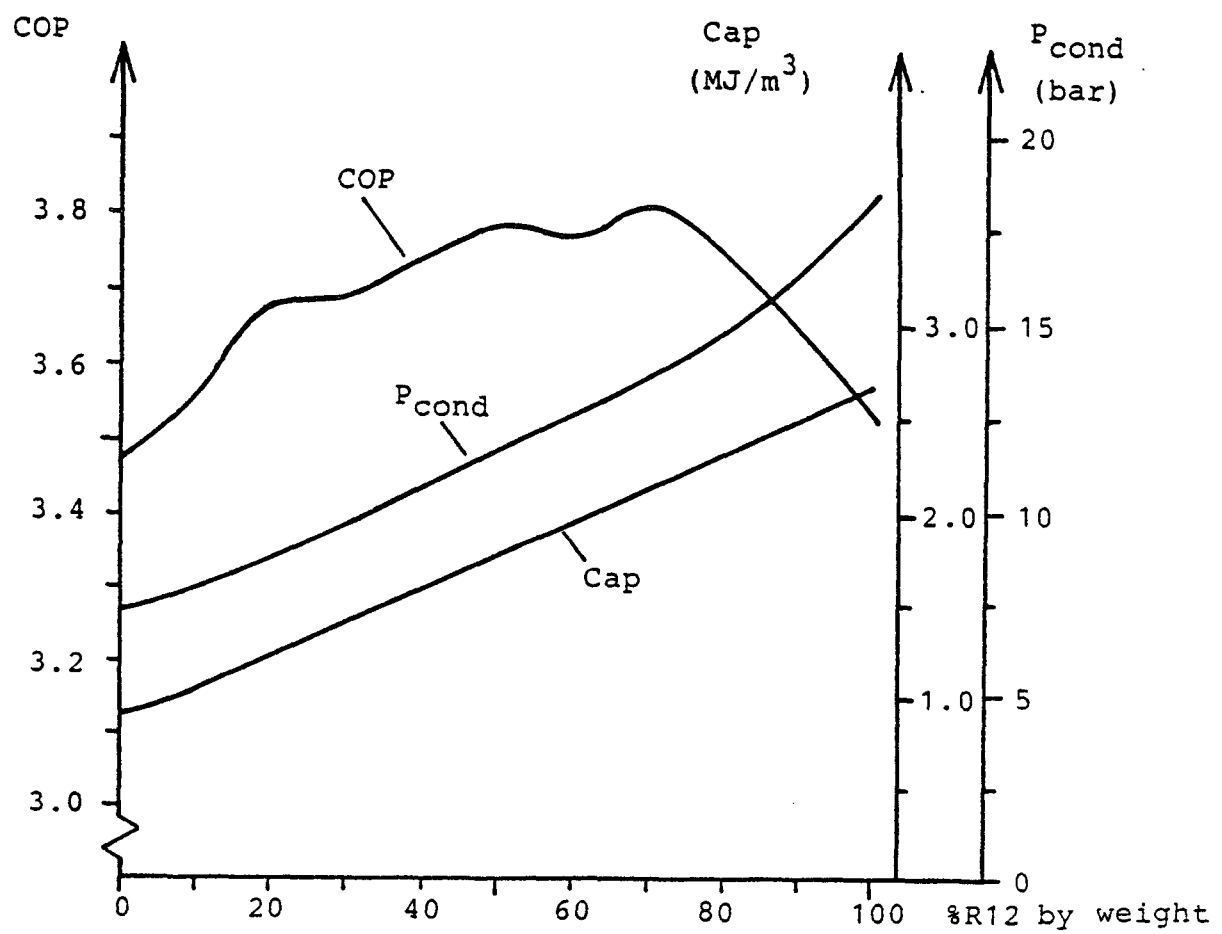

Figure 5.3.14 Space heating case, R12/R114, maximal

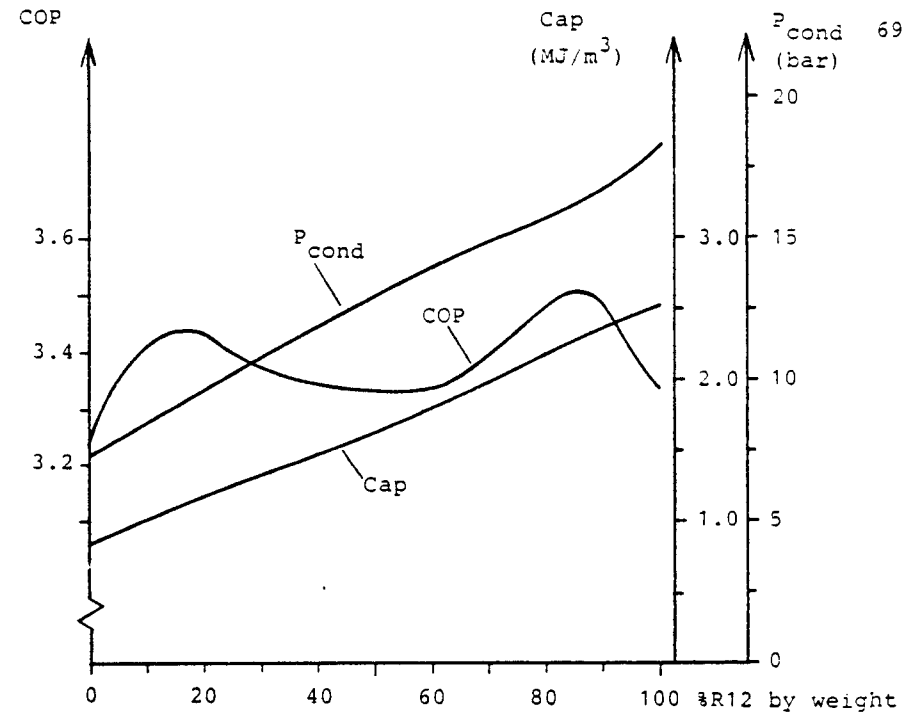

Figure 5.3.15 Space heating case, R12/R114, smaller temperature gradients on heat source and heat sink.

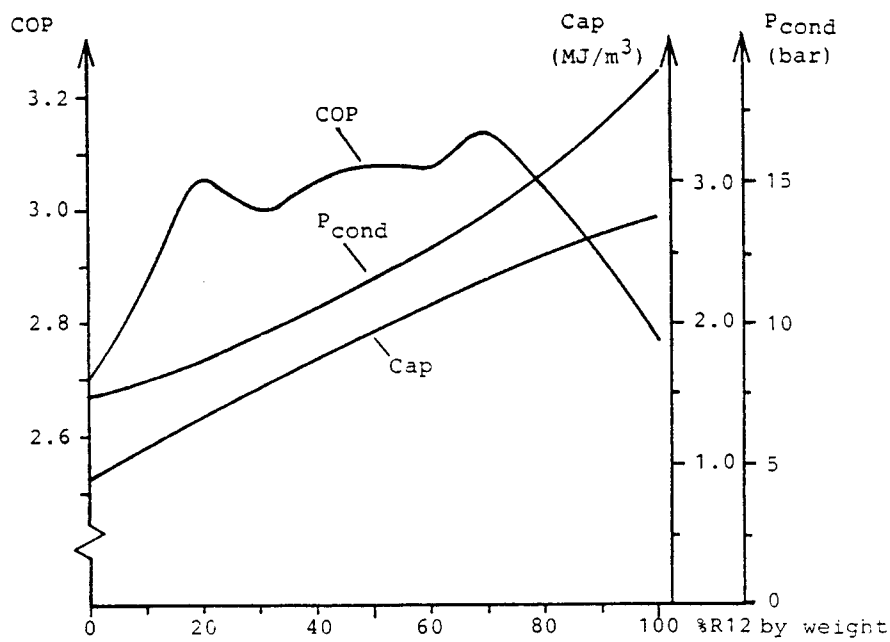

Figure 5.3.16 Space heating case, R12/R114, compressor efficiency equal to 0.6 .

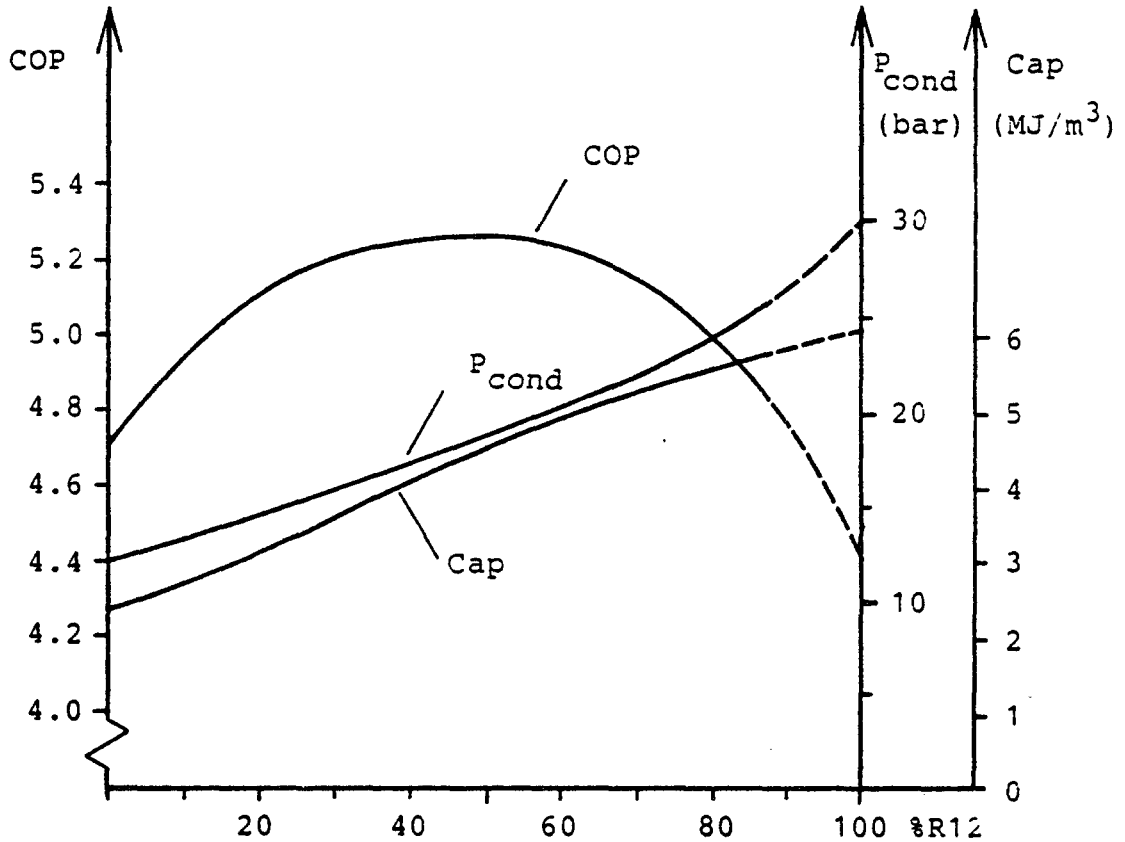

Figure 5.4.1 Industrial case, R12/R114.

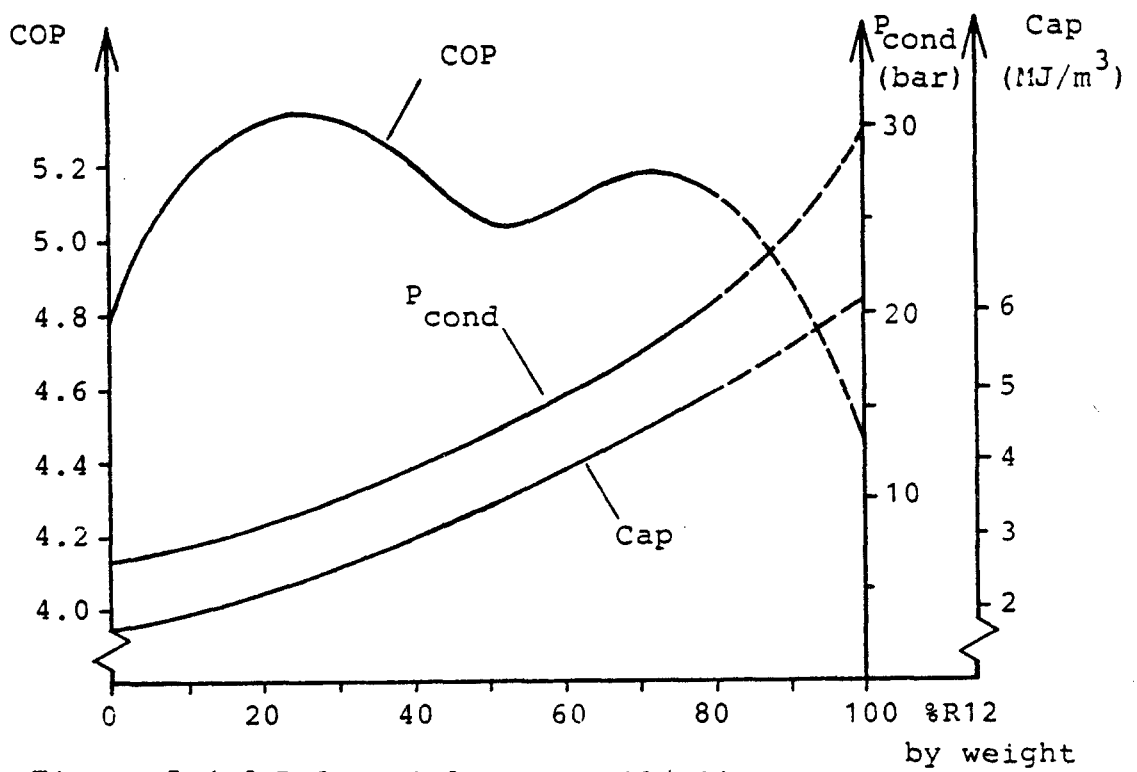

Figure 5.4.2 Industrial case, R12/R11.

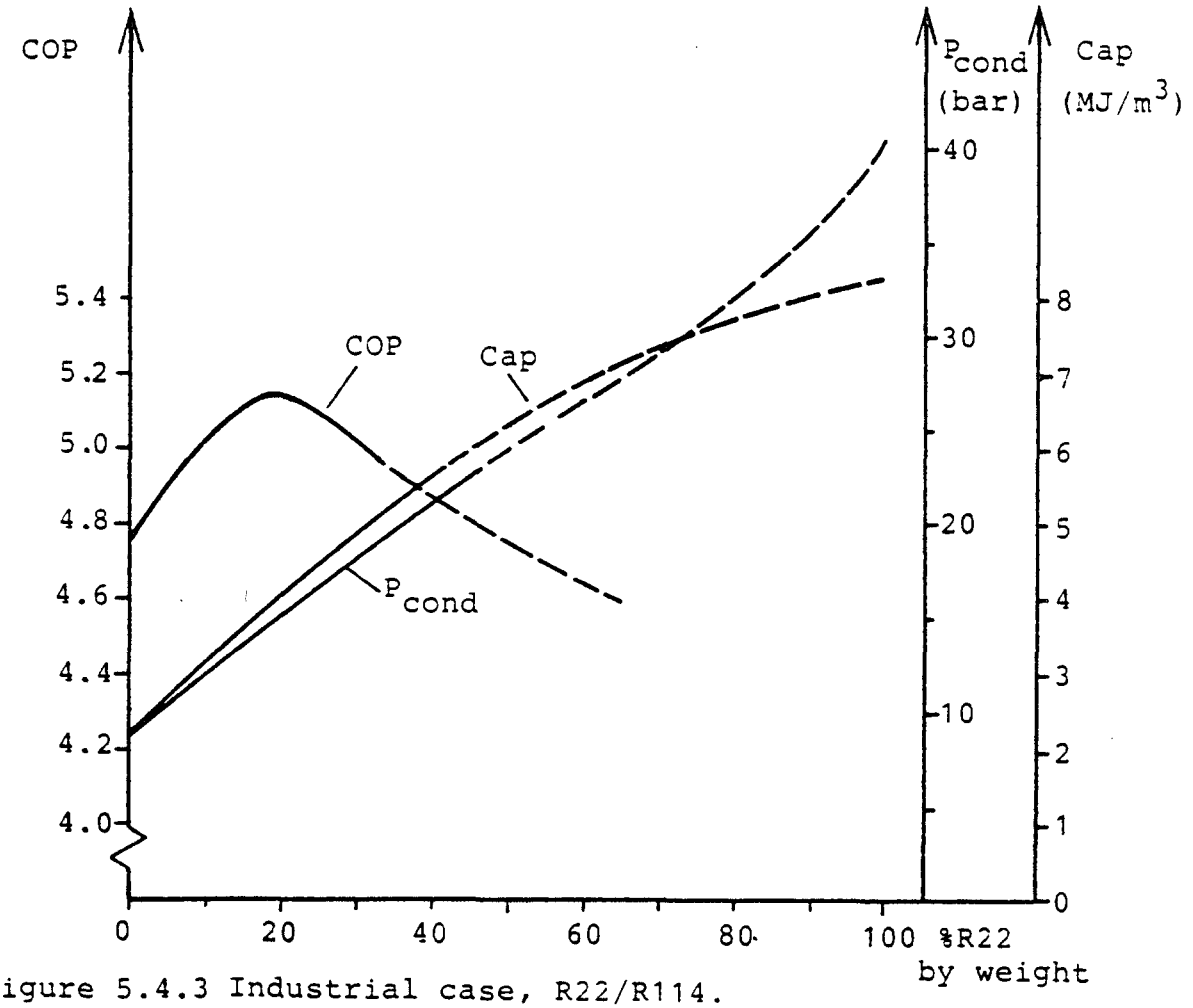

Figure 5.4.3 Industrial case, R22/R114.

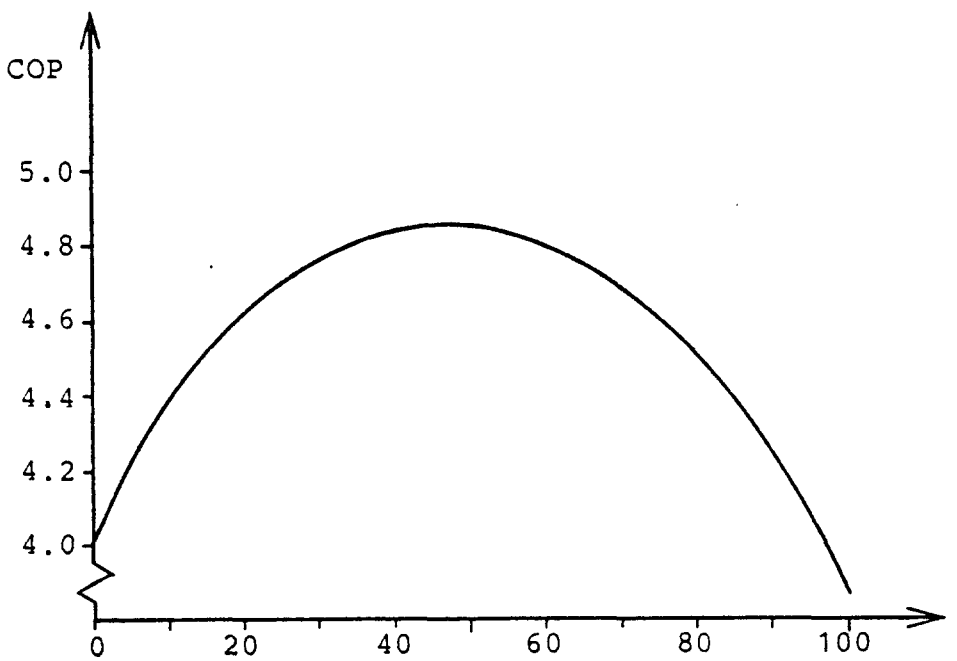

Figure 5.4.4 Industrial case R12/R114. %R12 by weight
No external subcooling.

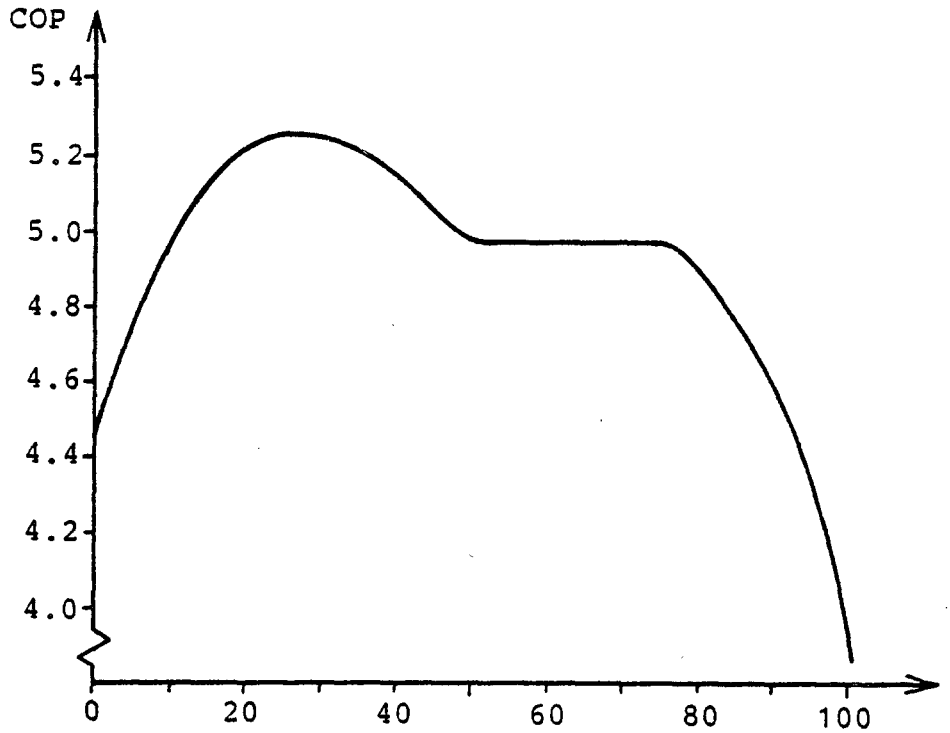

Figure 5.4.5 Industrial case, R12/R11. %R12 by weight
No external subcooling.

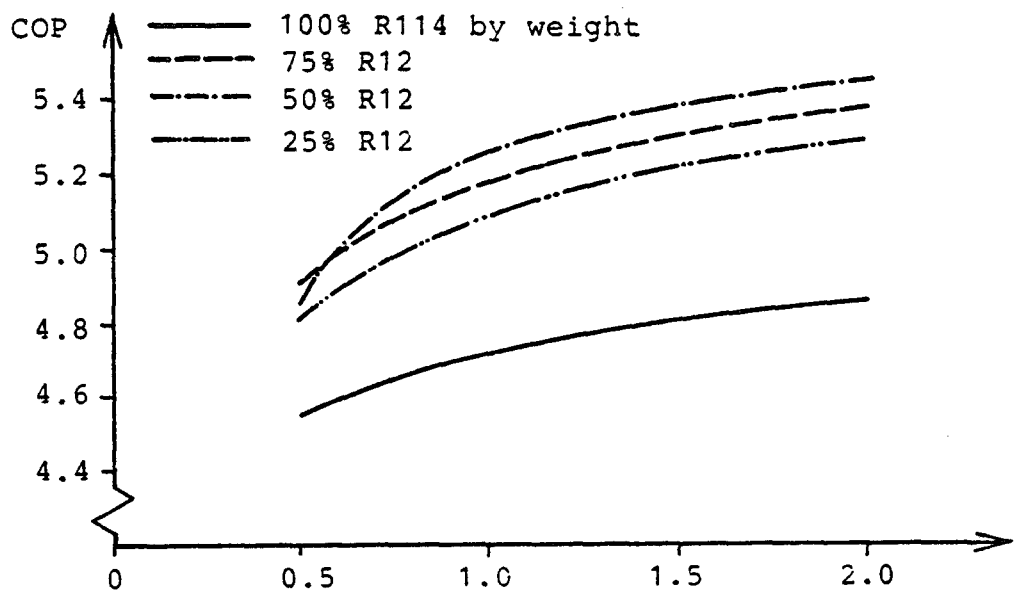

Figure 5.4.6 Industrial case R12/R114. Influence of evaporator area.

Relative evaporator area

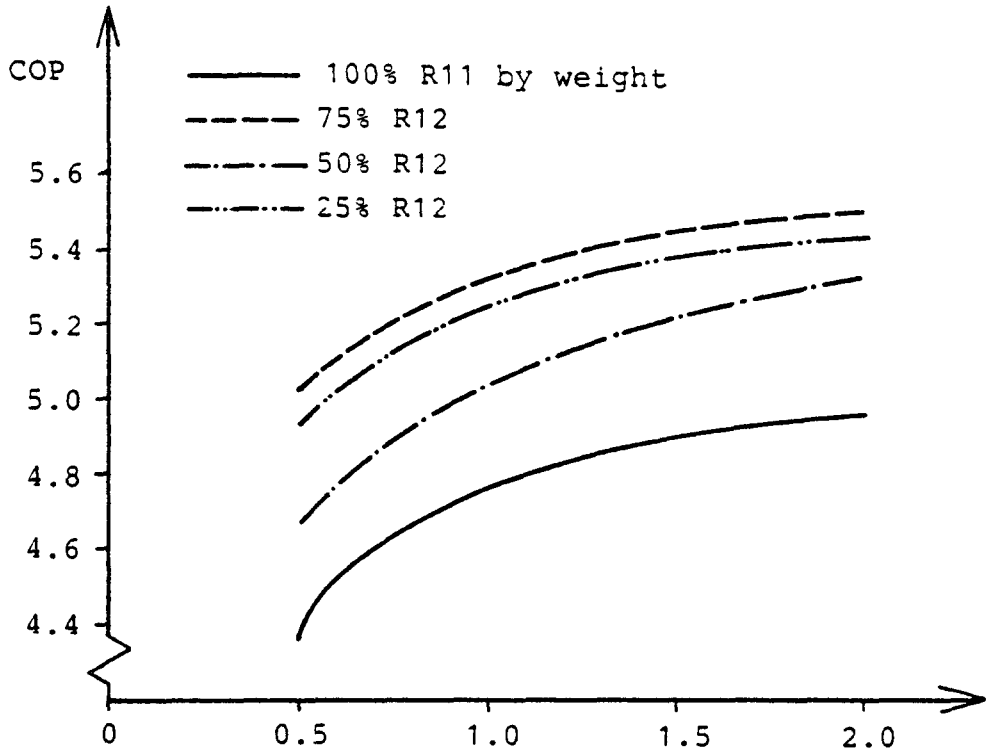

Figure 5.4.7 Industrial case R12/R11. Influence of evaporator area.

Relative evaporator area

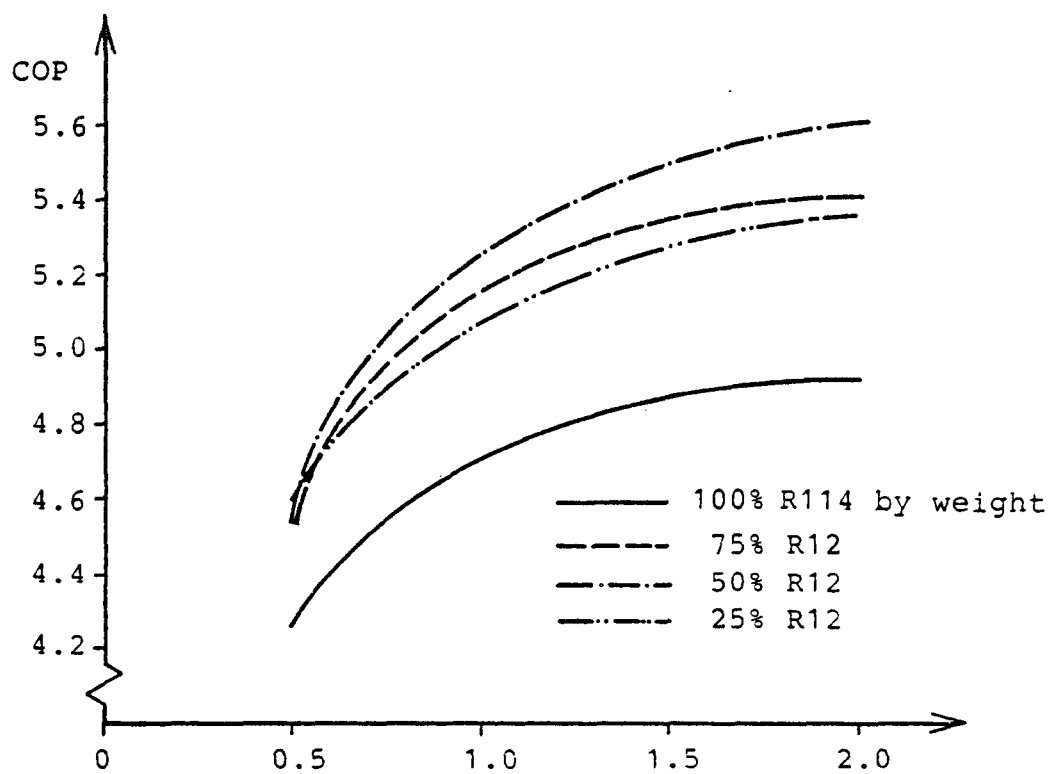

Figure 5.4.8 Industrial case R12/R114. Influence of condenser area.

Relative condenser area

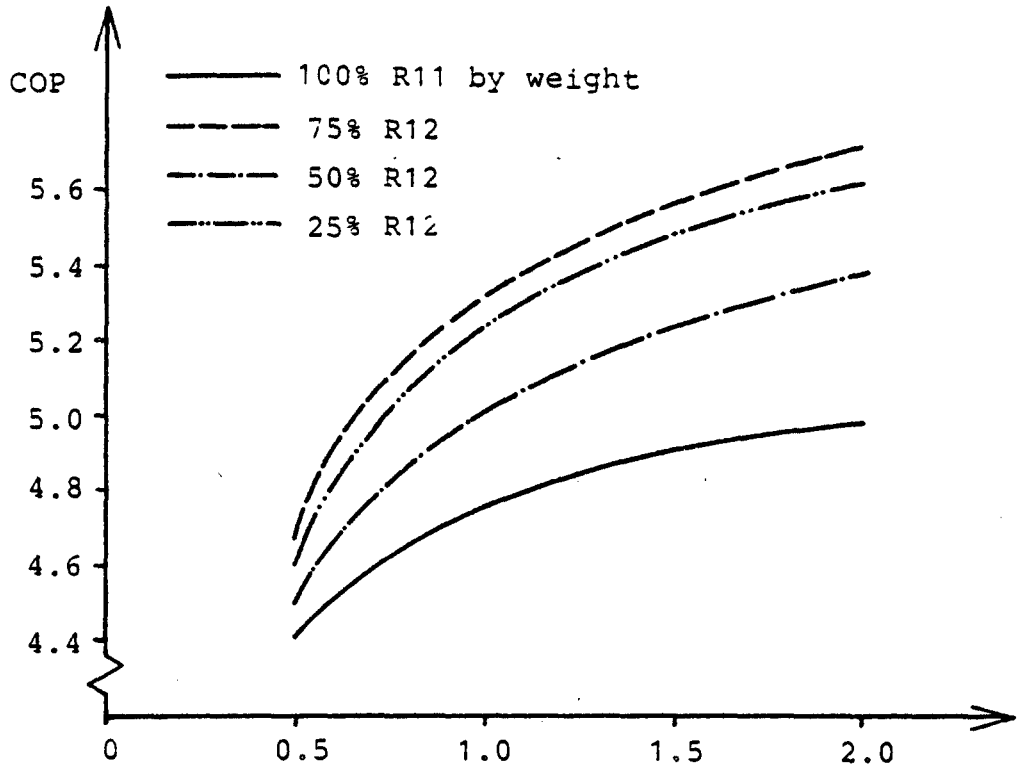

Figure 5.4.9 Industrial case R12/R11. Relative condenser area Influence of condenser area.

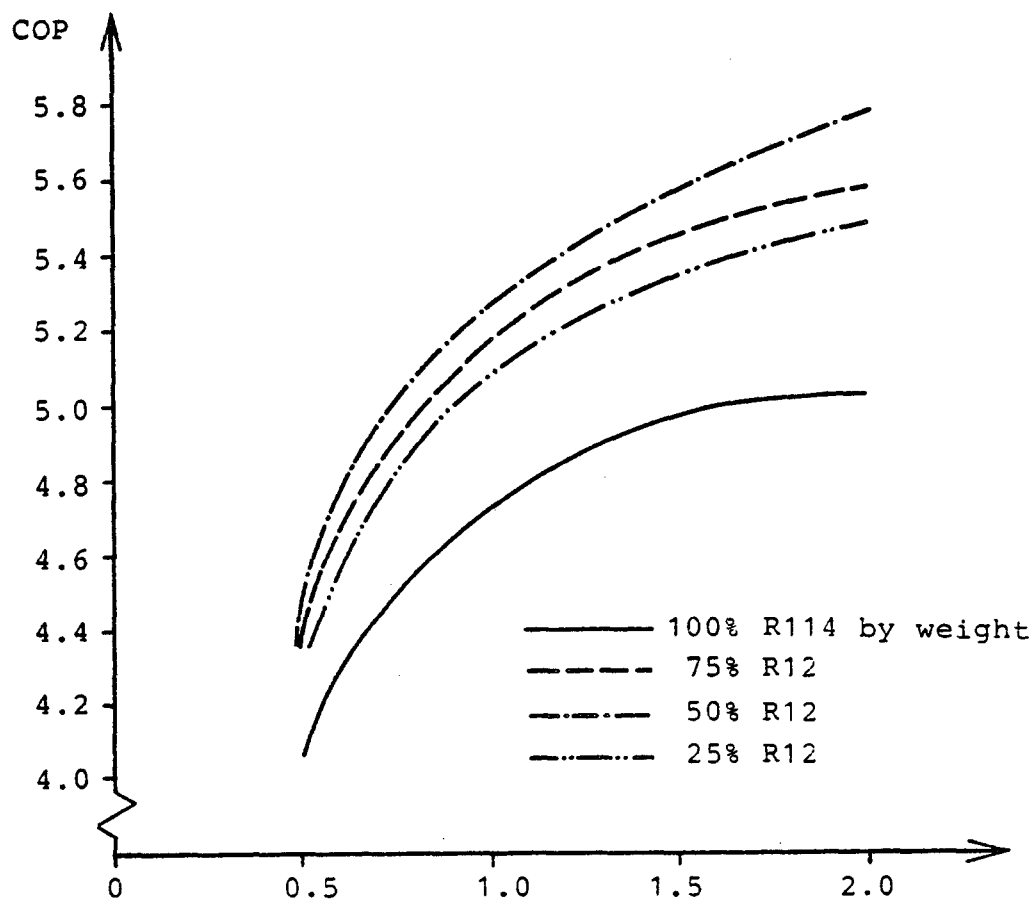

Figure 5.4.10 Industrial case R12/R114. Influence of evaporator and condenser area. Relative evaporator and condenser area

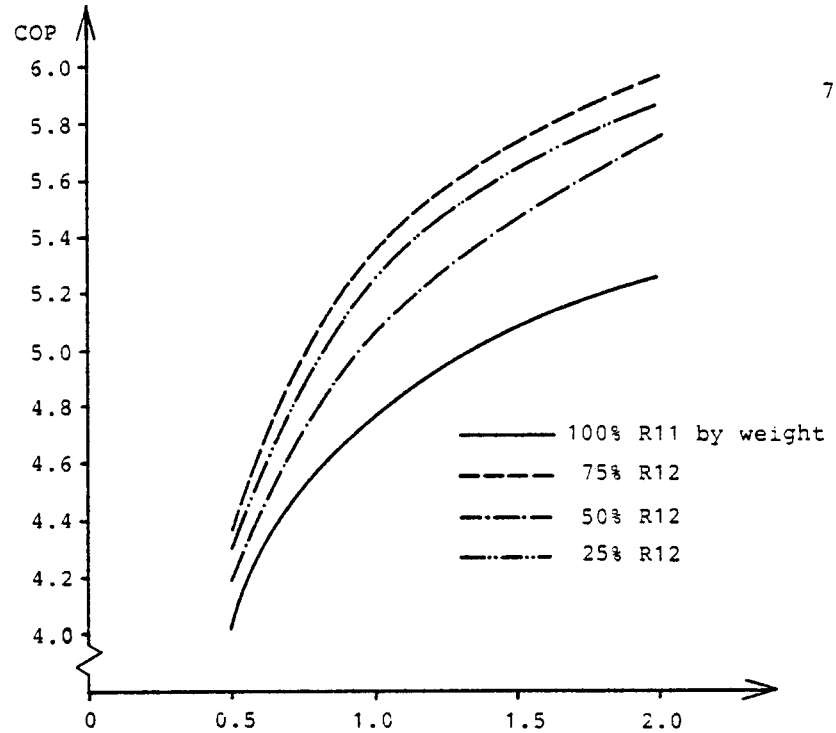

Figure 5.4.11 Industrial case, R12/R11.Influense of evaporator and condenser area.

Relative evaporator and condenser area

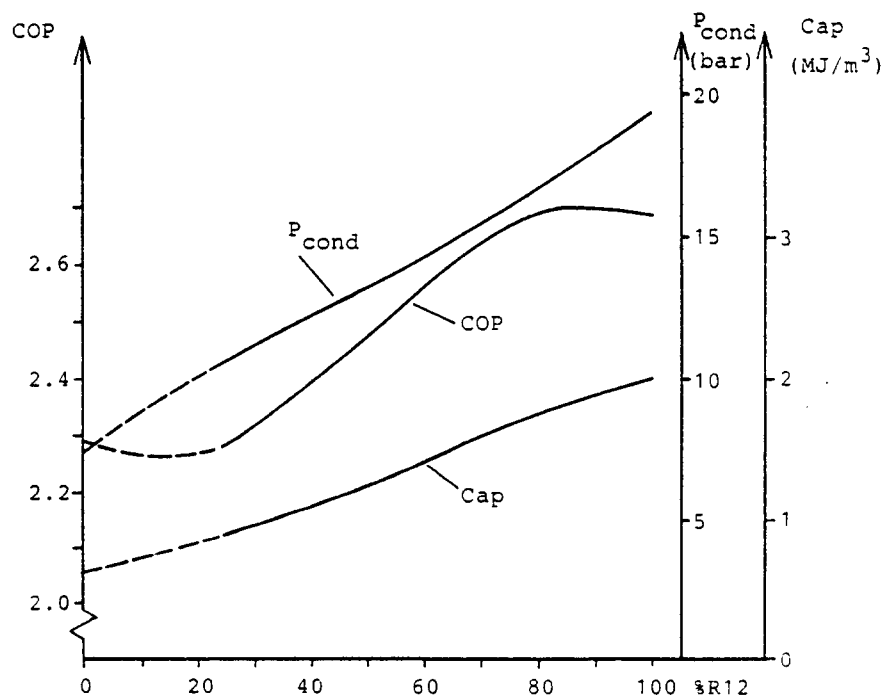

Figure 5.4.12 Space heating case R12/R114.

by weight

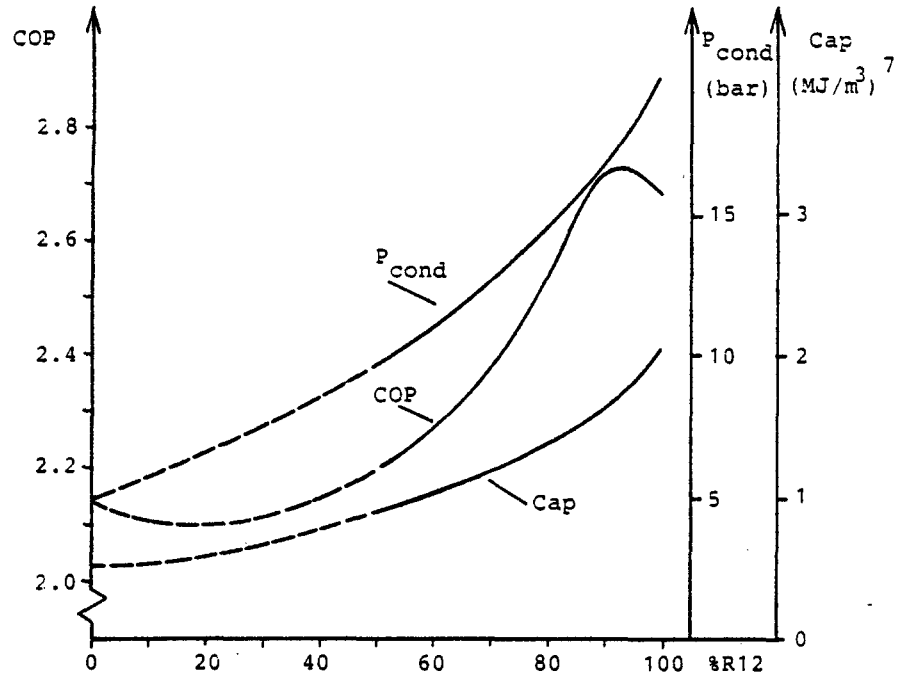

Figure 5.4.13 Space heating case R12/R11.

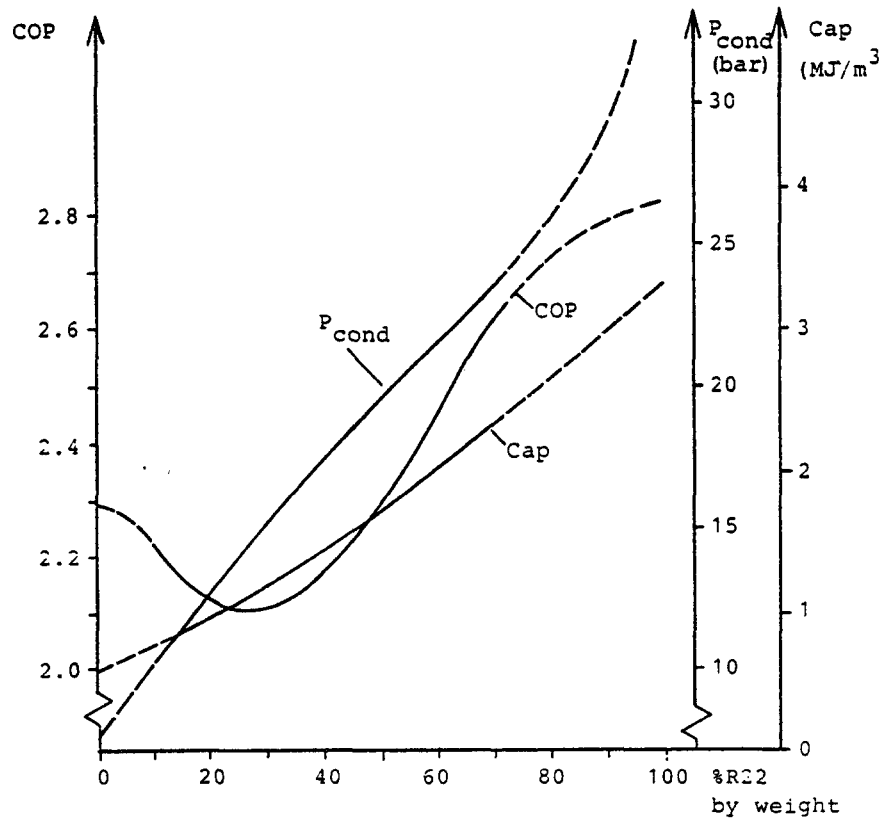

Figure 5.4.14 Space heating case R22/R114.

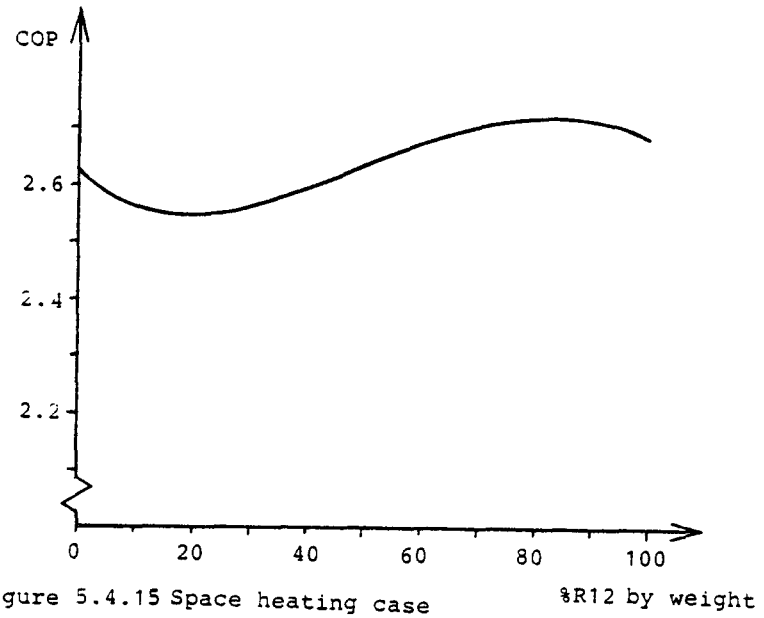

Figure 5.4.15 Space heating case R12/R114. Constant compressor efficiency.

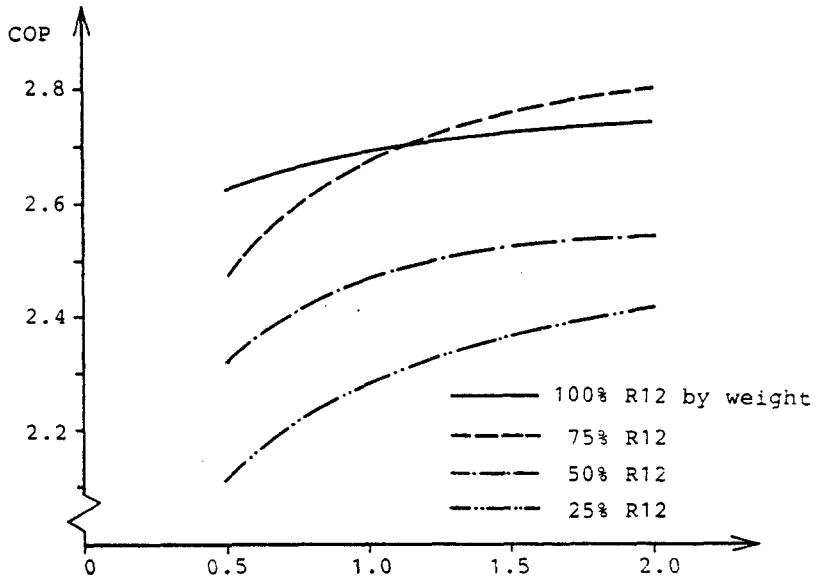

Figure 5.4.16 Space heating case R12/R114. Influence of evaporator size.

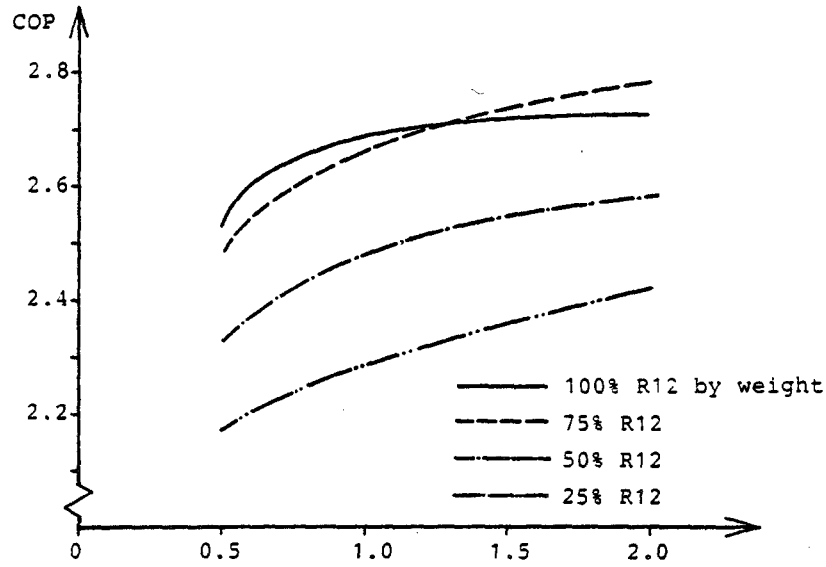

Figure 5.4.17 Space heating case, R12/R114. Influence of condenser size.

Relative condencer area

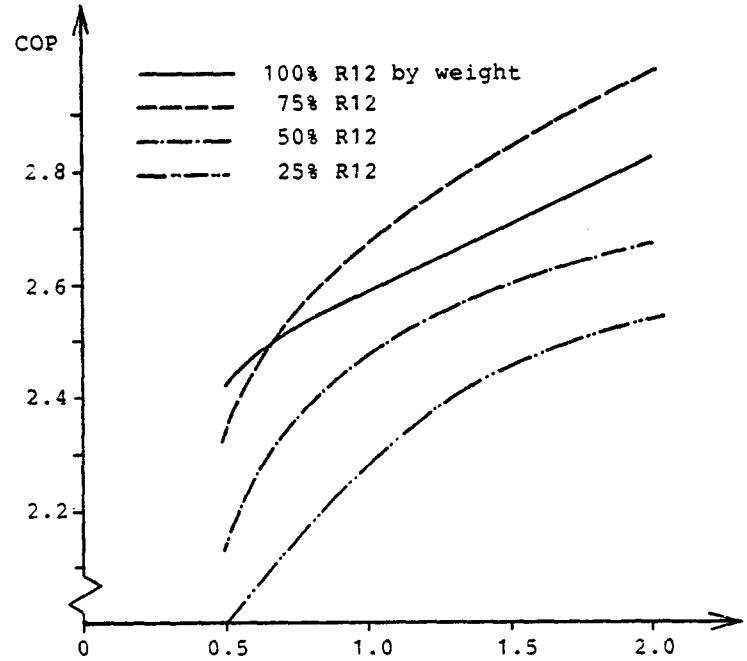

Figure 5.4.18 Space heating case, R12/R114. Influence at evaporator and condenser size.

Relative evaporator and condenser area

6. TECHNICAL-ECONOMICAL EVALUATIONS ON COMPRESSION HEAT PUMPS WORKING WITH NONAZEOTROPIC MIXTURES OF FLUIDS

6.1 Introduction

In general the decision to install a heat pump or not is based upon economical considerations. As a matter of fact a heat pump represents a capital investment which has to be compensated for by its savings. To be able to determine the most economical installation a number of characteristic values are in use that show the effectiveness of the investments. These values, for instance the payback period, the net present value or the rentability, all relate the savings in different definitions to the investment costs.

Expecially in the process industries the environmental conditions for heat pump applications often show a number of peculiarities. In these cases "tailor-made" heat pumps may represent more efficient installations than those offered in series by heat pump manufacturers. For these special applications the machines have to be laid out in such a way that the capital investments and the running costs can be foretold. This design procedure requires the assumption of many parameters that more or less influence the economy of the heat pump. For instance one must be aware of the best possible usage of the various apparatus types, tube diameters and other geometrical dimensions must be chosen as well as process parameters that show the best economy of operation. Hence the layout of the machine is an optimization problem where particular economical characteristic values are maximized or minimized and the degrees of freedom are in the fields of the process caluclation and the equipment design.

Such an optimization also can be regarded as a possibility to define several design parameters. If one for instance requires particular assumptions as e.g. that they result in a minimum payback period,

they are in many cases unambiguously defined. The prior condition for that, of course, is the knowledge of the mathematical relationship between the parameters and the payback period. The importance of the parameter definition that just has been portrayed, in particular at the economical comparison of heat pumps that operate with different heat carrier fluids, is shown in section 3. For example an evaporator of a heat pump running with the refrigerant R114 can be designed in a way that it causes a higher pressure loss than the one running with R11, although all environmental conditions for both systems are identical. Just the different pressure losses may result in entirely different geometrical dimensions of the two apparatus. The consequences may be so farreaching that for two different working fluids even different types of apparatus represent the solutions with minimum payback periods (for example a pool boiling evaporator for the one fluid and a once-through type for the other one).

In this way the working fluid is somehow the basis of the heat pump design. The requirements for its thermodynamical behaviour are that a high performance can be achieved and small and compact apparatus can be designed. Unterleutner /6.1/ has worked out some criteria from which appropriate fluids or mixtures can be chosen for particular conditions. The question, how far compression heat pumps running with nonazeotropic mixtures of fluids can compete with those running with pure fluids, can be answered only by comparison of economically optimized systems.

In order to be able to carry out a comprehensive optimization, special emphasis has to be laid on the mathematical model. This model must comprice the process simulation, the apparatus design as well as the cost estimation and calculation of economy. In chapter 2 a computer program package for the design of heat pumps is described. Beside a brief description of the mathematical relationships the parameters that influence the economy are specified. It turned out that many parameters can be evaluated if only single

parts of the whole system (evaporator, condenser, heat exchanger, etc.) are investigated. In this way the number of degrees of freedom for the overall optimization can be reduced. All the details concerning the mathematical models as well as the technical and economical optimization of different compression heat pump systems are described in the author's dissertation /6.16/.

6.2 Mathematical model

6.2.1 General Arrangement

To set up a clear and flexible computer model it seemed to be convenient to develop individual programs both for the simulation of thermodynamic processes and for equipment design and the economy in general (fig. 6.1). These programs are arranged like a modular system, which means, that single parts can be changed or added easily. In this way compression heat pumps as well as absorption heat pumps can be simulated and a number of different apparatus types can be designed. The calculation of economy for particular cases of application can be adapted to the system.

The three programs are executed in series and they read the input-data from only one file. On that file the different data are summarized into logical units that are identified by keywords. In this way the file is clearly arranged and the whole program system can be used as if there would only be one program. For the apparatus design and the economy additional data are needed from the previous programs. These are transfered automatically.

The maximization of specific values is done by a general optimization algorithm, which also is programed in an independant program. In this way not only characteristic values from the economy calculation can be maximized, but also values from the process simulation or the apparatus design. The maximization routine changes the specific values in the input data file that represent the degrees of freedom of the optimization. The special information, which data shall be modified and the ranges within which the modification shall happen, are given in a separate input file.

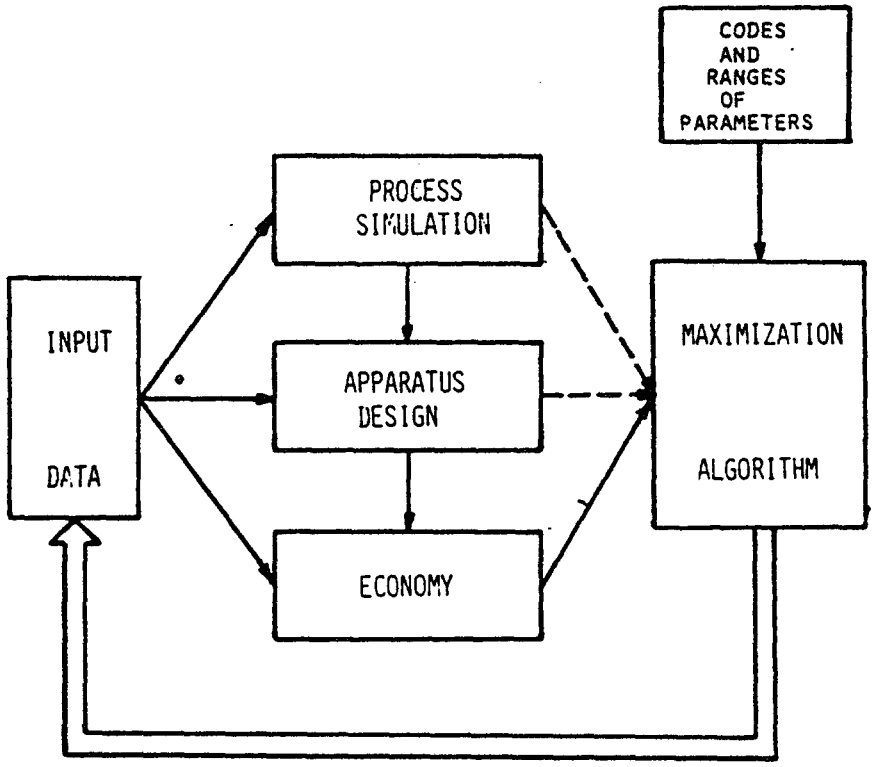

Fig. 6.1: General arrangement of programs

6.2.2 The Process-Simulation Program

This program is a modified form of a general purpose flowsheeting program. It solves the system of nonlinear equations of the steady state process-flowsheet with a sequential method. The organization is such, that the simulation of each apparatus is done by one separate subroutine (fig. 6.2). These subroutines can be called in different orders, which means, that flowsheets with different configurations of apparatus can be simulated. The thermodynamic data are obtained by calling a defined subroutine to which different data systems can be adapted. For compression - as well as absorption heat pump media the following systems exist:

- LEE KESLER PLÖCKER equation for hydrocarbons and refrigerants (pure substances and mixtures)
- SCHULZ/ZIEGLER equation for NH_3-H_2O

- Fitting procedure for H_2O-LiBr (HODGETT)
- EBWR (MORSY) (pure refrigerants)
- BWR-BWRS (pure fluids and mixtures)
- REDLICH KWONG modifications (SOAVE, PENG-ROBINSON)
- ACTIVITY COEFFICIENTS (WILSON, NRTL, UNIQUAC/UNIFAC)
- VDI - steam tables for water

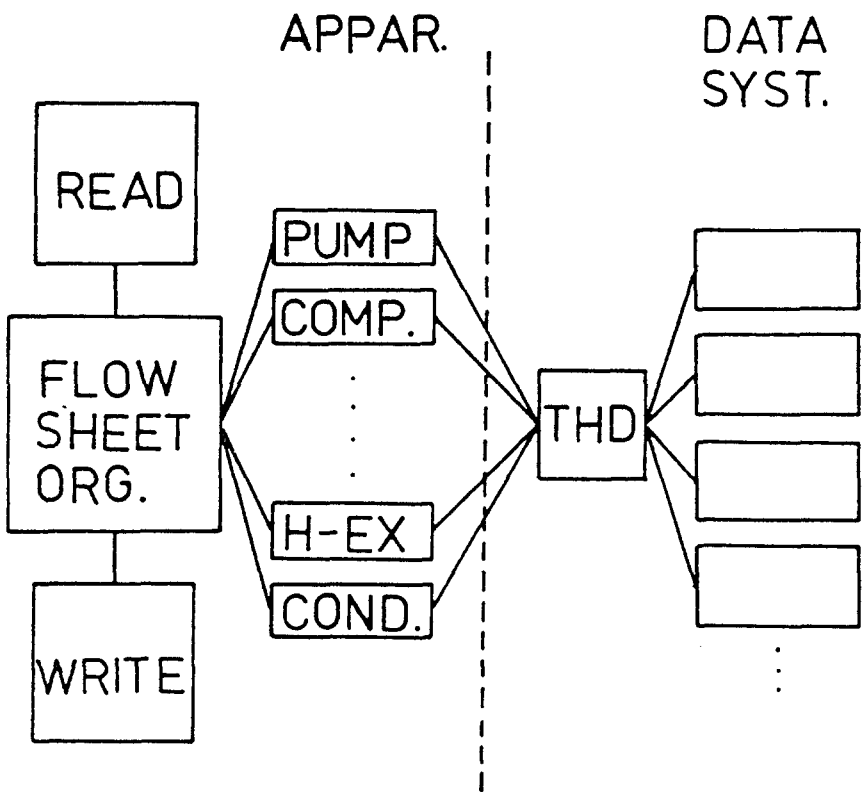

Fig. 6.2: Organisation of process simulation program

6.2.3 The Apparatus Design Program

This program reads the geometry-data of the apparatus and the external application conditions, like temperatures of heat sink and heat source etc., from the input data file. The process data are transfered from the process simulation program. Again there are subroutines for the design of the different apparatus. In this way alternatively models

for different apparatus types can be linked to the system (for example pool boiling evaporator, boiling within horizontal tubes or falling film evaporator).

For the thermodynamic data the same data systems that are used at the process simulation program can be used here. The transport properties are obtained from a little data bank.
In the following the parameters are pointed out that influence the heat transfer in the single apparatus. As far as the equations are concerned they are not listed in this article but only referred to by means of stating the sources they are taken from.

6.2.3.1 Evaporator

In the case study in chapter 3 a horizontal tube bundle evaporator is used (fig. 6.4) with the evaporating medium inside the tubes and the heat source on the shell side. The heat transfer coefficient and the pressure loss of an evaporating refrigerant inside a horizontal tube are calculated by equations of Chawla /6.2, 6.3/. In this method two different heat transfer mechanism are separately described by equations. At low velocities a sort of pool boiling of a flowing media dominates and at high velocities there is a forced convective boiling. Both heat transfer mechanism as well as the pressure loss strongly depend on the velocity. Higher velocities cause higher pressure losses but result in better heat transfer. On the other hand increasing pressure loss causes a decreasing coefficient of performance (COP) of the heat pump. For that reason this parameter can be optimized only including economical aspects that relate the COP to the size of the apparatus. Considering, that by the pressure drop the velocity is defined which result in the number of tubes, one can see, that the optimal main dimensions of the apparatus are determined with regard to the economy of the whole system at one particular place of application.

The influence of the heat transfer hindering effects at the evaporation of binary mixtures is calculated by the method of Stephan and Körner /6.4, 6.5/

The temperature difference between refrigerant and heat source behaves similarly to the pressure loss. Here again the COP and the heat transfer area show contrary tendencies.
As a general guidline for the choice of the tube dimension it was found out that at constant pressure drop smaller tubes lead to better heat transfer. The shell-side heat transfer is calculated according to the descriptions in 6.2.3.3.

6.2.3.2 Condenser

As a condenser type a horizontal tube bundle condenser is chosen (fig. 6.4). The heat transfer coefficient of condensing vapour on the outside of horizontal tubes are described by semiempirical equations similar to those of the falling film theory derived by Nusselt. This relationship and some correction factors taking into account the gas velocity and superheated inflow are summarized in /6.5/. The tube side heat transfer is calculated according to the descriptions in 6.2.3.3.

It turned out that long tube bundles with only a few tubes result in better heat transfer coefficients than short bundles with many tubes. The reason for that is the falling film that gets thicker if there are more tubes one above each other, but which represents a heat conduct resistance. The definition of the tube-number happens similarly to the evaporator by the pressure loss of the tube side flow, which is the heat sink here. The shell-side pressure loss can be regulated by the tube pitch.

6.2.3.3 Heat Exchanger

The internal heat exchanger and the subcooler are designed as shell and tube heat exchangers. Here the forced convective heat transfer in the tubes is calculated by an equation of Hausen /6.6, 6.5/ and the shell-side heat transfer by the method of Bell-Delaware /6.7, 6.5/. Primarily for the latter one a number of geometrical dimensions are required which in many cases can be calculated from correlations. At given heat flux and mean logarithmic temperature difference the following data have to be assumed in order to define the shell and tube heat exchanger in its main dimensions:

- The tube dimension
- The pressure losses of the shell- and tube side flow
- The tubes layout (30 ^{o} staggered, 45 ^{o} staggered or 90 ^{o} in-line)
- The segmental baffle cut (mostly given in percent of the shell diameter)

The pressure losses are attained by variing the velocities. On the tube side the velocity is changed by changing the flowrate in one tube. For that reason the tube side pressure loss also defines the number of tubes. The shell-side velocity is changed by the tube pitch.

At constant pressure losses and tube diameters the influences of the tubes layout and the segmental baffle cut on the heat transfer coefficient are very low. At the internal heat exchanger the vapour shall flow on the tube side. That is because of the facts that the heat transfer to the gas represents the main resistance in the heat exchanger and that at equal pressure losses the heat transfer on the tube side flow is better than that on the shell side flow.

6.2.3.4 Expansion Valve

For the dimensioning of the expansion valve instructions and layout data from Eckardt /6.8/ are used.

6.2.4 The Economy Program

This program calculates economical characteristic values (payback period, minimum annual heat costs) that show the competitivity of the heat pump to some other equipment. Here primarily the costs of the apparatus have to be determined. These are calculated from the following cost functions that have been developed on the Institute of chemical engineering/Technical University of Graz.

<u>compressor costs:</u>

$$pr_{comp} \left[10^{3}S\right] = 512{,}037 * \dot{V} + 78{,}222 \qquad *)$$

$\dot{V}$.....suction volume of compressor $\left[m^3/s\right]$

<u>electric motor:</u>

$$pr_{motor} \left[10^{3}S\right] = 1{,}08 * e^{(-0{,}035 + 1{,}027 \ln P_{el})}$$

P_{el}..... electric power $\left[kW\right]$

<u>condenser and evaporator (horizontal tube bundle):</u>

$$pr_{cond,\ evap} \left[10^{3}S\right] = 1{,}62 * e^{(a + b \ln A)}$$

$$a = e^{(0{,}9044 + 0{,}1159 \ln p)}$$
$$b = e^{(-0{,}4775 - 0{,}0485 \ln p)}$$

(if $p < 5 \rightarrow p = 5$)

A .....heat exchange area $\left[m^2\right]$

p .....pressure $\left[bar\right]$

<u>shell and tube heat exchanger:</u>

$$pr_{sthe} \left[S\right] = 0{,}7128 * e^{(a + b \ln A)}$$

$$a = e^{(2{,}1687 + 0{,}0308 \ln p)}$$
$$b = e^{(-0{,}35303 - 0{,}014762 \ln p)}$$

(if $p < 5 \rightarrow p = 5$)

A ..... heat exchange area [m

p ..... pressure $\left[bar\right]$

---

*) see footnote page 12

expansion valve:

$$pr_{valve} \left[10^3 S\right] = 1{,}08 * e^{(-0{,}5158 + 0{,}8021 \ln DM)}$$

DM .....layout diameter [mm]

To get the total investment costs of the heat pump some additional costs have to be taken into account. These, as well as the calculation of the payback period are descripted in chapter 6.3.1.

6.2.5 The Maximization Algorithm

For the maximization a simple alogorithm was chosen that is known as Fibonacci search /6.9, 6.10/. This is a one dimensional algorithm, which means that in case of multi dimensional optimizations a number of cycles have to be repeated where one parameter after the other is maximized with the remaining parameters kept constant. This may lead to long run-times. There would be better algorithms for this problem, but until now it did not seem worth the effort to program one of them.

6.3. The Economical Comparison of pure- and two Component Compression Heat Pumps for one Particular Application

6.3.1 Requirements on the System, Boundary Conditions and General Definitions

An industrial heat pump shall be applied to heat water from 70°C to 90°C with the required heat output of 5000 kW. As heat source liquid water with 60°C is used that shall be cooled down at 15 degrees. These temperature levels correspond to one of the IEA case studies worked out in the step 1 work of this Annex.

To get as fair comparison conditions as possible it seems convenient to define the pressure losses of the heat source and heat sink and to keep them constant for all calculations.

With the pressure losses the velocities of heat source and heat sink are defined, that influence the heat transfer coefficient in evaporator and condenser. The environmental conditions that are kept constant for all calculations are shown in fig. 6.3.

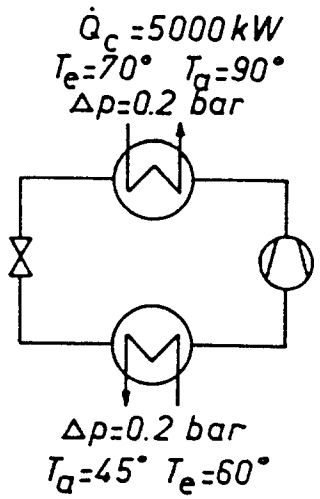

Fig. 6.3: Environmental conditions for case study

To be able to calculate the payback period of the heat pump it is considered, that the recovered heat could have been produced by a conventional heating device. In this case the savings mainly are related to the costs of heating fuels conserved and result from the difference between the running costs of the conventional heater and those of the heat pump.

If for the running costs only energy costs are taken into account one gets for an oil fired heater:

$$rc_{oil}\left[\frac{S}{a}\right] = \frac{Q_H\left[kW\right] \cdot 3600 \cdot b\left[\frac{h}{a}\right]}{H_u\left[\frac{kJ}{kg}\right] \cdot \eta_{heater}} \cdot pr_{oil}\left[\frac{S}{kg}\right] \qquad (1)$$

rc_{oil}.....annual feel costs (Austrian Shillings per year)
Q_H.....heat output (= 5000 kW)
b.....running hours per year
H_u.....caloric value (oil = 41 900 $\frac{kJ}{kg}$)
η_{heater}.....thermal efficiency of heating device
pr_{oil}.....prize for oil

At 4000 running hours per year, a thermal efficiency of 0,85 and the prize of 3,8 S*) for one kg oil equation (1) results in $rc_{oil} = 7682.10^3$ Austrian Shillings annual fuel costs for the oil heater.

The running energy costs for the heat pump depend on the coefficient of performance (COP). For electrically driven heat pumps one gets:

$$rc_{hp}\left[\frac{S}{a}\right] = \frac{Q_H\left[kW\right]}{COP\ \eta_m} \cdot b\left[\frac{h}{a}\right] \cdot pr_{el}\left[\frac{S}{kWh}\right]$$

η_m.....efficiency of power transmission and electric motor

The prize for the electricity (pr_{el}) is assumed 1 S per kWh in all calculations exept those where another prize is seperately indicated. The efficiency η_m is generally assumed 0.9.
For the investment costs (ic_{hp}) the prizes for the apparatus (plus electric motor) are sumed up and some additional costs must be added to them. These are given as a percentage of the apparatus costs and assumed as followes /6.11/:

*) 20 Austrian Shillings = 1 US $

6 %	for assembly
19 %	for casing
10 %	for isolation
18 %	for control equipment
17 %	for auxillary equipment
15 %	profit
3 %	for operating installation costs
88 %	sum of additionals

The statical payback period of the heat pump is:

$$t\left[a\right] = \frac{ic_{hp}}{rc_{oil}\left[\frac{S}{a}\right] - rc_{hp}\left[\frac{S}{a}\right]}$$

The apparatus types are chosen due to the considerations in chapter 6.2 and outlined in fig. 6.4. Here the condenser and the subcooler are designed as separate apparatus. The heat sink flows in counter currant on the shell side of the shell and tube subcooler and in series, again in counter currant, tube side through the condenser, which is specified as horizontal tube bundle condenser. The 0.2 bar pressure loss of the heat sink (fig. 6.3) have to be partitioned on the two apparatus and it is generally assumed, that the shell side flow in the subcooler effects 0.15 bar pressure loss and the tube side flow in the condenser 0.05 bar.

As an internal heat exchanger a shell and tube heat exchanger is specified. The apparatus operates in counterflow and according to the descriptions in 6.2.3.3 the subcooled liquid flows shell side of the tube bundle. The evaporator type is a dry (once trough) horizontal tube bundle evaporator with the evaporating medium inside the tubes and the heat source in cross-counterstream on the shell side. This evaporator type enables the adaptation of the temperature profile during evaporation of a nonazeotropic mixture to the cooling of the heat source, with which the effect of the COP-increase with nonazeotropic mixtures can be realized.

The expected power of the compressor is approximately 1000 kW. In this case a screw compressor can be used. One quite important design data for the whole heat pump, that is very difficult to estimate, is the isentropic efficiency of the compressor. This value primarily depends on the pressure ratio and the solubility of the compressed medium in the oil that is injected to reduce the leakage losses. The quantitative influence of several fluid properties on the isentropic efficiency is unknown. For lack of better data the isentropic efficiency is estimated from the pressure ratio (which in all cases lies between 2.8 and 3.6) and some data valied for R22 /6.12/. As an average η_{is}= 0.75 is taken and kept constant for all heat carrier media.

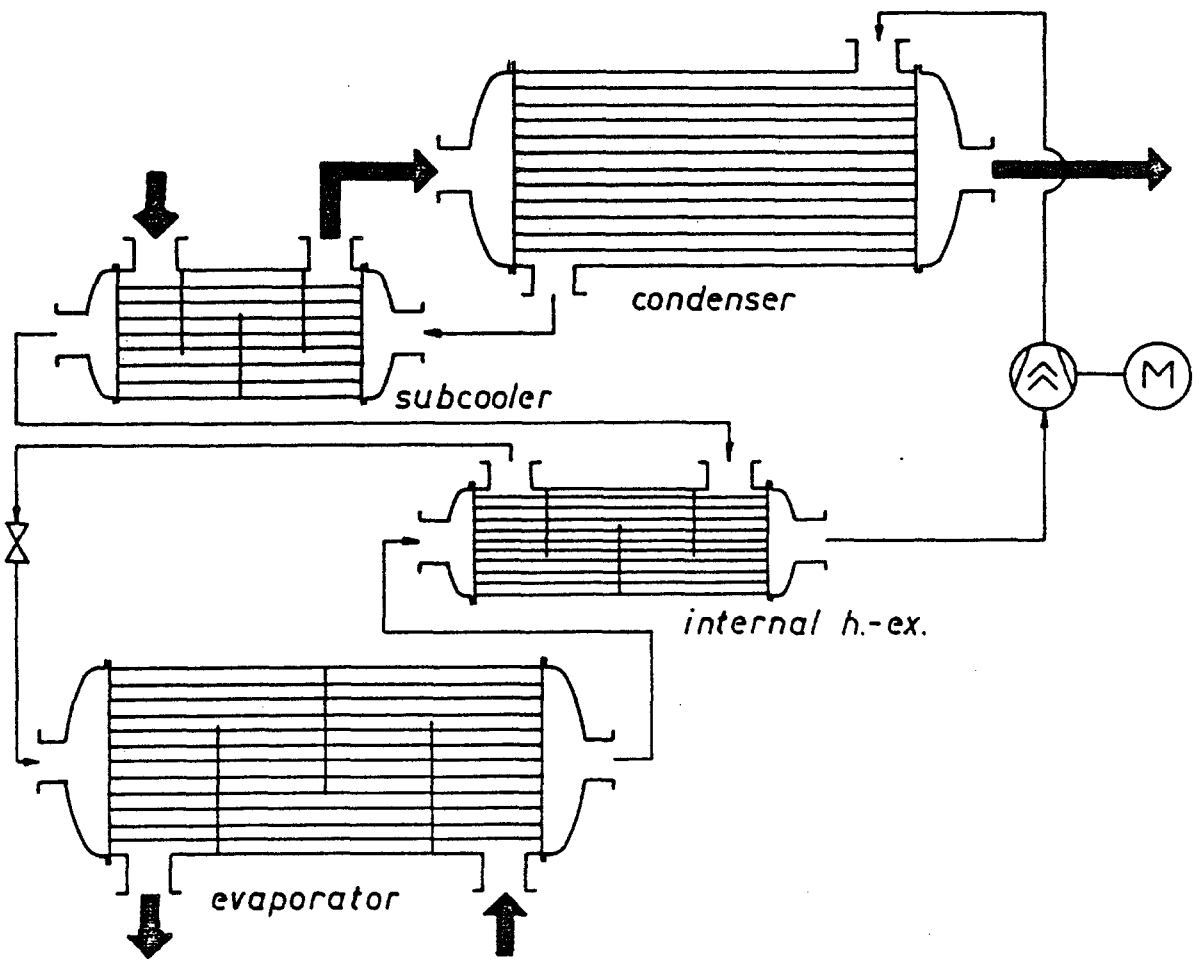

Fig. 6.4 : Configuration of apparatus

In Chapter 6.2.3.3 the data are listed that define a tube bundle heat exchanger in its main geometrical dimensions, actually the heat exchange area, the number of tubes as well as the length and the diameter of the tube bundle. In table 6.1 all these necessary assumptions for the four tube bundle apparatus are listet (notice the chapters 6.2.3.1, 6.2.3.2, 6.2.3.3).

apparatus	counter- or currantstream	pressure losses (bar) shell side	tube side	tube dimension	tubes layout	seg.baffle cut as % of shell diam.
condenser	counterstream	calculated	0.05	10x1.5	30° staggered	70%
subcooler	counterstream	0.15	0.05	18x2	30° staggered	50%
evaporator	counterstream	0.2	optimiz. parameter	10x1.5	30° staggered	40%
internal h-ex	counterstream	0.05	0.05	30x2.5	30° staggered	50%

Table 6.1: Assumptions for the tube bundle heat exchangers

Actually all pressure losses on the refrigerant side have to be determined by economical considerations. Higher pressure losses in all cases mean worse COP but better heat transfer. At the variation of each of the pressure losses it turned out that only the one on the tube side of the evaporator noticeably influences the payback period. The pressure loss on the shell side condensation is small and hardly influences neither the heat transfer coefficient nor the COP. Not 100-percent optimally designed subcoolers and internal heat exchangers are of no consequence, as these apparatus are rather cheep compared with the rest of the equipment. By definition of the pressure losses the degrees of freedom in the economical optimization can be reduced.

The parameters, that by no means can be estimated are the temperature differences in the condenser and the evaporator (fig. 6.5). Especially in the case of nonazeotropic mixtures it is absolutely necessary to define these parameters with regard to economical aspects, for instance in such a way that they result in a minimum payback period.

Additionally the superheating in the internal heat exchanger may noticeably influence the payback period. Here for instance the gas-outlet temperature does not influence only the heat transfer area of the internal heat exchanger, but also the suction volume of the compressor. For the sake of simplicity the gas-superheating is defined as 10 degrees in all calculations. It is considered, that this parameter can be optimized separately at the best cases after having optimized the other parameters.

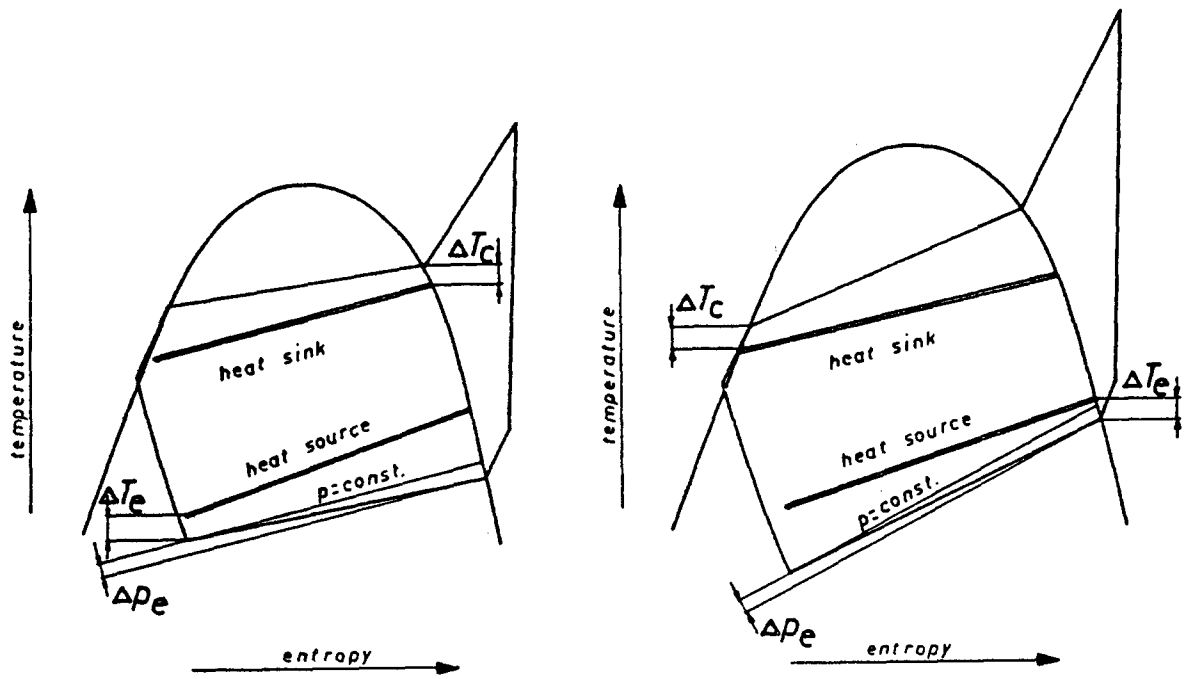

Fig. 6.5: Degrees of freedom in the economical optimization

6.3.2 Results

Below a number of heat pump layouts for the conditions described in 6.3.1 and different heat carrier fluids are discussed. In all the calculations the pressure loss of the evaporator and the temperature differences in condenser and evaporator were kept variable and such determined that they result in a minimum payback period of the whole installation.

For reasons of handling the program not directly the logarithmic mean temperature differences are optimized, but the minimum temperature differences on the inlet or the outlet (fig. 6.5). The table 6.2 shows the results for six refrigerants. The media were chosen due to the conditions that the evaporation pressure lies above 1 bar and the condensation pressure does not exceed 30 bar.

medium	variables									heat exchange area [m²] costs [10³ $]						t
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	T_{d-com}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	[a]
R11	0.11	1.85	2.23	1.92	6.89	3.59	117.8 (107/13/)	1760	5.27	1530	744 1491	745 1414	83 157	39 93	11352	3.27
R21	0.15	2.12	2.57	3.19	11.19	3.51	132.4 (121/14/)	2907	5.20	959	594 1443	480 1098	59 139	25 79	9562	2.80
R114	0.17	2.00	2.61	3.63	12.06	3.32	95.7 (121/13/)	2824	5.08	985	986 1918	505 1131	173 295	54 134	11034	3.34
R12	0.05	1.85	3.27	10.3	29.38	2.85	115.8 (121/13/)	7042	4.95	442	1144 2555	302 975	178 374	43 143	11141	3.49
n-pentane	0.05	1.85	1.19	1.29	4.84	3.75	95.1	1210	5.40	2192	742 1410	646 1303	97 162	39 86	12171	3.41
n-butane	0.15	1.50	2.62	4.16	13.12	3.15	100.1	3267	5.26	863	640 1558	398 953	84 185	29 90	9412	2.74

Table 6.2 : Economical comparison of heat pumps with different working media

The table shows the variables representing the degrees of freedom of the economical optimization (fig. 6.5) and the following process parameters: The inlet pressures of evaporator and condenser (p_e, p_c), the pressure ratio (π), the compressor discharge temperature, the volumetric efficiency ($\eta_v = Q_H/\dot{V}_{in-comp}$) and the coefficient of performance ($COP = Q_H/W_{comp}$). Furthermore the costs of the apparatus are listed and the heat exchange area of the heat exchangers, the total investment costs (see 6.3.1) and the payback period.

The refrigerants R21 and n-butane are by far the best from the economical point of view. It seems, that with these two media there is a good compromise between volumetric efficiency and COP. Regarding the halocarbons there has to be paid attention to the compressor discharge temperatures. At R11 and R21 these temperatures exceed limits that are proposed in literature (the values put in brackets are the maximum temperatures for continuous exposure in the presence of oil, steel and copper).

Applying a screw compressor the discharge temperature can be lowered by cooling the gas during compression with the oil that is injected to reduce the leakage losses /6.15/. For that reason it does not mean that R11 and R21 cannot be used in practice because of their high compressor discharge temperatures. Lowering this temperature by 10 to 15 deg hardly effects any noticeable changes in the apparatus costs and the payback periods. Pentane and butane are less advantageous, as they are highly inflammable and explosive. R114 is the substance that today is used without exception in the present temperature range.

From the economical point of view R114 is rather poor, compared with R21 the investment costs are 15 percent higher and due to the worse COP the payback period is even 20 percent longer.

At the consideration of mixtures one can start out from the pure fluids and try to improve specific fluid properties. Improvements can be achieved by

1) increasing the volumetric efficiency
2) getting a better temperature adaptation to the environmental conditions by nonazeotropic evaporation and condensation which may rise the COP
3) lowering or rising of pressure levels in condenser and evaporator and lowering of the pressure ratio
4) lowering the compressor discharge temperature.

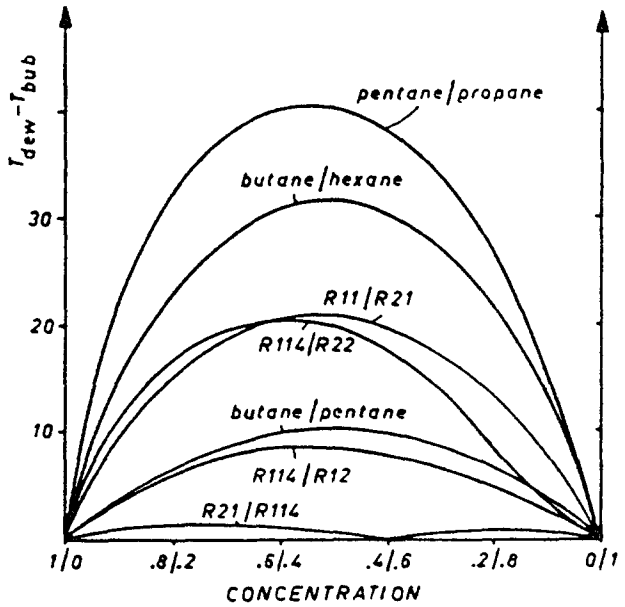

Fig. 6.6: Differences between dewpoint- and bublepoint temperatures for nonazeotropic mixtures at 40°C

The choice of the mixtures shall be restricted in that way, that only halocarbons and hydrocarbons are mixed among themselves.

Starting out from the best pure fluid R21 one can see, that the compressor discharge temperature can be lowered by adding R114. The table 3 shows the results for mixtures of R21-R114. The composition is always given in molar concentration. Here again in all cases the minimum payback periods are calculated by optimization of the variables shown in the table.

R21/R114	variables Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap T_i-T_o	T_{d-co}	η_v	COP	heat exchange area [m^2] / costs [10^3 s] comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R21	0.15	2.12	2.57	3.19	11.19	3.51	+ 1.60	132.4	2907	5.20	959	594 1443	480 1098	59 139	25 78	9562	2.80
0.8/0.2	0.22	2.12	2.81	3.64	12.22	3.36	+ 1.13	119.9	3105	5.18	902	679 1582	440 1044	74 167	31 92	9703	2.86
0.6/0.4	0.22	2.65	2.42	3.81	12.82	3.36	+ 1.69	110.6	3144	5.12	894	799 1748	385 968	108 217	36 103	10002	3.00
0.4/0.6	0.18	2.04	2.42	3.95	13.02	3.30	+ 1.51	104.1	3205	5.18	877	889 1859	536 1170	136 255	43 117	10635	3.14
0.2/0.8	0.22	1.57	2.81	3.93	12.80	3.26	+ 1.87	99.7	3048	5.11	916	909 1875	544 1180	147 268	49 127	10834	3.25
R114	0.17	2.00	2.61	3.63	12.06	3.32	+ 1.74	95.7	2824	5.08	984	986 1918	505 1131	173 295	54 134	11034	3.34

Table 6.3 : Economical comparison of mixtures R21-R114

The system has an azeotrop at about 40 % R21, but shows almost azeotropic character at all concentrations. For that reason the variables do not vary very much and the differences between evaporator inlet- and outlet temperatures (evap T_i-T_o) are almost constant (fig. 6.6). As one con see from table 6.3 no economical improvements can be achieved by the refrigerant mixture. The investment costs as well as the payback periods are continuously increasing. The reason for that are increasing costs for the heat exchangers and decreasing COPs. The high compressor discharge temperature of R21 can be lowered under the proposed limitby adding only small amounts of R114 (less than 20 %).

This is an advantage, but no decisive criterion why the mixture should be used as was pointed out before.

R114/R12	variables										heat exchange area [m²] / costs [10³ $]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap $T_i - T_o$	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R114	0.17	2.00	2.61	3.63	12.06	3.32	+ 1.74	95.7	2824	5.08	934	986 1918	505 1131	173 295	54 134	11034	3.34
0.2/0.8	0.23	3.04	2.23	4.65	13.85	2.98	- 2.65	98.9	3521	5.35	807	1195 2213	707 1371	155 283	64 156	11589	3.28
0.4/0.6	0.07	3.73	2.23	5.78	16.33	2.82	- 6.25	102.4	4335	5.60	664	1293 2389	785 1491	142 277	78 185	11808	3.18
0.6/0.4	0.07	3.73	3.27	7.07	19.95	2.82	- 6.09	107.0	5102	5.39	578	1196 2391	534 1242	115 253	66 175	11205	3.14
0.8/0.2	0.06	2.92	2.58	8.60	23.84	2.77	- 4.08	110.6	6097	5.27	499	1271 2570	403 1102	150 316	60 171	11295	3.26
R12	0.07	1.85	3.27	10.3	29.38	2.85	+ 0.20	115.8	7042	4.95	442	1144 2555	302 975	178 374	43 143	11141	3.49

Table 6.4 : Economical comparison of mixtures R114-R12

To raise the volumetric efficiency R12 can be added to R114. In table 6.4 the payback periods show a minimum at 40 % R12. Due to the nonazeotropic evaporation and condensation the COP can be improved (notice 6 deg temperature difference between evaporator inlet and outlet temperature including pressure drop). In this case the minimum payback period appears at a higher minimum temperature difference in the evaporator (ΔT_e) which compensates the lower mean logarithmic temperature difference and the heat transfer hindering effects of the two component boiling. At least because of the latter effects larger heat transfer areas are required and the investment costs cannot be lowered.

R11/R12	variables Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evan $T_i - T_o$	T_{d-co}	η_v	COP	heat exchange area [m^2] costs [10^3 s] comp	cond	evap	sub	h-e	ic_{hp}	t_a
R11	0.11	1.85	2.23	1.92	6.89	3.59	+ 1.85	117.8	1760	5.27	1530	744 1491	745 1414	83 157	39 93	11352	3.27
0.8/0.2	0.22	6.42	3.58	2.82	8.56	3.03	- 8.29	117.8	2347	5.51	1167	952 1780	666 1326	27 77	50 116	10827	2.96
0.6/0.4	0.33	6.96	4.73	4.13	10.99	2.66	- 13.96	117.8	3184	5.78	880	1180 2101	1038 1710	-	69 154	11424	2.97
0.4/0.6	0.37	7.31	5.15	5.58	14.62	2.62	- 14.95	118.7	4098	5.59	705	1352 2392	799 1498	10 44	71 169	11439	3.09
0.2/0.8	0.13	4.81	3.58	7.52	19.96	2.65	- 11.78	116.8	5555	5.63	538	1416 2618	577 1320	60 164	79 197	11463	3.07
R12	0.05	1.85	3.27	10.3	29.38	2.85	+ 0.20	115.8	7042	4.95	442	1144 2555	302 975	178 374	43 143	11141	3.49

Table 6.5 : Economical comparison of mixtures R11-R12

With the system R11-R12 in the equimolar region an optimal temperature adaptation to the cooling of the heat source can be achieved. The table shows, that there is a minimum payback period when adding 20 % to 40 % of R12 to R11. The optimal temperature adaptation to the heat source is at 60 % R12. Here the COP is even smaller than at concentrations slightly different because of the higher temperature differences (ΔT_e and ΔT_c), as well as a higher pressure loss in the evaporator (Δp_e). One remarkable thing is that the optimization algorithm has calculated the temperature differences at all concentrations so that they result almost in the same mean logarithmic temperature differences. The investment costs do not vary very much, the increasing volumetric efficiency almost makes up for the prize-increase due to the larger heat transfer areas at nonazeotropic mixtures.

The mixtures of butane-pentange show only small improvements which for practical use are of little importance.

Butane/ Pentane	variables										heat exchange area [m^2] costs [10^3 s]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap $T_i - T_o$	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
pentane	0.05	1.85	1.19	1.29	4.84	3.75	+ 1.24	95.1	1210	5.40	2192	742 1410	646 1303	97 162	39 86	12171	3.41
0.8/0.2	0.13	3.00	3.04	1.75	5.78	3.30	- 2.98	97.4	1524	5.56	1759	736 1438	643 1300	58 117	43 95	11262	3.05
0.6/0.4	0.15	3.00	2.62	2.23	6.77	3.03	- 6.31	98.3	1908	5.82	1421	885 1640	673 1333	55 117	50 109	10990	2.84
0.4/0.6	0.16	3.00	2.62	2.79	8.21	2.94	- 6.37	99.3	2293	5.78	1192	913 1723	512 1185	57 126	47 113	10468	2.73
0.8/0.2	0.16	2.50	3.04	3.43	10.34	3.01	- 4.07	100.4	2747	5.52	1010	760 1628	441 1046	62 141	40 104	9809	2.68
butane	0.15	1.50	2.62	4.16	13.12	3.15	+ 1.13	100.1	3267	5.26	863	640 1553	398 953	84 185	29 90	9412	2.74

Table 6.6 : Economical comparison of mixtures butane-pentane

As can be seen in fig. 6.6 a good temperature adaptation also can be achieved with the systems R114-R22, propane-pentane and butane-h xane. In these cases only the concentrations with small amounts of the second component are investigated.

R114/R22	variables										heat exchange area [m^2] costs[10^3s]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap T_i-T_o	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R114	0.17	2.00	2.61	3.63	12.06	3.32	+ 1.74	95.7	2824	5.08	984	986 1918	505 1131	173 295	54 134	11034	3.34
0.8/0.2	0.05	6.37	3.00	5.31	15.16	2.85	-10.05	101.4	4032	5.57	711	1448 2502	759 1443	54 142	83 189	11783	3.19
0.4/0.6	0.14	7.23	5.50	7.51	20.79	2.77	-14.12	109.5	5263	5.33	563	1249 2470	639 1390	21 83	80 200	11364	3.24

Table 6.7 : Economical comparison of mixtures R114-R22

Pentane/ Propane	variables										heat exchange area [m^2] costs[10^3s]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap T_i-T_o	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
pentane	0.05	1.85	1.19	1.29	4.84	3.75	1.24	95.1	1210	5.40	2190	742 1410	646 1303	97 162	39 86	12171	3.41
0.9/0.1	0.11	5.00	1.77	1.70	5.37	3.16	- 7.05	96.7	1492	5.78	1795	1199 1876	629 1282	9 33	52 106	11891	3.10
.85/.15	0.17	6.19	1.42	1.98	6.24	3.15	-11.30	100.8	1730	5.78	1558	1002 1735	859 1533	-	71 136	11650	3.04
0.8/0.2	0.20	8.54	0.92	2.18	7.24	3.32	-13.75	104.8	1851	5.42	1460	925 1701	656 1314	-	76 149	11165	3.11

Table 6.8 : Economical comparison of mixtures pentane-propane

Butane/ Hexane	variables										heat exchange area [m^2] costs [10^3 s]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	$T_i^{evap}-T_o$	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t_a
Butane	0.15	1.50	2.62	4.16	13.12	3.15	+ 1.13	100.1	3267	5.26	863	640 1558	398 953	84 185	29 90	9412	2.74
0.9/0.1	0.22	3.50	5.58	3.48	9.75	2.80	-12.43	105.1	2717	5.73	1019	886 1752	517 1146	22 68	60 136	10082	2.65
0.8/0.2	0.33	4.23	6.93	2.72	7.67	2.82	-17.83	107.1	2075	5.53	1320	1124 1916	629 1282	-	67 138	11179	3.05

Table 6.9 : Economical comparison of mixtures butane-hexane

Unterleutner /6.1/ described in detail, that nonazeotropic mixtures at low concentrations show nonlinear temperature-enthalpy curves during evaporation and condensation. At small amounts of a more volatile substance the temperature-enthalpy curve is bent to higher temperatures and with small amounts of a less volatile substance the other way round. In the equimolar region the curves mostly show linear behavior. In fig. 6.7 the temperatur-enthalpy diagrams of evaporator and condenser for the systems 0,9/0,1 butane/hexane and 0,85/0,15 pentane/propane are outlined. Because of these effects the ΔT_c rises so much when only small amounts of propane are added to pentane and the ΔT_e even decreases. In the system butane-hexane the ΔT_e increases much more than the ΔT_c. The described effects also can be observed at R11-R12 and R114-R12, but in these cases the consequences are not of much importance as the boiling points of the two components are not that far apart.

The tables 6.7, 6.8 and 6.9 show, that in the present case of application the nonlinear behaviour of the temperature-enthalpy curves in the condenser and evaporator are of disadvantage.

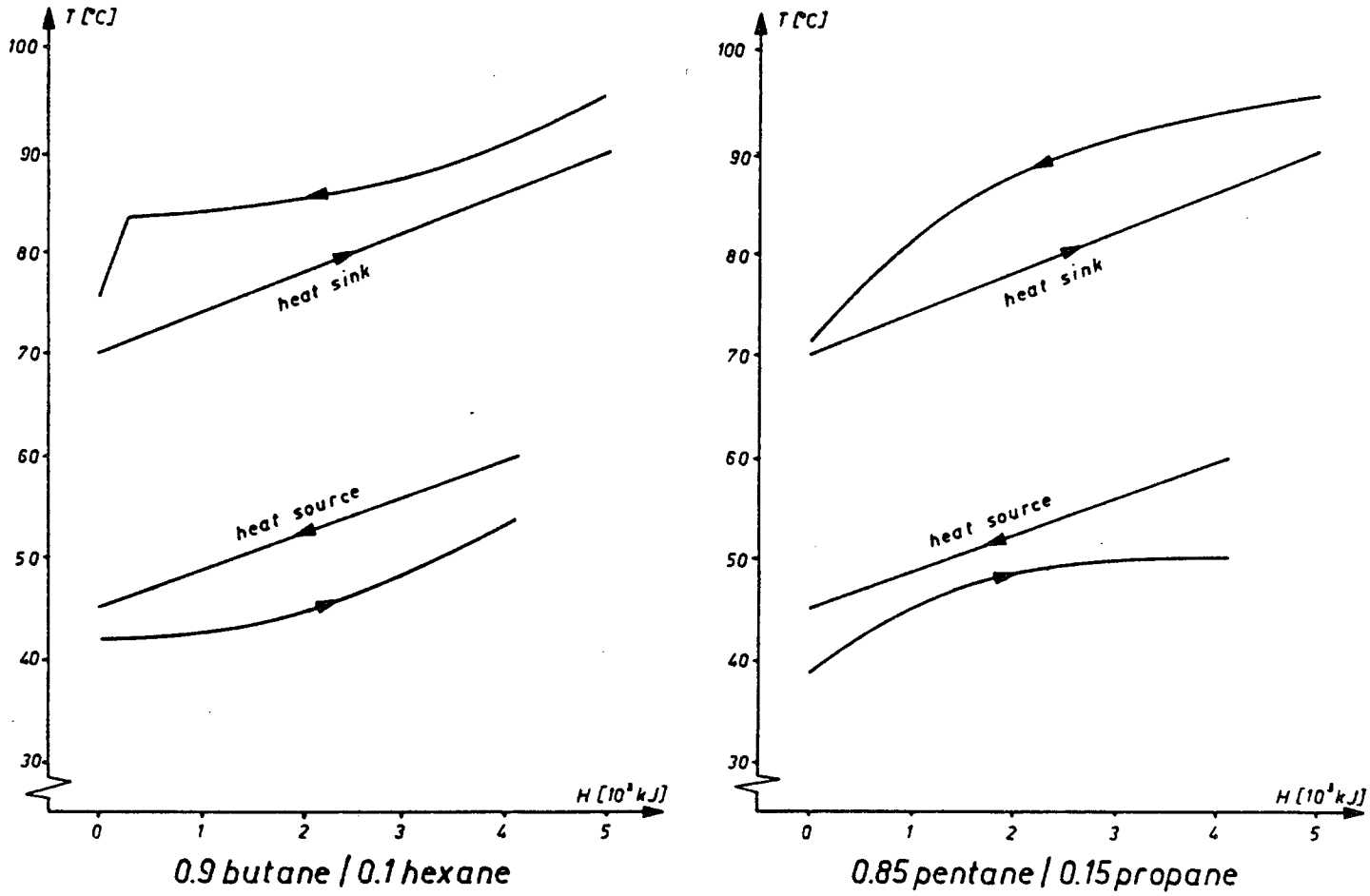

Fig. 6.7 : Temperature - enthalpy diagrams

6.3.3 Critical Reflections on the Results

Considering the heat pump layouts with the pure fluids (table 6.2) it is remarkable, that the minimum payback periods appear at rather small minimum temperature differences in the evaporators and condensers (ΔT_e and ΔT_c). These would be rather difficult to realize in practical designs. For that reason further calculations were made with the same refrigerants, but with ΔT_e and ΔT_c kept constant at 4^oC. The pressure loss in the evaporator again was varied until a minimum payback period was reached. The table 6.10 shows the results which now seem to be of more practical importance.

medium	variables									heat exchange area [m^2] costs [10^3s]						t
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	[a]
R11	0.08	4.	4.	1.79	7.17	3.99	120.7	1672	4.92	1609	597 1326	503 1128	59 125	32 82	10754	3.39
R21	0.12	4.	4.	3.02	11.54	3.82	135.6	2777	4.91	999	502 1324	344 908	45 117	21 71	9152	2.90
R114	0.14	4.	4.	3.43	12.41	3.62	96.7	2673	4.75	1038	828 1770	356 962	134 249	47 122	10547	3.51
R12	0.05	4.	4.	9.78	29.78	3.04	117.3	6667	4.65	462	970 2350	174 710	137 314	37 131	10337	3.55
n-pentane	0.05	4.	4.	1.18	5.09	4.31	96.6	1085	4.83	2436	568 1213	293 828	64 121	30 71	11557	3.75
n-butane	0.09	4.	4.	3.89	13.49	3.47	101.4	3067	4.87	913	546 1435	240 739	66 157	24 79	8999	2.89

Table 6.10 : Economical comparison of different media with T_e and T_c kept constant

In comparison with table 6.2 the payback periods here are 3,5 % to 9 % longer. The pressure losses at minimum payback periods are up to 37 % smaller. Beside these details it can be seen that in table 6.10 the refrigerants appear in the same order as in table 6.2 regarding their economical effectiveness. The quantitative differences between the payback periods have changed a little.

For the same conditions ($\Delta T_e = \Delta T_c = 4^oC$, p_e optimized) also the calculations for the mixture R114-R12 were repeated (table 6.11). Now 10 % reduction in the payback period can be achieved with the nonazeotropic mixture compared with pure R114, in contrast to 6% in table 6.4. This is because of the temperature differences ΔT_e and ΔT_c that in table 6.4 are higher at 0.6/0.4 than at pure R114.

R114/R12	variables										heat exchange area [m²] costs [10³ $]						t
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap T_i-T_o	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	k_{hp}	[a]
R114	0.14	4.	4.	3.43	12.41	3.62	+ 1.43	96.7	2673	4.75	1038	828 1770	356 962	134 249	47 122	10547	3.51
0.8/0.2	0.22	4.	4.	4.51	14.39	3.19	- 2.42	100.4	3356	5.00	840	957 1976	523 1153	107 223	54 140	10825	3.34
0.6/0.4	0.05	4.	4.	5.72	16.94	2.96	- 6.09	104.1	4273	5.31	676	1063 2165	702 1397	97 216	65 165	11210	3.20
0.4/0.6	0.05	4.	4.	7.01	20.25	2.89	- 6.15	107.8	5050	5.27	584	1099 2293	495 1188	101 223	61 167	10938	3.16
0.2/0.8	0.05	4.	4.	8.36	24.51	2.93	- 4.00	112.5	5882	4.96	515	1080 2373	293 917	115 266	50 153	10634	3.32
R12	0.05	4.	4.	9.78	29.78	3.04	+ 0.24	117.3	6667	4.65	462	970 2350	174 710	137 314	37 131	10337	3.55

Table 6.11 : Economical comparison of mixtures R114-R12 with ΔT_e and ΔT_c kept constant

Maybe the weakest point in the whole economical optimization that has been described in this paper are the functions for the determination of the apparatus costs. The correlation of prizes to continual functions is a strong idealization, and on top of that cost-functions, like those quoted in chapter 6.2.4, that have been derived by different people often show considerable differences. Nevertheless these functions do not influence only the value of the payback period, but also the location of its minimum refered to the variables ΔT_e, ΔT_c and Δp_e.

To find out how the whole mathematical model reacts upon modifications in the cost-functions the following investigations were made:
In the tables 6.12, 6.13 and 6.14 results of economical optimization for the refrigerants R11, R21 and R114 are listed, that have been calculated under the same conditions as those in table 6.2, but with the following modifications of the prize-functions: Once the compressor costs were doubled (doubling of the prize from the correlation in chapter 6.2.4) with all the remaining functions kept the same (table 6.12), one the prizes of the evaporators were doubled (table 6.13) and once the prizes of the condensers (table 6.14).

medium	variables									heat exchange area costs [10^3s] [m^2]						t
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	τ_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	[a]
R11	0.11	1.77	1.88	1.92	6.84	3.56	117.4	1760	5.30	3062	785 1535	741 1410	91 166	39 94	14309	4.10
R21	0.17	1.77	1.88	3.22	11.02	3.42	131.4	2907	5.28	1914	664 1530	497 1120	69 155	26 81	11556	3.33
R114	0.15	1.35	2.23	3.70	11.96	3.23	95.5	2907	5.22	1919	1025 1976	720 1386	188 312	58 141	13346	3.89

Table 6.12 : Economical comparison of different media at doubled compressor-prizes

medium	variables									heat exchange area [m^2] costs [10^3S]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R11	0.17	3.04	1.88	1.85	6.84	3.70	118.6	1639	5.03	1638	784 1534	384 1934	91 166	35 86	12739	3.90
R21	0.15	3.04	2.58	3.10	11.19	3.60	133.2	2825	5.09	983	593 1442	372 1897	59 139	24 76	11156	3.36
R114	0.22	2.62	2.23	3.57	11.96	3.35	95.3	2747	4.99	1011	1025 1977	385 1935	189 312	52 130	12772	3.95

Table 6.13 : Economical comparison of different media at doubled evaporator-prizes

medium	variables									heat exchange area [m^2] costs [10^3S]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R11	0.11	1.77	3.96	1.92	7.16	3.72	119.9	1754	5.10	1536	606 2675	713 1378	60 126	35 87	13536	4.07
R21	0.17	1.77	3.27	3.22	11.36	3.52	133.3	2907	5.14	959	543 2756	488 1109	51 127	24 77	12052	3.59
R114	0.17	1.35	3.96	3.70	12.40	3.35	96.9	2857	5.01	974	830 3546	643 1299	137 250	52 132	14335	4.42

Table 6.14 : Economical comparison of different media at doubled condenser-prizes

The following tendences can be observed:

Doubling the evaporator costs, both, ΔT_e and Δp_e, increase to compensate the higher evaporator prizes by smaller heat exchange areas (notice 22,6 % to 48,5 % smaller heat exchange areas of evaporators in table 6.13 compared with table 6.2). This results in a worse COP. In the case of the higher condenser costs the same effect can be observed with ΔT_c, but as here only one parameter directly influences the heat exchange area and the COP, the diminutions of these values compared with those in table 6.2 are smaller (8,6 to 18,5 %). At the higher compressor costs the temperature differences in the heat exchangers both diminish. There is no parameter that directly influences the compressor costs and at higher investment costs the minimum payback period appears at higher COP. The same tendency could be observed if the additionals to the apparatus costs (chapter 6.3.1) would be raised.

The results of the tables 6.12, 6.13 and 6.14 shall be analysed in two ways. The one thing that seems to be of interest is how the economical effectivenesses of the refrigerants among themselfs have changed. In the table 6.15 the payback periods for R11, R21 and R114 are gathered from the tables 6.2, 6.12, 6.13 and 6.14 and the relative differences for the refrigerants are expressed as percentage.

	normal prize-funct t [a]	devia tions %	ic-como *2 t [a]	devia tions %	ic-evap *2 t [a]	devia tions %	ic-cond *2 t [a]	devia tions %	medium devia. %
R 11	3.27	+16.8	4.10	+23.1	3.90	+16.1	4.07	+13.4	+17.3
R 21	2.80	basis	3.33	basis	3.36	basis	3.59	basis	basis
R 114	3.34	+19.3	3.89	+16.8	3.95	+17.6	4.42	+23.1	+19.2

Table 6.15 : Differences in the payback periods at normal- and modified prize correlations

Furthermore the results of the tables 6.12, 6.13 and 6.14 shall be compared with those that are achieved if one directly applies the modified cost functions (doubled compressor costs, - evaporator costs and -condenser costs) to the layout data of the table 6.2. There results (investment costs and payback periods) are listed in table 6.16. Here the payback periods hardly differ from those in the tables 6.12, 6.13 and 6.14, which means, that in the present case of application the differences in ΔT_e, ΔT_c and Δp_e caused by the repeated optimization do not effect any serious differences in the payback periods.

	COP	normal prize-func.		ic-comp *2		ic-cond *2		ic-evap *2	
		ic_{hp}	t	ic_{hp}	t	ic_{hp}	t	ic_{hp}	t
R 11	5.27	11352	3.27	14228	4.10	14155	4.08	14011	4.04
R 21	5.20	9562	2.80	11365	3.33	12276	3.60	11626	3.41
R 114	5.08	11034	3.34	12885	3.89	14641	4.43	13160	3.99

Table 6.16 : Investment costs and payback periods at normal- and modified prize correlations based on results from table 6.2

From these investigations (table 6.12 to 6.16) the consequences can be appraised if individual cost functions are applied to any results published in this paper. As a matter of fact all absolute values of the investment costs and payback periods should be treated with care, but the comparison of the economical effectiveness of the different refrigerants and refrigerant mixtures seem to be of general meaning, at least concerning the described conditions of application.

Furthermore the consequences of a lower prize for electricity shall be shown. In the tables 6.17 and 6.18 the results of optimization calculations are listed where the prize for electricity is assumed 0.5 S per kWh (at all other calculations 1 S per kWh is assumed).

Now the payback periods diminish considerably, but again the relative differences of the pure fluids are approximately the same. As the COP is inversely proportional to the running costs per year (eq2) at low electricity costs a high COP is not as important as at high ones. The minimum payback periods all appear at higher values for ΔT_e, ΔT_c and Δp_e. These considerations also imply, that the COP increase that can be achieved with non-azeotropic mixtures, at low electricity prizes is not as attractive as at high ones (notice 2,5 % reduction of the payback period in contrast to 6 % reduction in table 6.4).

medium	variables									heat exchange area [m^2] costs [10^3 S]						
	Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	T_{d-co}	η_v	COP	comp	cond	evap	sub	h-e	ic_{hp}	t_a
R11	0.17	3.46	4.65	1.83	7.27	3.98	122.0	1613	4.75	1667	572 1298	335 894	53 117	29 87	10446	1.95
R21	0.22	3.46	4.65	3.06	11.7	3.82	137.0	2732	4.80	1041	459 1264	274 796	41 110	20 68	8904	1.66
R114	0.28	3.04	3.96	3.53	12.4	3.51	96.7	2631	4.70	1051	831 1773	292 825	134 250	45 120	10415	1.96

Table 6.17 : Economical comparison of different media at electricity prize of 0.5 S per kWh

R114/R12	variables Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap $T_i - T_o$	T_{d-co}	η_v	COP	heat exchange area [m^2] / costs [10^3 S] comp	cond	evap	sub	h-e	ic_{hp}	t [a]
R114	0.28	3.04	3.96	3.53	12.40	3.51	+ 2.87	96.7	2631	4.70	1051	831 1773	292 825	134 250	45 120	10415	1.96
0.8/0.2	0.35	5.08	6.31	4.35	15.09	3.47	- 0.95	102.5	3086	4.50	908	791 1800	283 811	78 181	42 120	10163	1.95
0.6/0.2	0.07	6.31	7.00	5.32	18.01	3.38	- 5.61	107.2	3906	4.66	734	831 1921	367 953	65 168	46 133	10228	1.93
0.4/0.6	0.07	5.69	6.92	6.67	21.47	3.22	- 5.84	111.1	4717	4.72	620	867 2046	315 911	68 182	46 139	10172	1.91
0.2/0.8	0.07	4.46	6.92	8.24	25.95	3.15	- 3.83	116.0	5682	4.59	528	847 2114	258 851	80 213	41 137	10138	1.93
R12	0.05	4.00	4.00	9.78	29.78	3.04	+ 0.24	117.3	6667	4.65	462	970 2350	174 710	137 314	37 131	10337	1.95

Table 6.18 : Economical comparison of mixtures R114-R12 at electricity prize of 0.5 S per kWh

It should be considered, that the real economy of an installation cannot be expressed with the payback period alone. If for instance two investments show equal payback periods usually that with the higher first costs is the more effective, as in this case the profit after the payback time is higher. This means that at two heat pumps with equal payback periods the one with the higher COP (which is the one with the higher investment costs) represents the more economical solution, as with that one more energy costs can be saved which effects the higher profits after the payback time. Considering the results of this paper under these aspects the heat pumps with the nonazeotropic mixtures of refrigerants gain on importance. It is obvious that if one uses the criterion of a maximum profit the whole optimization of the machine has to be carried out under this principle. This means that the parameters Δp_e, ΔT_e and ΔT_c have to be varied until a maximum profit within a particular service live of the installation is achieved. This is equivalent to the demand of a minimum annual heat costs (c_{qa}) that can be calculated from the following equation:

$$c_{qa}\left[\frac{S}{a}\right] = rc_{hp}\left[\frac{S}{a}\right] + ic_{hp}\left[S\right] \cdot k_a\left[\frac{1}{a}\right] \qquad (4)$$

c_{qa}.....annual heat costs

rc_{hp}....running annual energy costs (equation 2)

ic_{hp}....investment costs

k_a.....annuity factor

To give an idea of the differences in the layouts if a minimum annual heat costs is used as optimization criterion, the results of such an optimization are shown in table 6.19. Here mixtures of R11/R12 were chosen, as with these an optimal temperature adaptation to the environmental conditions is possible and for that reason a high increase in the COP can be expected. In the calculations the annuity factor k_a was chosen 0.15 (15 years service live of the machine). The investment costs (ic_{hp}) and the running energy costs (rc_{hp}) were determined under the conditions of the calculations in chapter 6.3.2. Now the layouts show higher coefficients of performance as the optima appear at lower values of Δp_e, ΔT_e and ΔT_c. As a matter of fact the price for that are higher investment costs.

R11/R12	variables Δp_e	ΔT_e	ΔT_c	p_e	p_c	π	evap T_i-T_o	T_{d-co}	η_v	COP	heat exchange area costs [m^2] [10^3 S] comp	cond	evap	sub	h-e	ic_{hp}	c_{qa} [10^3 S]
R11	0.11	1.54	1.19	1.94	6.73	3.47	+ 1.96	116.5	1779	5.40	1519	779 1525	870 1546	112 192	41 97	11653	5864
0.8/0.2	0.14	5.50	1.50	2.94	8.17	2.78	- 9.83	114.3	2525	6.12	1094	1249 2057	1329 1971	52 119	73 149	12320	5478
0.6/0.4	0.29	6.50	2.50	4.23	10.46	2.47	-14.80	114.6	3311	6.31	850	1777 2611	1462 2082	—	101 198	12914	5458
0.4/0.6	0.29	5.50	3.00	5.76	13.96	2.42	-15.88	115.3	4310	6.18	673	1940 2888	1189 1888	15 60	111 227	12950	5536
0.2/0.8	0.05	3.00	2.50	8.07	19.51	2.42	-12.15	114.69	5882	6.14	512	1615 2795	849 1652	79 195	117 255	12353	5472
R12	0.05	1.00	1.50	10.51	28.43	2.70	+ 0.23	113.7	7246	5.20	432	1357 2773	394 1135	230 440	52 162	11859	6050

Table 6.19 : Economical comparison of mixtures R11/R12 with annual heat costs minimized

6.4 Conclusions

In the present case study rather high temperatures were chosen at which exclusively R114 is used today. The comparison with other high temperature media shows that from the economical point of view the use of R114 is relatively bad. For instance R11, R21 and n-butane result in shorter payback periods. Certainly it must be considered here that at the use of n-butane additional costs for safety precautations have to be taken into account, as this substance is hightly inflammable and explosive. At R11 and R21 only cooled compressors can be used in the present temperature range, as otherwise these substances superheat that high that there might be problems with their thermal stability.

The azeotropic mixture 40 % R21, 60 % R114 (by mole) also shows good performance conditions. With this mixture a high coefficient of performance can be achieved and the superheating of the medium is rather low.

Considering certain nonazeotropic mixtures with which temperature adaptations to the cooling of the heat source and the heating of the heat sink are possible, also economical improvements can be achieved. By mixing 40 % R12 (by mole) to R114 the payback period decreases 6 % compared with R114 (table 6.4), or by adding 20 % R12 to R11 the reduction of the payback period is 9.5 % compared with R11 (table 6.5). The improvements of the systems pentane-butane (table 6.6), R114-R22 (table 6.7), pentane-propane (table 6.8) and butane-hexane (table 6.9) are of less importance. It turns out, that from the economical point of few, mixtures that allow optimal temperature adaptations are not doing as well as it was expected (e.g. table 5, 40 % R11 - 60 % R12). The reason for that are the minimum temperature differences in the condenser and evaporator (ΔT_c,ΔT_e) as well as the pressure drop in the evaporator (Δp_e) that have to be increased in order to avoid too high investment costs. This effects that the increase of the coefficient of performance, gained by the nonazeotropic mixture, is not that high that a trastical reduction of the payback period can be achieved.

Furthermore the investment costs of the heat pump cannot be lowered by means of using nonazeotropic mixtures in the particular case that has been investigated. Only twice (80 % R11 - 20 % R12 and pentane-propane) a small reduction of the investment costs can be detected due to the increase of the volumetric efficiency. Lowering the pressur ratio seems to be of secondary importance as it hardly can be carriet out so far that a noticeable increase of the isentropic efficiency in the compressor can be expected.

Comparing all heat pump layouts the one with pure R21 represents the optimal economical solution, not counting systems with mixtures of pentane-butane or butane-hexane. If one cannot apply R21 (e.g. because of its high discharge temperature at an uncooled compressor) and the nonazeotropic mixtures shall be compared with pure R114, up to 10 % reduction in the payback periods can be achieved (R114-R21, R114-R12, R11-R12).

As a matter of fact all conditions might differ on other places of application. The question which working medium represents the most economic solution in heat pump systems for particular applications, can be answered only by comparison of economically optimized designs with the different substances.

6.5 References

/6.1/ UNTERLEUTNER K.; Thermodynamische Eignung von Arbeitsmedien und deren Gemische für Kompressionswärmepumpen; (Thermodynamical Applicability of Working Media and Mixtures for Compression Heat Pumps); Dissertation, Techn.Univ. Graz (Austria) 1983

/6.2/ CHAWLA J.; Wärmeübergang und Druckverlust bei der Kältemittelverdampfung im waagrechten Strömungsrohr; (Heat Transfer and Pressure Loss at Evaporation in Horizontal Tubes); Chem.Ing.Techn. Vol. 40, No. 5, 1968

/6.3/ VDI Wärmeatlas 2. Auflage 1974

/6.4/ STEPHAN K. and KÖRNER M.; Berechnung des Wärmeüberganges bei der Verdampfung binärer Gemische; (Calculation of Heat Transfer in Evaporating Binary Liquid Mixtures); Chem.Ing.Techn. Vol. 41, No. 7, 1969

/6.5/ Heat Exchanger Design Handbook; VDI-Verlag 1983

/6.6/ HAUSEN H.; Darstellung des Wärmeüberganges in Rohren durch verallgemeinerte Potenzbeziehungen; (Description of Heat Transfer in Tubes by General Power-Formulations); Z.Ver.Dtsch.Ing., Beiheft Verfahrenstechn. No. 4, 1943

/6.7/ BELL K.J.; Final Report of the Cooperative Research Program on Shell- and Tube Heat Exchangers; University of Delaware Eng.Exp.Sta.Bull. 5, 1963

/6.8/ SIEMERS H. (ECKARDT); Neue K_v-Wert- und K_{vs} Wert-Berechnungsmethoden für flüssige und kompressible Fluide; (New K_v value and K_{vs}-value Calculation Methods for liquid and compressible fluids); Sonderdruck aus "JCE-BERICHTE VII" 1980

/6.9/ KIEFER J.: Sequential Minimax Search for a Maximum; Proc. Am.Math.Soc. 4 (1953), 502-506

/6.10/ HOFFMANN/HOFMANN; Einführung in die Optimierung; (Introduction into Optimization); Verlag Chemie GmbH 1970

/6.11/ HELFRICH F., SCHUBERT W.; Ermittlung von Investitionskosten, Einfluß auf Wirtschaftlichkeitsrechnung; (Determination of Investment Costs, Influence on Economy-Calculation); Chemie-Ingenieur-Techn. Vol. 45, No. 13, 1973

/6.12/ PAUL J.; Schrauben und Kolbenverdichter im Vergleich; (Comparison of Screw- and Pistoncompressors); Kälte und Klimatechnik, 12, 1981

/6.13/ ASHRAE Standard 34-67, AMSI 1379, 1

/6.14/ REAY D.A.; Heat Pumps - design and applications; Pergamon Press 1979

/6.15/ RINDER L.; Schraubenverdichter; (Screwcompressor); Wien, New York: Springer 1979

/6.16/ OFNER H.; Dissertation, Techn.Univ.Graz (Austria) (will be published 1985)

7. DISCUSSION AND CONCLUSIONS

Nonazeotropic Mixtures

In the preceding chapters of this report and in the State-of-the-art report it has been shown that the comparison between pure fluids and mixtures is very complex and that general statements about advantages and disadvantages with mixtures can be made only to a very limited extent. The benefits with mixtures must be considered from application to application.

A discussion about general experiences and conclusions, interesting applications, and need for further work in the field of mixtures has been presented in the State-of-the-art report, see pp. B118-B120. All statements made there are still valid. In the Step Two part of the Annex (this report), improved knowledge in several areas has been obtained. Below, some conclusions and remarks, resulting from this work, are made.

Technical aspects

According to the results presented in Chapter 5, it is very important to compare pure fluids with mixtures at given outer conditions and to take all influences in the heat exchangers, compressor, etc. into account. It was shown there that the COP always increases with decreasing difference between the condenser and evaporator temperatures and with increasing temperature gradients of both the heat sink and the heat source. In order to get improvements in the COP, these latter gradients must normally be around 10 K or higher. For a given mixture, the composition giving the highest COP is always the one(s) giving the same temperature gradient as the heat sink and

the heat source. At favourable conditions for mixtures improvements of the COP on the order of 10-15 % seem to be possible in reality. The improvement is among other things influenced by the heat exchanger sizes used in the comparison.

According to the same chapter, the heating capacity can be increased considerably in many cases. This is always true when a pure fluid can be replaced by a more volatile mixture, e.g. replacing R114 with an R12/R114 mixture. Improvements of at least 100 % should be possible to achieve. It should also be possible in some applications to combine improvements of the COP and of the capacity. Such applications should be of very high interest for mixtures.

Economical aspects

Considering certain nonazeotropic mixtures with which temperature adaptations to the cooling of the heat source and the heating of the heat sink are possible, economical improvements can be achieved. E.g. by mixing 40 % R12 (by mole) to R114 the payback period of a 5 MW-unit decreases 6 % compared with R114, or by adding 20 % R12 to R11 the reduction of the payback period is 9.5 % compared with R11 (Table 6.5). It turns out, that from an economical point of view, mixtures that allow optimal temperature adaptations are not doing as well as it was expected. The reason for that are the minimum temperature differences in the condenser and evaporator (ΔT_c, ΔT_e) as well as the pressure drop in the evaporator (Δp_e) that have to be increased in order to avoid too high investment costs. Therefore the increase of the coefficient of performance, gained by the nonazeotropic mixture, is not so high that a drastical reduction of the payback period can be achieved.

In some cases it is possible to reduce even the investment costs of the heat pump due to the increase of the volumetric efficiency.

Comparing all layouts of large heat pumps the one with pure R21 represents the optimal economical solution, not considering systems with mixtures of pentane-butane or butane-hexane. If one cannot apply R21 (e.g. because of its high discharge temperature at an uncooled compressor) and the nonazeotropic mixtures shall be compared with pure R114, up to 10 % reduction in the payback periods can be achieved.

As a matter of fact all conditions might differ on other places of application. The question which working fluid represents the most economic solution in heat pump systems for particular applications, can be answered only by comparison of economically optimized designs with the different substances. No calculations have been made on small scale units.

Need for further work

The perhaps most important further work needed at this stage is to use the present knowledge about mixtures, presented in this report, in technical-economical studies on interesting applications for heat pumps with mixtures. As was discussed in the State-of-the-art report, it is thereby important also to take other aspects than the COP and the heating capacity into account. Such aspects are (see pp. B8-B12 in the State-of-the-art report):

- continuous variation of the heating capacity with the load

- adjustments of pressure levels and the compressor discharge temperature

- adjustment of series manufactured equipment to various heating capacity needs

- allowment of higher temperature gradients in the heat sink and the heat source, without decreasing the COP (e.g. in air-to-air heat pumps).

As mixtures offer possibilities for influences of all the variables above, they should in the future be an important means for a substantial improvement in the flexibility, when a heat pump shall be adjusted to a heating system. Today´s big problems with too small or too large heat pumps for a given object, with control, noise due to the need for high air flows, short compressor lives due to many annual starts and high pressure ratios, etc. should be possible to decrease considerably. This indicates that the possibilities for mixtures are not only in applications with high temperature gradients of the heat sink and the heat source, e.g. tap water heating, industrial applications, and air-to-air heat pumps, in which low air flows are required for e.g. noise reasons.

Other areas, where further work is needed, are the ones already mentioned in the State-of-the-art report (p. B120):

- measurements and development of estimation methods for state and transport properties

- measurements of compressor efficiencies for various types of compressors

- measurements of heat transfer coefficients in the condenser and the evaporator

- problems related to lubricating oils, especially chemical stability at high temperatures and solubility

- system simulations, containing appropriate types of mixtures, compositions, and component designs.

Of these areas, the one about state and transport properties has been considered by the Working Group to be appropriate for further international collaboration. Therefore, the group has suggested an extension of this Annex, which has been submitted to the Executive Committee of Advanced Heat Pumps. In this extension, entitled "State and Transport Properties of High Temperature Working Fluids and Nonazeotropic Mixtures", work shall be done in three areas:

- Identification of data of importance via a sensitivity analysis in system simulation programs.

- Methods for assessment of data.

- Measurements of data.

The work is scheduled to go on for 18 months and can start in late autumn 1985.

High-temperature Working Fluids

Not much has been reported on high temperature working fluids, through Step Two of the Annex. Some work has been done, by various research groups on the use of water and alcohols with oil-free compressors.

The search for new fluids has brought up some interesting substances, but no final choice has been made.