Combined Air-conditioning and Tap Water Heating Plant, Using CO_2 as Refrigerant for Indonesian Climate Condition

By
Willy Adriansyah

Thesis submitted in partial fulfillment of the requirements
for the degree of Doktor Ingeniør (Ph.D.)

Norwegian University of Science and Technology
Faculty of Mechanical Engineering
Department of Refrigeration and Air-conditioning

April 2001

Abstract

A combined air-conditioning and water heating system using carbon dioxide as refrigerant has been investigated theoretically and experimentally. A computer program simulates the combined system has been developed and verified with experimental data. Effects of the following parameters to the system performance were examined: ratio of hot water load to rejected heat from air-conditioning system (load ratio), evaporation temperature, cooling medium temperature, inlet water temperature, hot water temperature, discharge pressure, and presence of internal heat exchanger. Main results were coefficient of performance and cooling capacity.

It was concluded that there is an optimum pressure where the system reaches the highest coefficient of performance. Variation of coefficient of performance of the combined system with discharge pressures is similar to that of the air-conditioning system without heat recovery.

Load ratio affects the performance of the air-conditioning side. Coefficient of performance of the air-conditioning side (cooling-COP) increased with increasing load ratio. Optimum discharge pressure was affected by load ratio.

Improvement of cooling-COP depends on both cooling medium and inlet water temperatures. The cooling-COP was lower at higher cooling medium temperature and higher inlet water temperature. When inlet water temperature is higher than cooling medium temperature, the cooling-COP will be lower compared to the air-conditioning system without heat recovery.

The system performance decreased as hot water temperature increased. The decrease is due to a need for higher discharge temperature to achieve maximum cooling-COP.

Internal heat exchanger plays an important role in achieving higher system performance. Coefficient of performance is higher for the combined system with internal heat exchanger. The length of internal heat exchanger affects the cooling-COP and the location of the optimum discharge pressure.

Estimation results used for calculating annual energy consumption in various type of buildings show that the largest energy saving can be achieve in hospitals, followed by in hotels and in multifamily buildings. Simple comparison of combined CO_2 system with separated R22 and stand-alone water heating system show better total system efficiency for CO_2 system.

Preface

This doctoral work has been carried out at Department of Refrigeration and Air-conditioning at the Norwegian University of Science and Technology (NTNU), during the period of September 1997 to April 2001.

My supervisor have been Professor Arne M. Bredesen at NTNU, associate Professor Jostein Pettersen at SINTEF Energy, and Professor Aryadi Suwono at Bandung Institute of Technology. I would like to thank them for supporting and encouraging me during this work. My gratitude goes to the late Professor Gustav Lorentzen who invented the system investigated in the present work.

The study has been funded for the first semester by Direktorat Jenderal Pendidikan Tinggi, Indonesia and the rest of the semesters by Ministry of Education, Research and Church Affairs (KUF) in cooperation with International Office of NTNU and The Norwegian Educational Loan Fund, through Quota Program. Their supports have been highly appreciated.

To my wife Yessi Novia, my daughter Nursifa Aghnia, and my son Muhammad Naufal, my gratitude goes to them all for providing me a warm and happy family and for their patience living in completely different culture.

Table of Contents

Nomenclature

Latin letters

A	Helmholtz energy	kJ/kg
COP	Coefficient of Performance	-
Cp	Isobaric Spesific Heat Capacity	kJ/kgK
d	Diamter	m
D	Coil Diameter	m
E	Exergy	kJ
f	Friction factor	-
h	Spesific Enthalpy	kJ/kg
H	Enthalpy	kJ
I	Irreversibility	kW
IHX	Internal Heat Exchanger	-
k	Thermal Conductivity	W/mK
m	Mass flow rates	kg/s
Nu	Nusselt Number	-
P	Pressure	Pa or bar
Ph	Discharge Pressure	bar
Pr	Prandtl Number	-
Q	Thermal Energy	kW
qe	Cooling capacity	kW
r	Radius	m
R	Spesific Gas Constant	kJ/kgK
s	Spesific Entropy	kJ/kgK
S	Entropy	kJ/K
T	Temperature	°C or K
v	Spesific Volume	m^3/kg
W	Shaft work	kW
wc	Compressor power consumption	kW
xr	load ratio	-

Greek letters

Δ	Finite Different	
ε	spesific exergy	kJ/kg
η	efficiency	-
ϕ	Dimensionless Helmholtz Energy	-
μ	Dynamic Viscosity	Pa.s
ρ	Density	kg/m^3

Subscript

a	air
co2	carbon dioxide
e	evaporator, exit
evap	evaporation
h	hydraulic
hrhx	heat rejecting heat exchanger
i	inlet, inner side
in	inlet
is	isentropic
mix	mixing point
o	ambient, outer side
out	outlet
r	thermal reservoir
sink	heat sink
vol	volumetric
w	water, wall
whhx	water heating heat exchanger
x	exergetic

Superscript

o	ideal condition, degree
Q	thermal exergy
r	residual

Summary

Introduction

Natural refrigerants have been gained attention over the last decade to be used as working fluids in refrigeration application. One of the natural refrigerants is CO_2 (carbon dioxide), which offers complete solution to current environmental problems such as global warming and ozone layer destruction. CO_2 has zero ozone depleting potential and negligible global warming potential. CO_2 has excellent properties to be used as refrigerant such as; non-toxic, non-flammable, the price is only a fraction of today refrigerant, excellent thermodynamic properties, and compact system components due to high density.

The most distinction of CO_2 properties compared to common refrigerants is the low critical temperature of 31°C and high critical pressure of 73.8 bar. For air-conditioning application in tropical countries, the outdoor air temperature will be close to the critical temperature of CO_2 most of the time, leading to transcritical operation to obtain better efficiency. The efficiency of CO_2 system can be increased by lowering cooling medium temperature. Since ground water temperature is lower than outdoor air temperature in tropical countries, the average cooling medium temperature can be lowered in situation where there is a simultaneous need of space cooling and hot water such as in hospitals or hotels.

The present work studies the potential of a combined air-conditioning and water heating system using CO_2 as working fluid. The water heating heat exchanger recovers part of rejected heat of the air-conditioning system to produce hot water. When inlet water temperature to the water heating heat exchanger is lower than cooling medium temperature of the heat rejecting heat exchanger, the average cooling medium temperature will be lower and the efficiency of the system will increase. This combined system offers energy saving by eliminating the need of energy to produce hot water.

Main objectives of this study were to investigate the combined air-conditioning and water heating system using CO_2 as working fluid theoretically and experimentally. Different operating conditions of experiments were chosen to locate vital parameters for the combined system performance. A computer model of the combined system was developed and verified with the experimental data. Thermophysical properties of CO_2 was also written in computer code and integrated with the model.

Combined CO_2 air-conditioning and water heating system

A computer program of thermophysical properties of CO_2 that can be integrated with other program such as spreadsheet program has been developed in this work. Extended equation of state from Span and Wagner is used to calculate thermodynamic properties and equation of state from Vesovic et al. is used to calculate transport properties.

Some promising applications of transcritical cycle using CO_2 as working fluid are heat pump water heater and mobile air-conditioning systems. Heat pump water heater is the most promising application compared to other refrigerants due to better match of refrigerant temperature and water temperature. Heat rejecting process in transcritical cycle takes place in supercritical region where temperature and pressure are independence properties. By regulating the discharge pressure, the gliding temperature can be increased and a better temperature match can be obtained. In air-conditioning application, the key point to get a higher efficiency of transcritical cycle is to achieved a small temperature different between cooling medium temperature and CO_2 temperature leaving gas cooler. This temperature different is called temperature approach. A lower efficiency of CO_2 air-conditioning system in a higher cooling medium temperature has been reported from several studies. This is due to a lower CO_2 compressor efficiencies than expected and improper component design leading to a higher evaporation temperature and higher approach temperature.

The other promising application of CO_2 transcritical cycle is combined air-conditioning and water heating system. There will be at least two gas coolers in this system, one for rejecting heat and the other for recovering heat to produce hot water. These gas coolers can be arranged in series or parallel. In series configuration, gas cooler as water heating heat exchanger is placed in front of gas cooler as heat rejecting heat exchanger. The approach temperature will become lower and will tend to zero. This arrangement is similar to a heat recovery using desuperheater in subcritical cycle. In parallel configuration, gas cooler as water heating heat exchanger is placed in parallel with gas cooler as heat rejecting heat exchanger. Hot gas CO_2 discharged from compressor is split into two streams, one stream enters heat rejecting heat exchanger and the other stream enters water heating heat exchanger. The split ratio of hot gas CO_2 depends on load ratio of hot water load to total rejected heat of air-conditioning system. The load ratio will determine the performance of the air-conditioning system.

Steady state modeling of combined air-conditioning and water heating system

A computer program models the combined CO_2 system has been developed in this work. The model consists of blocks of component model including compressor, gas coolers, and internal heat exchanger. The gas cooler block can be used to model water heating heat exchanger or heat rejecting heat exchanger. The gas cooler blocks can be arrange in series or parallel. It is also possible to choose whether to use internal heat exchanger or not. The simulation results are in good agreement with the experimental results. The maximum deviation of the simulation program to the experimental results is 5% for coefficient of performance and 7% for cooling capacity.

Test facility

A test facility consisted of four loops was constructed. The loops are: a CO_2 loop, a glycol loop simulating heat source, a water loop simulating heat rejecting system, and a water loop simulating hot water system. Heat from electrical element is transferred to glycol and evaporates CO_2 in the evaporator. Saturated vapor CO_2 is heated in low pressure side of internal heat exchanger before being compressed by compressor. Part of energy content of hot gas CO_2 discharged from the compressor is transferred to heat water and the rest of it is dissipated to the cooling medium. Subcooling of CO_2 is done in high pressure side of the internal heat exchanger before entering expansion valve. High pressure CO_2 then expands to evaporation pressure.

The following parameters were varied in the experiments: evaporation temperature, cooling medium temperature, inlet water temperature, hot water temperature, and load ratio. The test rig was also ran with and without internal heat exchanger.

Result and discussion

In parallel gas coolers configuration, the load ratio affects the characteristic of the combined system. The dependence of cooling-COP (coefficient of performance of the air-conditioning system) of the combined system on discharge pressure is similar to that of air-conditioning system without heat recovery. The cooling-COP increases as load ratio increases. There is an optimum discharge pressure where the cooling-COP reaches the highest value. This optimum pressure depends on the load ratio. At 25°C cooling medium temperature, there is no significant effect of heat recovery on air-conditioning side performance. At 30°C or higher cooling medium temperature, cooling-COP increases as load ratio increases. At certain load ratio, there are minimum optimum discharge pressures at 30°C and 35°C cooling medium temperatures.

Improvement of the air-conditioning side is lower in series gas coolers configuration compared to parallel configuration. Cooling medium temperature will dictate the cooling-COP in series configuration and reach the highest value when the approach temperature approaches zero. The effect of heat recovery on optimum discharge pressure is insignificant in series configuration. In parallel configuration, the improvement will depend on inlet water temperature, cooling medium temperature, and load ratio. The largest improvement is achieved when all rejected heat from air-conditioning system is utilized to heat water.

Evaporation temperature affects the performance of combined system. The optimum pressure is lower for a lower evaporation temperature. The cooling-COP is lower at a lower evaporation temperature. There are two main reason associated with a lower cooling-COP by lowering evaporation temperature. The first one is that for the same discharge pressure the specific refrigerating capacity is lower at a lower evaporation temperature. The second one is associated with higher specific compression power followed by lower compressor performance at a higher pressure ratio.

Inlet water temperature is another important parameter in combined system. The performance of air-conditioning system will improve if the inlet water temperature is lower than the cooling medium temperature for 60°C hot water temperature. When the inlet water temperature is the same as or higher than the cooling medium temperature, the performance of the air-conditioning system will decrease. In case of hot water storage system, the inlet water temperature to the water heating heat exchanger should be maintained as low as possible. If one hot water tank is used, it is better to use hot water tank that can establish stratification.

Increasing hot water temperature will degrade the performance of air-conditioning system. This is because a higher discharge pressure is needed to heat water to a higher temperature. Since a higher discharge pressure is needed as hot water temperature increases, the compressor power consumption will increase while the cooling capacity is about the same for the same inlet water temperature.

The presence of internal heat exchanger is important to get higher system performance. A higher cooling-COP for the system with internal heat exchanger is due to a higher increase in the specific cooling capacity compared to the increase in specific power consumption over the system without internal heat exchanger. CO_2 temperatures before throttling valve is significantly lower in a system with internal heat exchanger due to subcooling effect that causes a large increase in specific refrigerating capacity (lower throttling losses) especially at a pressure close to critical point.

The length of internal heat exchanger affects the system performance. The cooling-COP will increase when the length of internal heat exchanger is added. The optimum discharge pressure will shift to a lower value as the length of internal heat exchanger increases. There is a limit of improvement even the length of heat exchanger is added beyond this limit. This limit is achieved when the temperature of the stream with lower specific heat capacity approaches the temperature at the hot end of the heat exchanger with countercurrent flows or the cold one with parallel flows.

The improvement of the air-conditioning system side can be seen more clearly when exergetic efficiency is plotted as a function of discharge pressure. Eventhough the shape of exergetic efficiency curves of air-conditioning side is similar to the shape of cooling-COP curves, the improvement of the performance of the air-conditioning system toward the same system without losses can be observed directly in exergetic efficiency curves. The exergetic efficiency of air-conditioning side increases as load ratio increases.

The system improvement can even be seen more clearly if one consider the whole system, i.e. both air-conditioning and water heating system. The system performance improvement shifts to a higher value as load ratio increases. This is because a lot of exergy needed to produce hot water, which is a form of low level energy, has been eliminated and is supplied from the rejected exergy from the air-conditioning system.

Using energy estimation program called eQuest (EQUEST, 2000), an estimation of energy consumption for several types of building located in Jakarta (Indonesia) can be performed. In this work, four types of building that is considered as having potential to save energy from domestic hot water side has been done. From the estimation result, the most potential application of combined air-conditioning and water heating system is in hospital, followed by in hotels and multifamily building. There is small energy saving that can be achieved in office buildings.

A simple system comparison between combined CO_2 system and separated R22 air-conditioning system and stand-alone water heating system shows a better system efficiency of CO_2 system compared to a separated system.

Main conclusion

- As in a transcritical system, there will be an optimum condition for a combined air-conditioning and water-heating system at which the system reaches the highest cooling-COP. The optimum condition is determined by geometrical parameters such as gas coolers configuration (series or parallel) and presence of internal heat exchanger, and by operational parameters such as cooling medium temperature, water inlet temperature, hot water temperature, evaporation temperature, and percentage of heat recovery.
- In parallel configuration, the optimum condition will be depending on the percentage of heat recovery. The performance of the air-conditioning side is determined by the heat sinks temperature. If the inlet water temperature to the water heating heat exchanger is higher than the cooling medium temperature, the air-conditioning side performance will become lower.
- The influence of heat recovery in series configuration on the performance of the air-conditioning side is insignificant and the performance of the air-conditioning side is dictated by cooling medium temperature. The location of the optimum discharge pressure in series configuration is not affected by heat

recovery and a higher inlet water temperature to the water heating heat exchanger can be tolerated without degrading the performance of the air-conditioning side.

- The optimum discharge pressure is lower at a lower evaporation temperature. The variation of the optimum discharge pressure with percentage of heat recovery is similar at all evaporation temperatures ran in the experiment.
- At all evaporation temperatures and cooling medium temperature of 30°C or higher, the cooling-COP is increased as percentage of heat recovery increases. At 25°C cooling medium temperature, the cooling-COP is decreased slightly.
- The location of the optimum discharge pressure is affected by the different between the optimum discharge pressure in air-conditioning mode and that in full recovery mode. If the different is not large there will be a minimum of optimum discharge pressure at certain percentage of heat recovery. As the different becomes larger, the optimum discharge pressure will vary linearly with percentage of heat recovery.
- The cooling capacity was increased at all percentage of heat recoveries and reached the highest value at full recovery mode.
- Producing hot water higher than 70°C with parallel configuration will deteriorate the performance of the air-conditioning side. At this situation, series configuration is a better option.
- Internal heat exchanger is important to improve the system performance. The optimum discharge pressure is lower and the cooling-COP is higher for the system with internal heat exchanger. The optimum cooling capacity of the system with or without internal heat exchanger is similar. The effect of heat recovery on the system with or without internal heat exchanger is also similar.
- The agreement between the experimental data and the simulation results is good. The average deviation is ±5%, which is within the average uncertainties of the measurement system.
- Exergetic efficiency of the combined system is better than exergetic efficiency of the separated R22 air-conditioning and water heating system.
- The most promising application of the combined air-conditioning and water-heating system is in hospitals, followed by in hotels and in multi family buildings.

1 Introduction

1.1 Background

Two problems of degrading the environmental quality within these two decades that have been turning into big issues are widening of ozone hole and global warming. Ozone layer is needed to reduce an ultra violet radiation from the sunlight that is harmful for human being while the main effect due to global warming is an increase of earth atmosphere temperature. Compounds that contain chlorine and bromine, such as CFC (chlorofluorocarbon) family and HCFC (hydrochlorofluorocarbon) family are regarded as agents for accelerating the destruction of ozone (Montreal Protocol, 1987). Global warming is a result of hindering heat radiation from the earth to the outside of atmosphere by a layer containing some gases. Some gases that have potential to hinder this radiation are CO_2, CH_4, N_2O, HFCs, PFCs, SF6 (Kyoto Protocol, 1997).

As a concern to these environmental issues, two agreements have been signed, the Montreal protocol in 1987 for banning production and consumption of ozone depleting compounds and the Kyoto Protocol in 1997 for reducing consumption of global warming substances.

Some industries including refrigeration, air-conditioning, and heat pump, as the main consumers of CFCs and HCFCs have been forced to search for its substitutes. There are two main ways in searching these substitutes, chemical way and natural way. In the chemical way, a new substance is developed with the objective is to have the characteristic as close as possible to the characteristic of substances being substituted in order not to make a big change in the system components. Looking back to history of refrigeration, this is the same way with what had been passed by CFCs and HCFCs development, but with different objectives. At that time, CFCs and HCFCs were the result of the need for local environment concern (human safety), while today the new synthetic refrigerant has been producing as a result of the need for global environmental concern.

In the natural way, the natural compounds that have been already circulating in earth atmosphere such as air, NH_3, SO_2, CO_2, hydrocarbon, and water are utilized as refrigerant. Due to its inherent characteristic, these compounds do not create harmful effect on environment such as ozone depletion and global warming problem.

CO_2 had been used since 1889 as refrigerant mainly in large capacities refrigeration system such as in marine refrigeration (Strømmen et al., 2000). Rapid drop in refrigerating capacity and very low coefficient of performance (COP) when passing hot weather areas had become a major factor to make this system unpopular and not so easy to operate. This performance degradation is due to the characteristic of CO_2 that has very low critical temperature of 31°C.

Operational problem came from the fact that some amount of CO_2 must be charged to the system in order to maintain the cooling capacity and this additional charge must be drained of the system again to reduce the pressure when passing mild weather areas (Shulters, 1944).

Since transcritical cycle introduced by the late Prof. Gustav Lorentzen, the difficulty when operating a CO_2 refrigeration system can be avoided (Lorentzen, 1995). Charging and discharging of refrigerant to control cooling capacity and COP can be done automatically. Hence, the system can be operated more or less as in common subcritical refrigeration cycle. Following this invention there have been a lot of effort done by research institute or related industries to exploit the potential of CO_2 as a promising refrigerant in some applications such as air-conditioning and heat pump.

In air-conditioning system with direct expansion, experimental results showed that CO_2 system has capacity and efficiency similar to that of R22 system (Aarlien and Frivik, 1998). It is worth to note that the CO_2 system in this experimental study was still in an early stage of research while the baseline R22 system was the state-of-the-art system. Experiments on mobile air-conditioning system have been performed and they showed the performance of CO_2 system was higher at low ambient temperature (below 35°C) but lower at higher ambient temperature (above 35°C) compared to R134a system (Furuya, 1999). The most promising result of the application of transcritical cycle due to its unique characteristic has been for hot water heat pump where the heat source is at relatively constant temperature (such as ambient air or ground water) and the heat sink is at large gliding temperature. In this situation, transcritical cycle is more efficient cycle compared to subcritical cycle (Neksa et al., 1998).

Considering the characteristic of CO_2 system which is somewhat inferior in air-conditioning mode while superior in heat pumping mode, it is possible that the performance of CO_2 air-conditioning system be improved if it is run in combined mode both for air-conditioning and water heating. Owing to its characteristic, transcritical CO_2 cycle is strongly affected by temperature of cooling medium where the system performance increases with decreasing cooling medium temperature. In combined air-conditioning and water-heating system, two heat sinks are available. In case of air-cooled air-conditioning, one heat sink is the ambient air and the other heat sink is water. If the inlet water temperature is lower than the ambient air temperature, the air-conditioning performance can be improved due to a lower average cooling medium temperature.

By utilizing rejected heat from the air-conditioning system for producing hot water, energy consumption of the water-heating system can be eliminated because normally the cooling load is higher than the hot water load. There can be two advantages of this combined system, energy saving from the hot water production and improvement of the air-conditioning system performance due to a lower cooling medium temperature.

Such combined system using CO_2 as refrigerant has been demonstrated in commercial and industrial refrigeration. In commercial refrigeration, Nekså reported on a combined refrigeration and water heating system in supermarkets. Waste heat from refrigeration system was utilized for space and tap water heating and 37% reduction in overall energy consumption could be achieved compared to R22 system without heat recovery (Nekså et al., 1998). A simultaneous refrigeration and water heating system in food industries has been tested successfully in New Zealand food processing industries (Yarral et al.1999).

Buildings like hotels, hospitals, or multi-family housings often need both air-conditioning and water-heating. Cooling is needed year-around in tropical countries like Indonesia, Malaysia, or Singapore. Hence, there is always abundant heat dissipated to the outdoor air from the air-conditioning system. Meanwhile, energi to produce hot water takes quite large portion of total energy consumption in these building types. However, since the energy needed for water heating is often less then dissipated energy from the air-conditioning system, the energy consumption for water heating can be eliminated by utilizing this rejected energy that otherwise lost. This combined system is suitable for these types of building.

There has been no information available regarding an air-conditioning system with heat recovery using CO_2 as refrigerant for tropical countries. Therefore, it is necessary to study a combined air-conditioning and water heating system for these areas. This study is focused on a combined system to investigate the potential of transcritical CO_2 system as both an air-conditioning and a water heating system. This potential will be analyze theoretically and experimentally.

1.2 Objectives

The potential of transcritical CO_2 cycle in some applications have been observed both theoretically and experimentally. Most of this research were focused on mobile air-conditioning and hot water heat pump system. The main objective of this work is to study a combined air-conditioning and hot water heating system using transcritical CO_2 cycle.

A model of this combined system is important to be developed that can be used as a tool in analyzing the system characteristic in a broad range of operating conditions. If the model can represent the system characteristic within a specified tolerance then experimental work can be focused just on crucial operating conditions and the other points can be simulated with the help of this model, hence reducing the number of experiments. Furthermore, the model can be used as a tool for designing a similar system with different component capacity and configuration. In this study, a model of the combined air-conditioning and water-heating system was developed and was verified by experimental data.

The objectives of the present work are as follows:
- Develop a model of the system.
- Design and build a test rig.
- Verify experimentally that the system can be realized within practical ranges of operation.
- Compare the model with the experimental results.
- Determine the crucial parameters for the system to operate in a high performance.
- Show the potential of the system through a practical application approach.

1.3 Outline of thesis

To achieve the objectives mentioned above, this thesis was arranged into some chapters.

Introduction of this work is given in Chapter 1 that also contains the background and the objectives of the study.

Chapter 2 gives an overview of thermophysical properties of CO_2 and transcritical vapor compression cycle. The application of the cycle to an air-conditioning and hot water heat pump system is described shortly while the principle of the combined air-conditioning and hot water heating system is explained briefly. Principle of exergy analysis is also given in this chapter.

In chapter 3, a steady state modeling of a combined system air-conditioning and water heating system is explained. Each main components of the system was modeled as a building block for the whole system and these component models were verified with the experimental data. The main component relevant to this work are; compressor, air-cooled gas cooler as heat rejecting heat exchanger, water-cooled gas cooler as water heating heat exchanger, and internal heat exchanger. Another important part of the system that has been modeled is a connection line, which was described by a simple model.

Description of the test rig is described in Chapter 4. Principal design and construction of the gas cooler is explained in detail while the compressor is only described shortly. The other components are also shown such as liquid separator and expansion device. System measurement and uncertainties are also explained here.

Experimental and simulation results are presented in Chapter 5. The experimental results covering important operating conditions were grouped into a relevant parameter such as the effect of: cooling medium temperature, inlet water temperature, hot water temperature, evaporation temperature, percentage of heat recovery, and influence of internal heat exchanger. Effect of gas coolers arrangement that can be parallel or series is studied through simulation.

A discussion of experimental and simulation results is presented in Chapter 6. The effect of different operating condition obtained from experiment and simulation is discussed in depth. In this chapter, the potential of the combined system is described by a practical example in some types of building. A simple analysis is shown to show its saving potential of energy consumption in these building types.

Finally the conclusion are drawn in Chapter 7 and proposition are made for further work.

2 Combined CO_2 Air-Conditioning/Water-Heating

Since transcritical CO_2 cycle has been considered as one of promising alternatives for some refrigeration applications that use CFCs or HCFCs as their working fluid, there have been many research activities that show its potential in low temperature refrigeration, space heating, and mobile air-conditioning. Research activities mostly concentrated on mobile air-conditioning because this sector consume a major amount of refrigerant and contribute to global warming and ozone destruction and should be looked for its solution in a relatively short time frame. In residential air-conditioning area, there have been only a few research activities exploiting the use of CO_2 as working fluid.

In some buildings like hotels or hospitals, which equipped with air-conditioning and consumed hot water, there is a potential to carry out energy saving. The ratio of hot water heating load to that of compressor power in hotels normally between 15% to 30% (Haughton, 1997). Hence, it is possible to recover a portion of rejected heat from the air-conditioning system to heat water so that the energy consumption for water heating can be eliminated. For a tropical country, this system offers a substantial saving since the air-conditioning is needed year around and hence, hot water can be produced for free. In this work, the potential of transcritical CO_2 cycle as a combined air-conditioning and water-heating system will be studied.

Several advantageous of CO_2 as working fluid are as follow:

- Zero ODP (ozone depleting potential)
- GWP (Global Warming Potential) is set to one (reference substance)
- Non toxic
- Non flammable
- Excellent thermodynamic properties
- The price is only a fraction of today refrigerant
- Compact system component due to a high density.

2.1 Thermophysical Properties Of CO_2

The primary distinction between CO_2 and other refrigerant are a low critical temperature of 31.0°C and a very high critical pressure of 73.8 bar. This characteristic leads to different consideration when designing an air-conditioning system using CO_2 as its working fluid, since most of the time the system will operate close to its critical region when rejecting heat to the ambient. As the gas cooling process is performed around the critical region, the thermophysical properties of CO_2 vary greatly.

It is important to have a reliable thermophysical properties of CO_2 for designing and investigating such a system. Relevant thermodynamic properties are temperature, pressure, density, specific heat capacity, enthalpy, and entropy. A relation between these properties can be expressed in an equation of state and an extensive equation of state from Span and Wagner (Span and Wagner, 1996) will be used since this equation is the latest comprehensive equation of state for CO_2. Transport properties that must be available are thermal conductivity and dynamic viscosity, which will be determined by equation of state from Vesovic (Vesovic et al., 1990). For water, which is used in this work as heat sinks, its properties are calculated based on equation from ASHRAE.

The equation of state from Span and Wagner can be expressed as dimensionless Helmholtz energy form with density and temperature as its free variable. The equation is as follow:

$$\frac{A(\rho,T)}{RT} = \Phi(\delta,\tau) = \Phi^{o}(\delta,\tau) + \Phi^{r}(\delta,\tau) \tag{2.1}$$

where $\delta = \rho/\rho_c$ and $\tau = T_c/T$. A is Helmholtz function and ϕ^{o} represent an ideal gas condition while ϕ^{r} represent a residual function as a departure function from ideal condition. All thermodynamic properties can be obtained by combining derivatives of Equation (2.1). Table 2-1 shows the relationship between some thermodynamic properties to Helmholtz function.

Table 2-1 Relationship between thermodynamic properties to Helmholtz function (Span et.al)

Property	**Relation to dimensionless Helmholts function**
Pressure: $P(T,\rho) = -\left(\frac{\partial A}{\partial v}\right)_T$	$\frac{P(\delta,\tau)}{\rho RT} = 1 + \delta \cdot \Phi^{r}_{\delta}(\delta,\tau)$
Enthalpy: $h(T,\rho) = A - T\left(\frac{\partial A}{\partial T}\right)_v - v\left(\frac{\partial A}{\partial v}\right)_T$	$\frac{h(\delta,\tau)}{RT} = 1 + \tau \cdot (\Phi^{o}_{\tau} + \Phi^{r}_{\tau}) + \delta \cdot \Phi^{r}_{\delta}$
Entropy: $s(T,\rho) = -\left(\frac{\partial A}{\partial T}\right)_v$	$\frac{s(\delta,\tau)}{R} = \tau \cdot (\Phi^{o}_{\tau} + \Phi^{r}_{\tau}) - \Phi^{o} - \Phi^{r}$

The validity of this equation of state is within the range of 216 K $\leq T \leq$ 1100 K and 0 MPa $\leq P \leq$ 800 MPa, with the uncertainty for transcritical CO_2 cycle within the operation temperature of 220 up to 423K and operating pressure between 1 up to 15 MPa are as follow:

- ± 0.05% for density
- ± 1.5% for specific heat capacity
- ± 1.5% for enthalpy
- ± 1.5% for entropy

The equation of state for transport properties from Vesovic can be expressed as composed of three parts. The first part is from a contribution of transport properties at the region close to zero density X°(ρ,T). The second part is from a contribution of critical enhancement at the vicinity of the critical point $\Delta X_c(\rho,T)$. And the third one ΔX (ρ, T) is a contribution of all effects to the transport properties outside the zero density region and critical region. The Vesovic equation is as follow:

$$X(T,\rho) = X^{o}(T,\rho) + \Delta X(T,\rho) + \Delta X_c(T,\rho) \tag{2.2}$$

The uncertainty of Vesovic equation depends upon a range of temperature and pressure. For a temperature range commonly encounter in refrigeration cycle of CO_2 of 220 up to 423K and a pressure range of 1 up to 15 MPa, the uncertainties of thermophysical properties of CO_2 are:

- ± 5% for thermal conductivity
- ± 5% for viscosity

The Vesovic equation of state for transport properties valid for pressure range up to 100 MPa and temperature ranges of 200-1500K and 200-1000K for viscosity and thermal conductivity, respectively.

A computer code has been developed in this work to calculate thermodynamic properties and transport properties of CO_2 based on these two equations of state. Because both equations use density and temperature as its input, while in computation normally temperature and pressure are used as inputs, the computer code must contain internal iteration so that various combination inputs can be used such as entropy and pressure for example. Some of important functions are listed in Table 2-2.

Figure 2-1 to Figure 2-5 below show some thermodynamic and transport properties of CO_2 calculated by the computer code developed in this work. From Figure 2-2 to Figure 2-5 it is clearly seen that the properties change strongly between 20°C and 50°C when pressure getting close to the critical point.

Table 2-2 Some thermodynamic properties functions

Function Name	Remark	Input parameters	Output parameters
V_tp	Calculate specific volume	T,P	v (kg/m^3)
H_tp	Calculate specific enthalpy	T,P	h (J/kg)
Cp_tp	Calculate specific heat capacity	T,P	Cp (J/kgK)
S_tp	Calculate specific entropy	T,P	s (J/kgK)
T_hp	Calculate temperature	h,P	T (K)
Dv_tp	Calculate dynamic viscosity	T,P	μ (Pa.s)
Tc_tp	Calculate thermal conductivity	T,P	k (W/mK)

More information about the computer code can be found in Appendix A.

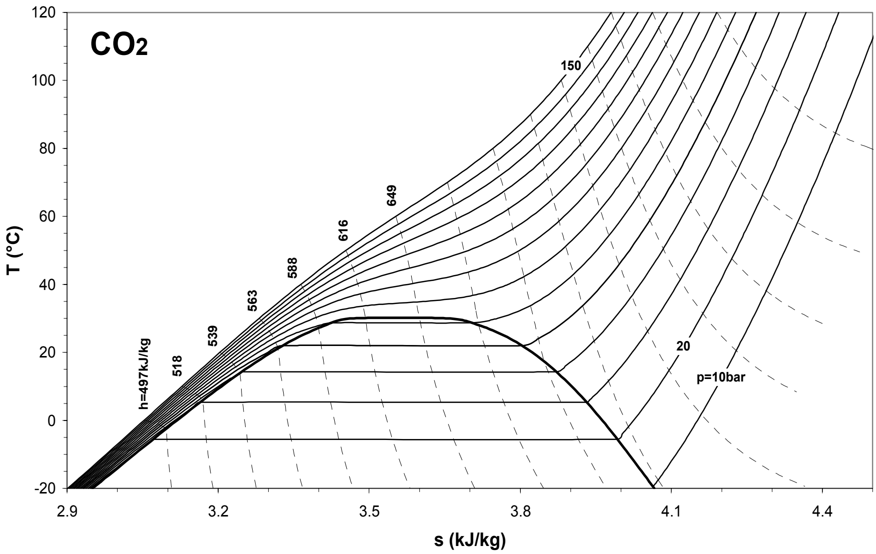


Figure 2-1 T-s Diagram of CO_2

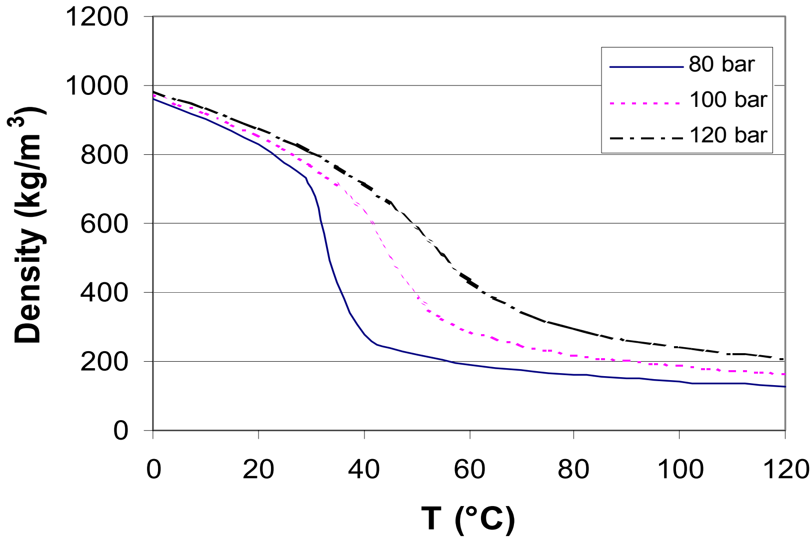


Figure 2-2 Density of CO_2 at various pressures

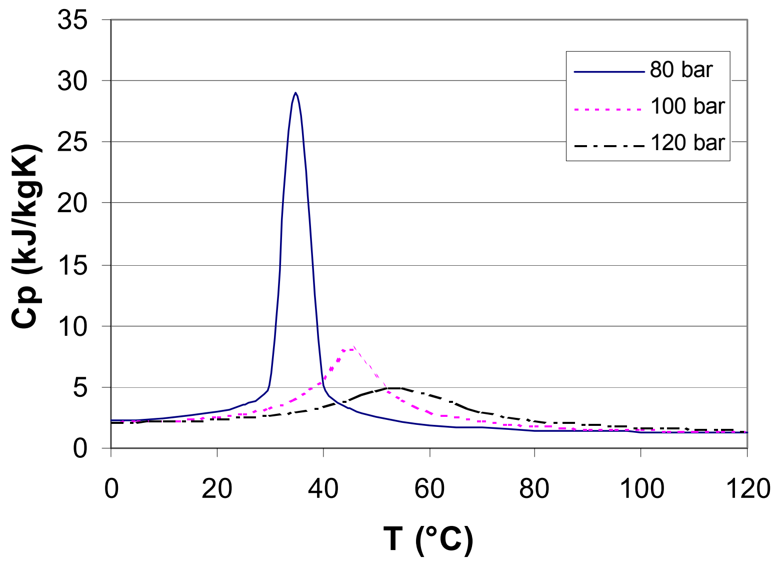


Figure 2-3 Specific heat capacity of CO_2 at various pressures

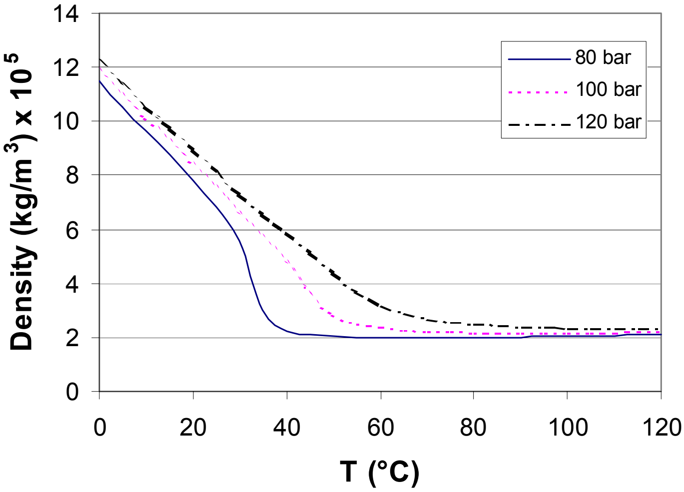


Figure 2-4 Dynamic viscosity of CO_2 at various pressures

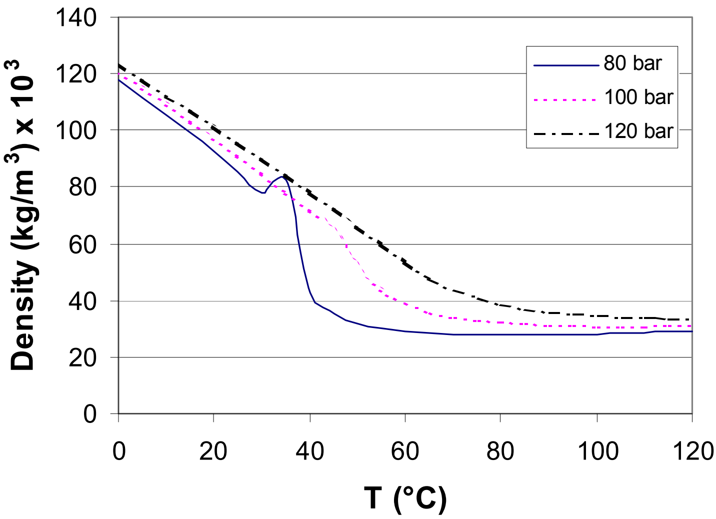


Figure 2-5 Specific thermal conductivity of CO_2 at various pressures

2.2 Transcritical CO_2 Cycle

Refrigeration system using CO_2 was commonly applied in marine sector. At that time, this machine was operated as subcritical cycle. There had been operating problem with this system when the ship was passing through hot water temperature where its cooling capacity drops rapidly (Lorentzen, 1995). To increase the cooling capacity, some additional CO_2 had to be charged into the system and then discharged when air temperature has decreased, which of course was not a good practice from operational practice point of view. This problem has been solved by the invention of Prof. Gustav Lorentzen who suggest transcritical cycle in place of subcritical cycle which make possible to operate the transcritical cycle like subcritical cycle without a need of charging and discharging CO_2 manually.

As expressed in its name, transcritical cycle operates in two-pressure level as in conventional cycle but the high side pressure is above its critical pressure. In this high-pressure side, heat rejection takes place in single-phase region where temperature and pressure becomes independent properties. While in subcritical cycle heat rejection takes place through condensation, in transcritical cycle this process is a sensible cooling which characterized by large gliding temperature in refrigerant side. That why an appropriate name for the heat rejection device of a transcritical cycle is gas cooler instead of condenser. Figure 2-6 shows flow diagram of a transcritical cycle along with its main component. The corresponding thermodynamic cycle on P-h diagram is shown in Figure 2-7.

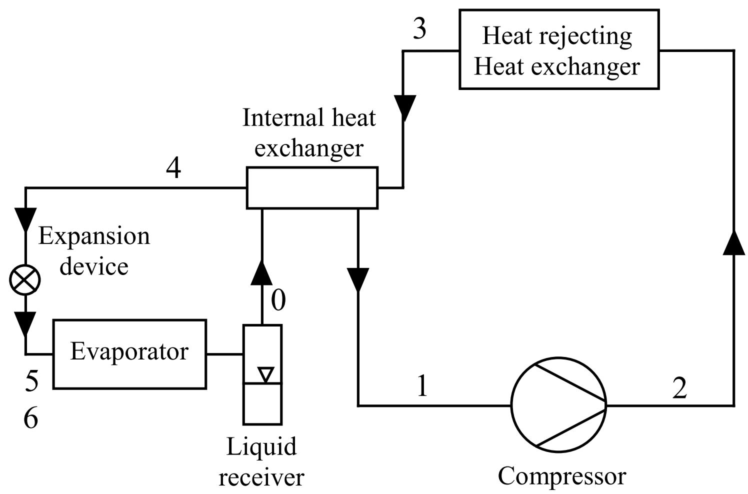


Figure 2-6 Flow diagram of a transcritical cycle

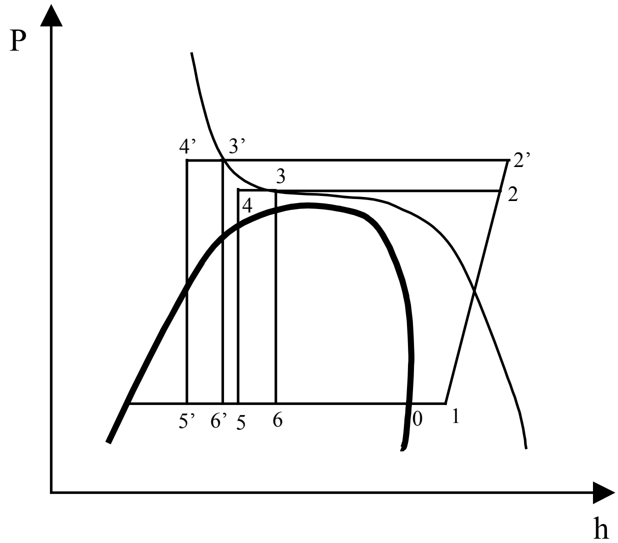


Figure 2-7 Transcritical cycle on P-h diagram

When cooling medium temperature is close to the critical temperature, vapor compression system using CO_2 should be operated at which heat is rejected above the critical pressure in order to get a higher cooling capacity. As shown in Figure 2-7, specific refrigerating capacity can be increase by increasing high side pressure from 2 to 2' where specific enthalpy at the outlet of heat rejection device reduce from 3 to 3' (or 6 to 6'). The change in cooling capacity become more pronounce when temperature at the cold end of the gas cooler is very close to critical region due to very flat isothermal line in P-h diagram. This control capacity through pressure regulation is a unique characteristic of transcritical cycle which can not be applied to subcritical cycle since its high pressure side is dictated by cooling medium temperature.

Figure 2-7 also shows that an increase in specific cooling capacity (from (h_0-h_4) or (h_1-h_3) to (h_0-$h_{4'}$) or (h_1-$h_{3'}$)) when high side pressure is increased from 2 to 2' is accompanied by an increase in specific compressor power consumption (from (h_2-h_1) to ($h_{2'}$-h_1)). However, the slope of isentropic line is almost constant while the slope of isothermal line changes as pressure changes. At a certain range, the slope of isentropic line is higher then that of the isothermal line, resulting an increase in coefficient of performance. As long as the slope of the isentropic line is larger than that of the isothermal line, the coefficient of performance will increase by changing the pressure. Increasing the pressure further will yield in a decrease in the coefficient of performance. Hence, there is an optimum pressure that gives the highest coefficient of performance.

Variation of specific cooling capacity, specific compressor power consumption, and coefficient of performance at various high side pressure is depicted in Figure 2-8.

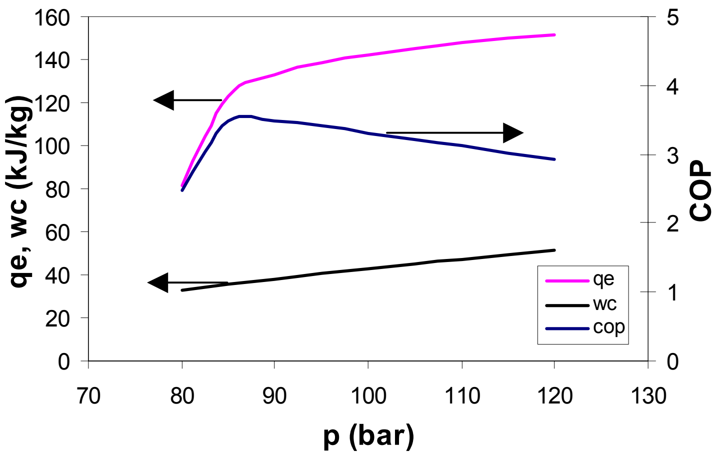


Figure 2-8 Performance variations at various high pressures
[T_{evap} = 0°C, T_{sink} = 30°C]

Owing to different shape of the isothermal lines, the optimum pressure will vary depended on the temperature at the end of gas cooling process. The optimum pressure will shift to a higher value when the temperature becomes higher as shown in Figure 2-9 below. As in subcritical cycle, the coefficient of performance decrease with increasing cooling medium temperature.

2.3 Air-Conditioning System

One of several promising applications of CO_2 transcritical cycle is for air conditioning system. In this application it is very important to design gas cooler which will keep approach temperature (temperature difference between CO_2 temperature and cooling medium temperature at the cold end of the gas cooler) as low as possible since rejected heat is not utilized and just dissipated to ambient. Practical approach temperature that can be achieved in air-cooled gas cooler was within 1-3 K (Pettersen et al., 1998) as long as pinch temperature (the smallest temperature different within a heat exchanger) occurs at the cold end of the gas cooler.

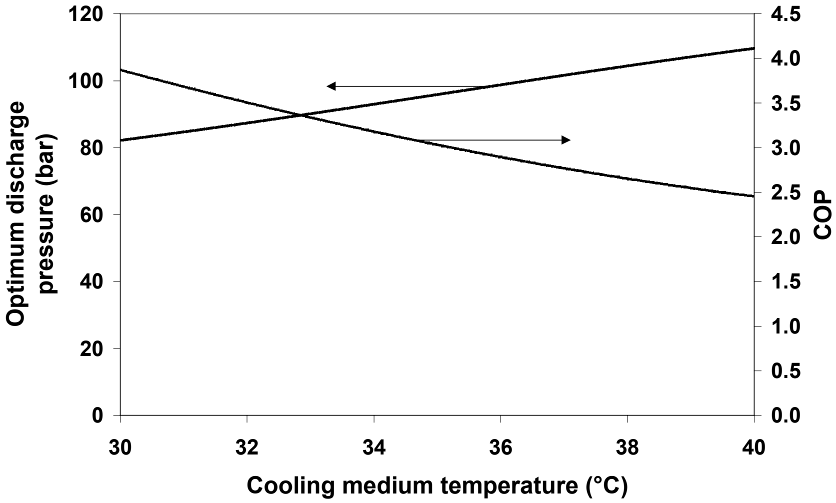


Figure 2-9 Effect of outlet temperature of gas cooling on optimum pressure and COP [T_{evap} = 0°C, T_{app} = 3 K, η_{isen} = 100%]

In subcritical cycle the approach temperature is rarely designed to fall within that small figure. Instead, it is designed based upon pinch temperature that occurs inside the condenser after desuperheating is complete, as shown in Figure 2-10. With commonly degree of subcooling around 5 K, the approach temperature for the condenser become higher within 10-15K depend on cooling medium temperature.

The different in approach temperature of transcritical cycle with that of subcritical cycle is one of the reasons why simple calculation is not adequate to explore the potential of CO_2 cycle compared to the subcritical cycle such as R22. For example, at 35°C ambient air temperature and 0°C evaporation temperature, a simple cycle calculation would give COP of 3.54 and 6.54 for CO_2 and R22 respectively, a 45.8% different.

In order to perform a realistic comparison between transcritical and subcritical cycle some of these factors should be taken into consideration, which for transcritical cycle:

- Compression process more efficient due to a lower pressure ratio.
- Smaller approach temperature in the gas cooler can be achieved.
- Higher heat transfer coefficient during gas cooling process.

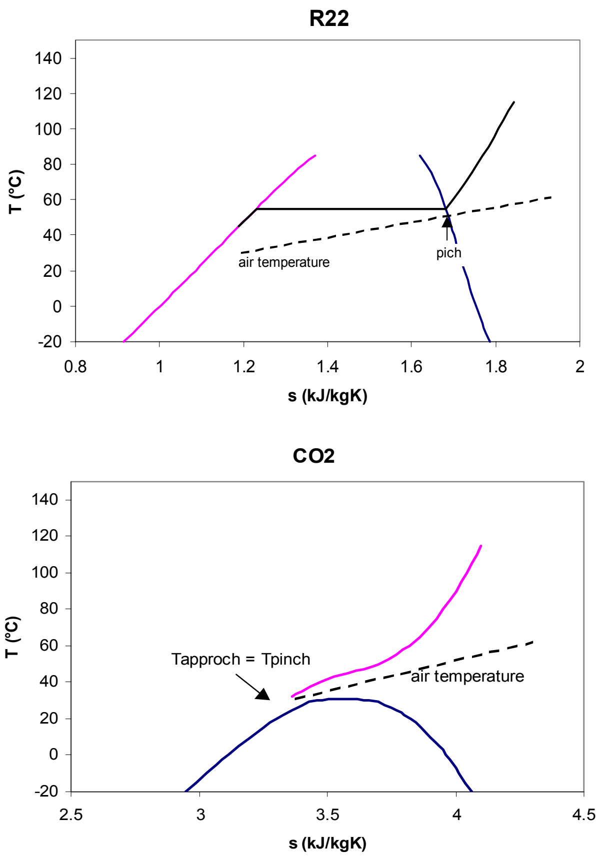


Figure 2-10 Location of pinch temperature in condenser and gas coolers

If all these factor are included in a more detail cycle calculation the different in performance between transcritical cycle and subcritical cycle would not that much as what will be obtained from a simple cycle calculation.

Both theoretical and experimental investigation of CO_2 system for residential air-conditioning application is still very rare. One of publication papers, which investigate experimentally the use of CO_2 in air-conditioning, was from Aarlien and Frivik. This work had been a performance comparison between CO2 and R22 cycle on a ductless air conditioning system. The experiment showed that at cooling mode, CO_2 system run at lower performance compared to R22 system. Depend on operating conditions, cooling COP of CO_2 system was 0.5% and 14.5% lower than that of R22 system at 25°C and 45°C outdoor air temperature, respectively (Aarlien et al., 1998).

It should be note here that on this experiment the main components of the system (compressor and heat exchangers) have been designed for automotive air-conditioning system and not for residential one. Some reasons to the lower performance of CO_2 system were low compressor efficiencies, lower evaporation temperature compared to R22 system and a poor temperature approach of the gas cooler. These facts gave some indication toward a better system performance if the CO_2 system can be designed properly.

2.4 Water-Heating System

Heat rejection that occurs in single-phase region is an ideal condition for water heating process with large temperature lift. As can be seen from T-s diagram in Figure 2-11, gas cooling process occurring in supercritical region will follow isobar line with decreasing temperature monotonously. If the energy released during this gas cooling process was used for water heating, it is possible to obtain high hot water temperature, which is difficult to be achieved in a subcritical cycle. Hot water temperature up to 90°C can be achieved without any operating problem (Nekså et al., 1998). Counter flow heat exchanger is an obvious choice for this purpose since it will give the highest effectiveness compared to other flow types.

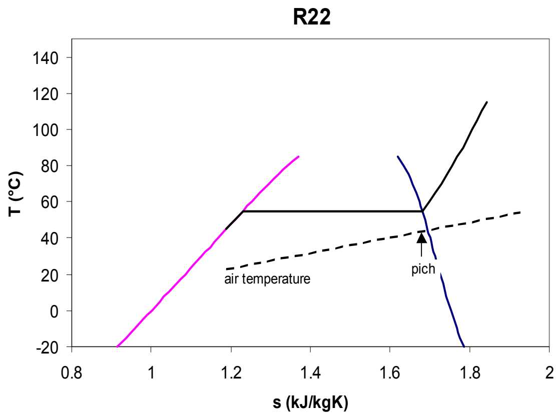
Experimental result obtained from a prototype hot water heat pump system using CO_2 as working fluid showed high system performances. At 0°C evaporation temperature and 7°C inlet water temperature, heating-COP of 4 can be achieved for 60°C hot water temperature (Nekså et al., 1998). Assuming compressor efficiency of 0.7 for R22 system with 5 K subcooling, it will give heating-COP about 3.5 at the same condition.

Figure 2-11 shows a comparison of water heating process from 10°C to 60°C between CO_2 system and R22 system. As in air-conditioning case, pinch temperature occurs inside the condenser of R22 system, which leads to a higher approach temperature, while in CO_2 system it occurs at the cold end of gas cooler and 2K approach temperature can be achieved easily.

2.5 Combined Air-Conditioning/Water-Heating System

Most of commercial buildings in tropical area need cooling and heating system year around. Cooling is needed to provide comfort for the occupant while heating is needed to produce hot water for various purposes. In some buildings such as hotels, energy consumption for air-conditioning system takes the largest part of total energy demand for operational activity while that for hot water is in a second place. Figure 2-12 shows a typical breakdown of energy consumption of a hotel in a hot climate area (Houghton, 1997).

As can be seen from this figure, around 44% of the total energy consumed by air-conditioning system while 19% of it consumed by hot water production system. It means that there is more than enough energy from the air-conditioning system that can be recovered to heat water.



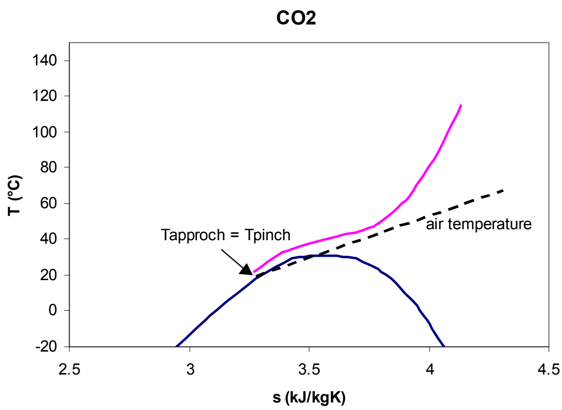


Figure 2-11 Comparison of water heating process in CO_2 system and R22 system.

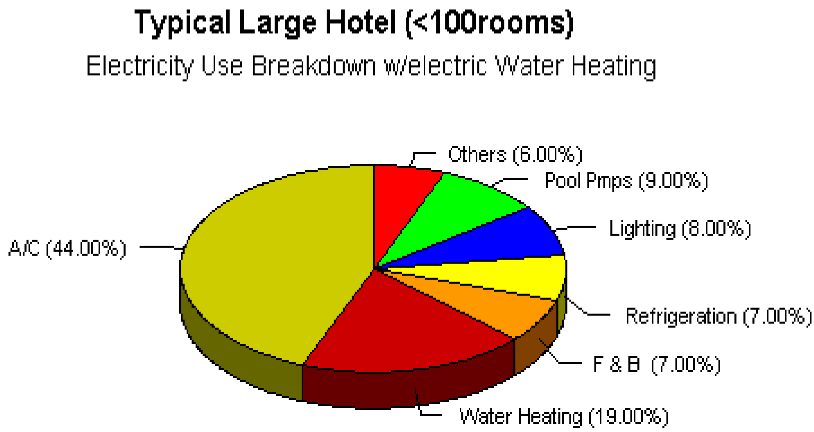


Figure 2-12 Typical energy consumption breakdowns in large hotels

2.5.1 Subcritical system

All vapor compression system used for cooling system in buildings is of subcritical cycle type. If heat recovery is being applied in these buildings for hot water production, usually a desuperheater is coupled into the refrigeration system as shown in Figure 2-13. This desuperheater can recover about 20% of total rejected heat (Olszewski, 1984). It also improves the refrigeration system performance due to the capacity of the heat rejecting device become larger.

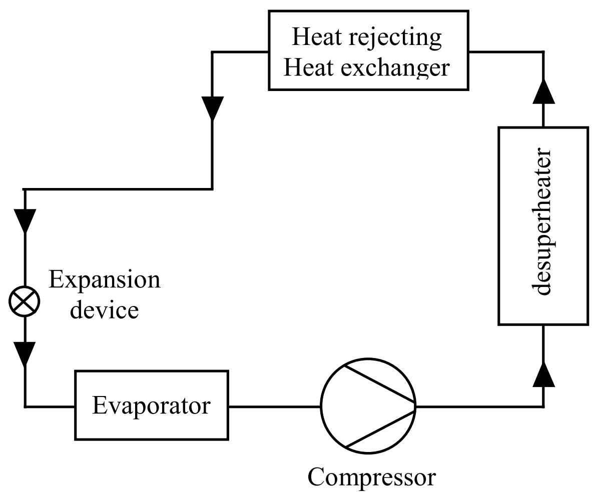


Figure 2-13 Heat recovery system in subcritical refrigeration cycle.

The principle of heat recovery system can be described as in Figure 2-13. Hot gas refrigerant is passed through the desuperheater in which hot water is being produced. The size of the desuperheater is usually designed just large enough to capture sensible heat of refrigerant so that the refrigerant state before entering the condenser is about at saturated vapor.

For a small capacity, the heat recovery system will not affect the cooling capacity and compressor power significantly. This is because usually the expansion device is of fixed flow area type and applying heat recovery will reduce pressure differential across the expansion valve resulting in a lower refrigerant flow rate. The cooling capacity will increase slightly while compressor power consumption will decrease slightly and the overall effect is a higher cooling-COP (Bong, 1988).

2.5.2 Transcritical system

Because of its excellent performance in hot water heat pump system while rather inferior in air-conditioning system compared to subcritical cycle, transcritical cycle would be an interesting option to be implemented in areas where there is a need for cooling and heating simultaneously. This combined system of air-conditioning and water heating offers both saving energy consumption for producing hot water and also improving performance of the air-conditioning side.

As has been stated before, approach temperature at the cold end of the gas cooler in a transcritical system plays an important role, which dictates the system performance. This approach temperature can be adjusted by:

- regulating high side pressure through adjusting of the expansion valve.
- regulating refrigerant or cooling medium mass flow rates.
- changing heat sink temperature.

By increasing high side pressure, the gradient of isobar line around the critical region will increase and the pinch temperature will move toward the cold end of the gas cooler thereby lowering the refrigerant temperature at the outlet of the gas cooler. This means an increase in specific refrigeration capacity.

Regulating mass flow rates of refrigerant or cooling medium will affect approach temperature. For constant refrigerant mass flow rates, increasing mass flow rates of cooling medium will reduce refrigerant temperature out of the gas cooler. If rejected heat is not utilized and just dissipated to the cooling medium, the temperature out of the gas cooler can be reduce as low as possible, hence reducing the approach temperature.

If there are different temperature level of heat sinks, such as in a building that needs simultaneously cooling and heating, the use of a lower heat sink temperature obviously will be an advantage. Instead of using a separated system for cooling and heating, heating can be provided from the cooling system by recovering part of rejected heat from the cooling system. This is an ideal situation for transcritical cycle since cooling process takes place in a relatively constant heat source temperature while heat rejection process is performed in a large gliding temperature.

In case where air is used as cooling medium and hot water is needed, there will be two temperature levels if both of them are utilized as heat sinks of a refrigeration system since normally ground water temperature is lower than air temperature in a tropical region. In a combined air-conditioning and water-heating system which uses air as the primary heat sink, an additional gas cooler is needed to transfer rejected heat to water, in addition to an air-cooled gas cooler. There can be two possible arrangements of these gas coolers, series and parallel. Figure 2-14 shows a schematic ideal system for a combination of cooling and heating purposes with parallel gas coolers configuration.

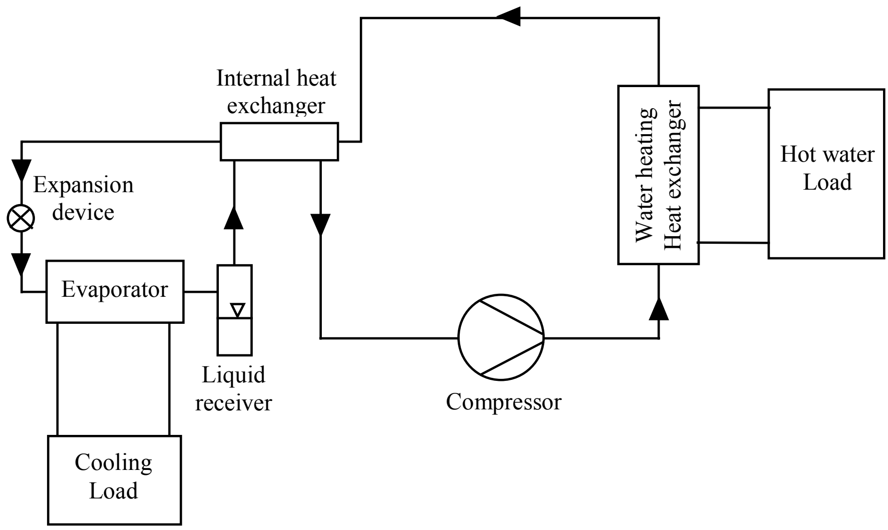
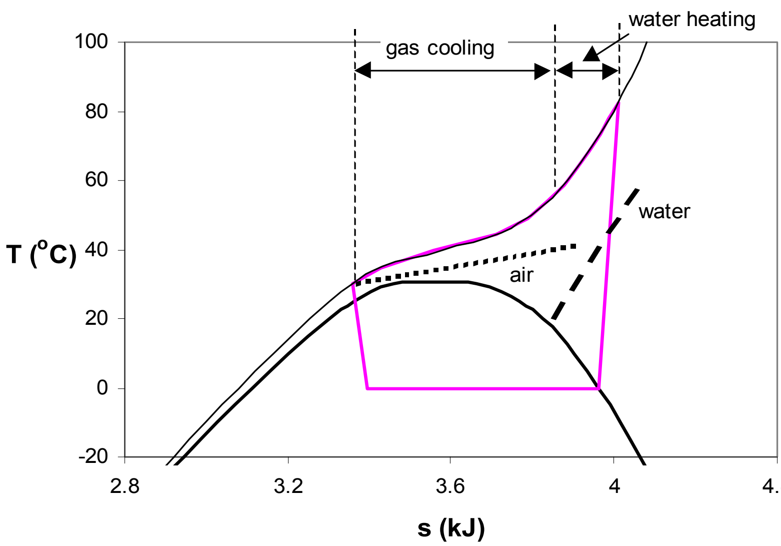


Figure 2-14 A system for simultaneous air-conditioning and water heating using CO_2 as working fluid

In a series arrangement, the additional gas cooler (a water heating heat exchanger) is placed between compressor and air-cooled gas cooler (a heat rejecting heat exchanger). This system acts in similar way as subcritical heat recovery system except the system performance now is dictated by ambient air temperature as long as the outlet temperature of refrigerant from the additional gas cooler is higher than the air temperature. As energy consumption for water-heating is normally less than total rejected heat from the air-conditioning system, the refrigerant temperature out of the additional gas cooler will always be higher than ambient air temperature and consequently the optimum pressure is determined by ambient air temperature. In such a system, the increase of system performance will be minute by only a few percent higher than the air-conditioning performance without heat recovery. As for example, at 30°C air temperature and 3K design approach temperature of the gas cooler, this approach temperature can be reduced to 0.1K giving an increase in coefficient of performance about 7.5%. The cycle in T-S diagram is given in Figure 2-15 and its flow diagram is given in Figure 2-16.



Figurse 2-15 T-s diagram of a combined system with series gas coolers arrangement

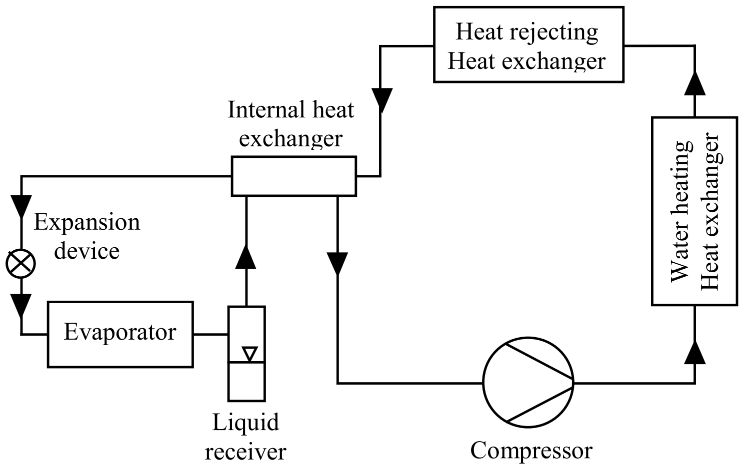


Figure 2-16 Flow diagram of a combined system with series gas cooler arrangement

Another possibility of placement of water heating heat exchanger is in parallel with the heat rejecting heat exchanger. In this configuration, refrigerant flow is divided into two passes, one into the heat rejecting heat exchanger and the other into the water heating heat exchanger. Figure 2-17 shows T-s diagram of a combined system with parallel gas cooler configuration and the layout of this system is shown in Figure 2-18.

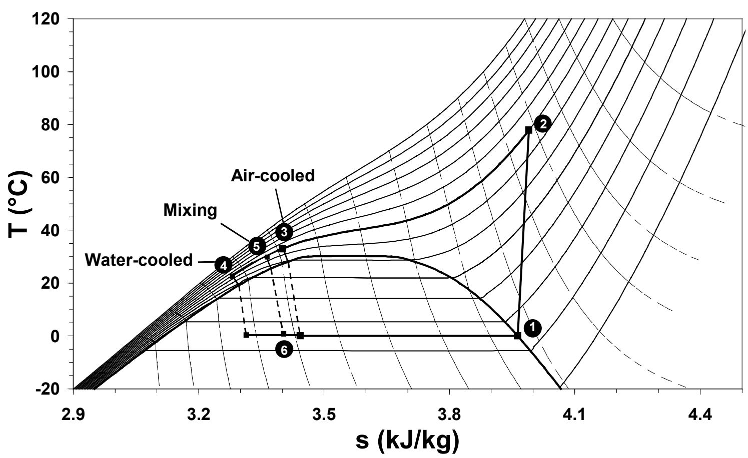


Figure 2-17 T-s diagram of a combined system with parallel gas cooler arrangement

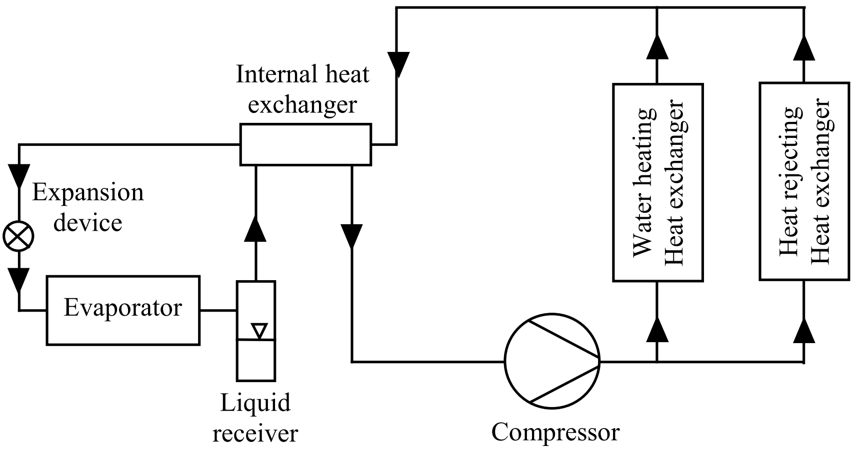


Figure 2-18 Flow diagram of a combined system with parallel gas cooler arrangement.

This arrangement gives a flexibility in term of capacity control. While in series arrangement the capacity is fixed for an operating condition, in parallel arrangement the capacity can be control by regulating the distribution of refrigerant flowing to both gas coolers.

Another advantage of parallel configuration is a higher improvement of the air-conditioning performance can be achieved. CO_2 temperature leaving the gas coolers system will become lower as percentage of heat recovery increases in case inlet water temperature is lower than ambient air temperature. As this temperature become lower, the air-conditioning performance becomes higher.

Despite a minor enhancement of the air-conditioning performance, there is an advantage of the series arrangement. The heat transfer area of the additional gas cooler required for recovering a portion of total rejected heat is smaller in series arrangement than in parallel arrangement. Since the entire refrigerant mass flow passes through the additional gas cooler in series arrangement, the required heat transfer area is smaller for the same hot water load.

2.6 Principle of exergy analysis of subcritical and transcritical cycles

This section explains the principle of exergy analysis that is used in the discussion part of the later chapter. Since both subcritical and transcritical cyles are compossed of several components, the analysis will be started from each components. The analysis of the cycles can be performed starting from the exergy inputs to the cycle until the exergy output from the cycle. After all exergy balances are calculated, the performance of the cycles can be compared.

Exergy of a steady stream of matter is defined as the maximum amount of work obtainable when the stream is brought from its initial state to the dead state by processes during which the stream may interact only with the environment (Kotas, 1995). The exergy of a substance can be determined by the following relation:

$$E = H - T_o S \tag{2.3}$$

For a process in a system, the exergy balance can be derived by combining the first law of thermodynamic (energy balance) with the second law of thermodynamic. The following equation expresses the exergy balance for an open system, steady flow case, control region (Kotas, 1995):

$$\dot{W}_x = \sum_{IN}(\varepsilon_i \dot{m}_i) - \sum_{OUT}(\varepsilon_e \dot{m}_e) + \sum_{r}(\dot{Q}_r \frac{T_r - T_o}{T_r}) - \dot{I} \tag{2.4}$$

where: $\dot{W}_x$ is shaft work defined as positif if work is transferred from the control region. $\dot{Q}_r$ is thermal energy reservoirs at reservoir temperature of T_r and defined as positif when heat is transferred to the control region. **ε** is spesific exergy, and $\dot{m}$ is mass flow rates. $\dot{I}$ is the expression for irreversibility rate for the control region. The irreversibilty rate can also be calculated as follows (Kotas, 1995):

$$\dot{I} = To\left[\sum_{OUT}(s_e \dot{m}_e) - \sum_{IN}(s_i \dot{m}_i) - \sum_{OUT}\frac{\dot{Q}_r}{T_r}\right] \tag{2.5}$$

From Eq. (2.4), the exergy balance and the irreversibility of each processes involved in the cycles can be derived.

To visualize exergy flows and exergy losses in a control region, Grassmann diagram is normally used. The diagram can show clearly how the exergy transferred from one component to the other components and what the magnitude of the exergy losses in each components. The following section describes exergy analysis of components in a refrigeration plant.

2.6.1 Compression process

The exergy of refrigerant increases in a compression process. Shaft work as exergy input is converted to exergy of refrigerant and part of it lost during the process. The control region, Grassmann diagram, and the process on T-s diagram for a single stage adiabatic compression process can be seen in Figure (2.19).

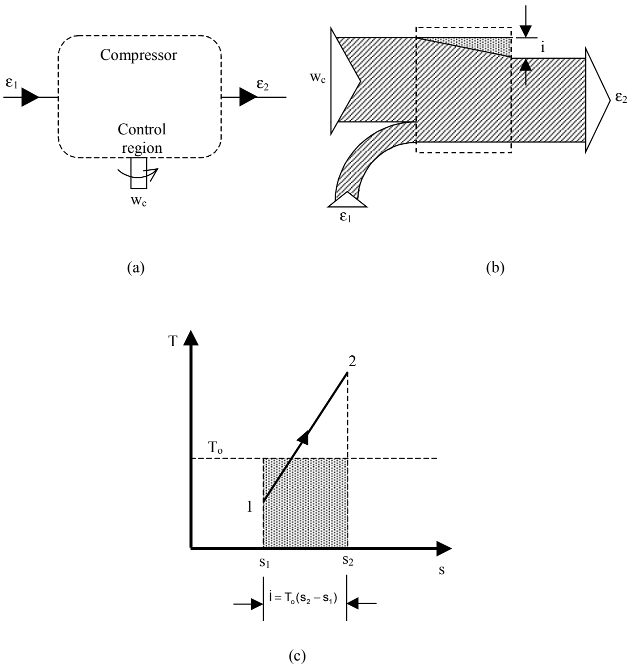


Figure 2-19 Compression process (a) control region
(b) Grassmann diagram (c) T-s diagram

Assuming single stage adiabatic compression process, the following exergy balance equation can be derived from Eq. (2.4):

$$\varepsilon_1 + w_c = \varepsilon_2 + i \tag{2.6}$$

And the irreversibility of the process can be calculated from Eq. (2.5), which becomes:

$$i = T_o(s_2 - s_1) \tag{2.7}$$

2.6.2 Heat transfer process

There are two kind of heat transfer that can be found in relation to exergy flows, the first is when heat is rejected from a stream to environment and the second is when heat is transferred from a stream to another stream. When heat of a stream is rejecting to the environment, the exergy transferred to the environment becomes zero. This heat transfer process can be found in a condenser of a subcritical cycle or in a gas cooler of a transcritical cycle. Control region and Grassmann diagram for heat transfer process can be seen in Figure 2-20.

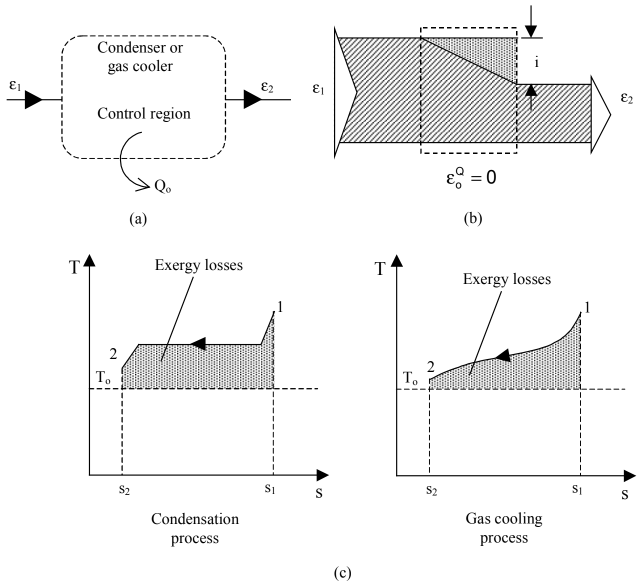


Figure 2-20 Heat transfer process in a condenser and gas cooler
(a) control region (b) Grassmann diagram (c) T-s diagramm

The exergy balance for a condenser or gas cooler is as follows:

$$\varepsilon_1 = \varepsilon_o^Q + \varepsilon_2 + i \tag{2.8}$$

where $\varepsilon_o^Q = 0$ since the exergy change of the stream is transferred to environment. The exergy losses can be calculated from Eq. (2.8) and the following relation can be obtained.

$$i = \varepsilon_1 - \varepsilon_2 = h_1 - h_2 - T_o(s_1 - s_2) \tag{2.9}$$

When heat is transferred from one stream to another stream such as in water heating process and in an internal heat exchanger, the exergy balance of such a process is given by the following equation:

$$\varepsilon_1 + \varepsilon_3 = \varepsilon_2 + \varepsilon_4 + i \quad (2.9)$$

and the exergy losses can be calculated from Eq. (2.9) or Eq. (2.5). The control region, Grassmann diagram, and T-s diagram is shown in Figure 2-21.

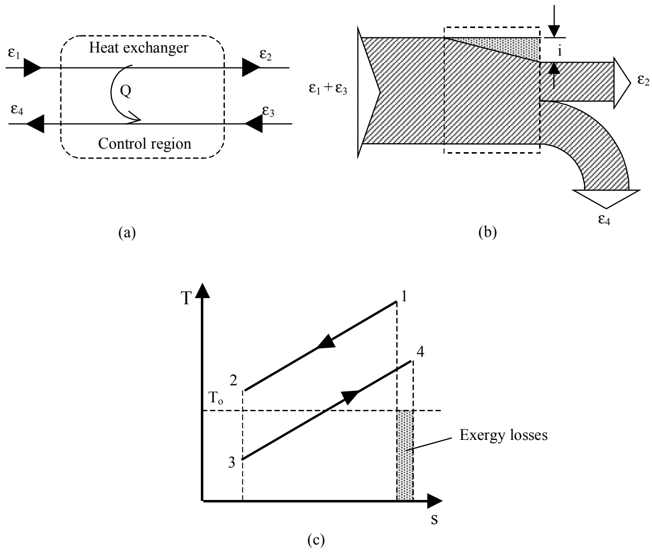


Figure 2-21 Heat transfer process in a heat exchanger (a) control region (b) Grassmann diagram (c) T-s diagram

2.6.3 Expansion process (throttling)

The purpose of expansion process in a vapour compression cycle is to reduce the pressure and temperature. From exergy view point, this process is solely an exergy destruction. If the process is assumed to be adiabatic, the exergy balance for the process becomes:

$$\varepsilon_1 = \varepsilon_2 + i \quad (2.10)$$

and the exergy losses can be calculated from Eq. (2.10) or Eq. (2.5). Figure 2-22 shows the control region and Grasmann diagram for throttling process.

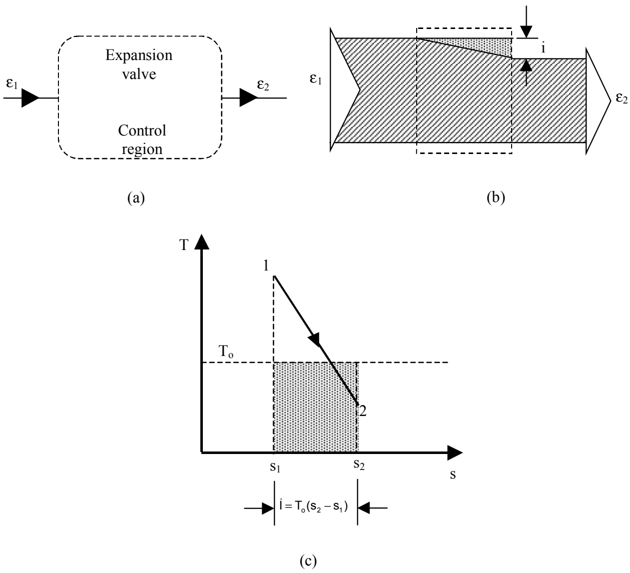


Figure 2-22 Throttling process (a) control region
(b) Grassmann diagram (c) T-s diagram

2.6.4 Evaporation process

There are two kind of heat transfer in an evaporation process regarding the heating medium, one when heat is absorbed by refrigerant from a space and the other when heat is absorbed from a secondary fluid. In evaporation process of vapour compression cycle, normally heat is absorbed in subenvironmental temperature and hence, the flow of energy and exergy is in reversed direction.

When heat is absorbed by refrigerant from a secondary fluid, the exergy balance is the same as Eq. (2.9) and exergy losses calculation can be done from Eq. (2.9) or Eq. (2.5) and the process is similar with Figure 2-21 with constant temperature for process from point 3 to point 4.

When heat is absorbed from a space, the direction of heat flow is from the space to the refrigerant. However, the direction of exergy flow is from the refrigerant to the space. Exergy that is transferred to the space is given by the following equation:

$$\varepsilon_e^Q = q_e \cdot (1 - \frac{T_o}{T_e}) \tag{2.11}$$

The exergy balance is this case is as follows:

$$\varepsilon_1 = \varepsilon_2 + \varepsilon_e^Q + i \tag{2.12}$$

Figure 2-23 below shows the exergy losses in T-s diagram when heat is absorbed from a space. Area **abcd** represents exergy transfer from refrigerant to the space (ε_1-ε_2), area **1b23** represents part of exergy that is received by the space (ε^Q), and the different between area **abcd** and **1b23** is the exergy loss (i).

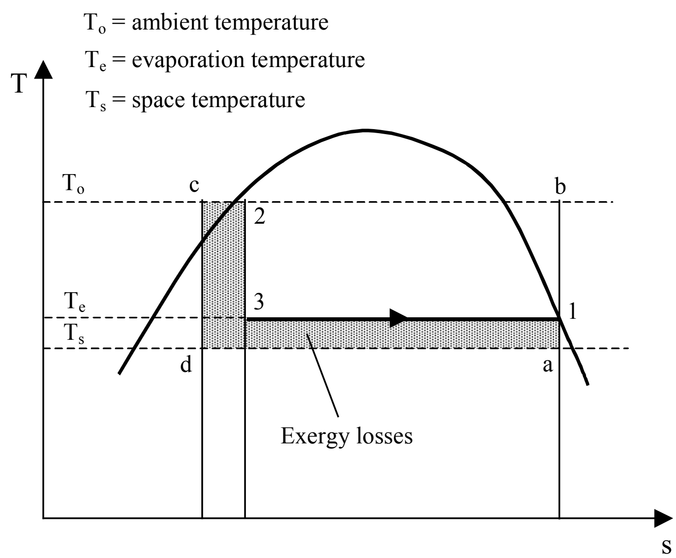


Figure 2-23 Exergy losses in evaporation process with constant cold space temperature

To apply exergy analysis in a vapor compression cycle, the state of refrigerant and other substances involved in the process must be known. When these information are available then the analysis of exergy in each component can be done and the exergetic efficiency of the plant can be determined. The following section will give short description on the exergy analysis of subcritical and transcritical cycle.

2.6.5 Subcritical vapor compression cycle

Figure 2-24 shows flow diagram of a subcritical cycle. Exergy analysis for each components of the plant is the same as explained in the previous section. Exergetic efficiency of the plant is defined as a ratio of desired exergy to input exergy. For a refrigeration plant, the desired exergy is exergy that is transferred to the space or secondary fluid and the input exergy is the electric power that is supplied to the compressor. Eq. (2.13) gives an expression of the exergetic efficiency.

$$\eta_x = \frac{\Delta E_{output}}{\Delta E_{input}} = \frac{\Delta E_{space}}{W_{compressor}} = \frac{\varepsilon_e^Q (1 - T_o / T_e)}{W_{compressor}} \tag{2.13}$$

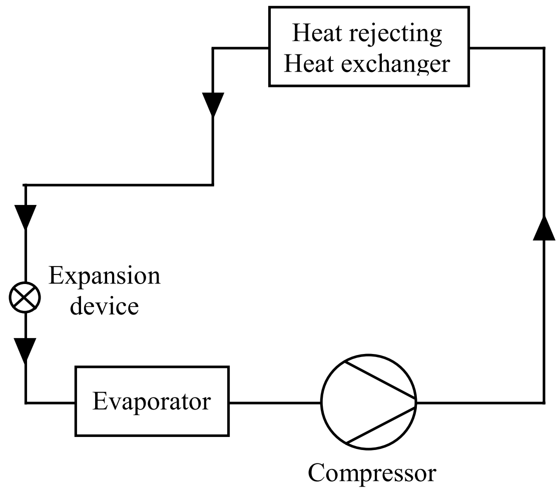


Figure 2-24 Flow diagram of subcritical cylce

To compare two different cycles, the exergy output of the plants can be taken as the exergy transferred from refrigerant, so that the heating medium temperature is not involved in the analysis. In this case, the exergetic efficiency becomes:

$$\eta_x = \frac{\Delta E_{output}}{\Delta E_{input}} = \frac{\Delta E_{evaporator}}{W_{compressor}} = \frac{E_{evap\_in} - E_{evap\_out}}{W_{compressor}} \tag{2.14}$$

Grassmann diagram of the subcritical cycle, which is independent of heating medium temperature, is shown in Figure 2-25.

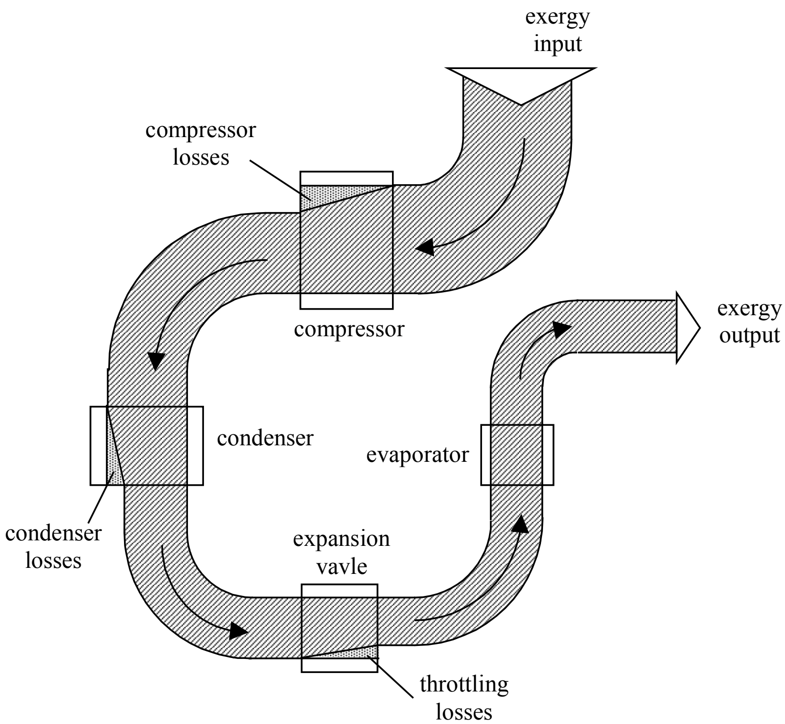


Figure 2-25 Exergy balance for the subcritical refrigeration plant

2.6.6 Transcritical cycle

The definition of the exergetic efficiency and the exergetic analysis of transcritical refrigeration cycle is the same as that of the subcritical cycle. The different is that in transcritical cycle shown in Figure 2-26, there are two additional components, internal heat exchanger and liquid receiver. Figure 2-26 and Figure 2-27 show flow diagram and Grassmann diagram of the transciritical cycle. The exergy analysis of the combined air-conditioning and water heating will be given in the discussion chapter.

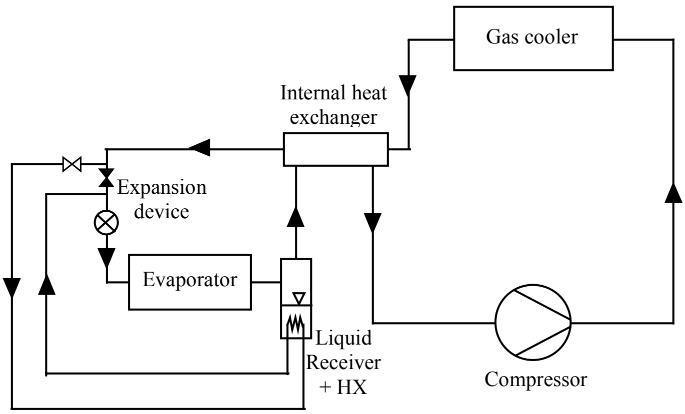


Figure 2-26 Flow diagram of transcritical cylce

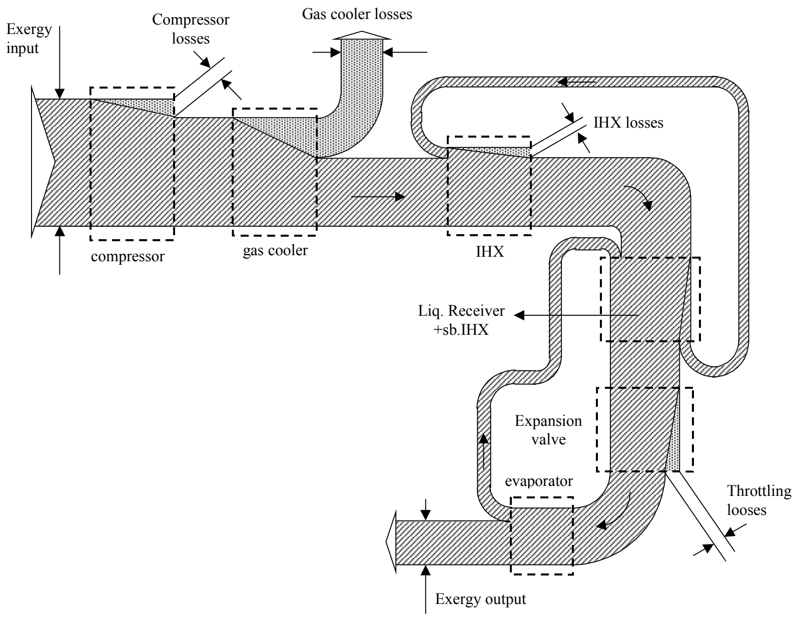


Figure 2-27 Exergy balance for the transcritical refrigeration plant

2.6.7 Combined Transcritical Air-conditioning and Water Heating Cycle

There are two exergy output in a a combined air-conditioning and water heating system, exergy transferred to evaporator and exergy transferred to water being heated. The definition of exergetic efficiency of a combined air-conditioning and water heating system becomes:

$$\eta_x = \frac{\Delta E_{output}}{\Delta E_{input}} = \frac{\Delta E_{evaporator} + \Delta E_{hotwater}}{W_{compressor}} = \frac{(E_{evap\_in} - E_{evap\_out}) + (E_{hotwater\_out} - E_{hotwater\_in})}{W_{compressor}} \quad (2.15)$$

The flow diagram and Grassmann diagram are given in Figure 2-28 and Figure 2-29.

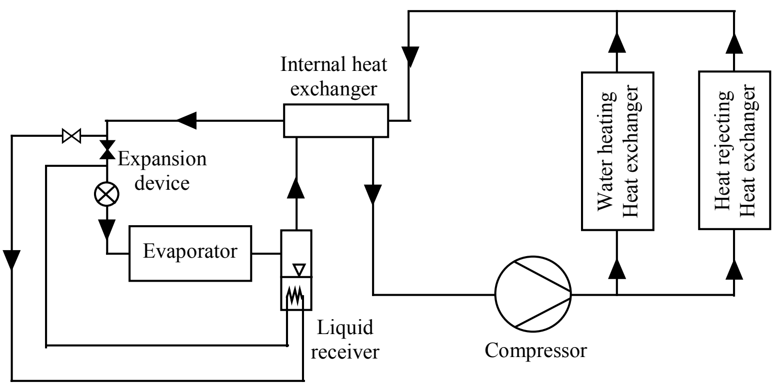


Figure 2-28 Flow diagram of a combined cycle with parallel heat rejecting devices

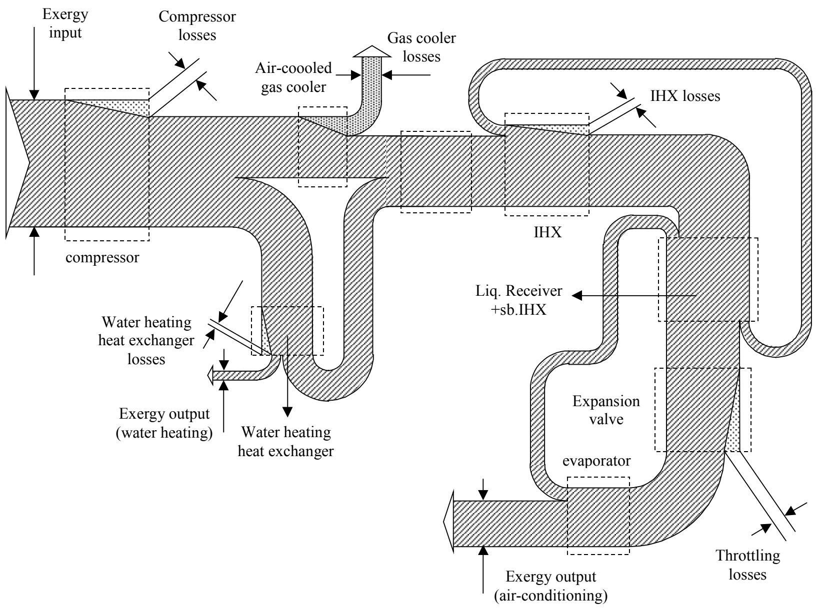


Figure 2-29 Exergy balance for the combined A/C and water heating system

3 Steady State Modeling Of The Combined Air-Conditioning/ Water-Heating System

In this work, a computer modeling of the combined system consist of a compressor, two gas coolers (water heating and heat rejecting heat exchangers), an internal heat exchanger, an expansion valve, and an evaporator will be developed. As the main objectives is to study the system characteristic in a wide range of the heat source temperature, evaporation temperature was chosen as a parameter by regulating heat source from electric heater. In such condition it is not necessary to model the evaporator since the evaporation temperature is constant, making the system modeling become simpler. Moreover, the integrated liquid receiver/submerged heat exchanger were not modeled in this work since the evaporator capacity was determined by the energy balance around the compressor and the gas coolers system. It was assumed that no heat loss from the evaporator, the internal heat exchanger and the integrated liquid receiver/submerged heat exchanger to the surrounding.

The other components that have been modeled are:

- compressor
- water heating heat exchanger
- heat rejecting heat exchanger
- internal heat exchanger

Schematic diagram of the combined system is shown in Figure 3-1. The principle operation of the system can be explained as followed. Starting from the compressor, CO_2 at suction temperature and suction pressure enters the compressor and then is compressed to a discharge pressure in supercritical region. This hot gas at supercritical pressure then is cooled in gas coolers where energy is transferred to the cooling media and the density of CO_2 becomes much higher. To obtain further cooling, the cooled CO_2 then enters an internal heat exchanger where CO_2 in the high-pressure side transfers heat to CO_2 in the low-pressure side ensuring a superheat state before entering the compressor. Before expansion process, CO_2 is cooled further in an integrated liquid receiver/submerged heat exchanger. This liquid like CO_2 then expands through an expansion valve reaching two-phase fluid before entering the evaporator. In evaporator CO_2 evaporates and leaves the evaporator as two-phase fluid due to the process in the integrated heat exchanger-liquid receiver. In the integrated liquid receiver/submerged heat exchanger, the two-phase CO_2 is evaporated by heat from high side pressure CO_2. The gas phase CO_2 is suck by the compressor through the internal heat exchanger while the remaining liquid phase CO_2 stays in the receiver. Then the saturated gas from receiver is heated in the internal heat exchanger by high pressure CO_2 before entering the compressor.

Two gas coolers is needed in heat recovery mode, one as heat rejecting heat exchanger where heat is dissipated to the environment and the other water heating heat exchanger where heat is transferred to water being heated. There are two possible arrangements of these gas coolers, that is series or parallel. In a series arrangement, the hot gas CO_2 transfers its energy to water being heated in the water heating heat exchanger and then enters the heat rejecting heat exchanger where heat from CO_2 is dissipated to the environment. In a parallel arrangement, the CO_2 discharged from the compressor must be split into two streams, one stream goes to the water heating heat exchanger and the other stream goes to the heat rejecting heat exchanger. By regulating the distribution of these streams, the capacity of both gas coolers can be adjusted.

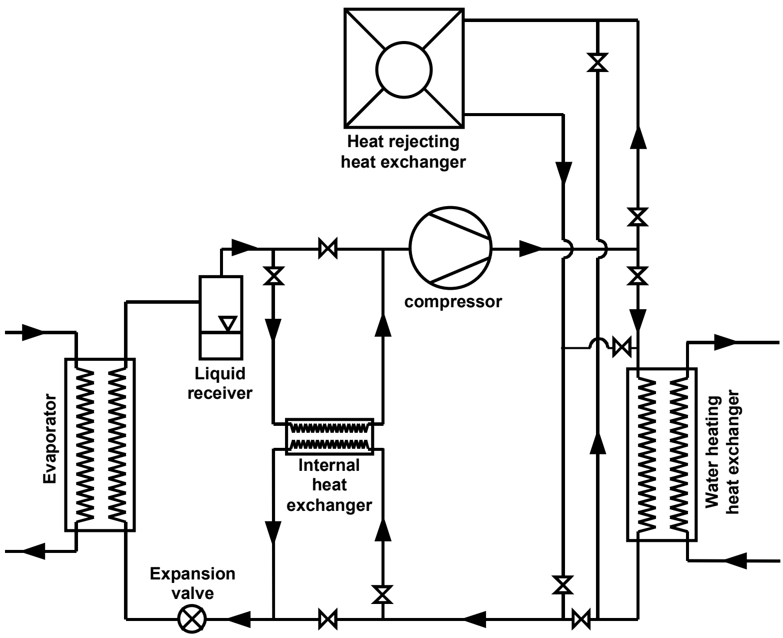


Figure 3-1 Schematic diagram of the combined system

3.1 Compressor Model

The compressor used in this work is of open type reciprocating compressor with one cylinder. The modeling of this compressor was of thermodynamic model type in which its characteristic was described by isentropic efficiency and volumetric efficiency. This model type can be used if experimental data of the compressor are available in which both efficiencies can be mapped into mathematical expressions.

Even the compressor head has been insulated, heat transfer from the compressor to the surrounding could not be avoided completely. It is assumed that 5% of total energy of compression process is dissipated to the surrounding considering the temperature of the compressor head was always higher than that of the surrounding in whole experimental conditions performed in this work.

The isentropic efficiency of the compressor is defined as a ratio of compressor power needed in an isentropic compression process to the shaft power connecting the compressor and the motor. The isentropic efficiency is described in a polynomial form as follow:

$$\eta_{is} = c_0 + c_1 \cdot pr + c_2 \cdot pr^2 + c_3 \cdot pr^3 + c_4 \cdot pr^4 \qquad (3.1)$$

For volumetric efficiency, the equation is as follow:

$$\eta_{vol} = c_0 + c_1 \cdot pr + c_2 \cdot pr^2 + c_3 \cdot pr^3 \qquad (3.2)$$

Based on experimental data of CO_2 heat pump water heater (Nekså, 1998), a set of constants can be found through least square method by fitting of the experimental data. Table 3-1 contains the constants for both efficiency functions and Table 3-2 shows inputs and outputs of the compressor model.

Table 3-1 Constants for isentropic and volumetric efficiency functions

Constant	$\eta$is	ηvol
c_0	0.22633019	0.96265832
c_1	0.60791411	-0.013978911
c_2	-0.20917691	-0.011966819
c_3	0.029189804	0.00095322459
c_4	-0.0014733007	-

Table 3-2 Input and outputs for compressor model

Inputs	Outputs
Suction temperature	Discharge temperature
Suction pressure	Refrigerant mass flow rates
Discharge pressure	Power consumption
Rotational speed	Heat loss

Refrigerant flow rates and compressor power is calculated based on equations (3.3) and equation (3.4):

$$\dot{m}_{co2} = \frac{V_{swept}}{v}\eta_{vol} \tag{3.3}$$

$$P_{comp} = \dot{m}_{co2}(h_{dis} - h_{suct}) \tag{3.4}$$

$$h_{dis} = \frac{h_{dis\_s} - h_{suct}}{\eta_{is}} \tag{3.5}$$

3.2 Water Heating Heat Exchanger Model

Water heating heat exchanger is a component that produces hot water. Considering large temperature change in both water side and refrigerant side it is obvious that a counterflow heat exchanger is an ideal choice. The water heating heat exchanger of the combined system was of helical double-pipe type where CO_2 flows in the inner tube while water flows in the annulus. There was only one pass in this gas cooler so that a high velocity can be achieved resulting in a high heat transfer coefficient. The entire outer part of the gas cooler was insulated and the gas cooler was put into a cylinder to reduce interaction with the surrounding.

Basic assumptions used in modeling the water heating heat exchanger are as followed:

1. there is no heat transfer with the surrounding,
2. refrigerant is pure CO_2 (without lubricant),
3. heat transfer process in one segment of the gas cooler is isobaric,
4. heat is transferred from CO_2 to water as the cooling medium,
5. gas cooler configuration is counterflow and of double-pipe type,
6. gas cooler is twisted to form coiled tube and mounted vertically.

Considering the large variation in the thermophysical properties of CO_2 in supercritical region, it is important to divide the gas cooler model into small heat exchangers. With this segmentation, the thermophysical properties can be considered constant during heat transfer process and a conventional method for calculating the heat transfer process in the gas cooler can be applied. In this study, UA-LMTD method was applied.

Since a constant thermophysical properties can not be achieved especially close to the critical region, the segmentation number should be based upon the accuracy and the calculation time needed to achieve a certain convergence criteria. The accuracy of the calculation for a segmentation number can be determined by comparing the result with the one that is obtained with a smaller segmentation number. If there is a large different between the results than it is necessary to divide the gas cooler into more segments. If the calculation results stay within a small percentage then the number of segmentation is considered as adequate.

For each element volume of the water-cooled gas cooler, the heat balance can be expressed by equation (3.6), (3.7), and (3.8).

$$\dot{Q}_{co2} = \dot{m}_{co2}(h_{co2\_in} - h_{co2\_out}) \tag{3.6}$$

$$\dot{Q}_{w} = \dot{m}_{w}Cp_{w}(T_{w\_out} - T_{w\_in}) \tag{3.7}$$

$$\dot{Q}_{gc} = UA_{i}LMTD \tag{3.8}$$

where LMTD is defined by equation (3.9),

$$LMTD = \frac{(T_{co2\_in} - T_{w\_out}) - (T_{co2\_out} - T_{w\_in})}{\ln\left(\frac{T_{co2\_in} - T_{w\_out}}{T_{co2\_out} - T_{w\_in}}\right)} \tag{3.9}$$

and U is overall heat transfer coefficient based on inside heat transfer area of the inner tube, A_i, and is calculated by equation (3.10).

$$\frac{1}{U} = \frac{1}{h_i} + \frac{A_i \ln\left(r_i / r_o \right)}{2\pi kL} + \frac{A_i}{A_o h_o} \tag{3.10}$$

Table 3-3 shows inputs and outputs for the water heating heat exchanger model.

Table 3-3 Input and outputs for water heating heat exchager

Inputs	**Outputs**
Refrigerant mass flow rates	Capacity
Water mass flow rates	Refrigerant outlet temperature
Refrigerant inlet temperature	Refrigerant outlet pressure
Refrigerant inlet pressure	Water outlet temperature
Water inlet temperature	Water outlet pressure
Water inlet pressure	
Detail geometry	

The condition for heat balance is expressed by equation (3.11):

$$\dot{Q}_{CO2} = \dot{Q}_{w} = \dot{Q}_{gc} \tag{3.11}$$

Heat transfer coefficient for CO_2 at supercritical region is rarely in literatures and to the base of the author knowledge there is no correlation specially developed for coiled tube double-pipe heat exchanger. However, gas cooling process solely involves single-phase fluids so that it can be expected that standard correlation for single-phase heat transfer process can be used with adequate accuracy. There has been some experimental results showing that Gnielinski's correlation can be applied for gas cooling supercritical CO_2 process (Pettersen, 2000). In this model, heat transfer coefficient for water, h_o, as well as for CO_2, h_i, was calculated based on Gnielinski's correlation for single-phase fluid (VDI, 1993):

$$\mathrm{Nu} = \frac{(f/8)\mathrm{Re}\,\mathrm{Pr}}{1.0 + 12.7\sqrt{(f/8)}\mathrm{Pr}^{2/3} - 1}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.14} \tag{3.12}$$

where,

$$f = 0.3164\,\mathrm{Re}^{1/4} + 0.03\sqrt{d/D} \tag{3.13}$$

d is outer tube diamter and D is coil diameter.

For water side, centering string will promote turbulency and hence increases pressure drop and heat transfer coefficient. This effect can be expressed by hydraulic diameter, which was calculated from its definition by taking into account the influence of the centering string.

$$D_h = \frac{4 \cdot \text{heat transfer area}}{\text{wet perimeter}} \tag{3.14}$$

The calculation of heat transfer coefficient was based on the assumption of constant thermophysical properties of CO_2 in a segment of the gas cooler. The thermophysical properties are determined as arithmetic average of the inlet and the outlet state.

Due to unknown state of the fluids in both end of the gas cooler for counterflow type, the calculation has to be performed in an iteration scheme. Before calculation was begun, a guessed temperature profile on both CO_2 side and water side was determined and linear temperature profiles are adequate in this case. From equation (3.6) to (3.9) it can be derived equation (3.15) and (3.16) for the outlet temperature of CO_2 and water (Stoecker, 1989).

$$T_{co2\_out} = T_{co2\_in} - \left(T_{co2\_in} - T_{w\_in}\right)\left(\frac{1 - e^{D_{co2}}}{w_{co2}/w_w - e^{D_{co2}}}\right) \quad (3.15)$$

$$T_{w\_out} = T_{w\_in} - \left(T_{w\_in} - T_{co2\_in}\right)\left(\frac{1 - e^{D_w}}{w_w/w_{co2} - e^{D_w}}\right) \quad (3.16)$$

where:

$$D_{co2} = UA\left(\frac{1}{w_{co2}} - \frac{1}{w_w}\right) \quad (3.17)$$

$$D_w = UA\left(\frac{1}{w_w} - \frac{1}{w_{co2}}\right) \quad (3.18)$$

$$w_{co2} = \dot{m}_{co2} Cp_{co2} \quad (3.19)$$

$$w_w = \dot{m}_w Cp_w \quad (3.20)$$

Starting the calculation from the first segment of the gas cooler, outlet temperatures of both fluids of the first segment become inlet temperature of the second segment, and so on, until the calculation for the last segment was performed so that the new temperature profile was obtained. The next step was the calculation of pressure drop for each element based on the calculated temperature profile.

The calculation process was repeated until the gas cooler load from the previous calculation closely match with the current calculation, which shows that the temperature profile satisfy all heat balance constrained mentioned before. The deviation in the gas cooler load between two calculations must be lower than the specified convergence criteria.

3.3 Heat Rejecting Heat Exchanger Model

A heat rejecting heat exchanger for the combined system in this study was designed with help of a computer program developed at Sintef Energy Research called hXsim (**h**eat e**X**changer **Sim**ulation). The heat exchanger is an air-cooled gas cooler with two circuits tube in fin. Detail geometry of the gas cooler along with inlet condition of CO_2 and air are given as inputs and the program will calculate the rating of the gas cooler along with the outlet condition of CO_2 and air. The principle calculation was the same as the calculation of the water heating heat exchanger model explained before with the possibility to choose different type of heat transfer and pressure drop correlation. It was reported that the agreement between experimental data with simulation data were within 10% (Skaugen, 2000).

Table 3-4 Design condition for the heat rejecting heat exchanger are as followed:

Mass flow rate of CO_2 : 0.167 kg/s	Inlet temperature of CO_2 : 90°C
Inlet pressure of CO_2 : 87 bar	Air velocity : 3 m/s
Inlet temperature of air : 30°C	

Figure 3-2 shows the side view of the heat rejecting heat exchanger from hXsim.

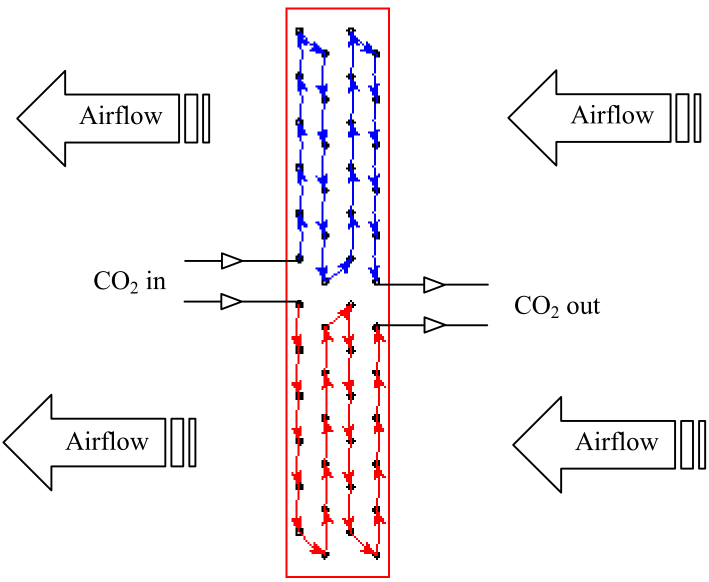


Figure 3-2 Side view of the heat rejecting heat exchanger

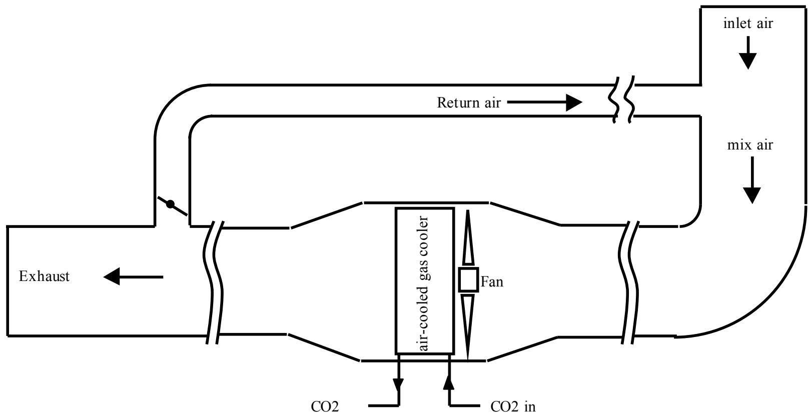
In this study, the air-cooled gas cooler as heat rejecting heat exchanger has been replaced by a water-cooled gas cooler. The reason for this was that by using water as the cooling medium, a wide range of operating conditions can be performed much easy compared to a system with air as the cooling medium. Furthermore the test rig become simpler. Performing experiment with an air-cooled heat exchanger would require a huge test rig especially for its duct system, and to control the inlet air temperature would not be an easy task. Figure 3-3 shows comparison of heat rejecting part of a test rig with air-cooled gas cooler and water-cooled gas cooler.

One important consideration to be taken into account to represent an air-cooled gas cooler characteristic is that the thermal mass of both the refrigerant and the cooling medium in the water-cooled gas cooler should be kept similar as in the air-cooled gas cooler. The refrigerant side volume of the water-cooled gas cooler has been kept the same as in the designed air-cooled gas cooler, while heat tranfer area of the water side was determined in order to have the same heat flux along the tube as in the designed air-cooled gas cooler. In order to have similar thermal mass, water mass flow rate was adjusted so that the outlet temperature and pressure of CO_2 are similar to what would be obtained if an air-cooled gas cooler was used. The outlet condition of the water side was not so important here since it will not affect the system performance. The most important parameter to be considered was in the refrigerant side. As long as the refrigerant state in the outlet of the gas cooler is similar, the system performance will be the same regardless what type of gas cooler is being used.

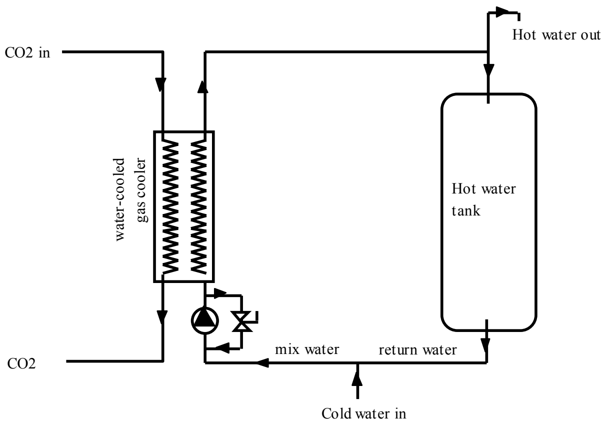
To design a water-cooled gas cooler that represents an air-cooled gas cooler, the process is described as followed. From the design stage with hXSim, total volume of CO_2 side and total heat transfer area of air side can be obtained. Pipe diameter for CO_2 side has been kept the same so that the pipe length was the same both in the designed air-cooled gas cooler and in the water-cooled gas cooler to have the same refrigerant volume. The total heat transfer area of water side of the water-cooled gas cooler representing total heat transfer area of air side of the air-cooled gas cooler is determined in such a way that it will give the same heat transfer rate. The total heat transfer area of water side was determined by the following relation:

$$\overline{h}_w A_w = \overline{h}_a A_a \tag{3.21}$$

where $\overline{h}_w$ is average heat transfer coefficient of the water and $\overline{h}_a$ is that of the air, A_w is total heat transfer area of water and A_a is total heat transfer area obtained from hXsim. The type of heat exchanger chosen for representing the air-cooled gas cooler is the same as the one for the water heating heat exchanger, that is coiled tube double-pipe type. The heat exchanger representing the air-cooled gas cooler consists of two similar sections. This was done in order to get similar pressure drop in the CO_2 side since the air-cooled gas cooler has been designed with two sections in the refrigerant side as shown in Figure 3-2. From now on, the heat exchanger representing the designed air-cooled gas cooler will be called the heat rejecting heat exchanger.



(a) Air-cooled gas cooler system



(b) Water-cooled gas cooler system

Figure 3-3 Comparison of air-cooled and water-cooled gas coolers system

As the type of heat rejecting heat exchanger is the same as that of water heating heat exchanger, all assumptions and calculation procedure as well as all correlation were the same as those in the water heating heat exchanger.

Figure 3-4 and Figure 3-5 show CO_2 temperature at the outlet of the gas coolers at various discharge pressure at air-conditioning mode without heat recovery and with 50% heat recovery, respectively. As can be seen from these figures, CO_2 temperatures at the outlet of the water-cooled gas cooler were similar with that at the outlet of the air-cooled gas cooler calculated by hXsim. It can be expected that the experimental data with the water-cooled gas cooler system can represent the situation for the system equipped with an air-cooled gas cooler.

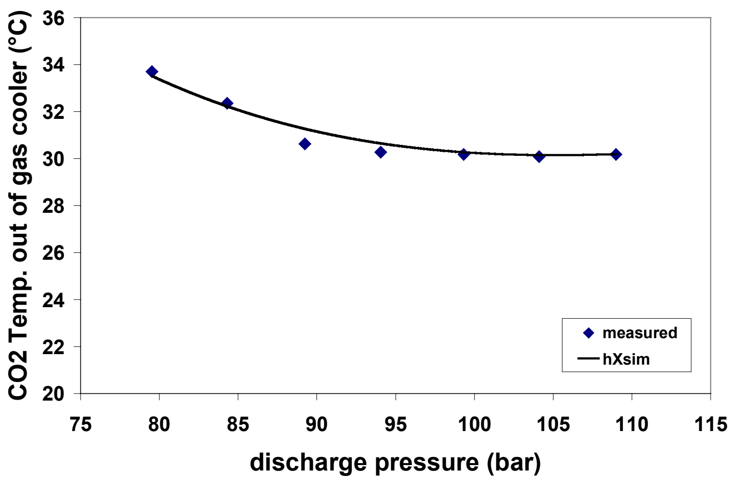


Figure 3-4 CO2 temperature at the outlet of the water-cooled and the air-cooled gas cooler system at air-conditioning mode
[T_{evap} = 0°C, T_{sink} = 30°C, A/C mode]

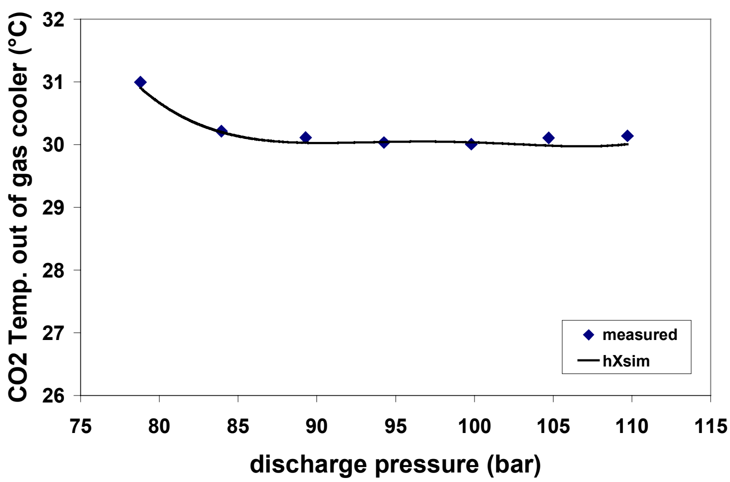


Figure 3-5 CO2 temperature at the outlet of the water-cooled and the air-cooled gas cooler system at combined mode
[T_{evap} = 0°C, T_{sink} = 30°C, xr = 0.5]

3.4 Internal Heat Exchanger Model

The internal heat exchanger transfers heat from CO_2 stream in the high-pressure side to CO_2 stream in the low-pressure side of the system. This heat transfer process is crucial for transcritical CO_2 cycle when operating at high ambient temperature (i.e. above the critical temperature of CO_2) which causes excessive throttling losses and very low specific refrigerating capacity. The internal heat exchanger used in this study is also of the same type as the water heating and the heat rejecting heat exchangers, i.e. coiled tube double-pipe heat exchanger. CO_2 in the low-pressure side flows in the annulus while that in the high-pressure side flows in the inner tube. Segmentation of the internal heat exchanger is also necessary to obtain a higher accuracy especially when CO_2 in the high-pressure side is in the vicinity of the critical region where its properties vary greatly.

The main assumptions for internal heat exchanger model are as followed:

- stream contains pure CO_2 (without lubricant)
- CO_2 entering low side pressure of the internal heat exchanger is at saturated vapor
- CO_2 entering high side pressure of the internal heat exchanger is at supercritical gas
- there is no heat transfer process between the internal heat exchanger and its surrounding
- in each segment, heat transfer process from the high side pressure to the low side pressure is occurred isobarically.
- the flow of fluids is counter flow
- the internal heat exchanger is of coiled tube type and installed vertically.

From these assumptions, the same correlation and calculation procedure can be used in modeling the internal heat exchanger because both CO_2 streams are single-phase fluid.

Table 3-5 shows inputs and outputs for modeling the internal heat exchanger. An example of temperature profile along the internal heat exchanger is shown in Figure 3-6.

Table 3-5 Input and outputs for internal heat exchanger model

Inputs	Outputs
Refrigerant mass flow rates	Capacity
HP refrigerant inlet temperature	HP refrigerant outlet temperature
HP refrigerant inlet pressure	HP refrigerant outlet pressure
LP refrigerant inlet temperature	LP refrigerant outlet temperature
LP refrigerant inlet pressure	LP refrigerant outlet pressure

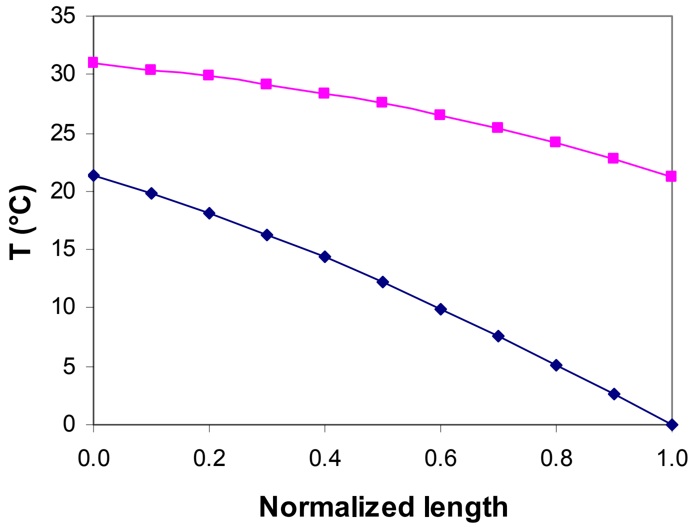


Figure 3-6 Temperature profile along the internal heat exchanger

3.5 Connecting pipe Model

The components of the test rig are connected by pipes. To return the lubricant back to the compressor it was necessary to install oil separator between the compressor and the gas coolers. Moreover, there were many measurement points mounted into the test rig for measuring temperature, pressure, and fluid flows. These accessories can influence the state of refrigerant from one component to the other one and this effect should be taken into account if the accuracy of the model needs to be improved. Connecting lines that contribute large change of refrigerant states are the suction line that connects the evaporator with the compressor and the discharge line that connects the compressor with the gas coolers.

Modeling of heat transfer process and pressure drop in the connecting lines is not easy due not only to the additional instruments installed along the lines but also a high oil concentration especially in the suction line. A portion of liquid collected in the liquid receiver that contains CO_2 and oil must be drained and feed into the internal heat exchanger to provide an automatic oil return. The simple approach to handle this problem is if experimental data are available for a particular test rig. With this data, a function based on curve fitting of these experimental data can be determined so that the characteristic of the system could be predicted more accurate.

Three equations have been developed in this work, which are for pressure drop in the suction line and discharge line, and temperature drop in the discharge line. Heat transfer in the suction line was not considered since the internal heat exchanger dominates heat transfer process and the different between suction temperature and the ambient air is small compared to the one in the discharge line.

From the curve fitting of experimental data, it can be obtained temperature drop and pressure drop as a function of Reynold number as followed:

$$\frac{\Delta T_{comp\_to\_gc}}{T_{crit}} = 4.00294\,\mathrm{Re}^{-0.459649} \tag{3.21}$$

$$\frac{\Delta P_{comp\_to\_gc}}{P_{crit}} = 2.43465.10^{-5}.\mathrm{Re}^{-0.481064} \tag{3.22}$$

$$\frac{\Delta P_{eva\_to\_comp}}{P_{crit}} = 2.6813.10^{-8}.\mathrm{Re}^{1.02943} \tag{3.23}$$

Equation (3.21) and (3.22) are for temperature drop and pressure drop from the compressor to the gas coolers, while equation (3.23) is for pressure drop from evaporator to the compressor.

3.6 Coupling of the Component Models

By so far, the steady state components model of the combined system has been described. As has been stated before, the evaporator is not needed to model because the evaporation temperature was used as a parameter by adjusting the amount of heat from an electrical heating element. Expansion process was considered to be adiabatic so that an isenthalpic model is sufficient for this component. By coupling all these component models, a complete refrigeration system modeling can be built and it is possible to arrange the gas coolers blocks to be in series or parallel.

3.6.1 Calculation algorithm

Calculation process referred to thermodynamic cycle as depicted in Figure 3-7. Because the least input required for the compressor model, the calculation was started from this block. A guessed value for suction temperature should be given before the calculation is begun while a guessed suction pressure can be chosen the same as saturation pressure at given evaporation temperature. The calculation was done in a sequential manner as followed: compressor $\rightarrow$ discharge line $\rightarrow$ gas coolers system $\rightarrow$ mixing $\rightarrow$ internal heat exchanger $\rightarrow$ suction line $\rightarrow$ compressor.

A new suction temperature and suction pressure then available from the first step of the calculation and these variables were compared with the previous values. If the deviation were smaller than convergence criteria then the calculation was stopped, but if not then the old suction temperature and pressure were replaced by the new ones and the calculation was repeated until the convergence criteria was fulfilled. Figure 3-8 shows a flow chart of the cycle calculation process. In the next chapter, the result from this simulation was validated with experimental data.

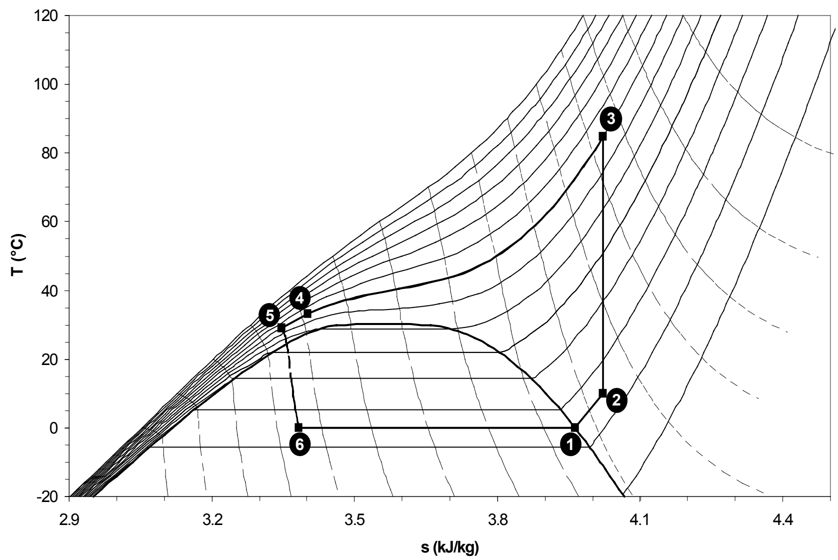


Figure 3-7 T-s diagram of the cycle calculation

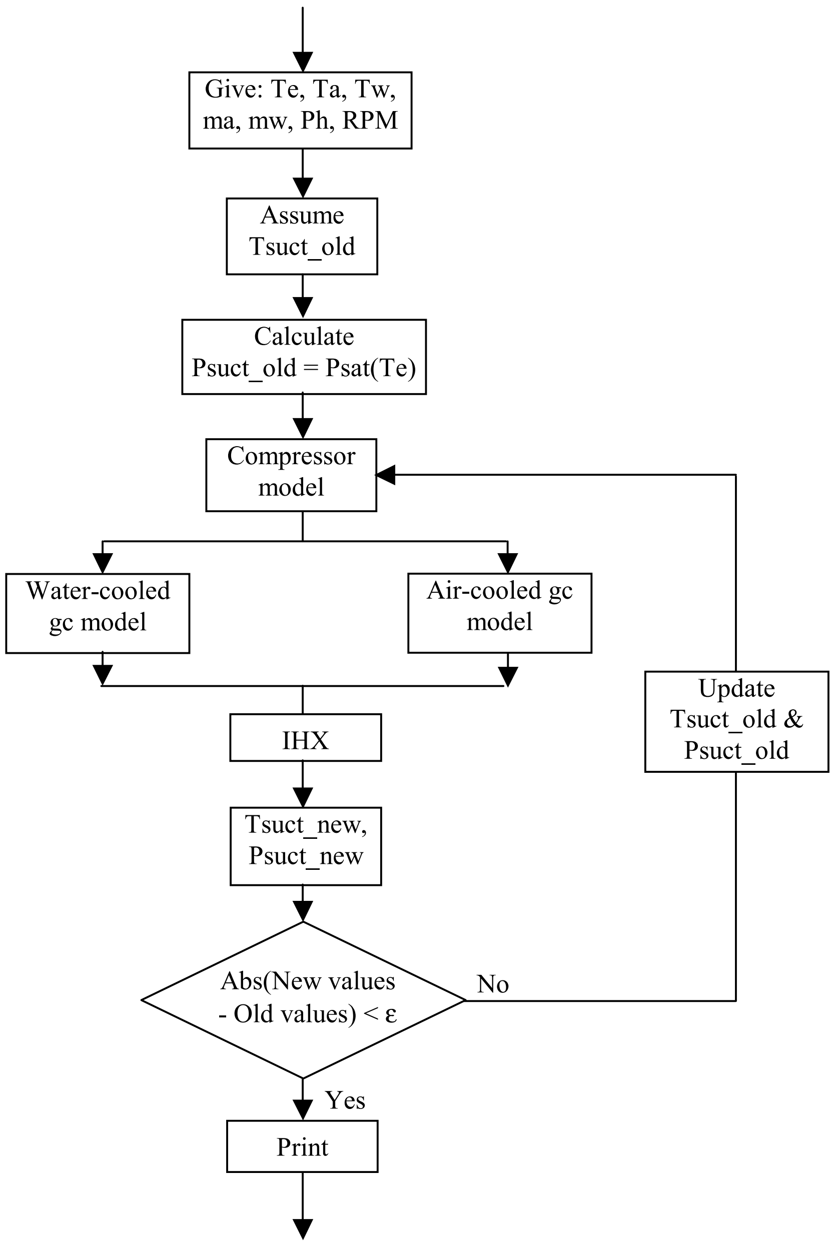


Figure 3-8 Flow chart of the cycle calculation

4 Prototype Of Combined Air-conditioning/ Water-heating System

Test rig developed in this work was a modified hot water heat pump system (Nekså, 1998) with an additional water-cooled gas cooler and a water loop as cooling medium. The original version of the test rig was designed for heat pump water heater using CO_2 as working fluid with heating capacity of 50 kW at 0°C evaporation temperature, 7°C inlet water temperature, and 60°C hot water temperature. In this work, a new water loop together with a water-cooled gas cooler was designed and built. The additional water-cooled gas cooler was intended to simulate different heat sink conditions.

The two water-cooled gas coolers can be arranged in series or parallel by switching the pipeline in order to observe the characteristic of the system in different gas coolers configuration. Figure 4-1 shows photograph of the test rig. Figure 4-2 shows part of the test rig that has been designed and built in this work.

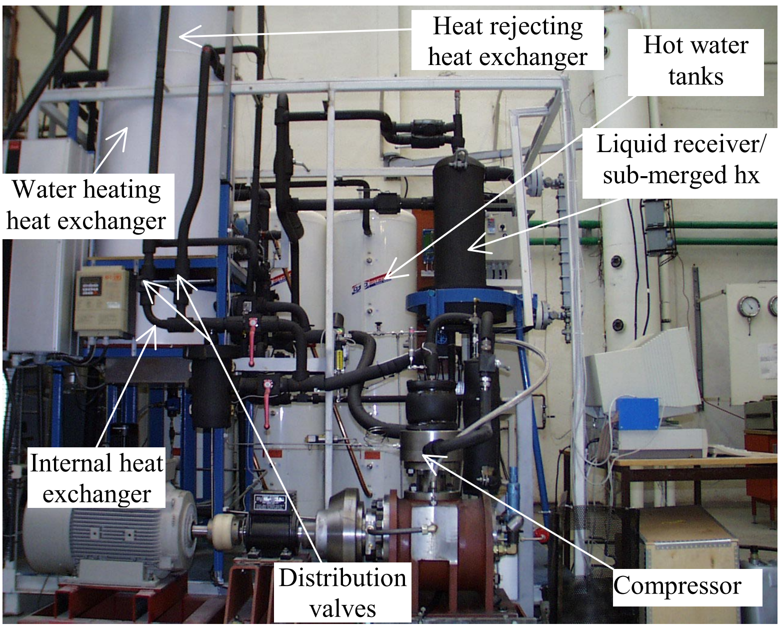


Figure 4-1 Prototype of the combined air-conditioning and water-heating system

The purpose of erecting the test rig has been to study experimentally the characteristic of transcritical CO_2 cycle at various operation modes. There were three kinds of operation modes that were run in this study:

1. Air-conditioning mode where all heat from the system was rejected to the ambient through the heat rejecting heat exchanger.
2. Full heat recovery mode where all rejected heat was utilized for producing hot water.
3. Partial heat recovery mode where only part of rejected heat was utilized for producing hot water.

Beside to investigate the system characteristic, the experimental data obtained from this work were also used for validating the simulation program described in Chapter 3. Agreement of the simulation program with the experimental data was studied to improve the accuracy of the component model through modification of mathematical model involved if necessary. It will be found later in this chapter that the agreement is well within a certain range so that the simulation program can be used as a basis for designing a similar system or for predicting the system behaviour not obtained from the experiment. The simulation program can also be used to predict the system behavior that is difficult to obtain from the experiment, at very high discharge temperature, for instance.

4.1 Process Description

The prototype of the combined air-conditioning and water-heating system consisted of four loops:

1. Glycol circuit as a heat source
2. Refrigerant circuit
3. Hot water circuit
4. Water circuit, simulating ambient condition, as a heat sink

To simulate a heat source, heat source circuit consisting of ethylene glycol, glycol pump, expansion tank, and electrical heater were employed in the system. Glycol was circulated through the evaporator by the glycol pump. The electrical heater heated glycol to simulate cooling load. In the evaporator, heat from the glycol was absorbed by CO_2 and then rejected in the high-pressure side to the ambient (simulated by the water circuit) or to the hot water circuit depend on the operating mode. The heat load from the electrical heater was controlled automatically so that the evaporation temperature can be set at any desired value.

Flow diagram and measurement points of the prototype system are shown in Figure 4-2. Referring to this figure, CO_2 from the liquid receiver is sucked by the compressor through low-side pressure of the internal heat exchanger so that CO_2 becomes superheat before entering the compressor. This superheat CO_2 then is compressed to a high-side pressure. The high-side pressure is controlled manually by opening the expansion valve, which is actuated pneumatically.

The high-pressure gas then is split into two streams, one stream goes to the heat rejecting heat exchanger and the other stream goes to the water heating heat exchanger. In the heat rejecting heat exchanger, CO_2 stream rejects heat to the water circuit as the heat sink, while in the water heating heat exchanger, CO_2 transfers its energy to water being heated to a target temperature. The target hot water temperature is achieved by regulating the water mass flow rates through a change in the water pump rotational speed. After giving its energy, these two CO_2 streams then mixes and then enters the high-side pressure of the internal heat exchanger where high pressure CO_2 is cooled down by low-side pressure CO_2.

To achieve efficient evaporation process it is necessary to keep the refrigerant out of the evaporator in two-phase region and this can be done by passing the high-side pressure CO_2 through the integrated liquid receiver/submerged heat exchanger. In this device, high-side pressure CO_2 is cooled down further by transferring its energy to evaporates part of liquid CO_2 in the liquid receiver. The evaporation process in the liquid receiver will determine the quality of CO_2 leaving the evaporator. A higher CO_2 being evaporated in the liquid receiver will shift the CO_2 leaving the evaporator into a lower quality making heat transfer process in the evaporator more effective due to a higher average evaporating heat transfer coefficient in the CO_2 side.

From the high-side pressure after being cooled in the liquid receiver, CO_2 enters the expansion valve as a liquid like fluid and expands to a two-phase region before entering the evaporator. Even CO_2 is in single-phase region before entering the expansion valve, its density is very high more like liquid and the expansion process will be more or less the same as that in subcritical cycle.

Two-phase CO_2 then enters the evaporator where it receives energy from glycol as the heat source. The evaporation process is the same as in the subcritical refrigeration cycle. From the evaporator, CO_2 enters the integrated liquid receiver/submerged heat exchanger as two phase fluid and evaporates there further. In this receiver, some of the liquid will stay at the bottom while some of it will evaporate. CO_2 vapor then is sucked by the compressor from the top of the receiver to the low-pressure side internal heat exchanger. Finally saturated vapor CO_2 is superheated in the low-side pressure of the internal heat exchanger before entering the compressor again, making a complete loop.

It can also be seen in Figure 4-2 that the test rig was also equipped with an oil separator in which lubricant brought by hot gas CO_2 is separated and collected. To drain collected lubricant back to the compressor; the oil separator is connected to the compressor crankcase via a steel pipe, which equipped with a valve. The oil can be drained automatically by adjusting the valve opening.

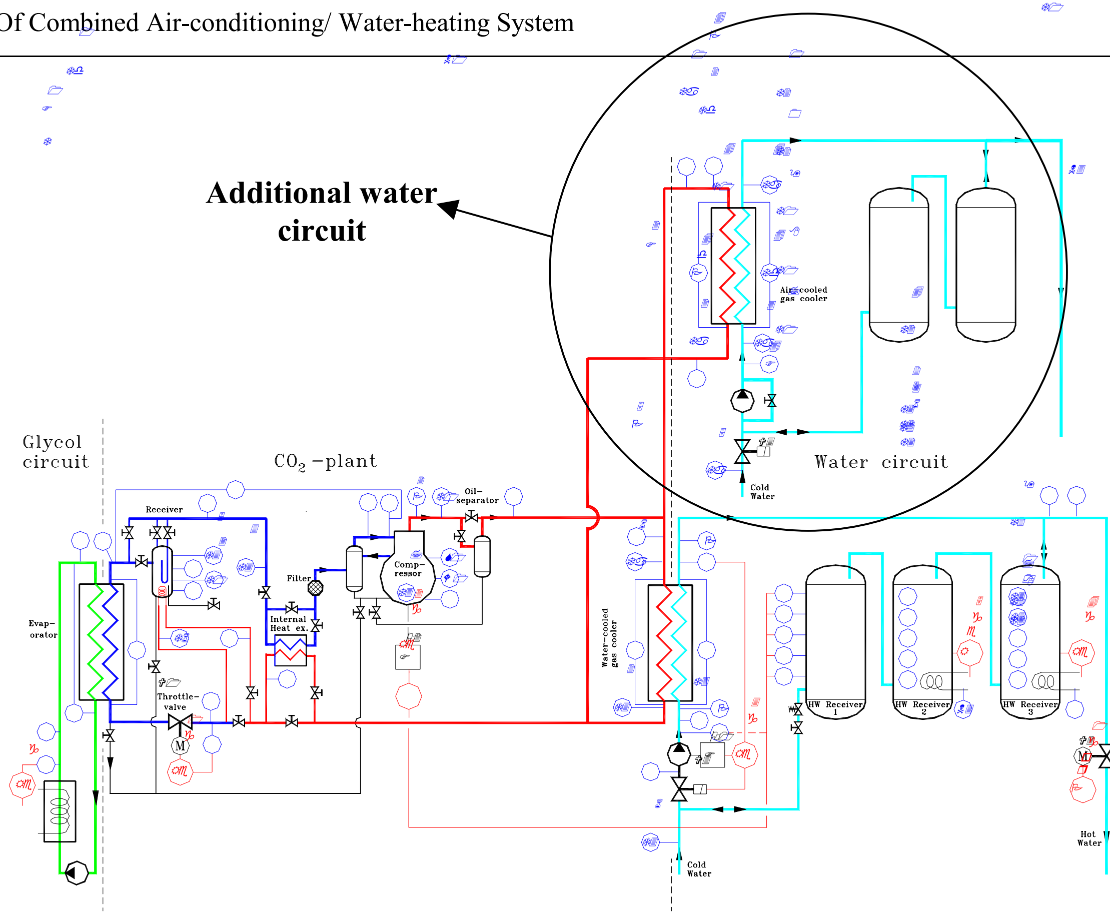


Figure 4-2 Flow diagram and measurement points of the prototype system

Since the lubricant will also accumulate in the lower part of the evaporator and liquid receiver, small pipes connect the lower part of these components to the compressor crankcase. The oil is drained when the system is not running.

4.2 Test Rig Components

4.2.1 Compressor

The compressor of the test rig is of open type one cylinder reciprocating compressor. To reduce leakage from the compression chamber to the crankcase three piston rings were installed. Design rotation was 890 rpm but due to a problem in the connection between compressor and motor shaft, the compressor was only run at 804 rpm in whole experiments. The compressor head discharging high temperature CO_2 was insulated to minimize heat loss to the surrounding. Table 4-1 shows the main technical data of the compressor and Figure 4-3 shows the compressor mounted in the test rig.

Table 4-1 Technical data for the compressor (Zakeri et al., 1999)

Type	Reciprocating
Number of cylinder	1
Cylinder bore diameter	50 mm
Piston stroke	80 mm
No. of revolution	600-1200 rpm
Swept volume	5.66-11.3 m^3/h
Design pressure:	
Low-side	75 bar
High-side	150 bar

Figure 4-3 The compressor

4.2.2 Water Heating Heat Exchanger

The water heating heat exchanger was of helical coil double-pipe type heat exchanger. High pressure CO_2 flows in the inner tube while water flows in the annuli. To center the inner tube, a small copper wire was twisted around the inner tube. Pressure drop in the annuli side will be higher due to the presence of this centering string but it will also promote convection heat transfer by increasing turbulency. Both inner and outer tubes are made of stainless steel. Table 4-2 shows main technical data of the water-cooled gas cooler and Figure 4-4 shows its cross section.

Table 4-2 Characteristic technical design data for the water heating heat exchanger (Zakeri et al., 1999)

Type	Helical coil co-axial heat exchanger
Material	Stainless steel
Diameter:	
Inner-tube (CO_2)	Ø 15 x 1,5 (mm)
Annuli (water)	Ø 26.9 x 1,6 (mm)
Test-pressure:	
Inner-tube (CO_2)	160 bar
Annuli (water)	16 bar
Length	54 m
Weight	115 kg
Heat transfer area:	
Inner-tube (CO_2)	2.04 m2
Annuli (water)	2.54 m 2
Coil diameter	0.7 m

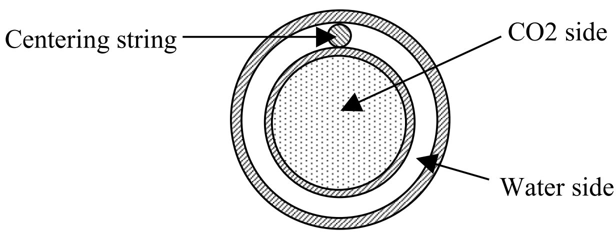


Figure 4-4 Cross section of the water heating heat exchanger

4.2.3 Heat Rejecting Heat Exchanger

Like the water heating heat exchanger, the heat rejecting heat exchanger was of helical coil double-tube type heat exchanger where CO_2 flows in the inner tube and water flows in the annuli. There is also a centering string between the inner tube and the annuli. Both the outer and inner tubes are made of stainless steel and the small centering string was made of copper. To obtain a similar pressure drop in CO_2 side with the air-cooled gas cooler designed with hXsim, the gas cooler has been composed of two identical units just like in the air-cooled gas cooler designed with hXsim program. Water flow rate has been adjusted to represent velocity of air and set constant in all experiment because the air velocity was chosen only at 3 m/s. Manufacturing of this gas cooler was conducted in the workshop at the department of refrigeration and air-conditioning, NTNU. Table 4.4 shows main dimension of the heat rejecting heat exchanger and Figure 4-5 shows its cross section. At certain stages of manufacturing process, leakage tests were done by pressurizing the heat exchanger with high-pressure nitrogen gas at 160 bar. The pressure in the tube was monitored for 24 hours to ensure there was no leakage from the tube. Figure 4-6 and Figure 4-7 show centering string and the heat rejecting heat exchanger before being insulated.

Table 4-3 Characteristic technical design data for the heat rejecting heat exchanger.

Type	Helical coil co-axial heat exchanger
Material	Stainless steel
Diameter:	
Inner-tube (CO_2)	Ø 12 x 1 (mm)
Annuli (water)	Ø 26.9 x 1,6 (mm)
Test-pressure:	
Inner-tube (CO_2)	160 bar
Annuli (water)	16 bar
Total length	60 m
Circuit Length	30 m
Number of circuit	2
Weight	115 kg
Heat transfer area:	
Inner-tube (CO_2)	2.04 m2
Annuli (water)	2.54 m 2
Coil diameter	0.7 m

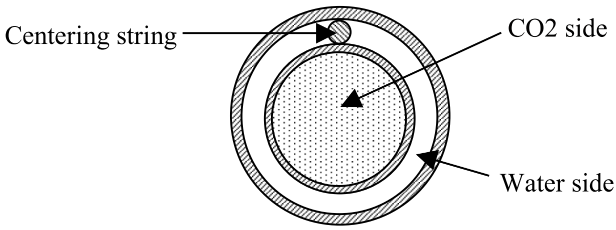


Figure 4-5 Cross section of the heat rejecting heat exchanger

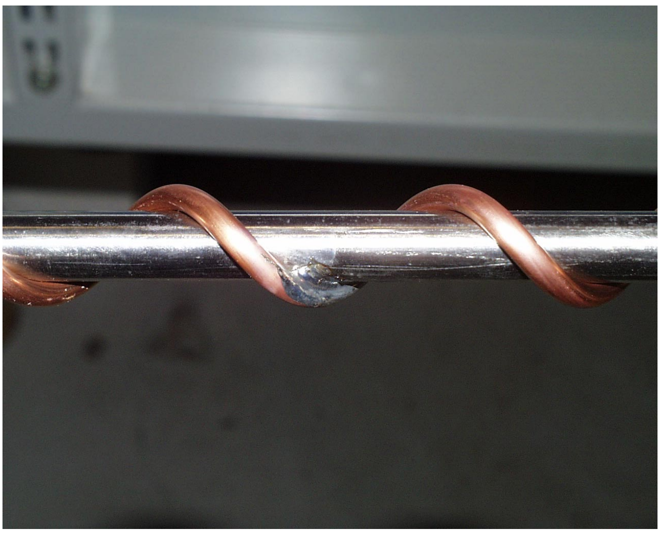


Figure 4-6 CO_2 tube with centering string

Figure 4-7 The heat rejecting heat exchanger

4.2.4 Internal heat exchanger

The internal heat exchanger installed in the test rig was the same type of both the gas coolers. Detail dimensions are given in Table 4-5 and its cross section is shown in Figure 4-8.

Table 4-4 Geometric data for internal heat exchanger (Zakeri et al., 1999)

Flowing medium, center tube	CO_2 low-pressure suction gas
Flowing medium, annuli	CO_2 high-pressure gas
Inside tube dimension (mm)	Ø15 x 1.5
Outside tube dimension (mm)	Ø26.9 x 1.6
Heat exchanger length (m)	6
Coil diameter (m)	0.7

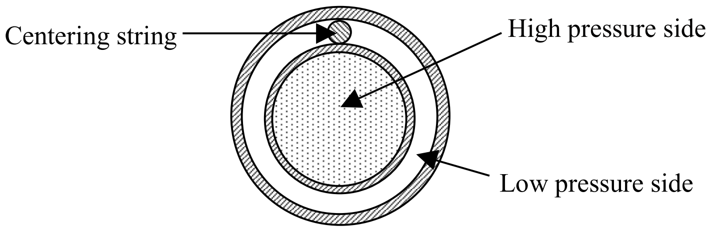


Figure 4-8 Cross section of the internal heat exchanger

4.2.5 Integrated liquid receiver/submerged heat exchanger

More compact component can be designed by integrating liquid receiver and submerged heat exchanger. The liquid receiver collects overcharge refrigerant and acts as a buffer when more refrigerant is needed to increase the high-side pressure. The submerged heat exchanger boils part of the liquid in the receiver to shift the refrigerant state into two phase region in the outlet of evaporator and cools high-side pressure CO_2 in the tube. The submerged heat exchanger was installed in the bottom of the receiver. When liquid level in the receiver becomes low, the submerged heat exchanger will not be covered by liquid CO_2 and this uncovered part will cause slight superheat when CO_2 leave the receiver. One important aspect when designing the liquid receiver is the entrance velocity of CO_2 from the evaporator that should be determine in order not to cause liquid droplet to be carried over from the receiver. This is crucial for the compressor especially when internal heat exchanger is not being used making a potential of compressing the liquid droplet that could damage the compressor. Figure 4-9 shows principal design of the integrated liquid receiver/submerged heat exchanger and its main technical data are given in Table 4-5.

Table 4-5 Design data for the liquid receiver/submerged heat exchanger (Zakeri et.al., 1999)

Evaporation Temperature : 0°C	m_{CO2}=0.125 (kg.s^{-1})	m_{CO2}=0.176 (kg.s^{-1})
Vapor velocity (m.s^{-1})	0.04	0.056
Internal heat exchanger (HX)		
Material	Stainless steel	
Design capacity	2200 W	
LMTD	8.5 K	
K-value	1500 W.m^{-2} .K	
HX surface area	0.173 m 2	
HX tube size	Ø15X1.5 mm	
HX tube length	4.6 m	

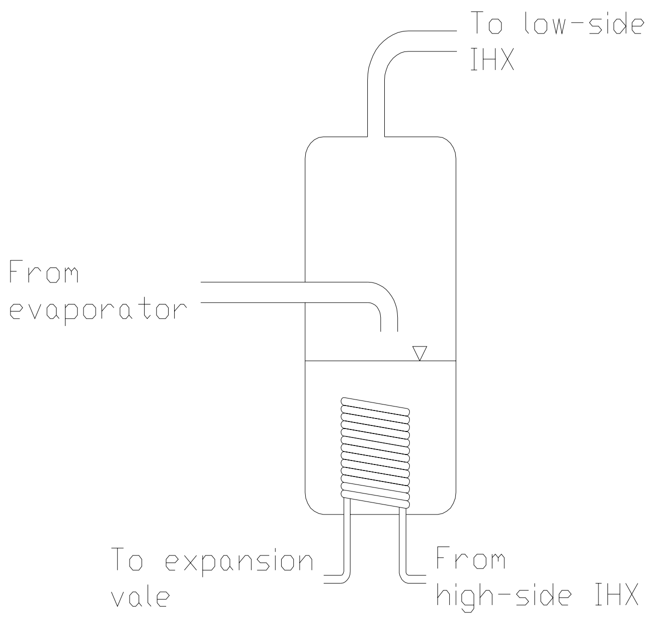


Figure 4-9 Integrated liquid receiver/submerged heat exchanger

4.3 Instrumentation And Measurement Accuracy

4.3.1 Temperature measurement

All temperature measurements were carried out by type-T thermocouple with accuracy ±0.5°C. These thermocouples were connected to a data logger that converts voltage signal from the thermocouples to temperature. The data acquisition system included a hardware and software compensation to provide the "ice-point" reference junction, so that there was no need for a second reference junction. Temperature points were made of a small diameter pipe placed in the center of measuring pipe to represent the measured temperature as an average value.

4.3.2 Pressure measurement

DRUCK pressure transducer measured absolute pressures, which send voltage signal to the data logger. To obtain more accurate measurement, differential pressure transducers were used to measure pressure drop across the heat exchangers. There were three differential pressures installed in the test rig, two in the gas coolers and one in the connecting line between evaporator and compressor. All the voltage signals from these transducers were sent to the scanner and converted back to pressure in the data logger. Measurement points was made of stainless steel pipe ¼" in diameter and installed vertically on the measuring pipe surface. Accuracy of these absolute transducers are ±0.1% of measured value and accuracy of the differential pressures were 0.04% of measured value.

4.3.3 Flow measurement

Turbine flow meters were installed in the test rig to measure water flows through the heat exchangers. There are two flow meters, one for the water heating heat exchanger and the other for the heat rejecting heat exchanger. The accuracy of the flow meters were 0.2% of calibration span, times water density.

CO_2 flow rates were determined through heat balance calculation in both gas coolers. To increase accuracy of calculation, 3-point thermopile were installed to measure temperature different across the gas coolers so that its accuracy becomes ±0.3°C. However, in the final report all uncertainties calculation were based on ±0.5°C temperature accuracy.

4.3.4 RPM and Torque meter

There was a rotation and a torque meter installed in the test rig through which the compressor power consumption was determined. The accuracy of the rotation meter was ±1 RPM and the accuracy of the torque meter was ±0.5% of measured value.

4.3.5 Uncertainty of the derivative values

Having had the accuracy of the main measurement equipment, the other uncertainties of the parameters can be determined from theory of measurement uncertainty. There are two definitions regarding the uncertainty of a measured value, absolute uncertainty and relative uncertainty. If there is a quantity f as a function of variable x and y, the absolute uncertainty of this quantity is defined as:

$$\Delta f(x,y) = \sqrt{\left(\frac{\partial f}{\partial x}\Delta x\right)^2 + \left(\frac{\partial f}{\partial y}\Delta y\right)^2} \tag{4.1}$$

where, Δx and Δy are the uncertainty of variable x and y, respectively.

The relative uncertainty of a quantity is defined as a ratio of its absolute uncertainty to its value as given by Equation (4.2).

$$\text{Relative uncertainty} = \frac{\Delta f}{f} \cdot 100\% \tag{4.2}$$

Compressor power consumption

The compressor power was calculated by the following formula,

$$W_{comp} = 2\pi T n \tag{4.3}$$

where, T = Torque (kgf.m)
n = rotational speed (RPM)

and its absolute uncertainty is:

$$\Delta W_{comp} = \sqrt{(2\pi n.\Delta T)^2 + (2\pi T.\Delta n)^2} \tag{4.4}$$

Air-cooled gas cooler and water-cooled gas cooler

The capacity of both gas coolers were determined by the following energy balance:

$$\dot{Q}_{gc} = \dot{m}_w.cp_w.(T_{w\_out} - T_{w\_in}) \quad (4.5)$$

so that its uncertainty becomes:

$$\Delta\dot{Q}_{gc} = \sqrt{(cp_w.(T_{w\_out} - T_{w\_in}).\Delta\dot{m}_w)^2 + 2(\dot{m}_w.cp_w.\Delta T)^2} \quad (4.6)$$

Refrigerant flow rates

The flow rates of CO_2 was determined through energy balance for the gas coolers as given by the following equation:

$$\dot{m}_r = \frac{\dot{Q}_{gc}}{h_{co2\_in} - h_{co2\_out}} \quad (4.7)$$

and the uncertainty will depend indirectly on temperature and pressure through enthalpy.

$$\Delta\dot{m}_r = \sqrt{\left(\frac{\Delta\dot{Q}_{gc}}{\left(h_{co2\_out} - h_{co2\_in}\right)}\right) + \left(-\frac{\dot{Q}_{gc}}{\left(h_{co2\_out} - h_{co2\_in}\right)^2}\Delta h_{co2\_out}\right) \\ = + \left(\frac{\dot{Q}_{gc}}{\left(h_{co2\_out} - h_{co2\_in}\right)^2}\Delta h_{co2\_in}\right)} \quad (4.8)$$

Evaporator

Uncertainties of evaporator capacity was calculated by neglecting heat loss from evaporators, internal heat exchanger, and integrated liquid/submerged heat exchanger. The following equation then can be obtained:

$$\dot{Q}_e = \dot{m}_r \cdot \left(h_{co2\_suct} - h_{co2\_gco}\right) \tag{4.9}$$

where h_{co2_suct} is enthalpy at the compressor suction port and h_{co2_gco} is enthalpy at the outlet of the gas coolers system. The absolute uncertainty becomes:

$$\begin{aligned}\Delta\dot{Q}_e &= \sqrt{((h_{co2\_suct} - h_{co2\_gco}) \cdot \Delta\dot{m}_r)^2 + (\dot{m}_r \cdot \Delta h_{co2\_suct})^2} \\ &= \overline{+ (\dot{m}_r \cdot \Delta h_{co2\_gco})^2}\end{aligned} \tag{4.10}$$

Coefficient of performance

The cooling coefficient of performance was defined as a ratio of cooling capacity (Q_e) to the compressor shaft power (P).

$$\text{cooling} - \text{COP} = \frac{\dot{Q}_e}{P} \tag{4.11}$$

Its absolute uncertainty is given as follows:

$$\Delta\text{cooling} - \text{COP} = \sqrt{\left(\frac{\Delta\dot{Q}_e}{P}\right)^2 + \left(-\frac{\dot{Q}_e}{P^2}\Delta P\right)^2} \tag{4.12}$$

Isentropic efficiency

Isentropic efficiency was defined as a ratio of compressor isentropic work to shaft power consumption as given by following equation.

$$\eta_s = \frac{\dot{m}_r \cdot (h_{co2\_dis\_s} - h_{co2\_suct})}{P} \tag{4.13}$$

where the nominator is the isentropic work. The absolute uncertainty then become:

$$\Delta\eta_s = \sqrt{\begin{aligned}&\left(\frac{h_{co2\_dis\_s} - h_{co2\_suct}}{P}\Delta\dot{m}_r\right)^2 + \left(\frac{\dot{m}_r}{P}\Delta h_{co2\_dis\_s}\right)^2 \\ &= + \left(\frac{\dot{m}_r}{P}\Delta h_{co2\_suct}\right)^2 + \left(\frac{\dot{m}_r \cdot (h_{co2\_dis\_s} - h_{co2\_suct}}{P^2}\right)^2\end{aligned}} \tag{4.14}$$

Volumetric efficiency

The volumetric efficiency was defined as a ratio of theoretical mass flow rates of refrigerant to actual mass flow rates of refrigerant.

$$\eta_v = \frac{\dot{m}_r / v_{co2\_suct}}{(V_{disp} \cdot 60 \cdot RPM)} \tag{4.15}$$

where V_{disp} is the compressor displacement and v_{co2_suct} is the spesific volume of CO_2 at suction port. Its absolute uncertainties is:

$$\Delta\eta_v = \sqrt{\begin{aligned}&\left(\frac{1/v_{co2\_suct}}{(V_{disp} \cdot 60 \cdot RPM)} \cdot \Delta\dot{m}_r\right)^2 + \left(\frac{-\dot{m}_r / v^2_{co2\_suct}}{(V_{disp} \cdot 60 \cdot RPM)^2} \cdot \Delta v_{co2\_suct}\right)^2 \\ &= + \left(-\frac{\dot{m}_r / v_{co2\_suct}}{60 \cdot RPM \cdot (V_{disp} \cdot 60 \cdot RPM)^2} \cdot \Delta V_{disp}\right)^2 \\ &= + \left(-\frac{\dot{m}_r / v_{co2\_suct}}{60 \cdot V_{disp} \cdot (V_{disp} \cdot 60 \cdot RPM)^2} \cdot \Delta RPM\right)^2\end{aligned}} \tag{4.16}$$

All uncertainties of the thermodynamic properties involved in the calculation were a function of temperature (T) and pressure (P) and were calculated by the following relationship:

$$\Delta f = \sqrt{\left(\frac{\partial f}{\partial T} \cdot \Delta T\right)^2 + \left(\frac{\partial f}{\partial P} \cdot \Delta P\right)^2} \tag{4.17}$$

The uncertainties of the experimental data was depend on the discharge pressure especially when running at a pressure close to critical region at which the uncertainties become higher. Most of the optimum condition were range from 85 to 105 bar discharge pressure and the relative uncertainties for cooling capacity, gas cooler capacity and cooling-COP were around 5%. Table 4-6 shows the uncertainties of the experimental result for three discharge pressures at 0°C evaporation temperature, 30°C cooling medium temperature, air-conditioning mode. A higher uncertainties at 80 bar compared to 85 bar can be seen from this table.

Table 4-6 Resulting relative uncertainties for two different discharge pressures

Parameter	**Ph = 80 bar**	**Ph = 85 bar**	**Ph = 110 bar**
Compressor power	±0.17 %	±0.17 %	±0.15 %
Air-cooled gas cooler capacity	±4.76 %	±4.16 %	±3.73 %
Evaporator capacity	±15.00 %	±4.66 %	±3.88 %
Refrigerant flow rates	±9.76 %	±4.35 %	±3.79 %
Cooling-COP	±15.00 %	±4.66 %	±3.88 %
Isentropic efficiency	±9.93 %	±4.69 %	±4.07 %
Volumetric efficiency	±9.77 %	±4.37 %	±3.81 %

4.4 Test Procedure

The test rig was equipped with control system connected to a computer. Compressor, water heating heat exchanger pump, and expansion valve could be controlled from the computer, but heat rejecting heat exchanger pump and evaporation temperature could not. The evaporation temperature was controlled manually by adjusting glycol temperature and the pressure of CO_2 out of the evaporator was checked whether it was corresponding with desired evaporation temperature or not. Water mass flow rates passing the heat rejecting heat exchanger was regulated by opening a bypass valve to give a flow rates that represent the air velocity. Water inlet temperature to the heat rejecting heat exchanger was controlled manually by mixing hot water in the hot water water tanks with cold water. Strong vibration has been experienced due to problem in the coupling connecting the compressor motor and the torque meter when running at higher than 890 rpm. Because of that, the whole experiments were only carried out at 804 rpm.

The test procedure were as followed:

1. The water heating heat exchanger pump, the heat rejecting heat exchanger pump, the glycol pump, the torque meter and rotation meter were turned on.
2. The expansion valve was fully opened.
3. Oil return valve from the bottom of the receiver to the low-pressure side internal heat exchanger was opened slightly.

4. The expansion valve was set at a desired high-side pressure and then controlled automatically.
5. The compressor was started at its lowest speed of 600 rpm.
6. Compressor Speed was increased to 804 rpm.
7. Inlet water temperatures to both gas coolers were controlled manually to a desired value by mixing hot and cold water through adjustment of the cold water valves manually.
8. Evaporation pressure was controlled manually by adjusting inlet glycol temperature to the evaporator.
9. Water mass flow rates to the water-cooled gas cooler was set to give a desired load ratio by regulating the rotational speed of the water pump in the water heating heat exchanger loop.
10. The distribution valves opening were adjusted to obtain target hot water temperature.
11. The test rig was run until all measuring points were stabilized.
12. All measuring data were logged for 15 minutes with 1 minute interval time.

5 Experimental Results

All experimental points carried out in this work can be grouped into two categories, one with internal heat exchanger and the other without internal heat exchanger. The gas coolers could also be arranged in series or in parallel. However, the series configuration was not performed in the experiment and was only studied through simulation results because of too large capacity of the water heating heat exchanger. It should be noted that at least four experiments were performed to establish the optimum discharge pressure at which the system gave the highest COP.

There are two definitions of the system performance in combined air-conditioning and water heating mode, i.e. cooling–COP and total-COP. Cooling-COP has been defined as a ratio of cooling load to compressor power consumption. If heat recovery is employed then total-COP has been defined as a ratio of cooling load plus hot water load to compressor power consumption.

$$\text{cooling - COP} = \frac{\text{cooling load}}{\text{compressor shaft power}} \tag{5.1}$$

$$\text{total - COP} = \frac{\text{cooling load} + \text{hot water load}}{\text{compressor shaft power}} \tag{5.2}$$

Another important parameter in a combined system is load ratio or percentage of heat recovery. Load ratio has been defined as a ratio of hot water load to total rejected heat of the system running without heat recovery. This definition makes the effect of heat recovery to the system performance is easier to be investigated especially when the system run at various load ratio at the same discharge pressure. In addition, the experimental results at various discharge pressures can be plotted at the same curve of the same load ratio as can be seen in this chapter. If load ratio is given in percent then it expresses the percentage of heat recovery.

After steady state has been established, all measurement points were recorded into a file by a data logger. The file was then transferred into a spreadsheet program to analyze the results. The calculation was performed automatically in a spreadsheet program and it made use of CO_2 library developed in SINTEF energy research that can be called through a macro program. The main experimental points are summarized in Table 5-1. Notice that **full** here means all rejected heat was transferred to the water heating heat exchanger and was utilized to heat water. The spreadsheet program along with the experimental data can be found in Appendix C.

Table 5-1a Experiment matrixes for parallel configuration with and without internal heat exchanger

With Internal Heat Exchanger		
Parameter	**Operating condition**	**Variation**
Discharge pressure (Ph)	T_{evap} = 0°C T_{w_in} = 20°C T_{w_out} = 60°C T_{sink} = 30°C xr = 0, 0.25, 0.5, 0.75, full	80 bar, 85 bar, 90 bar, 95 bar, 100 bar, 105 bar, 110 bar
Inlet water temperature (T_{w_in})	T_{evap} = 0°C, T_{w_out} = 60°C T_{sink} = 30°C xr = 0.25	20°C, 30°C, 40°C
Hot water temperature (T_{w_out})	T_{evap} = 0°C T_{w_in} = 20°C T_{sink} = 30°C xr = full	60°C, 70°C
Evaporation temperature (T_{evap})	T_{w_in} = 20°C T_{w_out} = 60°C T_{sink} = 30°C xr = 0,0.25,0.5,0.75,full	0°C, 5°C, 10°C
Load ratio (xr)	T_{evap} = 0°C T_{w_in} = 20°C T_{w_out} = 60°C T_{sink} = 30°C	0, 0.25, 0.5, 0.75, full
Cooling medium temperature (T_{sink})	T_{evap} = 0°C T_{w_in} = 20°C T_{w_out} = 60°C xr = 0, 0.25	30°C, 35°C

Table 5-2b Experiment matrixes for parallel configuration with and without internal heat exchanger

Without Internal Heat Exchanger		
Parameter	**Operating condition**	**Variation**
Discharge pressure (Ph)	T_{evap} = 0°C T_{w_in} = 20°C T_{w_out} = 60°C T_{sink} = 30°C xr = 0, 0.25, full	80 bar, 85 bar, 90 bar, 95 bar, 100 bar, 105 bar, 110 bar
Inlet water temperature (T_{w_in})	T_{evap} = 0°C T_{w_out} = 60°C T_{sink} = 30°C xr = 0.25	20°C, 30°C
Load ratio (xr)	T_{evap} = 0°C T_{w_in} = 20°C T_{w_out} = 60°C T_{sink} = 30°C	0, 0.25, full

Because there was no measurement points after mixing of CO_2 streams coming from the heat rejecting heat exchanger and the water heating heat exchanger, the state of CO_2 after mixing in parallel configuration was determined by assuming adiabatic mixing process. This assumption was based on the fact that the connecting line was very short about 10 cm from both measurement point at the outlet of the gas coolers to the mixing point and the pipes were well insulated. The relationship below shows how the temperature after mixing was determined through a 'T_hp' function in the CO_2 thermophysical library developed in this work.

$$h_{mix} = \frac{\dot{m}_{co2\_hrhx} \cdot h\_tp(T_{co2\_hrhx}, p_{mix}) + \dot{m}_{co2\_whhx} \cdot h\_tp(T_{co2\_whhx}, p_{mix})}{\dot{m}_{co2\_whhx} + \dot{m}_{co2\_whhx}} \quad (5.3)$$

$$T_{mix} = T\_hp(h_{mix}, p_{mix})$$

CO_2 flow rates in each gas coolers was calculated from heat balance between CO_2 side and water side since there was no refrigerant flow meters installed in the test rig.

5.1 System Performance With Internal Heat Exchanger

Internal heat exchanger plays an important role in a transcritical cycle because it improves the system performance but it makes the system more complex. Therefore, it was necessary to carry out the experiment to study how the internal heat exchanger affects the system.

5.1.1 Discharge pressure effect

Basic characteristic of transcritical cycle is shown in Figure 5-1. Discharge pressure was varied from 80 bar up to 110 bar with 5 bar step. The evaporation temperature was held constant at 0°C, and the inlet water temperature to the heat rejecting heat exchanger was held constant at 30°C. As can be seen, compressor power consumption increases linearly from 7.1 kW to 9.5 kW with increasing pressure while cooling capacity increases from a lower pressures and then become more or less constant at 24.2 kW as discharge pressure increases. The cooling-COP first increases with increasing pressure from 2.66 to 3.0 and then at 88.8 bar starts to decrease to 2.54 as pressure increase to 110 bar. The high side pressure at 88.8 bar here is the optimum condition where cooling-COP reaches the highest value.

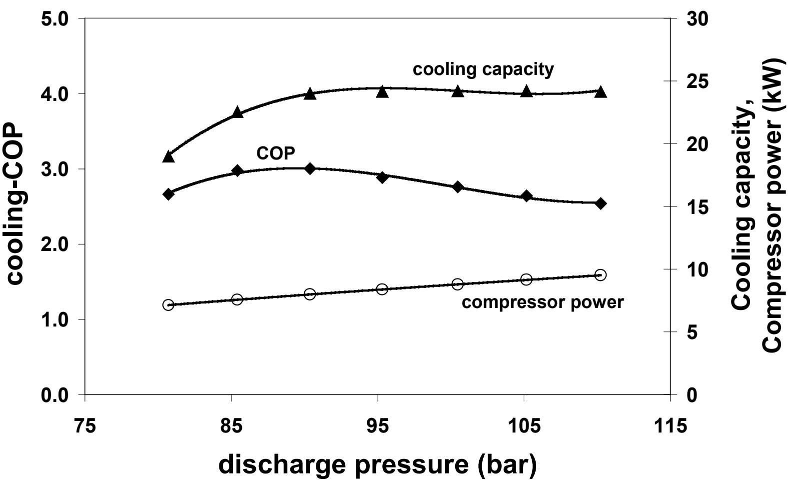


Figure 5-1 Discharge pressure influence on system performance
[$T_{evap.}$ = 0°C, T_{sink} = 30°C, A/C mode]

5.1.2 Effect Of Load Ratio

The result obtained from the experiment on air-conditioning mode without heat recovery was used to determine operating conditions in combined air-conditioning and water heating mode. In case of parallel configuration, load ratio (or percentage of heat recovery) was set at desired value, 0.25 for example, and then for a discharge pressure, 80 bar for instance, hot water load can be determined from the following relationship:

$$\dot{Q}_w = xr \cdot \dot{Q}_{o\_A/Cmode} \tag{5.4}$$

where xr = load ratio, and $Q_{o\_A/Cmode}$ = total rejected heat of air-conditioning without heat recovery at the same discharge pressure. Hence, for a certain inlet water temperature and hot water temperature, the required water mass flow rates can be calculated as followed:

$$\dot{m}_w = \frac{\dot{Q}_w}{Cp_w \cdot \Delta T_w} \tag{5.5}$$

By adjusting the distribution valves opening to control mass flow rates of CO_2 entering the water heating heat exchanger, hot water temperature can be controlled at a desired value, 60°C for example. Figure 5-2 shows cooling-COP at various load ratios.

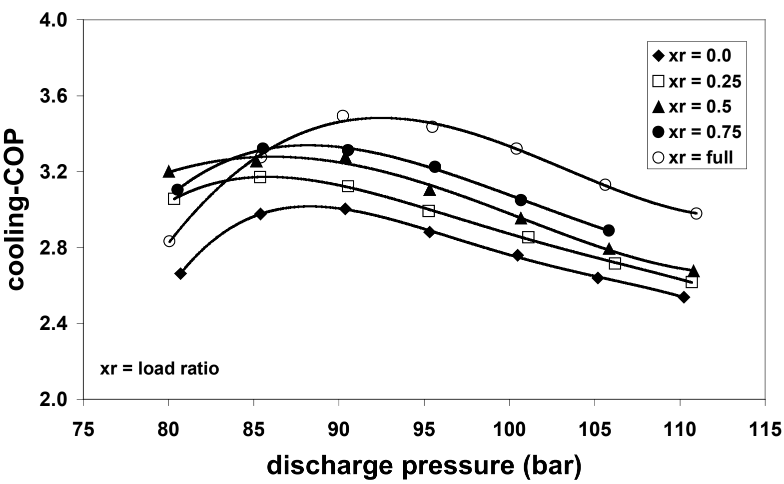


Figure 5-2 Load ratio effect on cooling-COP at various discharge pressures [T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

From Figure 5-2 it can be seen that at all load ratios, the cooling-COP trend was the same. The main different was the location of the optimum pressure. At the optimum condition, cooling-COP increases as load ratio increase but the optimum pressure first decreases and then increases as load ratio increases. The location of the optimum condition was given in Table 5-2 below.

Table 5-3 Optimum pressure and cooling-COP at various load ratios

Load ratio	Optimum pressure (bar)	Optimum cooling-COP	%
0.00	88.8	3.00	0.00
0.25	84.7	3.17	5.7
0.50	85.7	3.27	9.0
0.75	87.6	3.33	11.0
Full recovery	91.9	3.50	16.7

An advantage of parallel configuration is that the cooling-COP can be increased by increasing load ratio. As shown in Table 5-1, the optimum cooling-COP was increased by 5.7% at 25% heat recovery up to 16.7% at full recovery mode.

Another important aspect is that the cooling load variation due to heat recovery. Figure 5-3 and Figure 5-4 show variation of cooling load and compressor shaft power with discharge pressure at various load ratios.

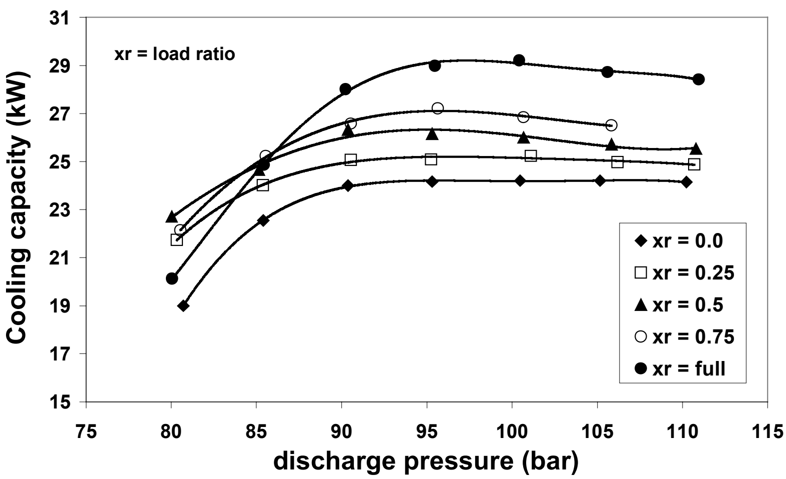


Figure 5-3 Variation of cooling load with load ratio as parameter
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

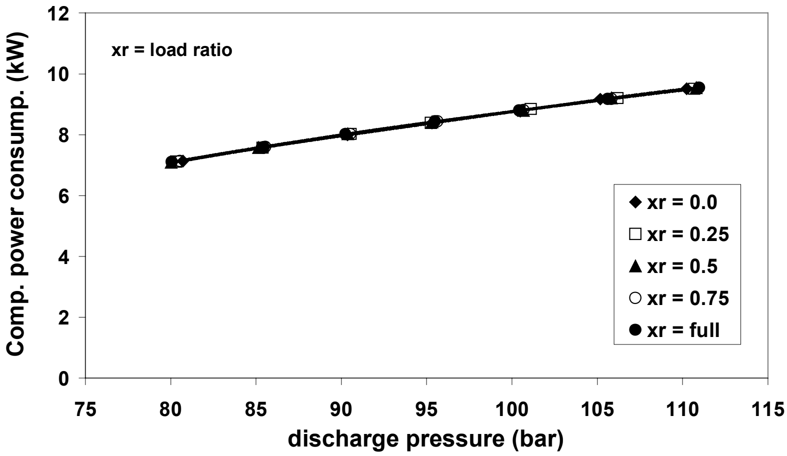


Figure 5-4 Variation of compressor shaft power at various load ratios
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

From Figure 5-3 and Figure 5-4, it can be seen that higher cooling load can be achieved at all load ratios while compressor shaft power at the same discharge pressure was the same for all load ratios. Tabel 5-3 below shows the value of the optimum cooling load and compressor shaft power at various load ratios. At discharge pressure of 80 bar, the cooling capacity of full recovery mode was lower than the others load ratio but still higher than that of without heat recovery.

Table 5-4 Optimum cooling load and compressor shaft power at various load ratios

Load ratio	cooling load (kW)	%	Comp. (kW)	%
0.00	23.6	0.00	7.8	0.00
0.25	23.8	0.85	7.5	-3.85
0.50	25.0	5.93	7.5	-3.85
0.75	25.9	9.75	7.7	-1.28
Full recovery	28.3	19.92	8.0	2.56

A lower compressor shaft power is needed if heat recovery is applied up to 0.75 load ratio but at full recovery mode it is higher than that of the system without heat recovery.

5.1.3 Effect Of Evaporation Temperature

The purpose of running experiment on different evaporation temperatures was to gain understanding if the system should be designed for different type of evaporators. For water-chiller system with 4°C supply water temperature and 11°C return water temperature, 0°C evaporation temperature is representative. A higher evaporation temperature can be expected for an air-conditioning system with an air cooler.

Figure 5-5 shows cooling-COP at various discharge pressures with evaporation temperature as parameter. As can be expected, higher cooling-COPs were obtained at higher evaporation temperatures except at 80 bar discharge pressure. At 80 bar discharge pressure, the cooling-COP at 10°C evaporation temperature was about the same as the one at 0°C evaporation temperature.

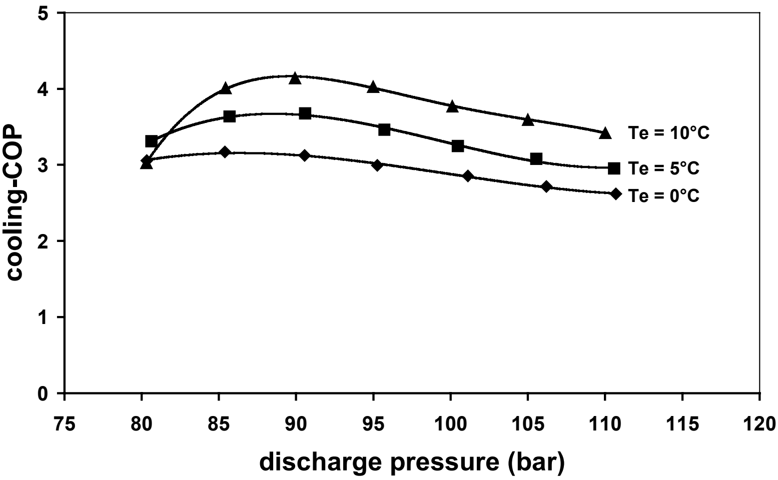


Figure 5-5 Influence of evaporation temperature to cooling-COP
[T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr = 0.25]

The optimum pressure increases as the evaporation temperature increases. Table 5-4 shows the cooling-COP and discharge pressure at the optimum conditions.

Table 5-5 Optimum cooling load and compressor shaft power at various evaporation temperatures

Evaporation Temp.	Popt.	Cooling-COP
0	85.42	3.16
5	88.19	3.69
10	89.68	4.16

Figure 5-6 shows cooling capacity for the same operating pressure mentioned above. At 80 bar discharge pressure, cooling load at 10°C evaporation temperature was lower than that at 0°C and 5°C one, and then increases rapidly as pressure increases.

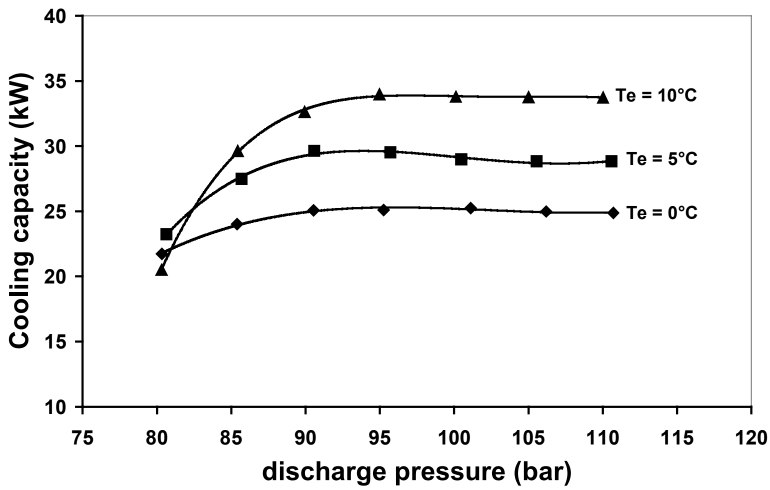


Figure 5-6 Cooling capacity at various evaporation temperatures
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr= 0.25]

5.1.4 Effect of Cooling Medium Temperature

In this experiment, the ambient air temperature was represented by the inlet water temperature to the heat rejecting heat exchanger. Strong influence of cooling medium temperature on system performance is a characteristic of transcritical cycle as can be seen in Figure 5-7 below. At air-conditioning mode, cooling-COP falls from 3.0 to 2.5 at 30°C and 35°C inlet water temperature, respectively. The optimum pressure shift to a higher value as inlet water temperature increase, that is 87 bar at 30°C to 97 bar at 35°C.

Heat recovery effect is clearly shown in the Figure 5-7. It reduces the optimum pressure and shifts the optimum pressure to a lower value. The shifting of the optimum pressure is clearer at higher cooling medium temperature. At 30°C cooling medium temperature, the optimum pressure shifted from 87 bar to 85 bar while at 35°C it shifted from 97 bar to 90 bar.

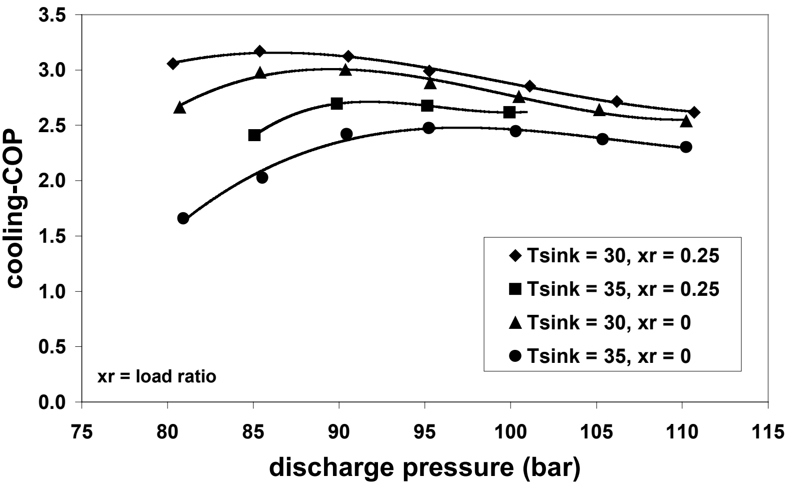


Figure 5-7 Influence of cooling medium temperature to cooling-COP [T_{evap} = 0°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

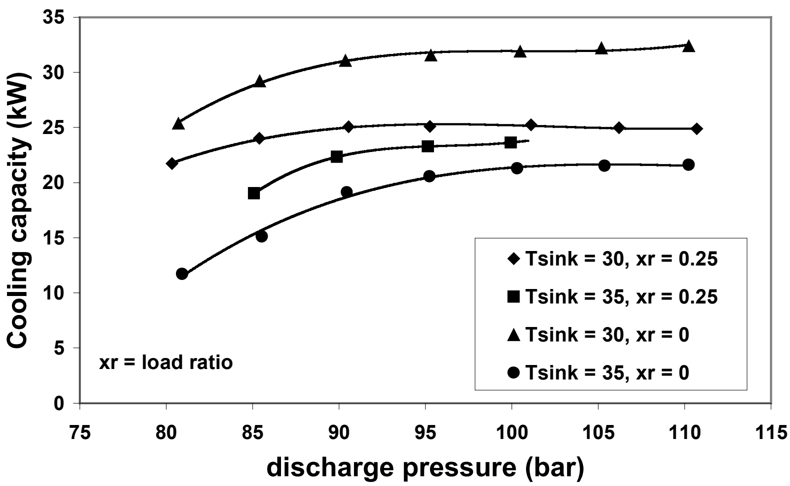


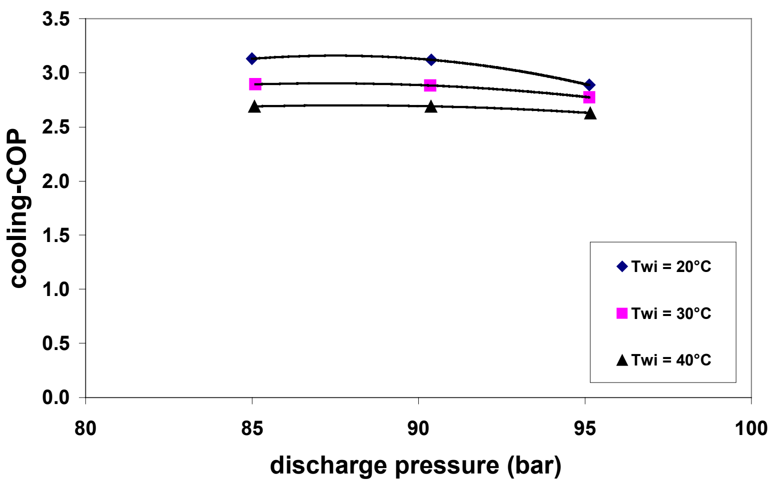
Figure 5-8 Cooling capacity at two cooling medium temperatures and load ratios [T_{evap} = 0°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

Cooling capacity decreases as cooling medium temperature increases (Figure 5-8). It can also be observed that performing heat recovery at higher cooling medium temperature will increase cooling capacity more than at lower cooling medium temperature.

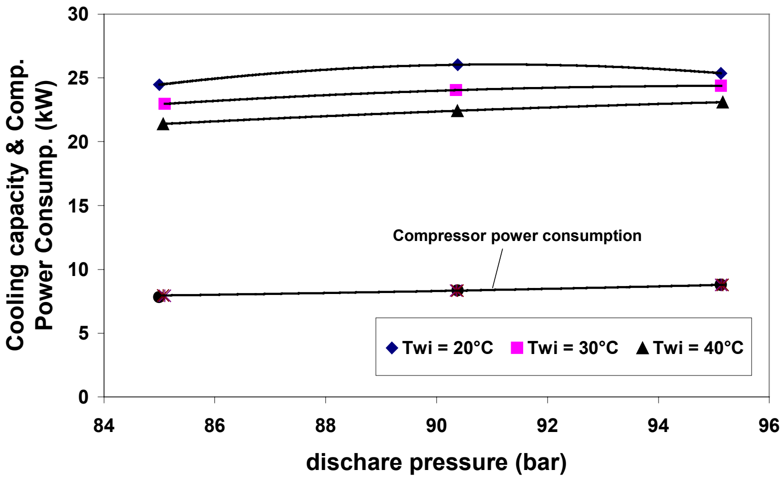
5.1.5 Effect of Inlet Water Temperature

When hot water system is of hot storage type then the water will be circulated around the water heating heat exchanger loop. Water temperature in the storage tank will gradually increase with time and at some time it will become higher than the ambient air temperature and the effect is that the cooling performance will become lower. Therefore, it is important to see how the inlet water temperature to the water heating heat exchanger will affect the system performance.

Figure 5-9a shows variation of cooling-COP and cooling capacity as the inlet water temperature to the water heating heat exchanger varies. It is clearly demonstrated that at the same discharge pressure, as the inlet water temperature increases both the cooling-COP and the cooling capacity decrease. The compressor shaft power at the same discharge pressure are basically the same for all inlet water temperature as shown in Figure 5-9b.



(a)



(b)

Figure 5-9 Effect of inlet water temperature to the system performance Cooling-COP (b) Cooling capacity & Compressor power consumption [T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr = 0.25]

5.1.6 Effect of Hot Water Temperature

Hot water temperature also has an influence on overall system performance. Experimental results showing the effect of hot water temperature to the system performance is depicted in Figure 5-10 for two different hot water temperatures. Increasing hot water temperature results on decreasing cooling-COP and shifts the optimum pressure to a higher value.

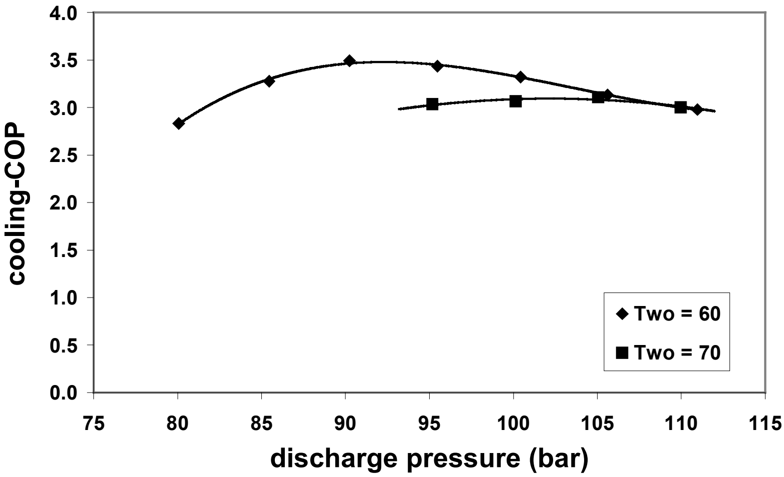


Figure 5-10 Influence of hot water temperature on cooling-COP
[T_{evap} = 0°C, T_{w_in} = 20°C, xr = full]

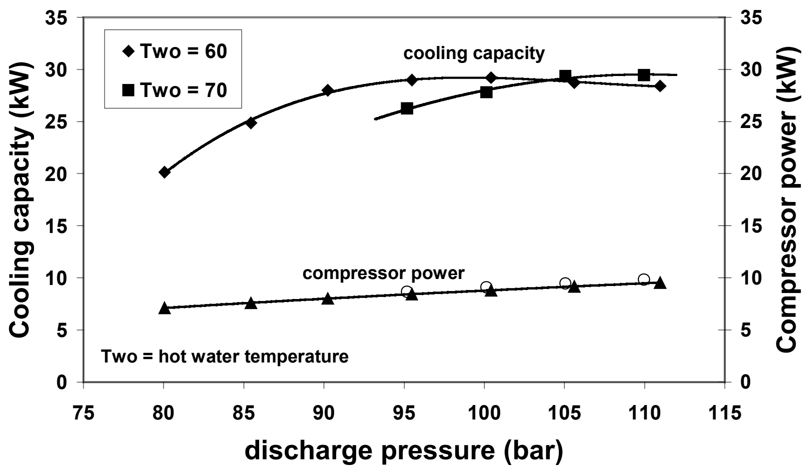


Figure 5-11 Influence of hot water temperature on system performance
[T_{evap} = 0°C, T_{w_in} = 20°C, xr = full]

From Figure 5-10 it can also be seen that the curve become flatter at higher hot water temperature compared to that at lower hot water temperature. For 60°C hot water temperature, cooling-COP reaches a value of 3.45 at 92 bar discharge pressure and drops to 3.10 at 104 bar discharge pressure for 70°C hot water temperature. The drop in cooling-COP can be seen more clearly from Figure 5-11. Here cooling capacity together with compressor power are plotted both for

60°C and 70°C hot water temperatures. As can be seen from this figure, the compressor power at the optimum pressure is lower for the lower hot water temperature and gives slightly lower cooling capacity compared to that for the higher temperature. For 60°C hot water temperature, the compressor power and cooling load at optimum conditions are 8.2 kW and 28.6 kW, respectively, while for 70°C hot water temperature, they are 9.4 kW and 28.9 kW, respectively.

5.2 System Performance With And Without Internal Heat Exchanger

In this subchapter, experimental results when the test rig running without internal heat exchanger are presented. There are three different operating conditions of the experiment that have been performed without internal heat exchanger, i.e. air-conditioning mode without heat recovery, combined mode at load ratio of 0.25 and full heat recovery mode at two different inlet water temperatures. Each operating condition is compared with the system having internal heat exchanger so that the effect of internal heat exchanger can be seen clearly.

5.2.1 Operating at Air-conditioning Mode

Figure 5-12 and Figure 5-13 shows the experimental results for the test rig with and without internal heat exchanger at two different cooling medium temperatures, 30°C and 35°C. At both cooling medium temperatures, the optimum cooling-COPs depend strongly on the presence of the internal heat exchanger where the cooling-COP is higher for the system with internal heat exchanger than the one without.

5.2.2 Operating at Combined Mode

When heat recovery is employed, the system performance will be affected by the present of the internal heat exchanger as shown on Figure 5-14. The cooling-COP is higher and the location of the optimum pressure was shifted to a lower value for the system with internal heat exchanger. For the operating condition shown in the figure, the optimum pressure for the system with internal heat exchanger is 85.4 bar giving 3.17 cooling-COP while for the one without was at 90 bar with 3.0 cooling-COP. Notice that the cooling-COP curve of the system with internal heat exchanger is flatter around the optimum point compared to the one without.

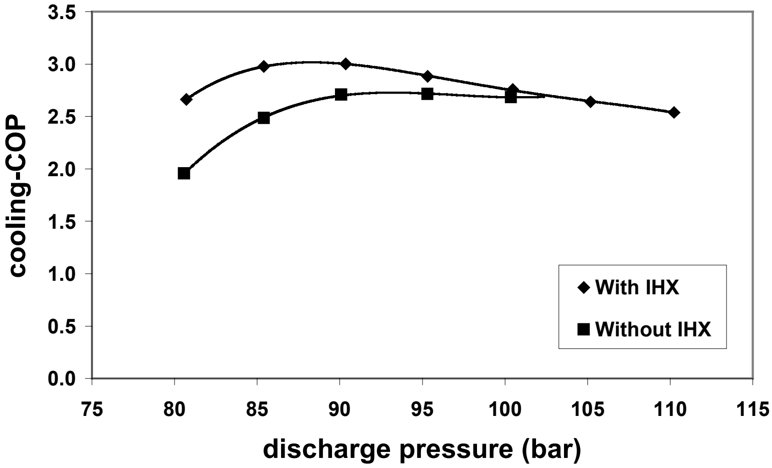


Figure 5-12 Influence of internal heat exchanger at 30°C cooling medium temperature [T_{evap} = 0°C, T_{sink} = 30°C, A/C mode]

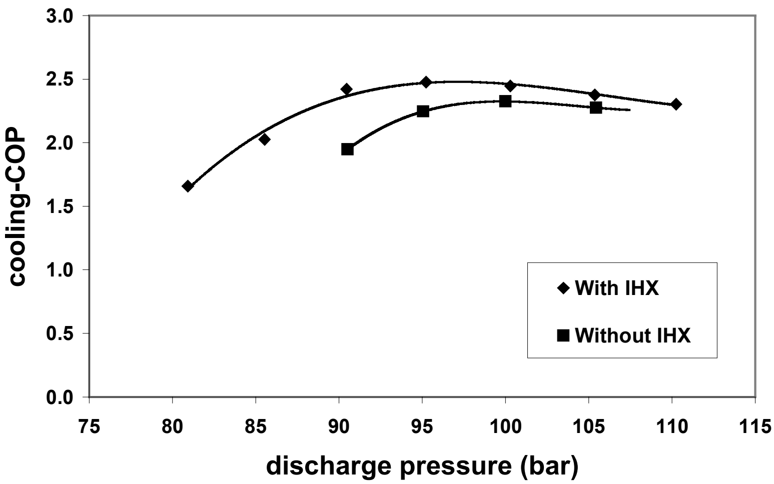


Figure 5-13 Influence of internal heat exchanger at 35°C cooling medium temperature [T_{evap} = 0°C, T_{sink} = 35°C, A/C mode]

Figure 5-15 shows that the cooling capacity of the system with internal heat exchanger at 85.4 bar is larger than that without internal heat exchanger, which is 24 kW compared to 22.8 kW. The compressor shaft power also slightly higher for the system without internal heat exchanger as can be seen in the lower part of Figure 5-15.

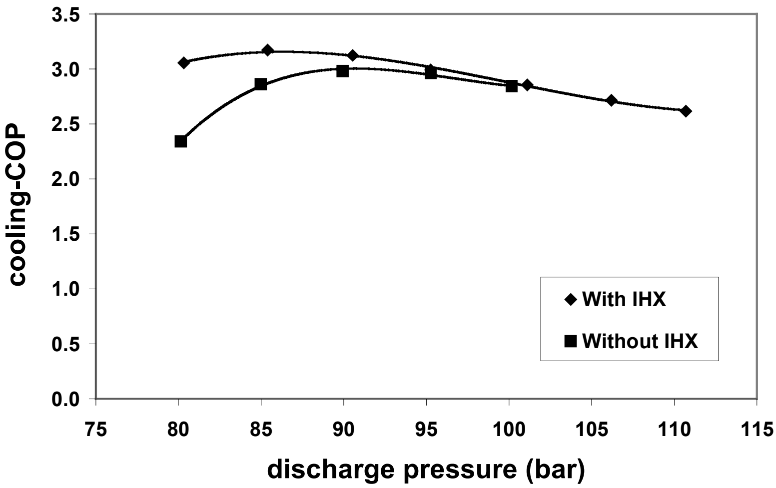


Figure 5-14 Influence of internal heat exchanger on cooling-COP ratio
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr = 0.25]

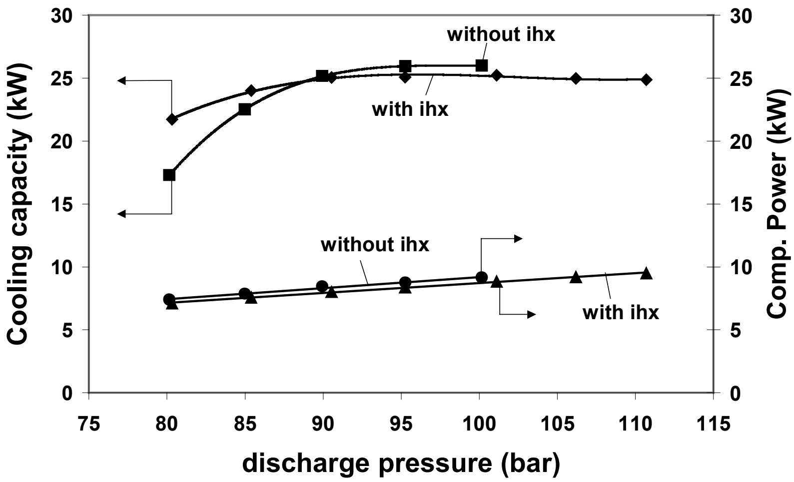


Figure 5-15 Influence of internal heat exchanger on system performance
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr = 0.25]

5.2.3 Operating at Full Recovery Mode

The last experiment was about the influence of inlet water temperature to the system performance. Referring to Figure 5-16 below, the shape of the cooling-COP curves of the system without internal heat exchanger were not affected by the change in inlet water temperature. The location of the optimum conditions for 60°C hot water temperature were about the same at 90 bar, but the optimum cooling-COP was lower with increasing inlet water temperature as the case of the system with internal heat exchanger. At 20°C inlet water temperature, the optimum cooling-COP was 3.0 and it was 2.8 at 30°C inlet water temperature. The cooling capacity became lower with increasing inlet water temperature. As shown in Figure 5-17, at 90 bar discharge pressure the cooling capacity at 20°C inlet water temperature was 25 kW and it fell to 23.3 kW at 30°C inlet water temperature. The compressor shaft power was not affected by the change of inlet water temperature for the same discharge pressure.

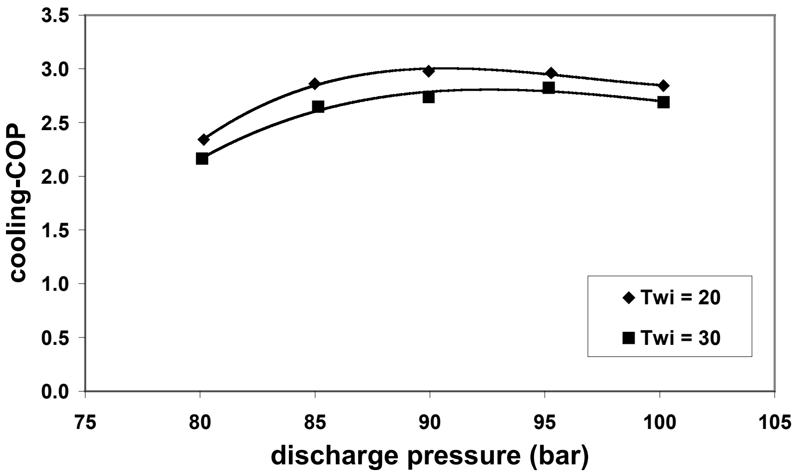


Figure 5-16 Influence of inlet water temperature on cooling-COP without Internal Heat Exchanger
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_out} = 60°C, xr = 0.25]

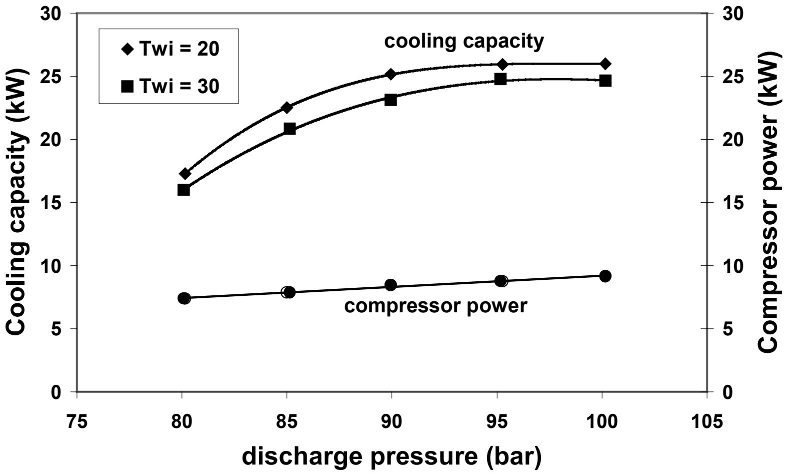


Figure 5-17 Influence of inlet water temperature to the system performance without Internal Heat Exchanger
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_out} = 60°C, xr = 0.25]

5.3 Effect Of Gas Coolers Configuration

Owing to large capacity of the water heating heat exchanger, the test rig could not be run as a combined system with series configuration of the gas coolers. To investigate the effect of gas coolers configuration, the simulation program were run for the series configuration under different discharge pressures.

The effect of the gas coolers configuration on cooling-COP is shown in Figure 5-18. As seen in this figure, the optimum cooling-COP of the series configuration increased slightly to 3.07 compared to 3.01 of the system without heat recovery. The optimum pressure also changed slightly from 87.9 for the system without heat recovery to 87.3 for the series configuration. In parallel configuration, the optimum cooling-COP increase to 3.24 at load ratio of 0.25 and the optimum pressure shifted to a lower value of 86.2 bar. The shape of the curves was not affected by the gas coolers configuration as can bee seen in this figure.

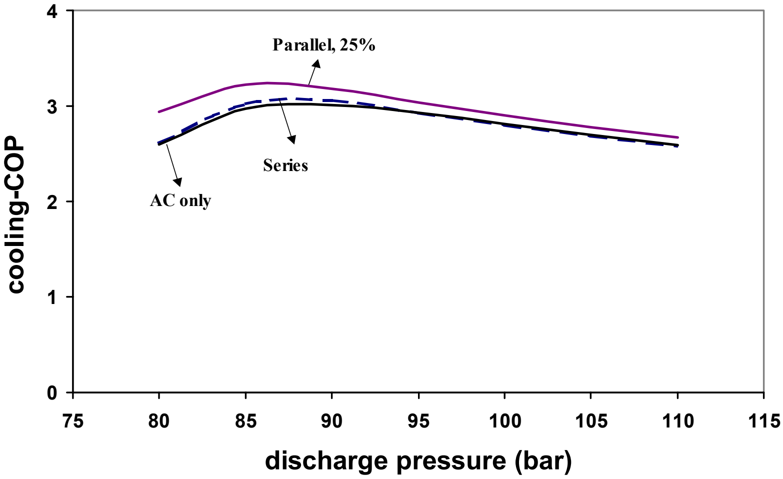


Figure 5-18 Cooling-COP of the A/C only, series, and parallel configuration [T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

The series configuration did not change the cooling capacity appreciably. At the optimum pressure, the cooling capacity of the series configuration was 23.6 kW as compared to cooling capacity of the system without heat recovery of 23.3 kW, while the cooling capacity of the parallel configuration at optimum discharge pressure was 24.6 kW. Figure 5-19 shows variation of cooling capacities at various discharge pressures and different gas coolers configurations. The compressor powers were the same for all configuration since in the simulation it was only a function of pressure ratio.

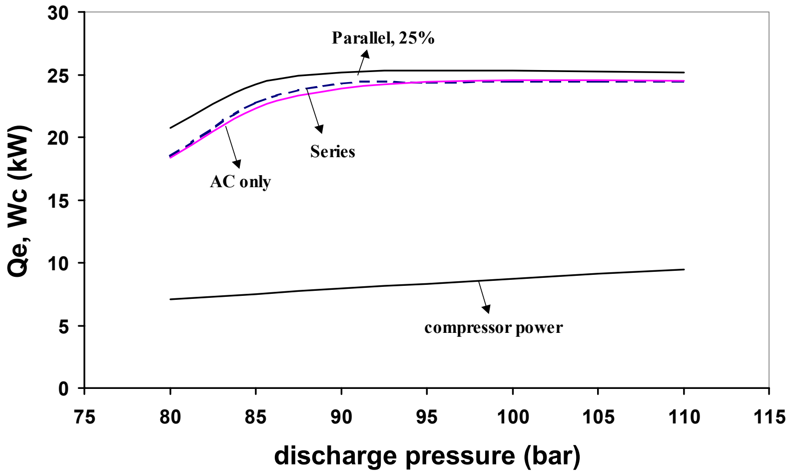


Figure 5-19 Cooling capacity and compressor shaft power
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

5.4 Total-COP

In term of overall performance index for a combined air-conditioning and water heating system, the utilization of rejected heat of the air-conditioning system for water heating should be taken into account when calculating the coefficient of performance. There are two limits for the total-COP, the lowest limit when there is no heat recovery and the highest limit when all rejected energy is utilized. Between these limits, the total-COP will vary depend on the percentage of heat recovery. Figure 5-20 shows total-COP as a function of discharge pressure at various heat recovery ratios.

As can be expected, the total-COP increase as heat recovery ratio increases. The trend of the curves were not affected by heat recovery ratios and it can be seen that the optimum discharge pressure were almost the same as the air-conditioning mode for all heat recovery ratios except for full recovery mode. The optimum discharge pressure and the optimum total-COP are given in Table 5-5.

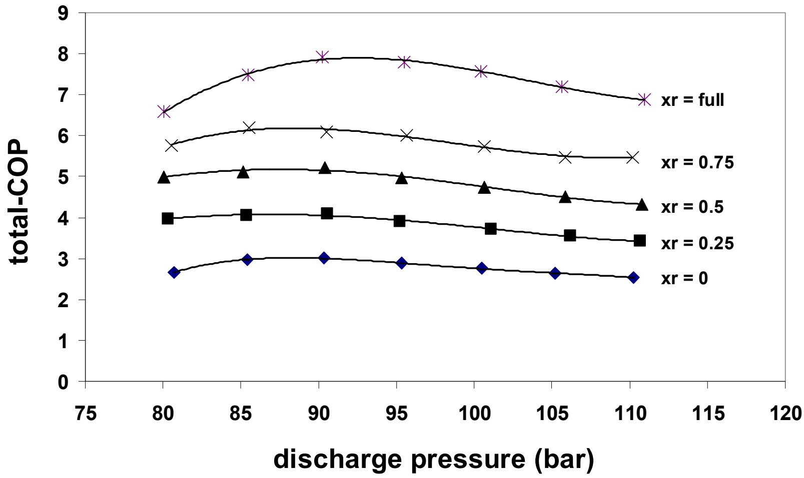


Figure 5-20 Total-COP at various heat recovery ratios
[T_{evap} = 0°C, T_{air} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

Table 5-6 Optimum discharge pressure and total-COP

Recovery ratio	Discharge Pressure (bar)	total-COP
0.00	88.1	3.00
0.25	88.0	4.06
0.50	88.0	5.18
0.75	87.9	6.20
1.00	92.3	7.92

5.5 Comparison of The Experimental and Modeling Results

The experimental results in this work were used to validate the system modeling of the combined air-conditioning and water heating system as explained in Chapter 3. Compressor, water heating heat exchanger, heat rejecting heat exchanger and internal heat exchanger model was validated with respect to its characteristic. Since the compressor was modeled as a black box system, its characteristics were represented by isentropic and volumetric efficiency. Capacity, temperature and pressure at the outlet of the gas coolers were used to validate the models, while due to the absence of pressure measurement only capacity and temperature at outlet of the internal heat exchanger were used as its validation parameters.

5.5.1 Component Validation

5.5.1.1 Compressor

A compressor is the heart of a refrigerating system and it is the most difficult component to be modeled. This is because it contains moving part compared to other components, which are static. As has been explained in Chapter 3, the compressor of the test rig has been modeled as a black box which is the simplest model and its characteristic are represented by isentropic and volumetric efficiency.

Figure 5-21 and Figure 5-22 show experimental data for the compressor covering almost all the experimental points performed in this work. The line on the figures is curve fitting lines obtained from previous work with the same compressor [Zakeri et al., 1999]. The figures on the low side are the deviation of the efficiencies from the experimental points. As can be observed from these figures, most of the data are scattered and 90% of its are lain within ±5% deviation lines for both isentropic and volumetric efficiency. The coefficient of the efficiency equations of the compressor can be found in Chapter 3.

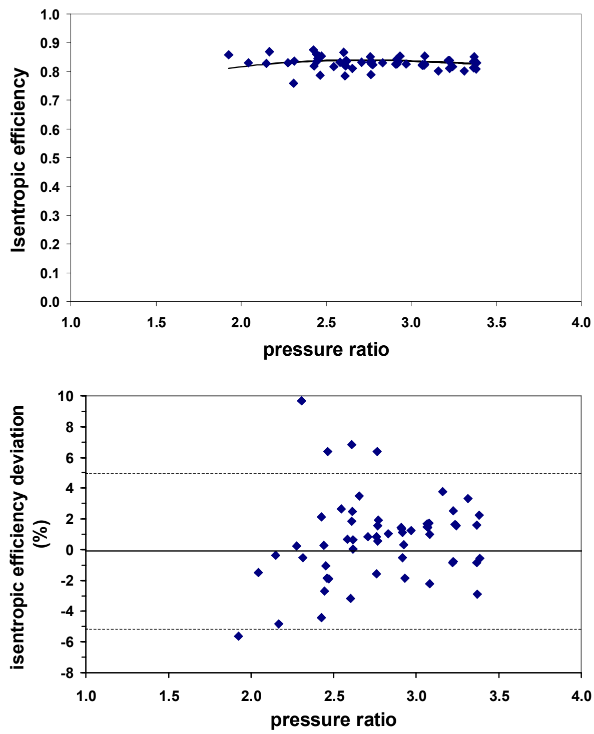


Figure 5-21 Isentropic efficiency

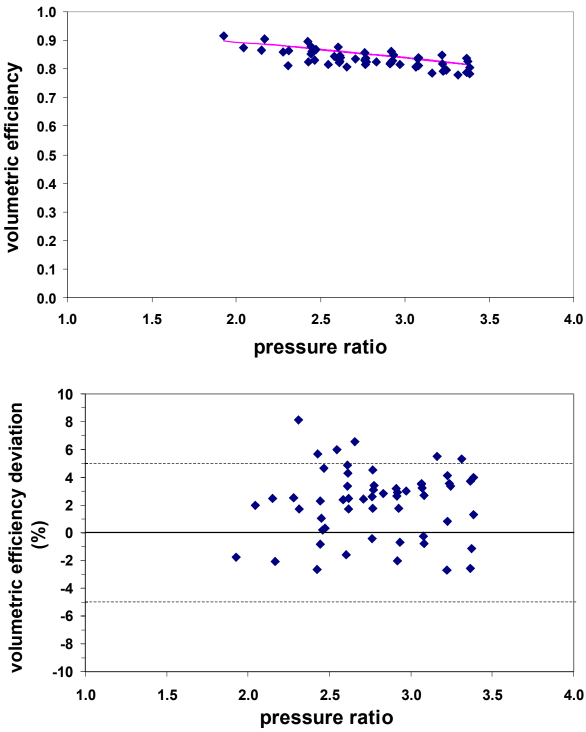


Figure 5-22 Volumetric efficiency

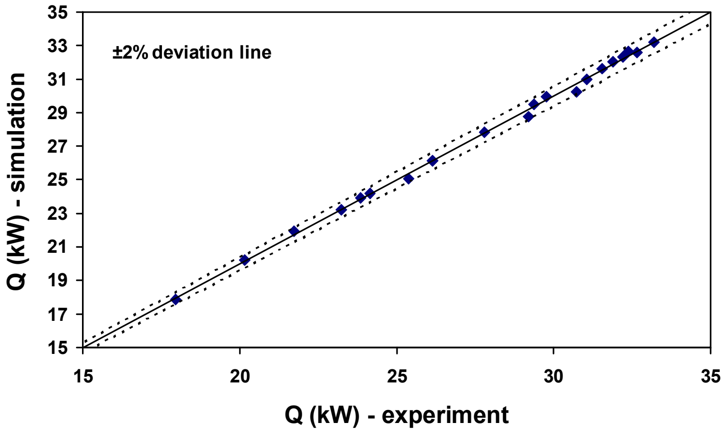
5.5.1.2 Heat Rejecting Heat Exchanger

The heat rejecting heat exchanger model explained in Chapter 3 gives capacity, temperature and pressure at the outlet of the gas cooler as its outputs. Temperature and pressure measurement point were installed only at the main inlet and outlet of the heat exchanger so that there was no information on the working fluid distribution flowing into each section of the heat rejecting heat exchanger. In the model, it was assumed a uniform streams distribution.

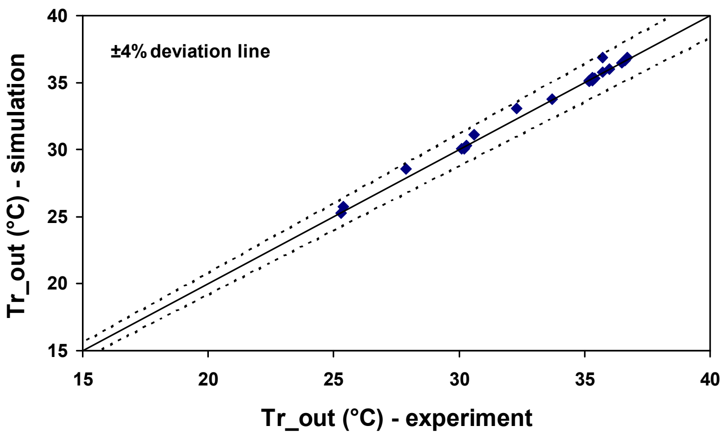
Figure 5-23 shows comparison of the model and experimental results. The agreements are listed in Table 5-6. The agreements are good enough considering the assumption of pure refrigerant and heat transfer correlation used in the model.

Table 5-7 Deviation of the model from the experimental data

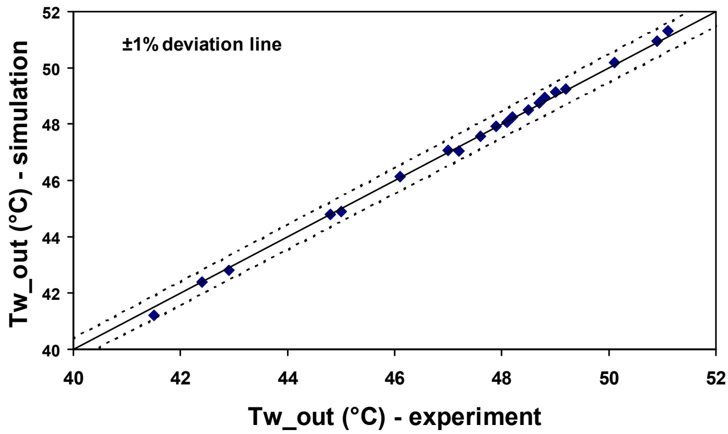
Quantity	**Deviation**
Capacity	±2%
CO_2 outlet temperature (°C)	±4%
CO_2 outlet pressure (bar)	±0.3%
Water outlet temperature (°C)	±1%


±2% deviation line
Q (kW) - simulation
Q (kW) - experiment
15
17
19
21
23
25
27
29
31
33
35
15
20
25
30
35

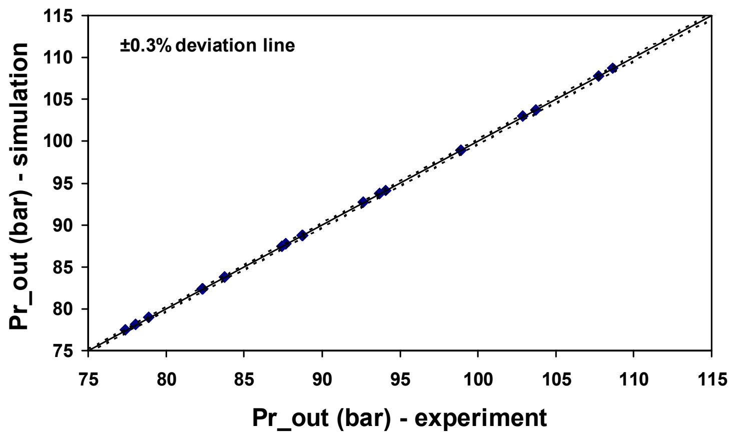
(a)


±4% deviation line
Tr_out (°C) - simulation
Tr_out (°C) - experiment
15
20
25
30
35
40
15
20
25
30
35
40

(b)



(c)



(d)

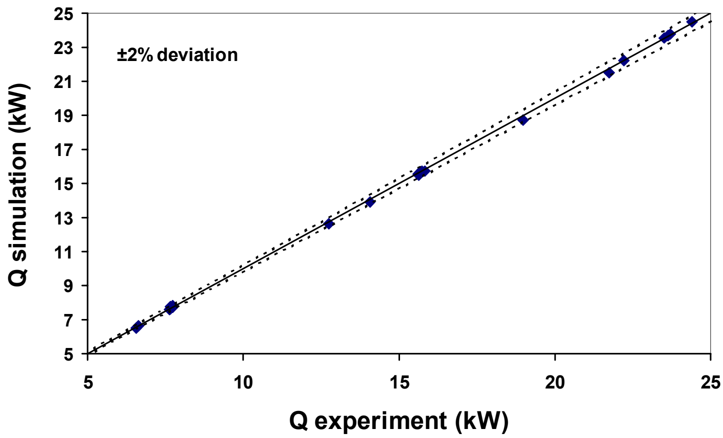
Figure 5-23 Validation of the heat rejecting geat exchanger model

5.5.1.3 Water Heating Heat Exchanger

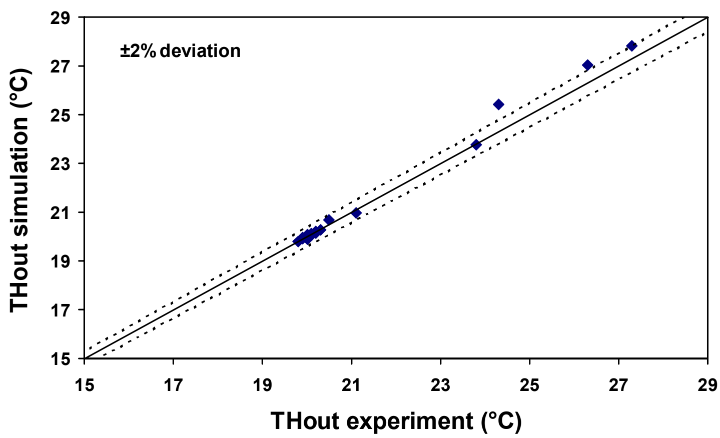
As in the heat rejecting heat exchanger model, the quantities which are verified for the water heating heat exchanger model are: capacity, CO_2 temperature and pressure at the outlet of the gas cooler, and water temperature at the outlet of the gas cooler. The heat exchanger consist of only one section so that it could be expected that pressure drop across the heat exchanger will be higher for the experimental data compared to the model because of the assumption of pure CO_2 in the model. The validation can be observed from Figure 5-24. Table 5-7 gives percent deviation for each parameter.

Table 5-8 Deviation of the water heating heat exchanger model from the experimental data

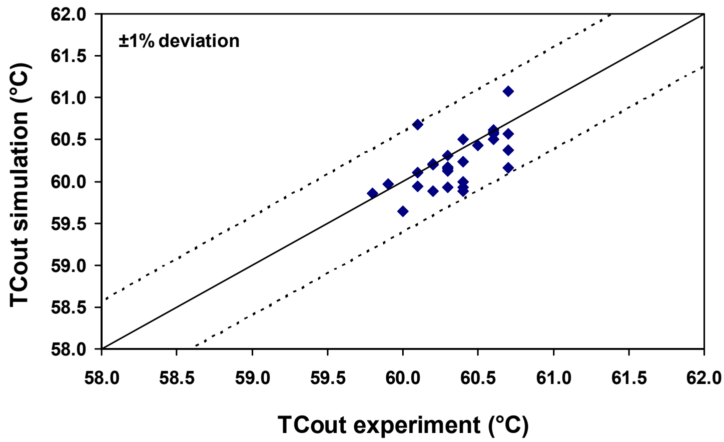
Quantity	**Deviation**
Capacity	±2%
CO_2 outlet temperature (°C)	±2%
CO_2 outlet pressure (bar)	±0.5%
Water outlet temperature (°C)	±1%



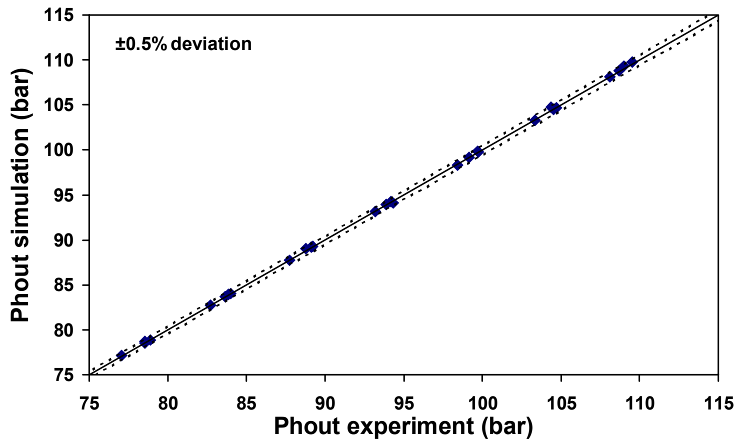
(a)



(b)



(c)

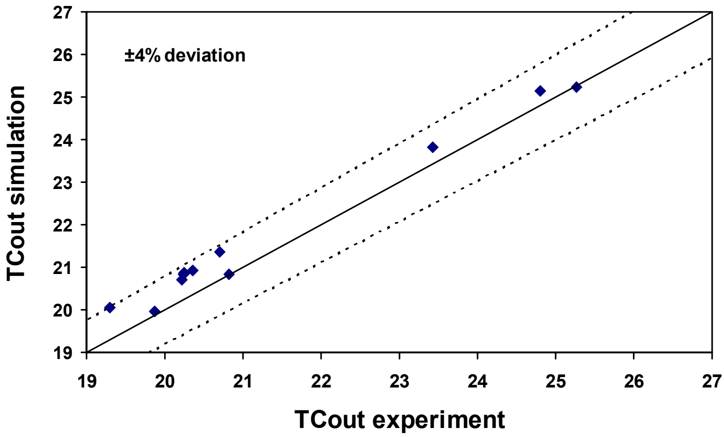


(d)

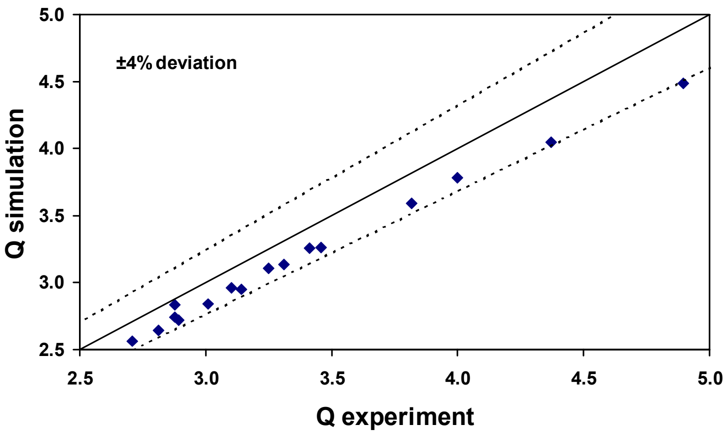
Figure 5-24 Validation of the water heating heat exchanger model

5.5.1.4 Internal heat exchanger

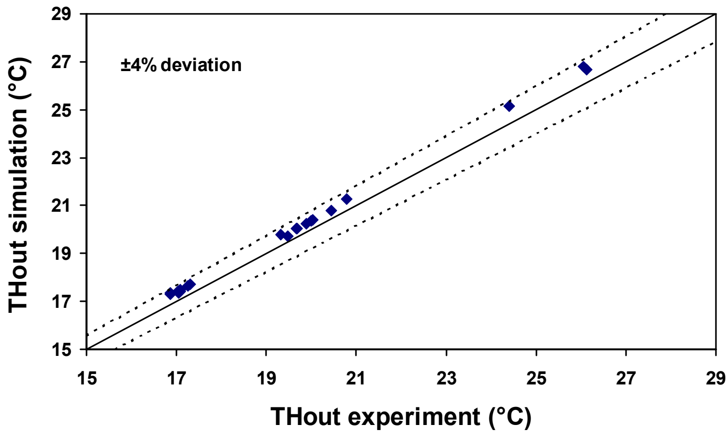
During experiments, part of oil and liquid CO_2 mixture was drained from the liquid receiver and fed to the low-pressure side of the internal heat exchanger. A lower pressure drop from the simulation could be expected compared to the experimental data. Furthermore, validation of the pressure drop could not be performed due to the absence of pressure measurement for the internal heat exchanger in the test rig. However, because the tube length of the internal heat exchanger was 6 m it can be expected that the pressure drop across the internal heat exchanger in the high-side and low-side pressure would affect the system performance insignificantly. Therefore, the parameters that are verified are only the capacity and the outlet temperatures in both the high-pressure and the low-pressure side. As can be seen Figure 5-25, all quantities are agreed within ±4% with experimental data. The capacities of the model were lower for all comparison points while the outlet temperatures of the high-side and low-side pressures of the model were higher.



(a)



(b)



(c)

Figure 5-25 Validation of the internal heat exchanger model

5.5.2 Performance comparison

In this section, the system performance from the experimental results and those from the simulation results are compared. Two main parameters that are important to be observed are cooling-COP and cooling capacity. Since heat recovery to heat water ideally should not disturb any operation of the air-conditioning unit, these two parameters can be used as indications. Therefore, comparison are focused on these parameters.

Figure 5-26 shows comparison of the experimental results with the simulation results. Discharge pressure ranges from 80 bar to 110 and it run in air-conditioning mode without heat recovery. From this figure, it can be seen that good agreement between experimental and simulation results were achieved where the cooling-COP stayed within ±4% deviation line. Location of the optimum point can also be determined from the simulation. For this run, the optimum discharge pressure was around 87 bar with cooling-COP of 3.0.

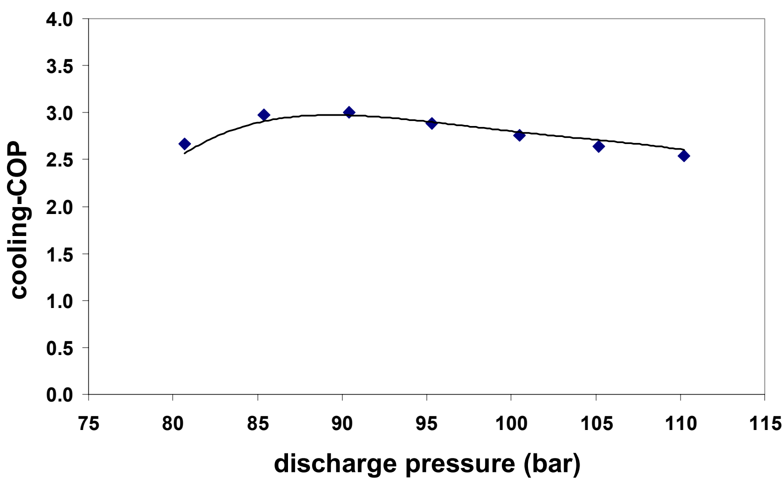
The second comparison was shown from Figure 5-27 to Figure 5-29 where the system run in combined mode with 25%, 50%, and 75% heat recovery. Except at the lowest pressure of 80 bar, the simulation results agreed with the experimental results and stayed within ±2% deviation line for 25% heat recovery. With inclusion of the point at 80 bar, the deviation still within ±4% which is a good agreement.

Predicted cooling-COPs at 25% heat recovery were in good agreement with that of experimental results as can be seen in Figure 5-27. The agreement between simulation and experimental data stayed within ±2% and location of optimum pressure could be determined as in case of the system without heat recovery.

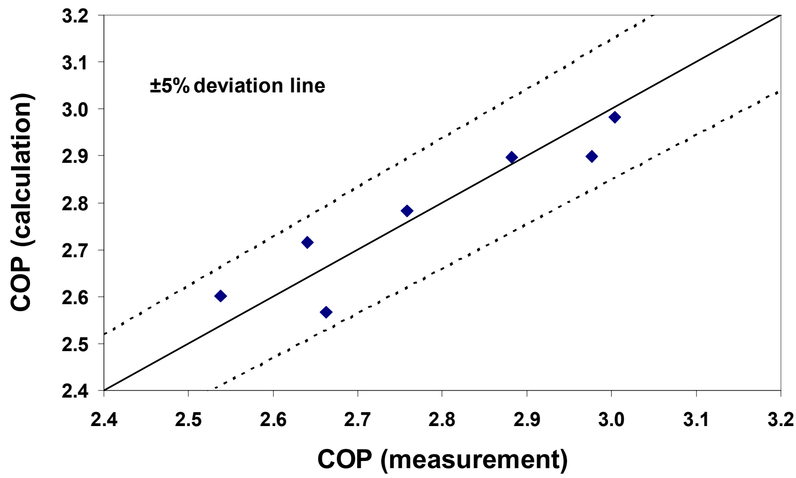
Figure 5-28 shows for 50% heat recovery where predicted cooling-COPs also in good agreement with the experimental data within ±2% while cooling capacity for pressure range of 85-110 bar, deviation of the predicted value lain within ±3% and for a lower pressure of 80 bar it deviated within ±6%. Note that for all operating pressures in this experiment, predicted cooling capacity were lower than that of experimental value as shown in Figure 5-28.

At 75% heat recovery, predicted cooling-COPs and cooling capacities were ±4% of experimental data as can be seen in Figure 5-29.

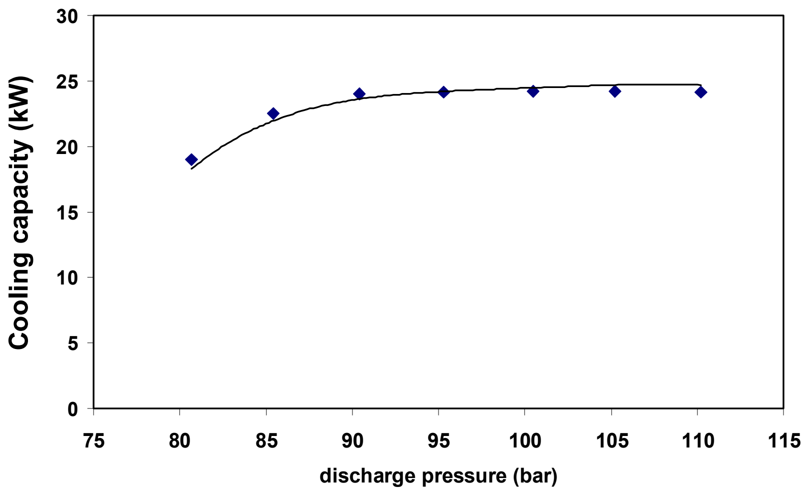
As in the case of 50% heat recovery, Figure 5-30 shows a lower predicted cooling capacity compared to the experimental data for the system running in full recovery mode. Furthermore the predicted cooling-COP were also lower for entire pressure range of 80 bar to 110 bar. From this figure it can also be observed that the simulation program was able to determined the location of the optimum pressure and the trend of cooling capacity curve was similar as compared to experimental data. The deviation were ±4% for cooling-COP and were ±7% for cooling capacity.



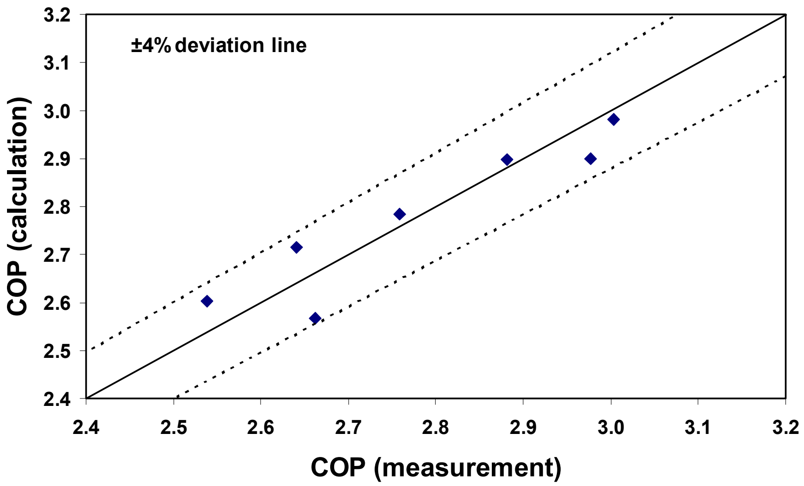
(a)


±5% deviation line
COP (calculation)
COP (measurement)
3.2
3.1
3.0
2.9
2.8
2.7
2.6
2.5
2.4
2.4
2.5
2.6
2.7
2.8
2.9
3.0
3.1
3.2

(b)

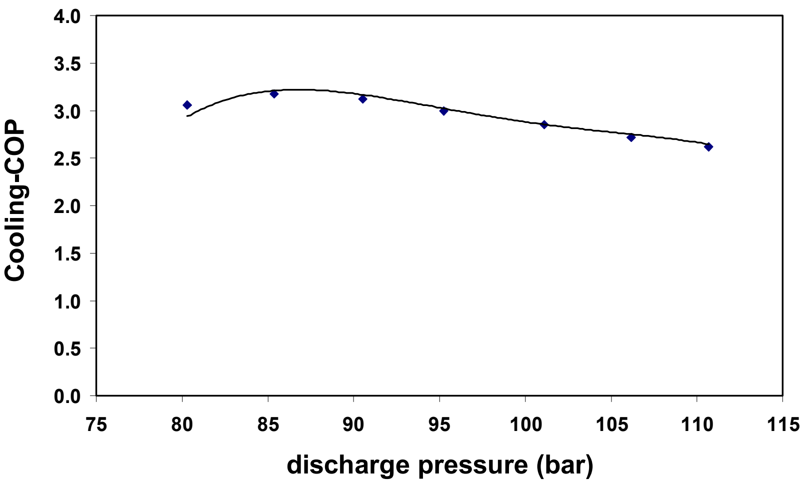

Cooling capacity (kW)
discharge pressure (bar)
30
25
20
15
10
5
0
75
80
85
90
95
100
105
110
115

(c)

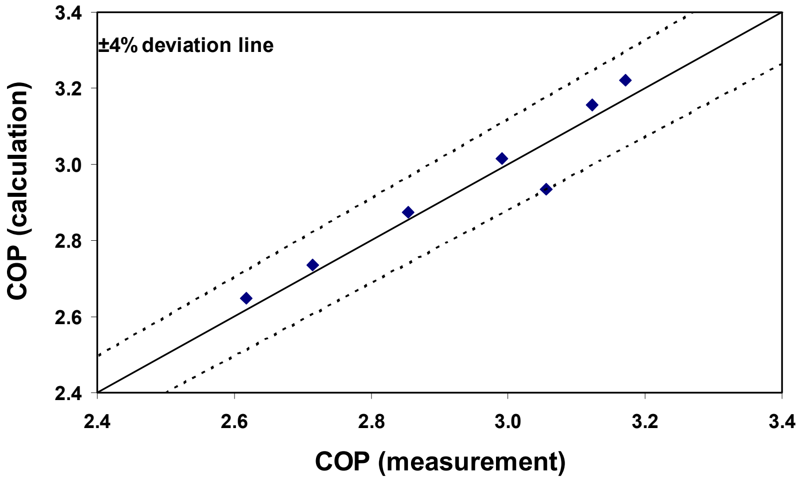


(d)

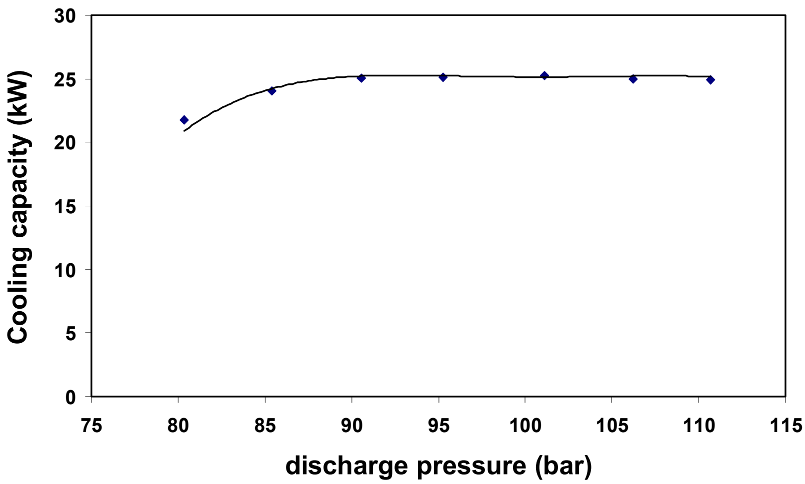
Figure 5-26 Cooling capacity comparison at evaporation temp. : 0°C, cooling medium temp. : 30°C, air-conditioning mode



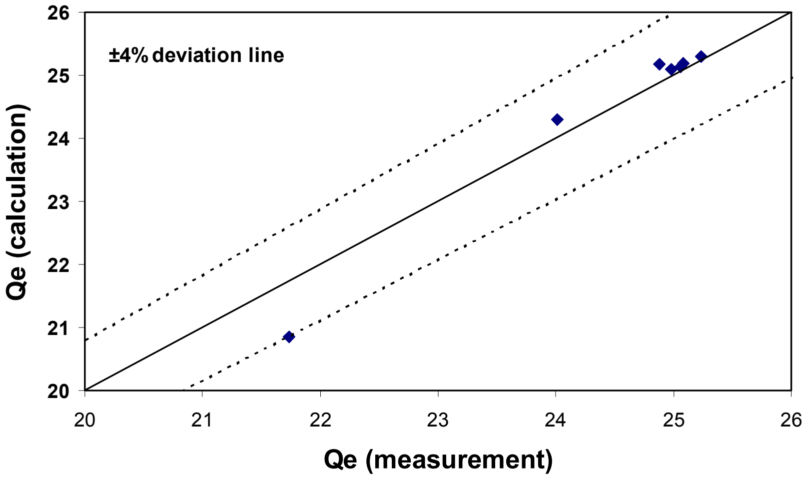
(a)


±4% deviation line
COP (calculation)
COP (measurement)
3.4
3.2
3.0
2.8
2.6
2.4
2.4
2.6
2.8
3.0
3.2
3.4

(b)

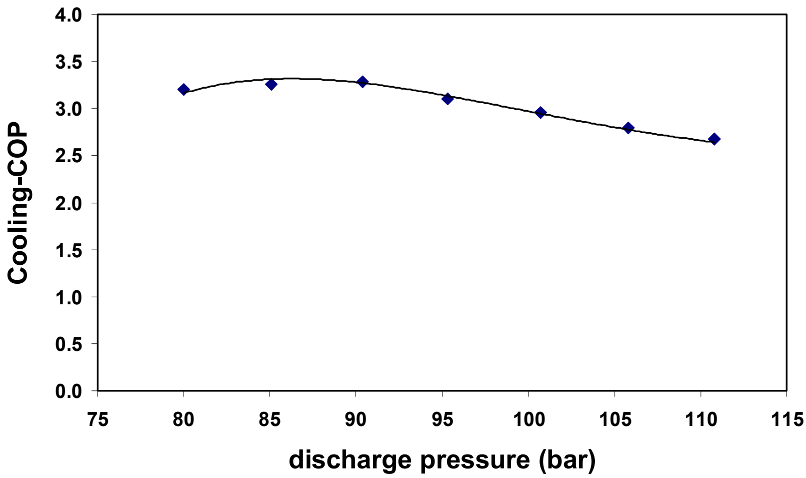

Cooling capacity (kW)
discharge pressure (bar)
30
25
20
15
10
5
0
75
80
85
90
95
100
105
110
115

(c)

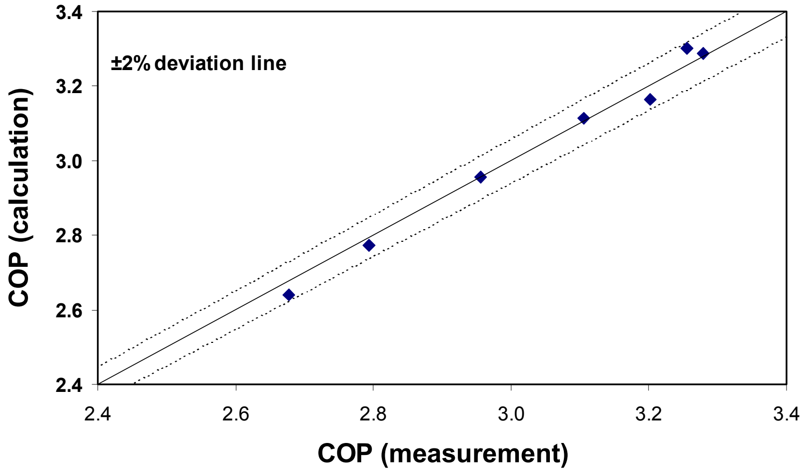


(d)

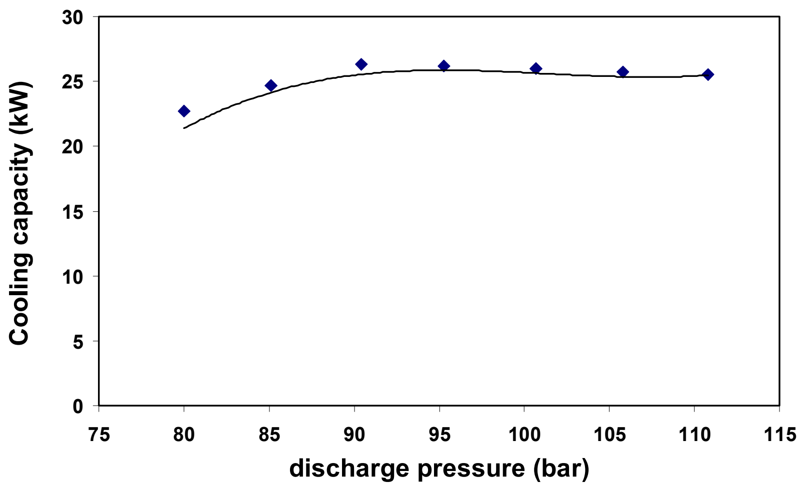
Figure 5-27 Performance comparison at evap. temp = 0°C, cooling medium temp. = 30°C, inlet water temp. = 20°C, hot water temp. = 60°C, 25% recovery



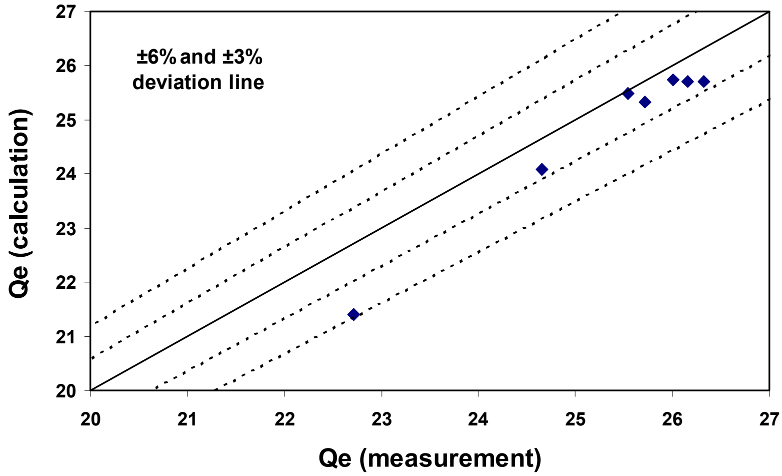
(a)


±2% deviation line
COP (calculation)
COP (measurement)
2.4
2.6
2.8
3.0
3.2
3.4

(b)

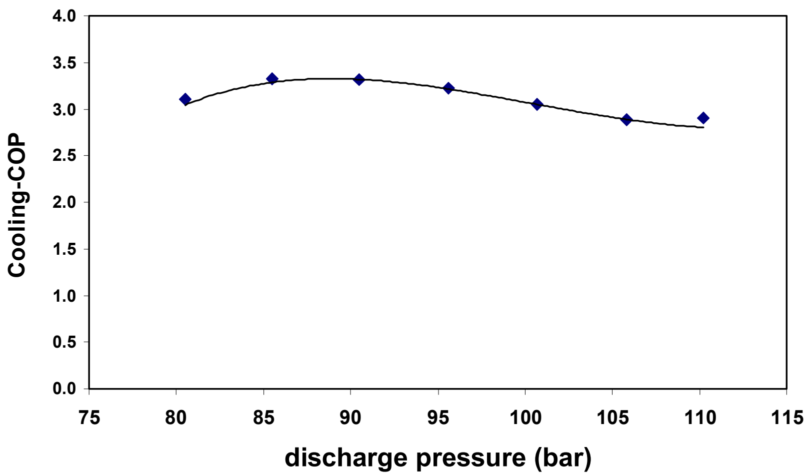

Cooling capacity (kW)
discharge pressure (bar)
0
5
10
15
20
25
30
75
80
85
90
95
100
105
110
115

(c)

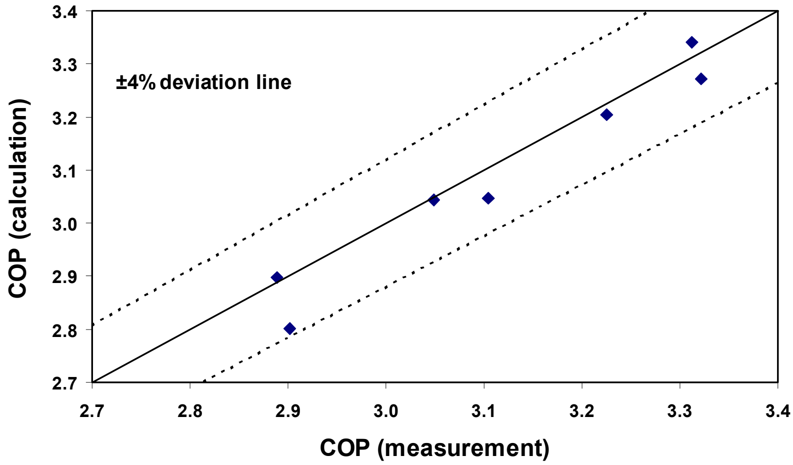


(d)

Figure 5-28 Performance comparison at evap. temp = 0°C, cooling medium temp. = 30°C, inlet water temp. = 20°C, hot water temp. = 60°C, 50% recovery

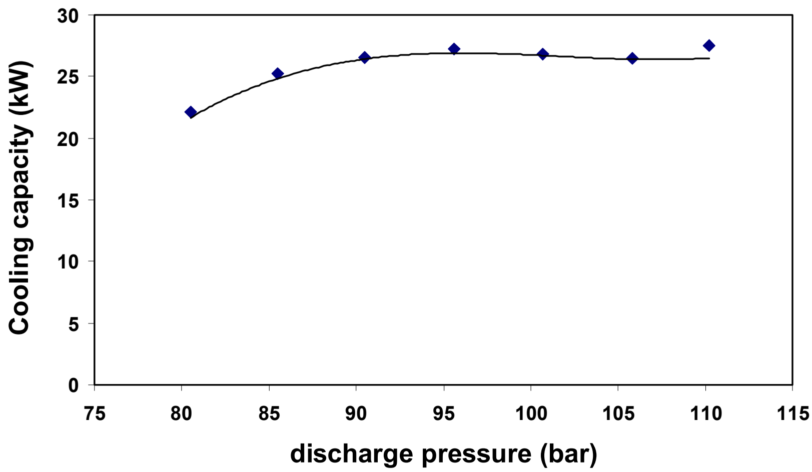


(a)



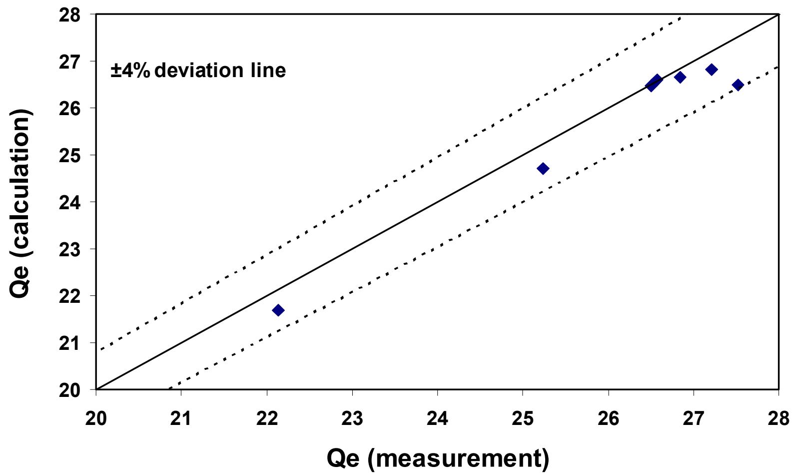
3.4
3.3
3.2
3.1
3.0
2.9
2.8
2.7
±4% deviation line
COP (calculation)
2.7
2.8
2.9
3.0
3.1
3.2
3.3
3.4
COP (measurement)

(b)



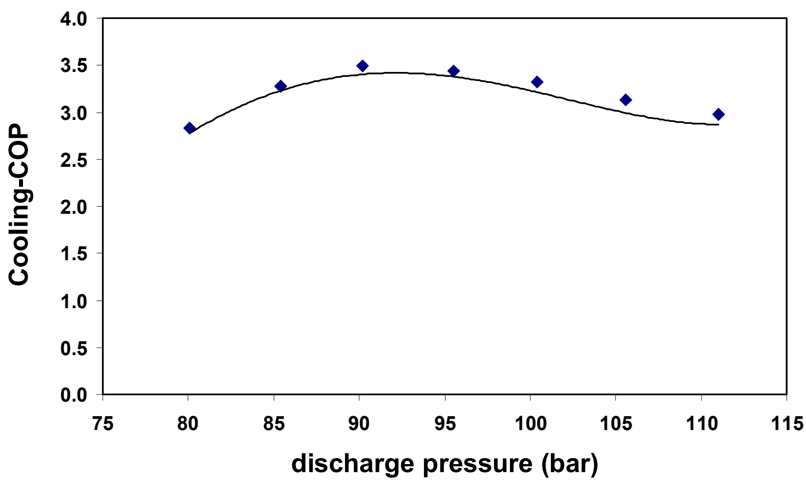
30
25
20
15
10
5
0
Cooling capacity (kW)
75
80
85
90
95
100
105
110
115
discharge pressure (bar)

(c)

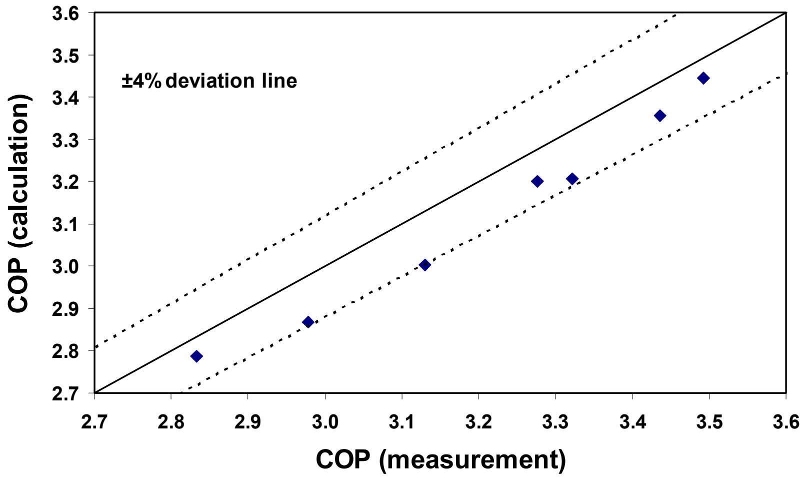


(d)

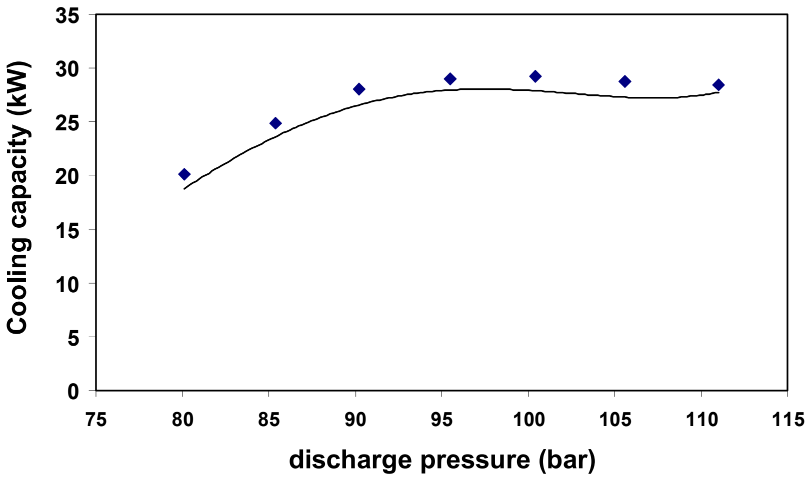
Figure 5-29 Evap. temp = 0°C, cooling medium temp. = 30°C, inlet water temp. = 20 °C, hot water temp. = 60°C, 75% recovery



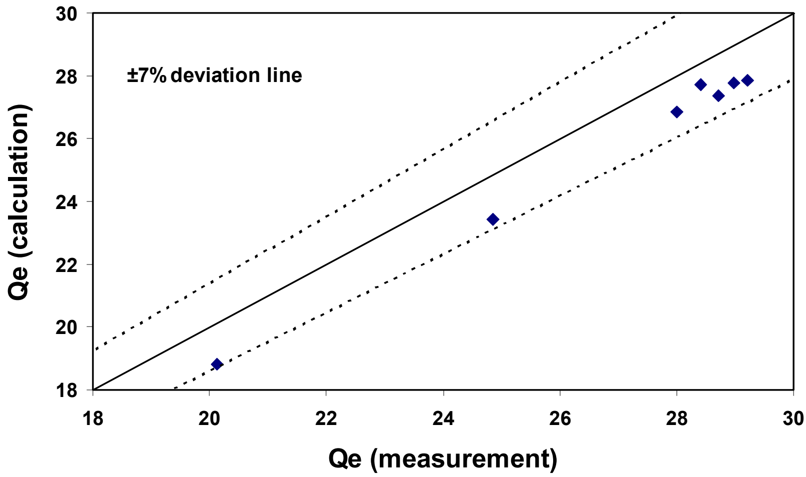
(a)


±4% deviation line
COP (calculation)
COP (measurement)
3.6
3.5
3.4
3.3
3.2
3.1
3.0
2.9
2.8
2.7
2.7 2.8 2.9 3.0 3.1 3.2 3.3 3.4 3.5 3.6

(b)


Cooling capacity (kW)
discharge pressure (bar)
35
30
25
20
15
10
5
0
75 80 85 90 95 100 105 110 115

(c)



(d)

Figure 5-30 Performance comparison at evap. temp = 0 °C, inlet water temp. = 20 °C, hot water temp. = 60 °C, full recovery

6 Discussion of Experimental Investigation And Application

In this section, a detail discussion of the combined air-conditioning and water heating system characteristic based on the experimental and simulation results are presented. The discharge pressure influence on the system performance is first explored as a basic characteristic of a transcritical cycle which should be adjusted in order the system to run in its high performance range. Then how load ratio affected the performance of the combined system is discussed briefly, as it is very important to have information on how to get benefit from this heat recovery system. Other parameters such as operating conditions as well as component design are observed in depth. Finally, exergy analysis and the application of the combined system in different types of building are discussed.

6.1 Basic characteristic

Before exploiting the combined system, it is necessary to look into basic characteristic of the transcritical cycle in air-conditioning mode without heat recovery. For air-conditioning mode without heat recovery, design condition was chosen at 0°C evaporation temperature and 30°C inlet water temperature to the heat rejecting heat exchanger. The same operating condition with addition of 20°C inlet water temperature to the water heating heat exchanger was chosen when operating the system as combined system.

As can be seen in Figure 5-1, the cooling-COPs varied with varying discharge pressures. At low discharge pressure of 80 bar, cooling capacity was at its lowest value. This is due to the specific cooling capacity was very low when operating the system close to the critical point of CO_2. As the isobar line of 80 bar is very flat, a small change in temperature causes a large change in specific cooling capacity. For the given water flow rate, the temperature approach at this pressure was 3.5K and the cooling capacity was 19 kW. As the discharge pressure increases, the approach temperature decreases toward 0 K and this small change in approach temperature causes a significant increase in cooling capacity. As an example, the cooling capacity increases by 18.6% when discharge pressure increases from 80 bar to 85 bar at which the approach temperature decreases from 3.5K to 2.2K.

Figure 6-1 shows reduction of approach temperature as a result of increasing the discharge pressure. From pressure of 80 bar to 90 bar, the approach temperature decreases rapidly and after that it decreases slowly and goes to about zero. This trend explains why the cooling capacity rises faster with increasing discharge pressure from 80 bar up to 95 bar and become about constant after that.

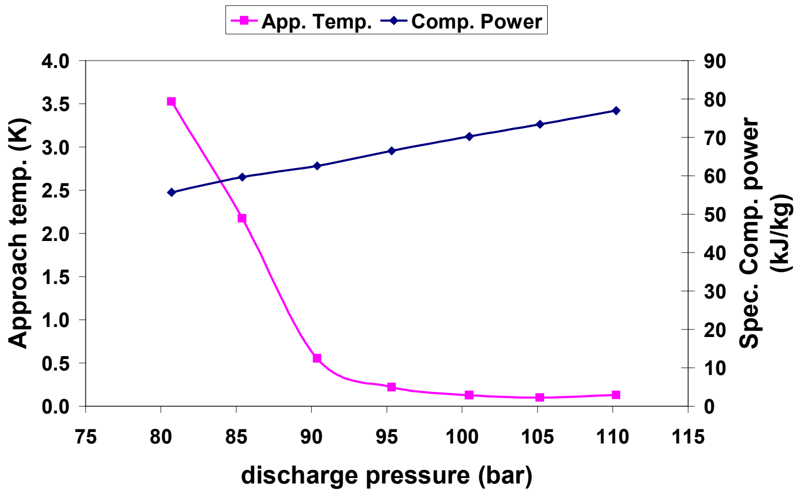


Figure 6-1 Temperature approach and specific comporessor power consumption as a function of discharge pressure.

Not like the cooling capacity behavior when varying the discharge pressure, compressor power did not show such a trend. It rose more or less linearly with increasing discharge pressure as can be seen from Figure 5-1. This was because the compressor power is mainly determined by refrigerant flow rates and specific power consumption where the CO_2 mass flow rates were almost constant at 0.127 kg/s for the range of the discharge pressure of 80-110 bar in this case. The specific power consumption increased linearly with increasing discharge pressure as depicted in Figure 6-1.

The characteristic of cooling capacity and compressor power make cooling-COP, which has been defined as the ratio of the cooling capacity to the power consumption behaved like the one shown in Figure 5-1. With increasing discharge pressure, cooling-COP first rose up to a certain value at certain pressure and then started to fall beyond this pressure. The pressure at which the cooling-COP reaches its maximum value is called the optimum pressure. Below the optimum pressure the increase of cooling capacity was higher compared to the increase in compressor power consumption, while above this pressure the increase of cooling capacity could not compete the increase in compressor power consumption anymore. At the design point, the optimum cooling-COP of the air-conditioning mode was 3.0 at 88.8 bar.

From this basic characteristic of transcritical cycle, it is very important to have high-side pressure control system to ensure the system will run around its maximum performance. In case of transcritical system without internal heat exchanger, high-side pressure control become more important as will become evident later in this chapter. However, at discharge pressure between 85 and 95 bar, the cooling-COP changes less than 1%, means that maintaining the discharge pressure around 92 bar is enough to get a transcritical cycle to run at high performance (Rieberer, 2000).

6.2 Influence of heat recovery on system performance

6.2.1 Parallel Gas Coolers Configuration

When the system is carrying out heat recovery at which part of its rejected heat is utilized for producing hot water, then its basic characteristic will be different from the system without heat recovery. As can be observed in Figure 5-2, the curves trend were still the same as of the system without heat recovery where the optimum point still exist at a certain pressure, but its magnitude were different depend on percentage of heat recovery.

The optimum pressure now not only depends on cooling medium temperature but also on load ratio (or percentage of heat recovery). From Figure 5-2 it was clearly showed that first the optimum pressure decreased with increasing percentage of heat recovery down to a certain value and then started to rise when percentage heat recovery was increased further. Dependence of the optimum pressure on load ratio is shown in Figure 6-2 with load ratio as its absisca at 0°C evaporation temperature. This behavior can not be predicted without a detail model of the gas coolers system because a simple theoretical analysis would tell that for a transcritical cycle, its optimum pressure will decrease with decreasing temperature of the cooling medium. However, in the case of a combined system in which its gas coolers configured in parallel, the optimum pressure will depend on heat transfer process in both gas coolers.

As mentioned in the previous chapter, the performance of a transcritical cycle is strongly affected by refrigerant temperature out of the gas coolers. When the system is running in air-conditioning mode without heat recovery, the optimum pressure was dictated by cooling medium temperature where at design condition it was 88.8 bar, while when the system was running in full recovery mode its optimum pressure was 91.9 bar. Since the water heating heat exchanger has been designed to be able to run under a wide range of load ratios (from 0 to full recovery), it turned out that its capacity become too large when operating in a low load ratio and in this situation the approach temperature will drop quickly by increasing discharge pressure. The capacity of the heat rejecting heat exchanger also become larger since part of heat that must be rejected has been taken away by the water heating heat exchanger. The total effect of these operations was a lower optimum discharge pressure.

If percentage heat recovery was increased further, the capacity of the heat rejecting heat exchanger become larger and larger. However, since its approach temperature was already happening in the cold end for the entire pressure range (80-110 bar), the optimum discharge pressure would not change very much with respect to the heat rejecting heat exchanger only when the percentage of heat recovery changes. The location of the optimum pressure now was strongly affected by heat transfer process in the water heating heat exchanger. The amount of heat that must be transferred in the water heating heat exchanger increased with increasing the percentage of heat recovery and therefore the optimum pressure shifted to a higher value. That was why the optimum pressure starts to rise toward the optimum pressure at full recovery mode after reaching a minimum value where the heat rejecting heat exchanger has already reached its lowest approach temperature.

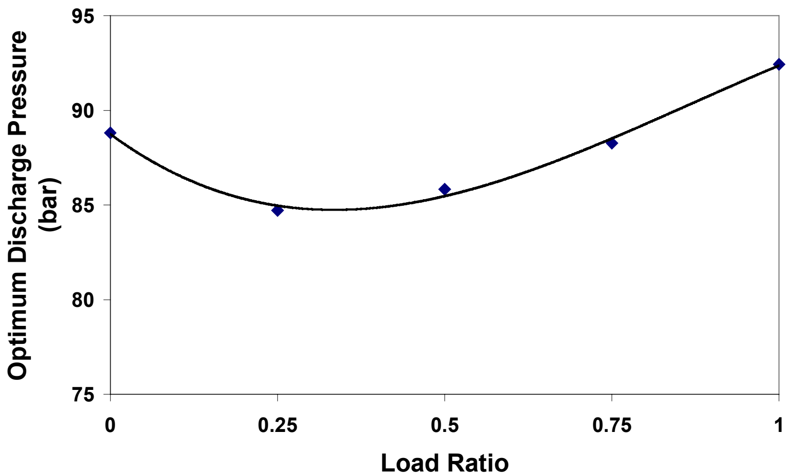


Figure 6-2 Dependence of optimum pressure on load ratio
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

Figure 6-3 shows the approach temperature in both gas coolers for four modes of operation: air-conditioning, 25% heat recovery, 75% heat recovery, and full recovery. As can be seen from this figure, at 25% heat recovery the approach temperature of the heat rejecting heat exchanger tended to become zero at about 85 bar and that of the water heating heat exchanger dropped rapidly from 12 K to 5 K at this pressure. At this pressure, the system reached its optimum conditions even though the approach temperature of the water heating heat exchanger was still 6 K (indicating that the pinch point occurs inside the gas cooler). At this pressure, the load of the heat rejecting heat exchanger was much larger compared to that of the water heating heat exchanger so that the heat rejecting heat exchanger dominated in reaching the optimum condition.

At 50% heat recovery, both gas coolers control the optimum condition because their capacities were not so different. The optimum discharge pressure at this load ratio was 85.8 bar. It was needed to run the system at 85.8 bar to pull the approach temperature of the water heating heat exchanger from 12K down to 5K. At this pressure, the approach temperature of the heat rejecting heat exchanger had already reached its lowest value. Eventhough the pinch temperature of the water heating heat exchanger was still inside the heat exchanger, the optimum condition had already achieved since both heat exchangers dictated this optimum point.

At 75% heat recovery, the heat rejecting heat exchanger practically did not control the optimum condition anymore as indicated by very low approach temperature at all pressure range in these experiments. The water heating heat exchanger in this condition determined the system performance. Because the water heating heat exchanger has been designed to captured rejected heat at full recovery mode, its capacity was still oversize at this percentage of heat recovery - where 25% of rejected heat was handled by the heat rejecting heat exchanger - resulting a lower optimum pressure compared to at full recovery mode. The optimum pressure for this operating condition was 87.6 bar at which the approach temperature of the water-cooled gas cooler was about 2 K.

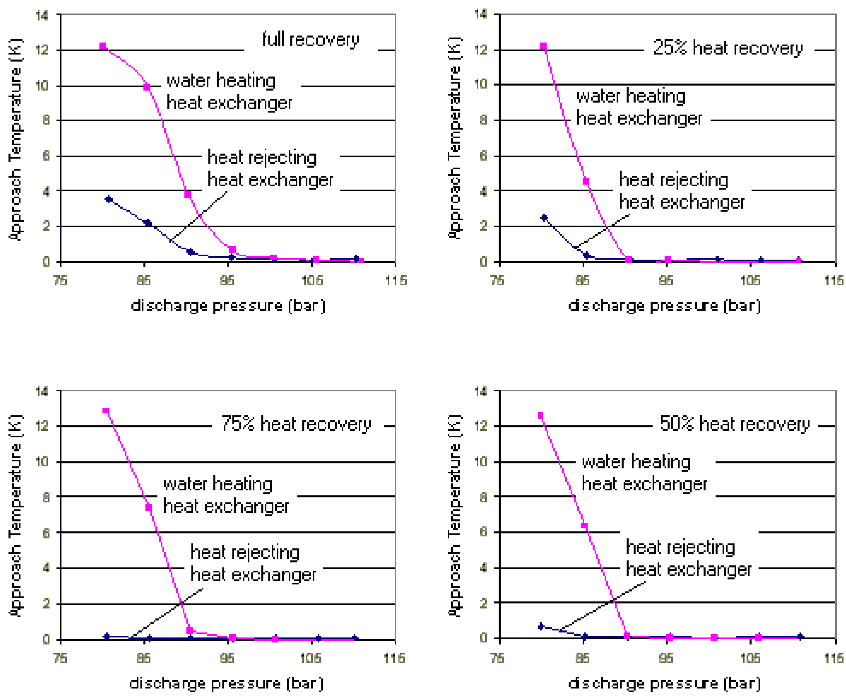


Figure 6-3 Approach temperature in both gas coolers at various percentage of heat recoveries

6.2.2 Series Gas Coolers Configuration

The situation was completely different when the two gas coolers arranged in series. The series arrangement was a common practice in heat recovery using desuperheater in conventional air-conditioning system. For transcritical cycle, the effect of adding additional gas cooler simply makes the capacity of heat rejecting heat exchanger larger, resulting in a lower temperature approach for the same discharge pressure.

At the heat rejecting heat exchanger, the approach temperature took place in the cold end of the gas cooler for the entire range of discharge pressure performed in the experiments. Because of this, arranging the water heating heat exchanger as heat recovery device in series did not affect the performance of the air-conditioning side appreciably. As can be observed in Figure 5-18, at design operating condition, the optimum pressure occurs almost at the same pressure and cooling-COP increase a bit from 3.01 to 3.07. Whereas if the water heating heat exchanger placed in parallel with the heat rejecting heat exchanger, an increase in cooling-COP to 3.24 occurred at lower discharge pressure.

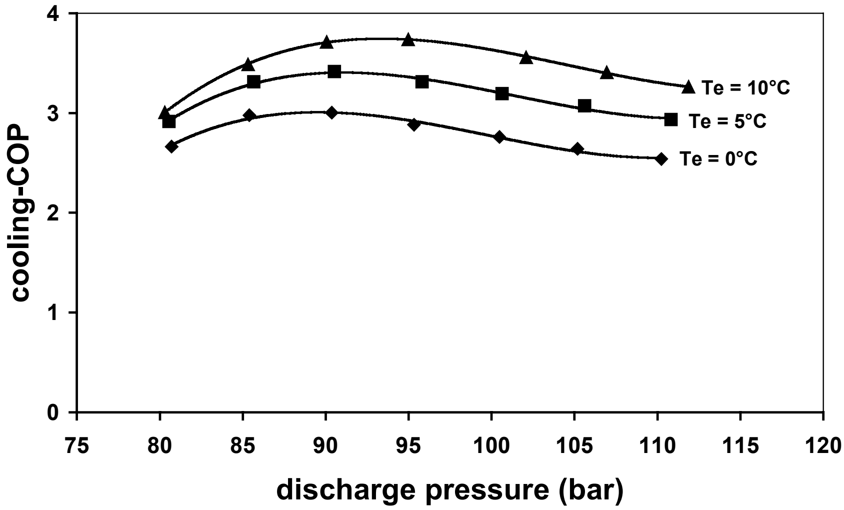
In spite of its minor contribution to an improvement of the system performance, series configuration requires a smaller heat transfer area of the water heating heat exchanger compared to parallel configuration. Because all refrigerant discharged from the compressor is utilized to produce hot water in case the water heating heat exchanger placed in front of the heat rejecting heat exchanger, heat transfer area needed for the same load ratio will be much smaller compared to the parallel configuration. For example, for the same tube size at 25% heat recovery, the series configuration would only need 6 m tube long to heat water from 20°C to 60°C while the parallel configuration would need 30 m tube long.

6.3 System performance at various evaporation temperature

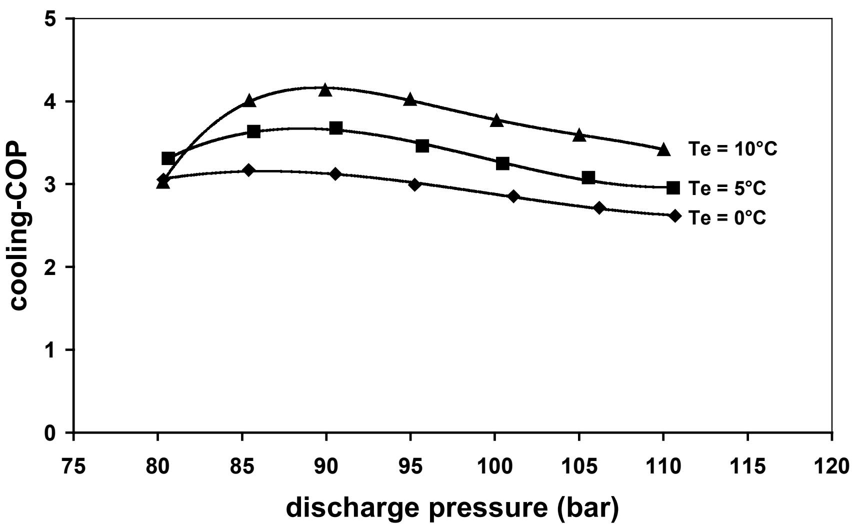
The shape of cooling-COP curves as function of discharge pressure is basically the same at various evaporation temperatures (see Figure 5-5). The main different is that the optimum pressure was lower for a lower evaporation temperature. As can be expected, the cooling-COP was higher at a higher evaporation temperature.

There are two main reason associated with a lower cooling-COP by lowering evaporation temperature. The first one is that for the same discharge pressure the specific refrigerating capacity was lower at a lower evaporation temperature. The second one is associated with higher specific compression power followed by lower compressor performance at a higher pressure ratio.

Cooling capacity becomes lower at a lower evaporation temperature because not only of a lower specific refrigerating capacity but also of a lower mass flow rates of CO_2 due to a lower density before entering the compressor. Moreover, the volumetric capacity of the compressor tends to be lower due to a lower volumetric efficiency at higher-pressure ratio.



(a)



(b)

Figure 6-4 Exp. result showing heat recovery effect on cooling-COP at various evaporation temperatures. (a) A/C Mode (b) 25% heat recovery [T_{evap} = 0, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

Figure 6-4 shows experimental results at 0°C, 5°C, and 10°C evaporation temperatures for the system running in air-conditioning mode and 25% heat recovery. Notice from Figure 6-b that at 80 bar discharge pressure, the cooling-COP at 10°C evaporation temperature was about the same as the one at 0°C evaporation temperature. This could be explained as followed. From the experimental result, it was found that the pressure at the outlet of the gas coolers was 78.8 bar, 78.8 bar, and 78.0 bar at evaporation temperature of 0°C, 5°C, and 10°C respectively, while the corresponding temperature was 32.5°C, 33.5°C, and 33.1°C, respectively. So there was no significant difference at the condition out of the gas coolers system. But the compressor performance at 10°C evaporation temperature was poor due to very low pressure ratio compared to the other evaporation temperatures at which the compressor performance were higher. Table 6-1 shows how the compressor performance varies at different evaporation temperatures and 80 bar discharge pressure.

Table 6-1 Compressor performance at 80 bar discharge pressure

Evap. Temp. (°C)	Pressure ratio	Vol. Efficiency	Isen. Efficiency
0	2.5	0.87	0.85
5	2.2	0.89	0.85
10	1.9	0.71	0.66

A need of higher optimum pressure as evaporation temperature increases can be explained by considering discharge gas temperature. As evaporation temperature increases, the discharge gas temperature will decrease (see Figure 6-5). To achieve the same hot water temperature, this will need a higher discharge pressure in the water-cooled gas cooler to get higher discharge gas temperature.

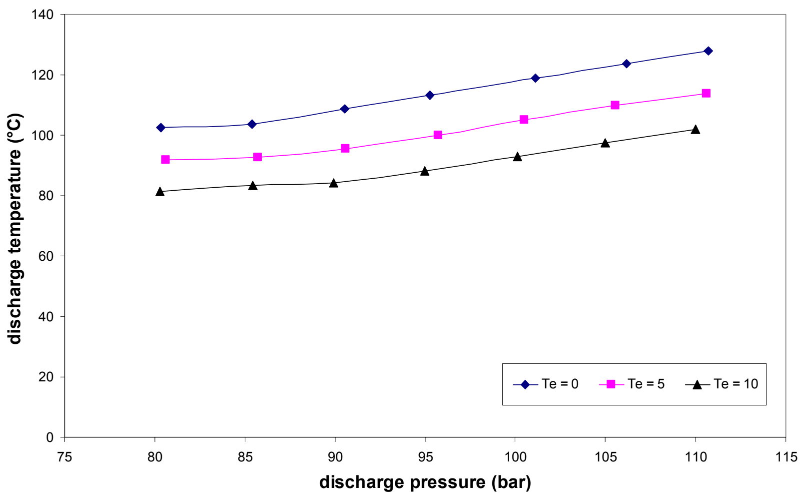


Figure 6-5 Discharge temperatures at various evaporation temperature

The optimum cooling-COP increased at all evaporation temperatures when heat recovery was employed especially at higher evaporation temperature as seen in Table 6-2. It can also be seen that the optimum pressures at all evaporation temperatures were lower for the system with heat recovery compared to the one without heat recovery.

Table 6-2 Optimum cooling-COP improvement by 25% heat recovery

Evap. Temp. (°C)	A/C mode		25% heat recovery		$(COP_{25\%}/COP_{A/C})$ *100%
	Popt.	COP	Popt.	COP	
0	88.7	3.01	84.7	3.14	4.3
5	91.2	3.41	88.6	3.67	7.6
10	93.7	3.74	89.8	4.15	11.0

A higher improvement by applying 25% heat recovery at higher evaporation temperature was associated with a lower CO_2 temperature leaving the gas coolers system. Table 6-3 shows an increase in temperature drop with increasing evaporation temperature at 90 bar discharge pressure. At 10°C evaporation temperature, the CO_2 temperature out of the gas coolers system drop from 33.9°C at air-conditioning mode to 30.2°C at 25% heat recovery mode. This yields in a higher cooling capacity, while the compressor power consumption was about the same, resulting a significant improvement in cooling-COP. This COP improvement become lower at a lower evaporation temperature as the drop in CO_2 temperature out of the gas coolers system was lower.

Table 6-3 CO_2 temperature leaving gas coolers system at 90 bar discharge pressure.

Evapaporation temp. (°C)	CO_2 Temp. (°C)		Temp. drop (K)
	A/C	Xr = 25%	
0	30.6	28.2	2.42
5	31.5	28.5	2.96
10	33.9	30.2	3.68

Table 6-4 Optimum cooling-COP improvement by 50% heat recovery

Evap. temp. (°C)	A/C mode		50% heat recovery		ΔCOP(%)
	Popt.	COP	Popt.	COP	
0	88.7	3.01	85.2	3.28	9.0
5	91.2	3.41	87.4	3.72	9.1
10	93.7	3.74	90.0	4.27	14.2

Similar behavior was observed for 50% heat recovery mode (see Table 6-4). The optimum cooling-COP at 0°C and 5°C evaporation temperature now increase by 9%. The same explanation applies for larger cooling-COP improvement at 10°C evaporation temperature.

6.4 Influence of inlet water temperature on the system performance

There are two types of water heating system, instantaneous-type and storage-type. Instantaneous-type is simple but only suitable for a relatively uniform load. One of its advantages if applied in a combined air-conditioning and water-heating transcritical cycle is a higher improvement in air-conditioning side for a certain inlet water temperature. Because hot water is directly consumed, the inlet water temperature to the water heating heat exchanger will always be the same as the water source temperature. However, as it is impractical for large load variation that is common in most situations, the instantaneous-type is limited to a small capacity and normally uses an electrical heating element.

When hot water system is of storage-type then the water will be circulated around hot water loop and in some systems the water is drawn from the bottom of the tank in charging period and from the top in discharging period. In a stand-alone system, water is heated by a heat source that could be electrical heating element or boiler and hot water temperature in the tank is used as a controlled parameter. Charging process is performed until a certain amount of water in the tank reaches set point temperature. During period of charging, the inlet water temperature to the heating system will increase and for a storage tank that designed with stratification this increase will occur at the end the charging period. The increase in inlet water temperature will occur earlier in a storage tank without capability of maintaining stratification where mixing of hot and cold water will take place during charging or discharging period.

As has been discussed before, applying heat recovery in a transcritical system will affect the performance of air-conditioning side of the system if the heat exchangers installed in parallel. The degree of improvement will directly depend on the inlet water temperature to the water heating heat exchanger. If hot water system is of storage-type then there will be a time when the inlet water temperature start to rise. At this time, the performance of air-conditioning side will start to decrease because CO_2 temperature leaving both the heat exchangers will become higher, resulting in a lower cooling capacity.

To investigate the respond of the system under varying inlet water temperature, the experiment should be done in transient state since the inlet water temperature changes from time to time. However, due to limitation of the instrument in the test rig this could not be performed in this work. Nevertheless, running the system in combined mode with constant temperature at different inlet water temperatures can give information on how the system will respond to such a change.

Figure 5-9 shows performance degradation as inlet water temperature to the water heating heat exchanger rose from 20°C to 40°C. The cooling capacity became lower for all discharge pressures as the inlet water temperature increased while the compressor power consumption remained the same at the same pressure. This results in a lower cooling-COP.

When inlet water temperature is higher than the cooling medium temperature, the performance of the system with heat recovery will be worse than the one without. At 30°C inlet water temperature, cooling-COP of the combined system was lower compared to that of the system without heat recovery for all discharge pressure. The optimum cooling-COP of the system without heat recovery was 3.01 while that of the combined system with 25% heat recovery was 2.9. The situation becomes worse when the inlet water temperature increase beyond the cooling medium temperature. At 40°C inlet water temperature and 30°C cooling medium temperature, the cooling-COP was drop to 2.7. Therefore, it is very important to keep the inlet water temperature to the water heating heat exchanger below the cooling medium temperature. Using a hot storage tank with stratification will help the combined system running in a higher performance for a longer time since the inlet water temperature to the water heating heat exchanger will be lower due to absent of mixing of hot and cold water.

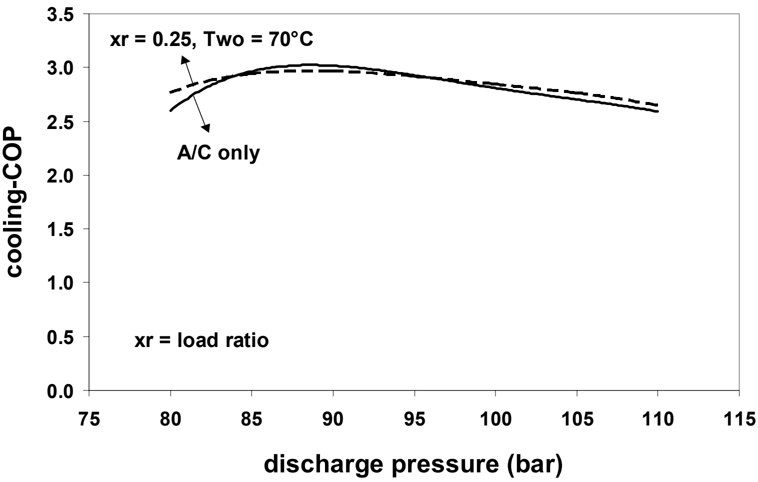
6.5 Influence of hot water temperature on the system performance

When there is a need in a higher hot water temperature, then it is also important to observe how this will affect the combined system performance. There were two set of experiments with two different hot water temperatures at full heat recovery mode. Figure 5-10 and Figure 5-11 shows the system performance at 60°C and 70°C hot water temperature with discharge pressure as parameter.

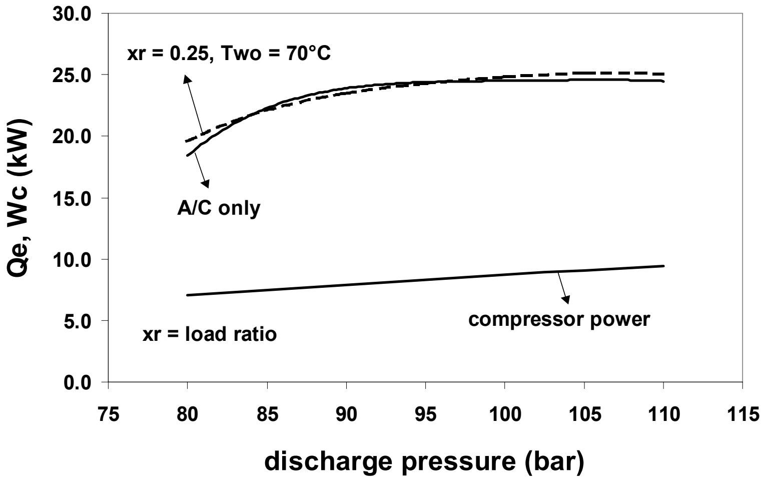
Referring to those figures, cooling-COP of the system was strongly influenced by hot water temperature. The optimum cooling-COP dropped from 3.42 to 3.1 when running with 60°C and 70°C hot water temperature respectively. The optimum pressure was also affected by hot water temperature where from those figures it can be seen that the optimum pressure was higher as hot water temperature increased. For 60°C hot water temperature, the optimum pressure was 94 bar while for 70°C was 104 bar.

A lower optimum cooling-COP at higher hot water temperature was a direct consequence of a need for higher optimum discharge pressure. Figure 5-11 shows how cooling capacity and compressor power varied with discharge pressure. A higher discharge pressure was needed to reach a higher hot water temperature because a higher hot water temperature requires a higher temperature glide on CO_2 side to heat the water up. A higher temperature glide can be achieved by increasing the discharge pressure so that the discharge temperature will also rise. From Figure 5-11 it can be seen that at the optimum pressure, the cooling capacity at both hot water temperatures were similar while the compressor power consumption was of course higher for 70°C hot water temperature since it needs a higher discharge pressure. That was why the cooling-COP becomes lower as hot water temperature increases.

Figure 6-6 shows simulation result for combined mode at 25% heat recovery. The curve of air-conditioning mode is experimental result for the same operating condition. As seen in this figure, around the optimum points the cooling-COP was similar for two operating modes. Cooling capacity and compressor power consumption are shown in Figure 6-6b where it can be observed that the cooling capacity and compressor power were similar as well. So there will be no effect on air-conditioning side performance when running the system with or without heat recovery for 20°C inlet water temperature. However, in case of a water heating system with storage tank at which there will be a time when inlet water temperature start to rise, the system performance with heat recovery will drop quickly due to a higher CO_2 temperature at the outlet of the gas coolers system.



(a)



(b)

Figure 6-6 Comparison of A/C mode and 25% heat recovery with 70°C hot water temperature. (a) cooling-COP (b) Evaporator capacity and compressor power. [T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

6.6 Influence Of Internal Heat Exchanger On System Performance

Effect of an internal heat exchanger is discussed first for the system at air-conditioning mode and then performance of the system with 25% heat recovery with and without internal heat exchanger are compared. Cooling medium temperature and inlet water temperature is chosen as variables. Effect of the length of internal heat exchanger is studied at the end of this subchapter.

In general, the purpose of an internal heat exchanger is to exchange heat from high-pressure side to low pressure side. The temperature of refrigerant at high-pressure side will decrease while at low-pressure side will increase. In case of a transcritical system with liquid receiver placed at the evaporator outlet, the state of the refrigerant leaving the liquid receiver will be at saturated vapor and the internal heat exchanger will make the refrigerant become superheat before entering the compressor. Since compressor normally designed to work with refrigerant vapor, the present of an internal heat exchanger is important for the system with liquid receiver to avoid compressor damage due to liquid droplet that enters the compressor. At high-pressure side, the refrigerant temperature after the internal heat exchanger will further decrease resulting in a lower throttling loss. Furthermore, a lower refrigerant temperature before throttling process will result in a lower refrigerant quality at inlet of the evaporator, which will make a better refrigerant distribution in the evaporator with multi circuits.

When the compressor uses immiscible lubricant, the lubricant will collect either in the liquid receiver or in the evaporator. The internal heat exchanger in this case can also be used to provide an automatic oil return to the compressor by drawing oil rich mixture from the bottom of liquid receiver or evaporator and injected to the low pressure side of the internal heat exchanger.

Specific refrigerating capacity will become higher with the present of internal heat exchanger but at the same time, refrigerant mass flow rates will become lower due to a lower density at compressor suction port. Thus, the total effect of internal heat exchanger to refrigerating capacity will depend on how the specific refrigerating capacity and the specific volume vary.

6.6.1 Air-conditioning mode

Figure 5-12 and Figure 5-13 show cooling-COP for the system running in air-conditioning mode at 30°C and 35°C cooling medium temperatures, respectively. At both temperatures, the cooling-COP of the system with internal heat exchanger was higher compared to that without internal heat exchanger. From those figures, it can also be seen that the optimum pressure was lower for the system with internal heat exchanger.

A higher cooling-COP for the system with internal heat exchanger can be explained by a higher increase in the specific cooling capacity compared to the increase in specific power consumption over the system without internal heat exchanger (see Table 6-5). CO_2 temperatures before throttling valve were significantly lower in a system with internal heat exchanger due to subcooling effect that causes a large increase in specific refrigerating capacity especially at a pressure close to critical point (see Table 6-6).

A reduction in the optimum pressure in a system with internal heat exchanger is a direct consequence of much lower temperature before throttling valve. Observing Table 6-6 it can be seen that the rate of temperature decrease was faster in the system with internal heat exchanger compared to the one without. This means that the optimum conditions was reached earlier in the system with internal heat exchanger. Furthermore, the temperature before throttling will reach a certain lowest value in the system without internal heat exchanger since it was dictated by the lowest temperature approach of the heat rejecting heat exchanger. Whereas in the system with internal heat exchanger this temperature can go down further depend on the effectiveness of the internal heat exchanger.

Another important point with the presence of internal heat exchanger is that the refrigerant flow rates become less sensitive to the discharge pressure as can be seen in Table 6-5. This is because in the system without internal heat exchanger, the state of refrigerant entering the compressor was about the same for all operating pressure and a reduction in refrigerant mass flow rates is due to a lower compressor performance as indicated by a lower volumetric efficiency. On the other hand, in the system with internal heat exchanger, the state of refrigerant entering the compressor will vary depend on heat transfer process in the internal heat exchanger. As discharge pressure increases, temperature of refrigerant before throttling valve decreases causing a decrease in temperature at the compressor suction port, which means an increase in density. The total effect is that the reduction in volumetric efficiency of the compressor (due to a higher pressure ratio) was counter balance by an increase in density, making slight change in refrigerant mass flow rates.

Table 6-5 Specific cooling capacity, power consumption, and refrigerant flow rates at various discharge pressures

Ph (bar)	With internal heat exchanger			Without internal heat exchanger		
	Spes. cooling capacity (kJ/kg)	Comp, power (kJ/kg)	Flow rates (kg/s)	Spes. cooling capacity (kJ/kg)	Comp. power (kJ/kg)	Flow rates (kg/s)
80	148.36	55.72	0.128	88.39	45.20	0.165
85	177.73	59.70	0.127	119.93	48.25	0.160
90	188.05	62.61	0.128	145.29	53.65	0.153
95	191.77	66.55	0.126	161.85	59.63	0.146
100	193.81	70.26	0.125	170.96	63.68	0.140

Table 6-6 CO_2 temperatures (°C) before throttling valve

Ph (bar)	With internal heat exchanger	Without internal heat exchanger
80	29.8	33.2
85	24.4	34.4
90	22.0	32.0
95	21.4	30.6
100	20.7	30.3

6.6.2 Combined mode

When heat recovery is applied in a system without internal heat exchanger, the system performance of the air-conditioning side will be enhanced as in the case for the system with internal heat exchanger. At 30°C cooling medium temperature and 25% heat recovery, the optimum cooling-COP was 3.17 for the system with internal heat exchanger compared to 2.98 for the one without (6.5%). At 35°C cooling medium temperature, cooling-COP was 2.59 for the one without internal heat exchanger while for the one with it was 2.69 (4.2%).

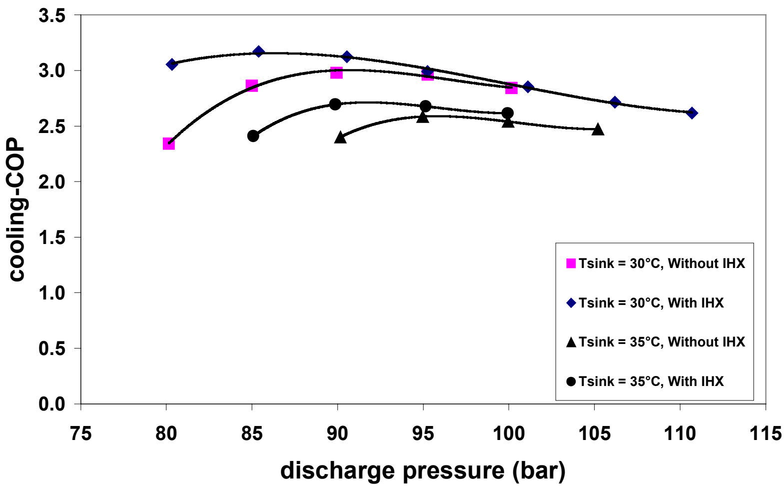


Figure 6-7 Cooling-COP at 25% heat recovery for the system with and without internal heat exchanger
[$T_{evap} = 0°C$, $T_{sink} = 30°C$, $T_{w\_in} = 20°C$, $T_{w\_out} = 60°C$]

Table 6-7 Internal heat exchanger effects on the optimum system performance

	Recovery (%)	cooling medium temp. (°C)	Pressure (bar)	Cooling-COP	Cooling capacity (kW)	Comp. power (kW)
Without internal heat exchanger	0	30	90	2.71	22.1	8.3
	25	30	90	3.00	25.3	8.3
	0	35	100	2.33	20.8	9.0
	25	35	95.8	2.59	22.7	8.8
With internal heat exchanger	Recovery (%)	cooling medium temp. (°C)	Pressure (bar)	Cooling-COP	Cooling capacity (kW)	Comp. power (kW)
	0	30	89.4	3.02	23.7	7.9
	25	30	85.3	3.17	24.0	7.6
	0	35	96.7	2.48	20.8	8.5
	25	35	91.4	2.72	22.9	8.4

Table 6-7 shows the effect of internal heat exchanger to the optimum system performance with and without heat recovery. In general, the optimum system performance both with and without internal heat exchanger increase by applying heat recovery. However, the optimum cooling-COP of the system with internal heat exchanger was higher compared to that of the system without internal heat exchanger. This is because, for the system with internal heat exchanger, the compressor power consumption was lower while the cooling capacities were not so different. The optimum pressures of the system with internal heat exchanger were lower compared to that of the system without internal heat exchanger resulting in a lower compressor power consumption when the system running at the same evaporation temperature. Since the optimum pressure was lower for the system with internal heat exchanger, the optimum cooling capacity will be a little bit lower at 30°C cooling medium temperature. It was about the same at higher cooling medium temperature of 35°C.

6.6.3 Effect of inlet water temperature

Figure 5-16 shows cooling-COP for the system without internal heat exchanger at 20°C and 30°C inlet water temperature at the same cooling medium temperature of 30°C and 25% heat recovery. Compared with Figure 5-9 for the system with internal heat exchanger and the same operating conditions, the effect of the inlet water temperature was the same. The cooling-COP of the system with and without internal heat exchanger will become lower as the inlet water temperature to the water-cooled gas cooler rose.

At 90 bar discharge pressure, the cooling-COP of the system with internal heat exchanger fell from 3.12 to 2.88 as the inlet water temperature rose from 20°C to 30°C. At the same condition, the cooling-COP of the system without internal heat exchanger fell from 2.98 to 2.74. Therefore, it is important to control this inlet water temperature in the combined system with or without internal heat exchanger when the hot water system will be circulated around the water-cooled gas cooler. The different between CO_2 temperature after the mixing point and the cooling medium temperature can be used as a control variable. As long as the cooling medium temperature is lower than CO_2 temperature by 2K, heat recovery can be performed without hurting the air-conditioning side. As this different becomes higher than 2K, the hot water circulation should be stopped and the water heating heat exchanger should be bypass. This way will ensure the heat recovery will not decrease the performance of the air-conditioning side.

6.6.4 Effect of internal heat exchanger length

The length of the internal heat exchanger will affect system performance through changing of refrigerant temperature at both ends of the heat exchanger when the flows are countercurrent or at cold end of the heat exchanger when the flows are parallel. A shorter heat exchanger will exchange heat of the refrigerant less and consequently will change refrigerant temperatures smaller. A longer heat exchanger will change refrigerant temperatures more. However, there will be a limit where heat that can be exchanged is fixed even the length of heat exchanger is added beyond this limit. This limit is achieved when the temperature of the stream with lower specific heat capacity approaches the hot end of the heat exchanger with countercurrent flows or the cold one with parallel flows.

For combined air-conditioning and water heating system, the length of internal heat exchanger will affect the system performance and the location of the optimum point. Figure 6-8 shows simulation results for cooling-COP of the system without heat recovery for various internal heat exchanger length.

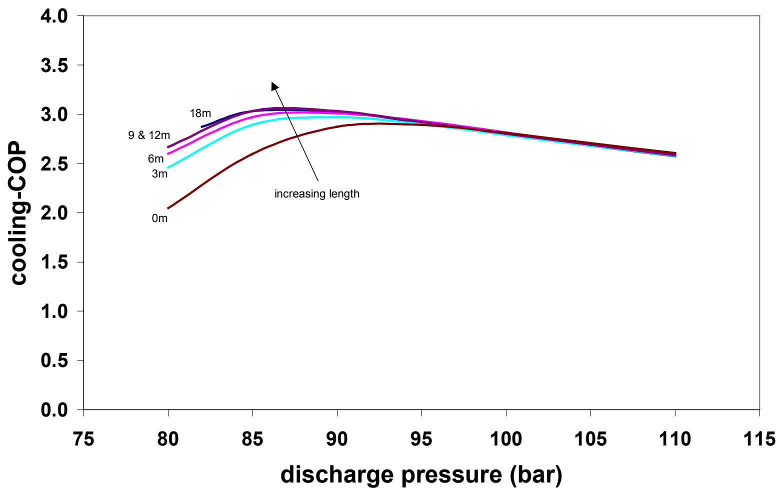


Figure 6-8 Effect of internal heat exchanger length on cooling-COP
[T_{evap} = 0°C, T_{sink} = 30°C, xr = 0]

As can be seen from this figure, the internal heat exchanger is important to the system performance. It will increase the cooling-COP and decrease the optimum discharge pressure. The longer the internal heat exchanger the more cooling-COP increases up to a certain length where further additional length will not make a significant improvement. Increasing the length from 9m to 12m for example will not make any improvement and is not economic at all. At the hot end of 9m internal heat exchanger, temperature of CO_2 in the low pressure side almost the same as temperature of CO_2 in the high pressure side. Further increase in the length of heat exchanger will just decrease temperature of CO_2 in the high pressure side a bit, resulting in a bit higher specific cooling capacity. Figure 6-9 shows temperature profile in the internal heat exchanger where the temperature of CO_2 in the low pressure side approaches the temperature of the CO_2 in the high pressure side. Similar trend has been observed for the system with 25% heat recovery and will not discuss further.

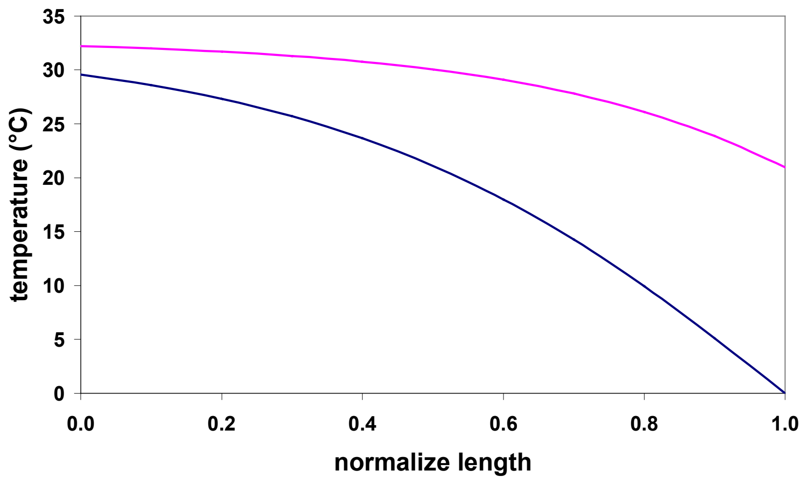


Figure 6-9 Temperature profile along the internal heat exchanger

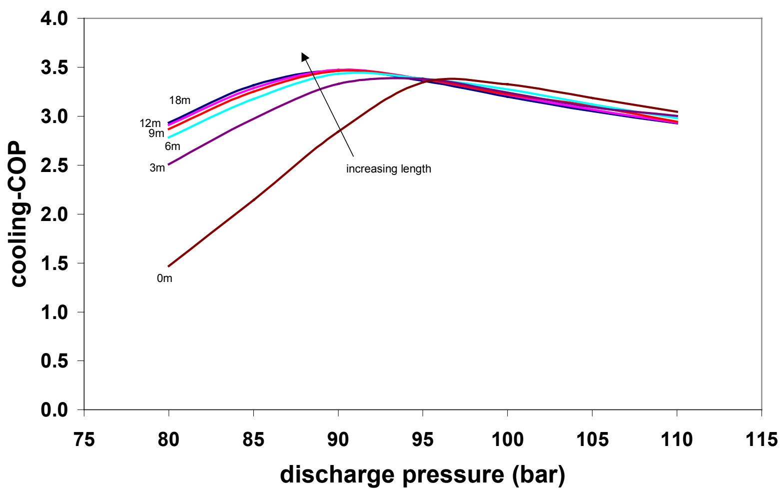


Figure 6-10 Effect of internal heat exchanger length on cooling-COP
[T_{evap} = 0°C, T_{w_in} = 20°C, T_{w_out} = 60°C, xr = 1]

The important of internal heat exchanger to the system performance most pronounce when the system is run at full recovery mode. In this mode, the internal heat exchanger will change the optimum point to a lower discharge pressure with slightly increase in cooling-COP. As can be seen in Figure 6-10, the optimum discharge pressure decreases with increasing the length of internal heat exchanger. This behavior can be explained by considering discharge temperature, which dictates the inlet temperature of CO_2 to the water heating heat exchanger.

The higher inlet temperature of CO2, the lower discharge pressure needed to heat water to a certain hot water temperature. As the length of the internal heat exchanger increases, the suction temperature of the compressor will also increase and consequently the discharge temperature will increase. Therefore, for the same hot water temperature, a lower discharge pressure will be required when the length of the internal heat exchanger increases.

The simulation program developed in this work can be used as a tool to optimize the length of internal heat exchanger. It is very important to design an internal heat exchanger with appropriate length since it will affect the system performance. A longer internal heat exchanger will improve the system performance up to a certain limit, however a longer heat exchanger means a higher cost. Therefore, it is necessary to make several designs with different internal heat exchanger length to find an optimal design. The program can be used if the information on the cost is available.

6.7 Gas Coolers Configuration

Gas coolers configuration will affect the performance of the combined air-conditioning and water-heating system. There can be two possibilities arrangement of these gas coolers (as water heating heat exchanger and heat rejecting heat exchanger), series or parallel. In series configuration there can be three possibilities in placing the water heating heat exchanger. These arrangements are:

1. in front of the heat rejecting heat exchanger,
2. in the back of it,
3. or the heat rejecting heat exchanger is placed in the middle of water heating heat exchangers.

Figure 6-11 shows possible arrangements of the gas coolers.

Placing the water heating heat exchanger in the back of the heat rejecting heat exchanger (see Figure 6-11b) will limit hot water temperature that can be achieved since CO_2 inlet temperature to the water heating heat exchanger will become lower. However, CO_2 temperature before throttling process will become lower depends on the inlet water temperature. On the other hand, if the water heating heat exchanger is placed in front of the heat rejecting heat exchanger (see Figure 6-11a), CO_2 temperature before throttling process will be dictated by the cooling medium temperature and hot water temperature can be set higher.

The other configuration is to divide water heating heat exchanger into two units and places heat rejecting heat exchanger between these water heating heat exchangers (see Figure 6-11c). In this configuration, CO_2 temperature before throttling process can be pulled down as close as possible to the inlet water temperature by adjusting the size of water heating heat exchangers unit or mass flow rates of water. To control load ratio the capacity of heat rejecting heat exchanger must be regulated by either reducing mass flow rates of the cooling medium or by bypassing part of CO_2 stream before entering heat rejecting heat exchanger.

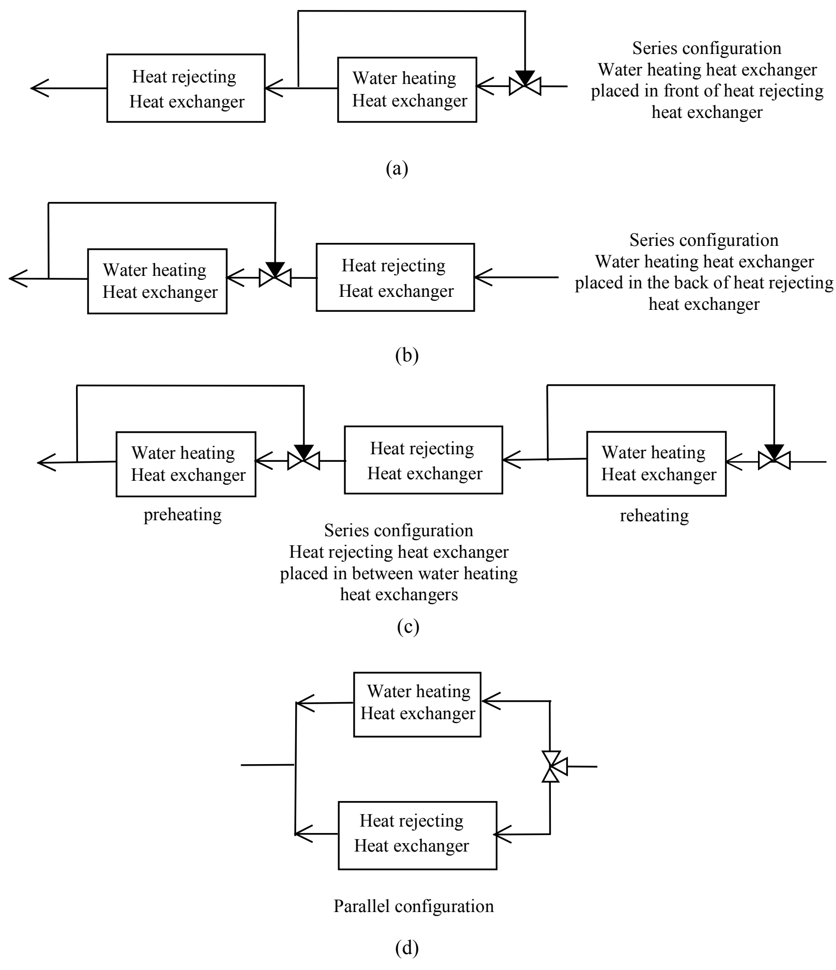


Figure 6-11 Possible configurations of gas coolers

In this work, only the series configuration with the water heating heat exchanger placed in front of the heat rejecting heat exchanger was examined. Since the desired hot water temperature was at least 60°C, this would be difficult to be achieved if the water heating heat exchanger was placed in the back of the heat rejecting heat exchanger. The system performance with either series or parallel configuration was strongly determined by the outlet temperature after heat rejection process in both gas coolers.

The effect of gas cooler configuration on system performance can be seen in Figure 5-18 and Figure 5-19 for 25% heat recovery. The optimum cooling-COP in the series configuration only increased slightly since the approach temperature in the heat rejecting heat exchanger was already low without heat recovery. This also explains why the optimum pressure was almost the same in the system with or without heat recovery. At design operating condition, the optimum pressure was about the same at 87.3 bar for the system with heat recovery and 87.9 bar for the system without. The optimum cooling-COP for the system with heat recovery was 3.07 compared to 3.01 for the system without heat recovery. Cooling capacity was also not affected significantly by applying heats recovery with series configuration. This is because there was small different in temperature before throttling process.

Since the system performance was only determined by the cooling medium temperature in the series configuration with water heating heat exchanger placed in front of heat rejecting heat exchanger, the inlet water temperature to the water-cooled gas cooler does not play an important role. If the water in storage tank will be circulated, heating process can be run continuously even the inlet water temperature start to rise more than the cooling medium temperature. For example, at 30°C cooling medium temperature, the inlet water temperature to the water heating heat exchanger can be let to increase even to 50°C without degrading the air-conditioning side performance.

The system performance will completely be different by applying heat recovery with parallel configuration, as have been discussed earlier in this chapter. Both cooling-COP and cooling capacity will increase as percentage of heat recovery increases if the inlet water temperature to the water heating heat exchanger can be maintained at lower value than the cooling medium temperature. When water is circulated and its inlet temperature goes higher than the cooling medium temperature, the cooling-COP and cooling capacity will become lower as a result of a higher temperature before throttling process. Therefore, it is important to have a good temperature control on the water side to ensure the system performance with heat recovery not become lower than the one without heat recovery.

6.8 Overall Assessment

This section describes system performance at all operating conditions range. The most important parameters that affect the system performance are evaporation temperature, cooling medium temperature, discharge pressure, and percentage of heat recovery. All these main parameter are plotted as a function of heat recovery ratio so that it is easy to observe how the system will respond to a heat recovery action. The points at 40°C cooling medium temperature were obtained from the simulation program.

Figure 6-12 shows optimum discharge pressure as function of heat recovery ratio at several evaporation temperatures. As seen in this figure, at all evaporation temperatures, the optimum discharge pressure first fell down with increasing heat recovery ratio and then rose to the optimum pressure for full recovery mode. Notice that at all evaporation temperatures, the full recovery mode must be run at a higher discharge pressure. Performing heat recovery at 60% ratio or lower will result in a lower optimum discharge pressure.

The optimum cooling-COP and cooling capacity are shown in Figure 6-13 and Figure 6-14. It can be seen that at all evaporation pressure the optimum cooling-COP and cooling capacity increased with increasing percentage of heat recovery. At full recovery mode, eventhough the optimum discharge pressure was higher, the cooling capacity of the system was also higher, which means a shorter operation time can be achieved.

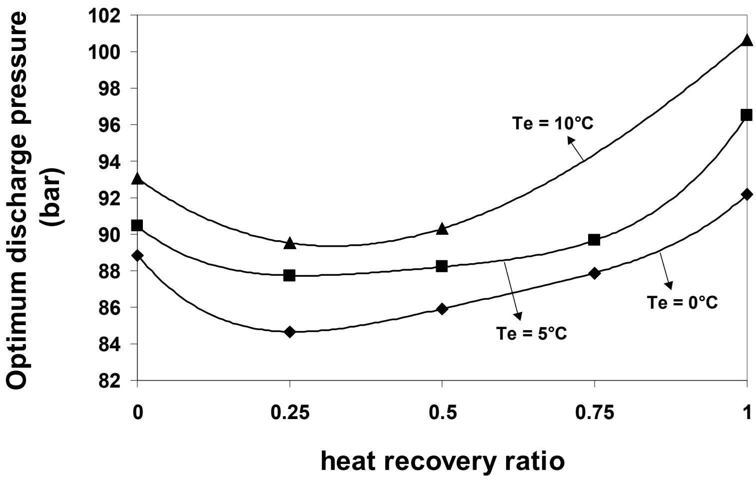


Figure 6-12 Optimum discharge pressures at various evaporation temperatures [T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

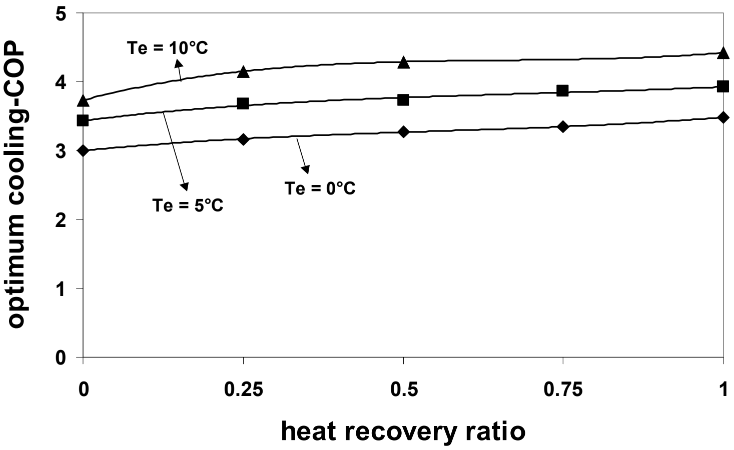


Figure 6-13 Optimum cooling-COPs at various evaporation temperatures
[T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

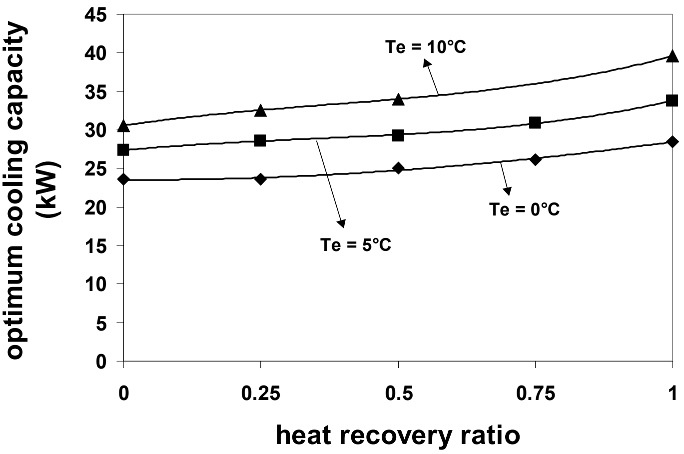


Figure 6-14 Optimum cooling capacity
[T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

The effect of cooling medium temperature can be observed in Figure 6-15. The optimum discharge pressure was determined by cooling medium temperature and inlet water temperature. At full recovery mode, the discharge temperature of course was solely be determined by the inlet water temperature since all rejected heat was dissipated to the water heating heat exchanger. At partial recovery mode, the variation of optimum discharge pressure with percentage of heat recovery was depended on cooling medium temperature.

At 40°C cooling medium temperature the optimum discharge pressure for air-conditioning mode was 103.2 bar while the optimum discharge pressure for full recovery mode was 91.6 bar. Since there was a large temperature different between cooling medium temperature and inlet water temperature, the temperature of CO_2 at mixing point become lower as percentage heat recovery increases and the optimum discharge pressure become lower proportional to percentage heat recovery. From Figure 6-15 it can be seen that as percentage heat recovery rises, the optimum pressure falls almost linearly and reaches its lowest value at full heat recovery. There was no minimum point at this high cooling medium temperature.

The minimum point can be obtained at cooling medium temperature lower than 35°C. Referring to Figure 6-15, the minimum optimum pressure will shift to a higher percentage of heat recovery as cooling medium temperature increases. At 35°C cooling medium temperature the minimum pressure occurred at around 50% heat recovery, while at 30°C it occurred at around 25% heat recovery.

At 25°C cooling medium temperature, there was no minimum pressure observed. This is because there was a large different of the optimum pressure. The optimum pressure at 25°C outdoor air temperature was 80.8 bar while that at full recovery mode was 91.6 bar. Therefore, the optimum discharge pressure will increase with increasing percentage heat recovery since the CO_2 temperature at the mixing point will always become higher.

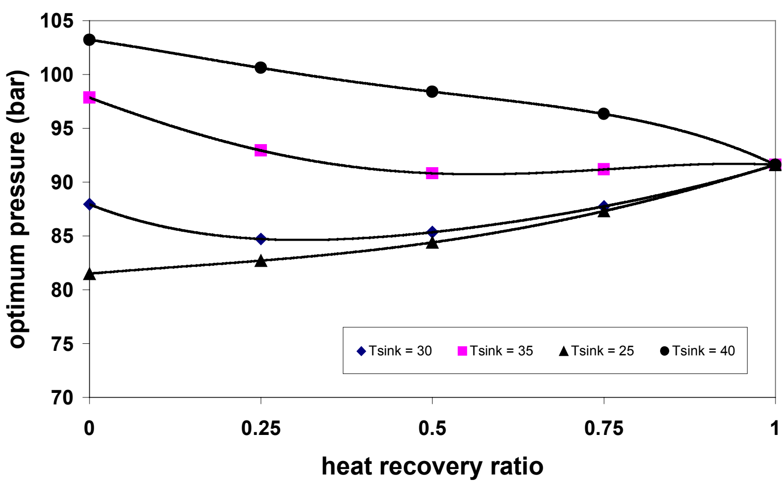


Figure 6-15 Variation of optimum pressure with heat recovery at various cooling medium temperatures
[T_{evap} = 0°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

The cooling-COP at 25°C cooling medium temperature decrease slightly with increasing percentage heat recovery as seen in Figure 6-16. This decrease in cooling-COP was due to a higher increase in compression power compared to the increase in cooling capacity. At 25% heat recovery for example, the contribution of heat recovery in reducing the temperature of CO_2 at the mixing point was small compared to the need in a higher pressure to achieve the optimum condition. This also apply to all percentage of heat recovery. Since a higher discharge pressure increases as percentage of heat recovery increases, the spesific power consumption becomes higher.

Despite this small COP reduction, the system performance of the combined system should be looked as a total system performance so that eventhough there is a slight reduction in cooling-COP, the total-COP will still be higher compared to a separate air-conditioning and water-heating system.

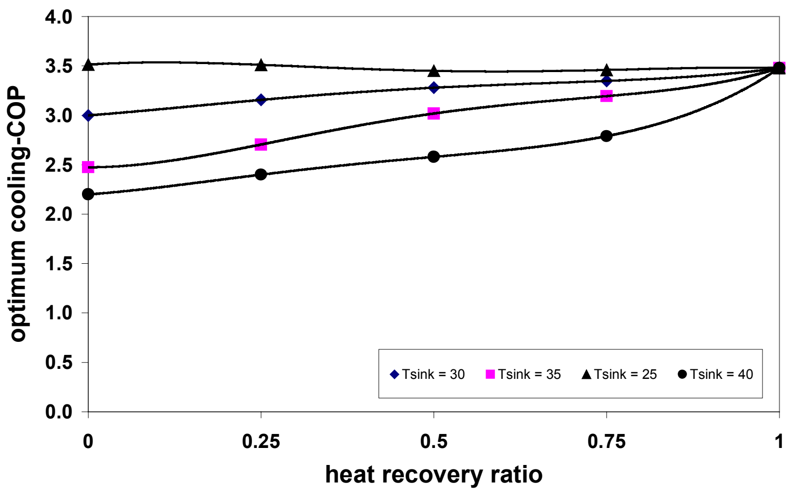


Figure 6-16 Variation of optimum cooling-COP with heat recovery at various cooling medium temperatures

Cooling capacity at the optimum conditions can be observed in Figure 6-17. At all percentage of heat recovery, the cooling capacity increased with increasing percentage of heat recovery. The highest cooling capacity was achieved at full recovery mode. For partial heat recovery mode, the cooling capacity increased almost linearly from air-conditioning mode to the full recovery mode.

Figure 6-18 shows the compressor power consumption as function of percentage of heat recovery at various cooling medium temperatures. The curves shape look similar to the curves of optimum discharge pressure (Figure 6-15). The reason is that since the mass flow rates of CO_2 vary very little at the same cooling medium and evaporation temperatures, the compressor power consumption was dictated by the different between CO_2 states at discharge and suction pressure. Since the evaporation temperature were the same, the compressor power than solely was determined by the discharge pressure. That is why the variation of compressor power consumption at optimum condition will look similar to the variation of optimum discharge pressure.

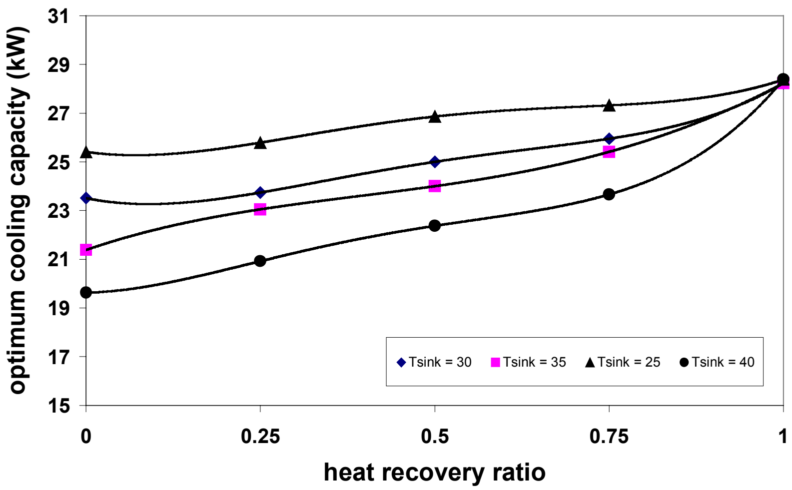


Figure 6-17 Optimum cooling capacity at various cooling medium temperatures

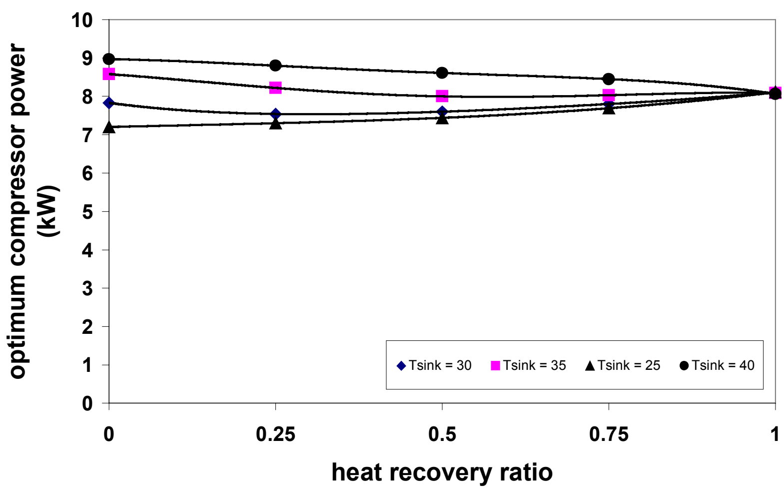


Figure 6-18 Compressor power at optimum conditions at various cooling medium temperatures

6.9 Exergy analysis of the combined system

Beside comparison based on COP or cooling capacity that shows improvement of the combined system, another method that shows direct indication of the system improvement is through exergy analysis. In this method, the system perfection is given as exergetic efficiency. This method is still not accepted in general practice because it involves an abstract quantity, entropy. However, it can show the system perfection more clearly because it defines efficiency of a system in more general sense, i.e. uses a scale from 0 to 1 (or 0% to 100%), rather than COP that can have a value higher than 1.

There are two main information that can be obtained from exergy analysis, exergetic efficiency and sources of exergy losses. Exergetic efficiency indicates the system perfection while the sources of exergy gives an information on which components of the system that cause degradation of the system performance. With this information, modification of the system can be concentrated just on the components that causes large exergy losses.

The spesific exergy content of a substance at a certain state relative to a reference state is defined as (Kotas,1995):

$$\varepsilon = h - T_o s - \beta_a \tag{5.1}$$

where,

$$\beta_a = h_a - T_o s_a \tag{5.2}$$

and its exergy content is expressed by,

$$E = \dot{m}.\varepsilon \tag{5.3}$$

The exergetic efficiency is given by:

$$\psi = \frac{\sum \Delta E_{out}}{\sum \Delta E_{in}} \tag{5.4}$$

For air-conditioning system, the definition of exergetic efficiency is as follows:

$$\psi_{cooling} = \frac{\Delta E_{evaporator}}{P_{compressor}} \tag{5.5}$$

For combined air-conditioning and water heating system, the definition of exergetic efficiency is:

$$\psi_{total} = \frac{\Delta E_{evaporator} + \Delta E_{hot\ water}}{P_{compressor}} \tag{5.6}$$

6.9.1 Exergetic efficiency of the combined system

Figure 6-19 shows exergetic efficiency of cooling as a function of discharge pressure for various heat recovery ratio. As can be seen, the shape of the curves are similar with those in Figure 5-2. However, the system perfection now can be observed more clearly. For example, at air-conditioning mode (xr = 0), its optimum exergetic efficiency is 34.5% and at 50% heat recovery (xr = 0.5), its optimum exergetic efficiency is 37.7%. It tells that the air-conditioning perfection can be improved by 3.2% in this case.

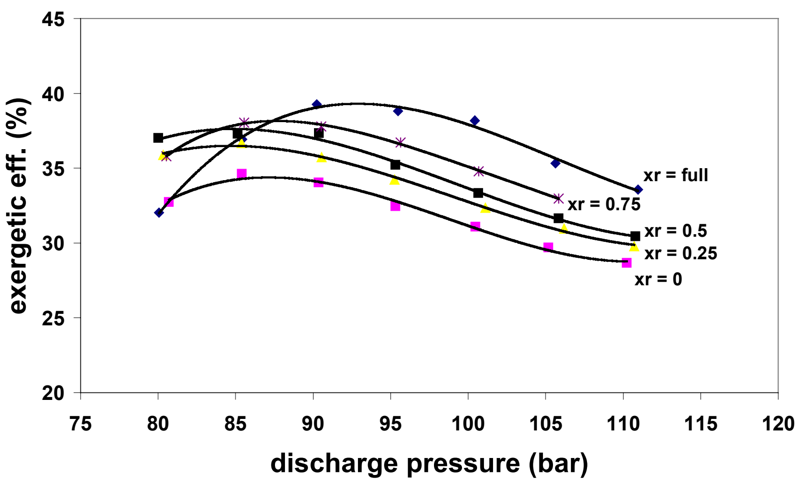


Figure 6-19 Exergetic efficiency of cooling at various heat recovery ratios
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

The system improvement can even be seen more clearly if one consider the whole system, i.e. both air-conditioning and water heating system. In a system that consist of a separated air-conditioning and hot water heating system, there will be two input of exergies, one for air-conditioning system and the other for hot water heating system. For a combined system, its exergy input is only for air-conditioning system because exergy input for water heating system is supplied from the air-conditioning itself.

Figure 6-20 shows total exergetic efficiency of the combined system. Now one can see directly how the system can save energy. As can be seen, the system perfection shifts to a higher value as percentage of heat recovery increases. This is because a lot of exergy needed to produce hot water, which is a form of low level energy in this case, has been eliminated and is supplied from the rejected exergy from the air-conditioning system. In this figure, it should be noted that at 0% heat recovery, the total exergetic efficiency is defined as a system without water heating so that the only exergy input is to drive the compressor.

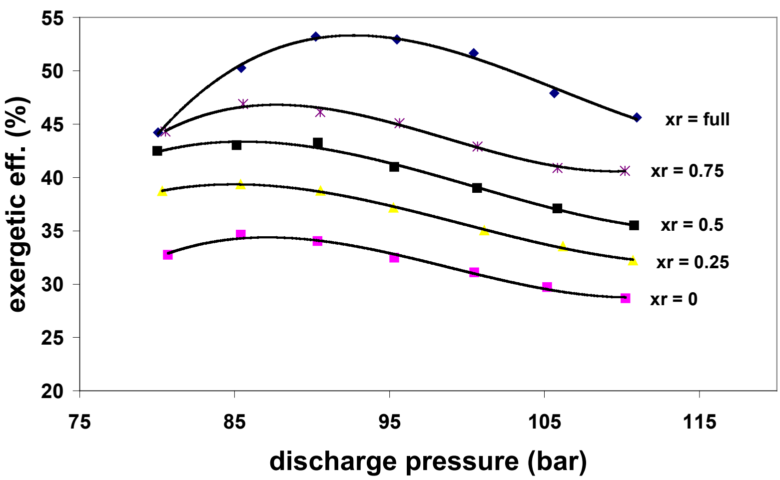


Figure 6-20 Total exergetic efficiency for various heat recovery ratio
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

To observe the efficiency of a separated system, one need information on efficiency of hot water system, and it is assumed that exergetic efficiency of water heating system is 100% (used electrical heater). The total exergetic efficiency of a separated air-conditioning and water heating system is as follows:

$$\psi_{separated} = \frac{\Delta E_{evaporator} + \Delta E_{hotwater}}{P_{compressor} + P_{waterheater}} \tag{5.7}$$

A large improvement in term of exergetic efficiency can be seen in Figure 6-21. In case of separated system of air-conditioning and water heating, large fraction of exergy from the heat source to produce hot water lost and only part of it is converted to exergy content of hot water. The exergetic efficiency of water heating in this case is only 3%, that is why the total exergetic efficiency of the system becomes very low.

If exergy is taken from rejected heat of air-conditioning side, there will be two advantageous. One comes from the increase in exergetic efficiency of the air-conditioning itself and the other comes from eliminating the need of exergy for producing hot water. From Figure 6-21 it can be seen how the combined system performs much better energy conversion compared to a separated air-conditioning and water heating system.

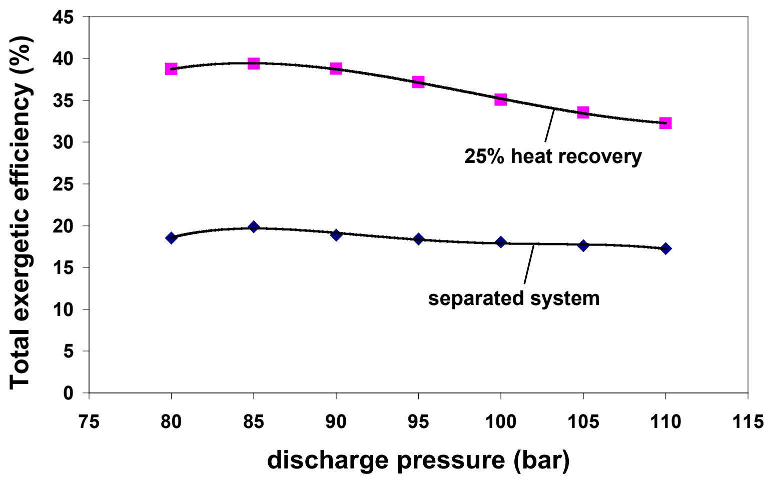


Figure 6-21 Total exergetic efficiency for system with and without heat recovery [T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C]

6.9.2 Distribution of exergy losses

Figure 6-22 shows how exergy losses is distributed among the system components at 90 bar discharge pressure. It gives an information on at which component exergy is destroyed and how much. The definition of exergy losses in this analysis is a relative value of exergy losses in each component to the exergy input, which is the compressor shaft power.

Compressor

Compression process is a complex process and its characteristic has been represented by isentropic efficiency. Since irreversibility of the compression process can be determined solely from the increase in entropy, the variation of exergy losses with percentage of heat recovery will be determined by the variation of isentropic efficiency. Experimental results to show the states in Figure 6-22a, showed a discrepancy in isentropic efficiency at the same pressure ratio and therefore the exergy losses also follow the same trend.

Connecting line between compressor and gas coolers

From the compressor and the gas coolers, there are oil separator and distribution valves. These accessories create irreversibilties by decreasing temperature and pressure of CO_2 before exchanging heat with cooling mediums. Around 2% of exergy is destroyed in this connecting line.

Heat rejecting heat exchanger

At air-conditioning mode without heat recovery, large part of exergy is destroyed in heat rejecting heat exchanger. This losses is inherent characteristic of air-conditioning system and can not be avoided since all rejected heat is just dissipated to the ambient. For transcritical cycle, the higher the pressure, the more exergy is destroyed in the heat rejecting heat exchanger.

Water heating heat exchanger

When rejected heat is captured to heat water, part of exergy that otherwise lost is transferred to water. Because now there are two heat exchangers, exergy losses is distributed between these heat exchangers. As can be seen from Figure 6-22a, exergy destroyed in the heat rejecting heat exchanger decreases as percentage of heat recovery increases. While in the water heating heat exchanger more exergy is destroyed as more exergy is transferred to water. The total exergy losses in these heat exchangers will be smaller than the one of the system without heat recovery because part of the rejected exergy now is utilized.

Producing hot water by recovering rejected heat of air-conditioning system is far more efficient in term of exergetic efficiency compared to other methods of heating. If electrical resistance heating is used to heat water from 20°C to 60°C at 15kW for instance, only 3% of the electrical energy is converted to exergy content of hot water, which is very unefficient process. On the other hand, if rejected heat of air-conditioning system at 90 bar is recovered to heat water for example, the exergetic efficiency of the process is 48%, far more efficiency than heating up the water by electrical heating element. This is because exergy content of hot water is very small as it is lowgrade energy, and producing it from highgrade energy such as electrical energy will cause large destruction of exergy.

Since rejected heat of air-conditioning system is also a lowgrade energy, recovering it to heat water will result in an efficient process.

Mixing process

The mixing process where CO_2 from heat rejecting heat exchanger and CO_2 from water heating heat exchanger mixes will also create another irreversibiltiy. Since the tubes were well insulated, heat loss to ambient was small and irreversibility is mostly determined by pressure drop in this process. Transcritical cycle is characterized by a more tolerable to pressure drop, and pressure drop in mixing process is much smaller compared to pressure drop occurs in the other components. No more than 0.5% of exergy is destroyed in mixing process in the experiments and therefore, the effect of mixing losses can be ignorred.

Expansion valve

The irreversibility in expansion process is determined by increase in entropy of CO_2 after the process. It will increase with increasing entropy. The increase in entropy will become less as temperature before throttling decreases. Since the temperature before throttling becomes lower as percentage of heat recovery increases, the throttling losses will also become smaller. This trend can be seen in Figure 6-22b where the throttling losses decrease as percentage of heat recovery increases.

Internal heat exchanger and integrated liquid receiver/submerged heat exchanger

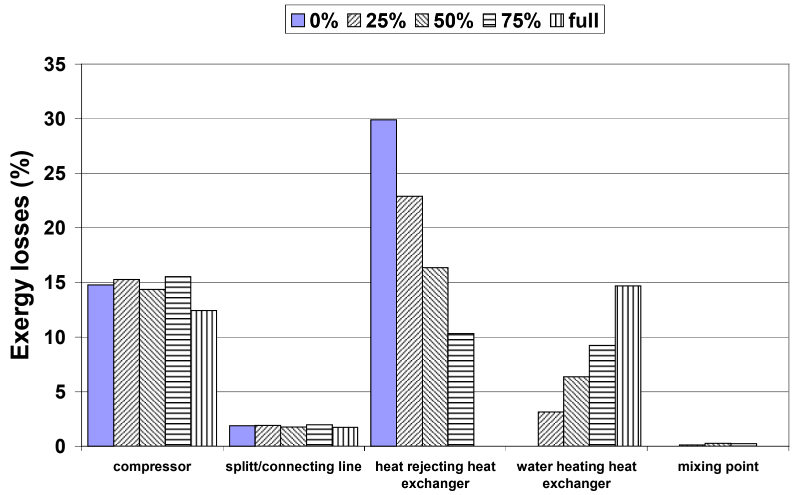
Temperature mismatch is the dominant factor of the irreversibility in the internal heat exchanger and the integrated liquid receiver/submerged heat exchanger. The higher the mismatch, the higher the irreversibilty of the component. There was no trend in the irreversibilty rates that occurs in these components as can be seen in Figure 6-22b. It was lower in partial recovery mode compared to air-conditioning mode without heat recovery or to full recovery mode. Around 1% exergy is destroyed in this component.

Evaporator

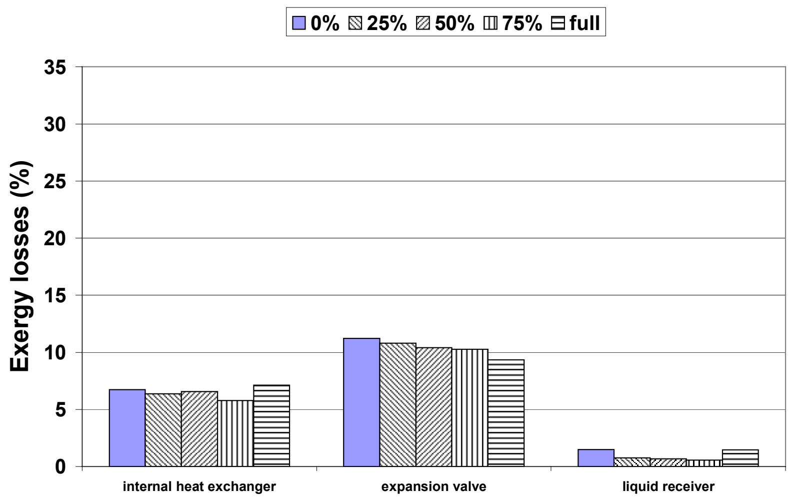
Since evaporation temperature was set as a parameter and the objective is to study heat recovery of the rejected heat, the exergy losses in the evaporator is not considered in this work. It is assumed that all exergy content of CO2 in evaporation process is transferred to the space and therefore there is no losses occurs in the evaporator. This approach will make comparison with other system such as conventional R22 air-conditioning system more easy since the heat source condition is not taken into consideration.

Exergy balance diagram (Grasmann Diagram)

Figure 6-23 and Figure 6-24 show Grassmann Diagram for the system without heat recovery and the one with 50% heat recovery.



(a)



(b)

Figure 6-22 Exergy losses distribution of the combined system
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, Ph = 90 bar]

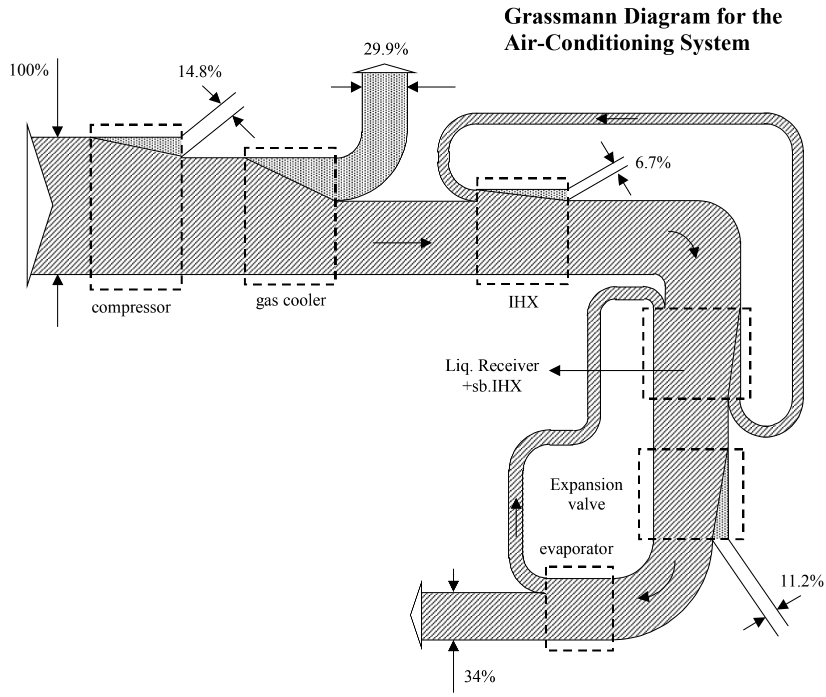


Figure 6-23 Grassmann Diagram for Air-Conditioning System
[T_{evap} = 0°C, T_{sink} = 30°C, Ph = 90 bar]

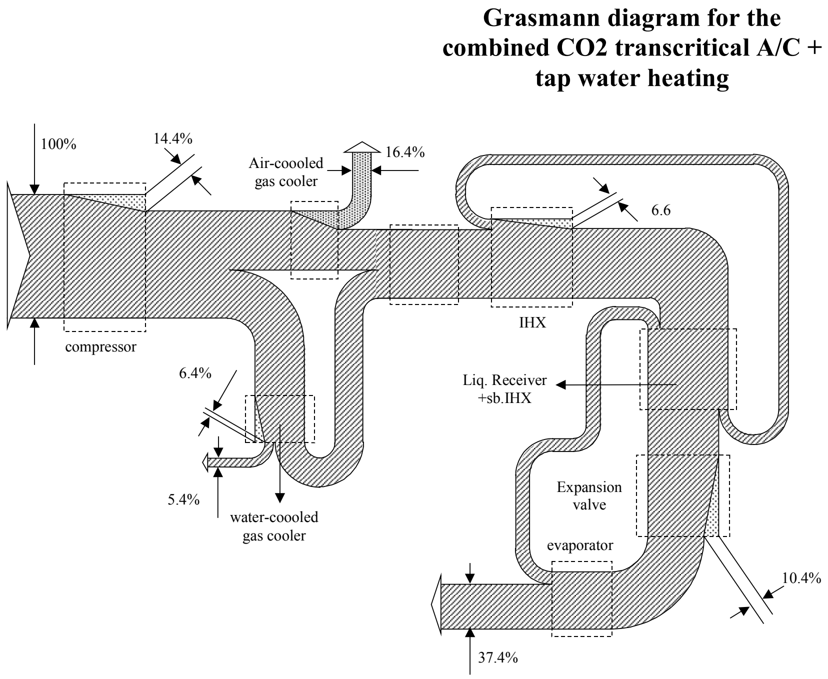


Figure 6-24 Grassmann Diagram for combined system
[T_{evap} = 0°C, T_{sink} = 30°C, T_{w_in} = 20°C, T_{w_out} = 60°C, Ph = 90 bar, xr = 0.5]

6.10 System Application

This section discusses the possibility to save energy consumption in buildings by recovering rejected heat from air-conditioning system to produce hot water. The energy consumption in different types of building were estimated by using a computer software. The estimation of annual energy consumption for particular building was done using computer software called eQUEST. This computer program was based on DOE-2.2 building simulation program from U.S. Department of Energy (EQUEST, 2000). The program provide information on energy consumption in a building for the main energy consuming parameters. Based on this information, estimation of the amount of energy to provide hot water and to run air-conditioning system can be performed.

In this study, the energy consumption estimation was done with the following main assumptions:

- Buildings are located in Jakarta, Indonesia.
- Weather data are taken from DOE weather data for Jakarta.
- A/C and hot water system are installed in the buildings.

The program comes with the ability to calculate ground water temperature, which is needed to determine energy consumption of hot water production. In case of Jakarta, the ground water temperature according to the program varies a little from month to month with average value of 21.3°C. In this study, the information on ground water temperature of Jakarta from the program will be used.

The distribution of energy consumption will depend on the building type. This building type can be represented by a different operation and occupancy schedule that can be given as input to the program. Regarding energy saving from the hot water side, the type of building that is relevant with this study are as follow:

- Hotel
- Hospital
- Office
- Multifamily

Table 6-8 gives the main dimension of the buildings used for estimating the energy consumption.

Table 6-8 Buildings main dimension

Type	Area (m^2)
Hospital	23224
Hotel	16721
Multifamily	16721
Office	11612

The program will calculate all energy consumption hourly and the result can be presented in average monthly or yearly. Figure 6-25 shows a result from the calculation based on the main assumption mentioned above with all other inputs taken from the default value provided by the program.

The main energy consuming parameters in a building vary in its magnitude depend on the type of building which are as follows: space cooling, space heating, lighting, pumps and fans, water heating, miscellaneous equipment. This different mainly is due to a different schedule of occupancy and equipment for different building type.

Table 6-9 shows estimation results of energy consumption in various buildings. The energy consumption here means required energy that must be supplied to run the components listed in the table.

Table 6-9 Breakdown of annual electrical energy consumption (kWh/m^2)

Component	Building Type			
	Hospital	Hotel	Family	Office
Space Cool	225.5	109.2	163.9	44.2
Heat Reject.	0.0	0.0	0.0	1.5
Hot Water	110.3	37.1	39.6	3.1
Vent. Fans	23.6	23.7	34.1	4.7
Misc. Equip.	118.0	18.3	29.3	23.9
Task Lights	0.0	0.0	0.0	2.4
Area Lights	104.8	22.7	36.4	20.7
Total	**582.1**	**211.0**	**303.3**	**100.6**

Among these, space cooling takes the first place and water heating consumes significant amount of energy as can be seen from Figure 6-25 to Figure 6-28. Except in an office that only needs a little part of total energy to produce hot water, there is a significant portion of energy consumption that can be saved by recovering rejected heat from space cooling equipment for producing hot water for other buildings.

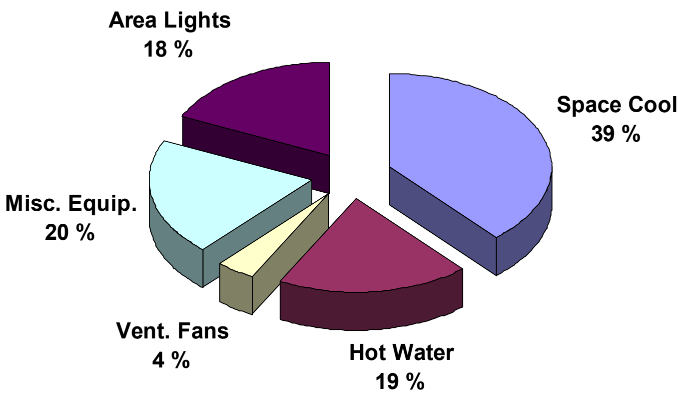


Figure 6-25 Breakdown of energy consumption in a hospital

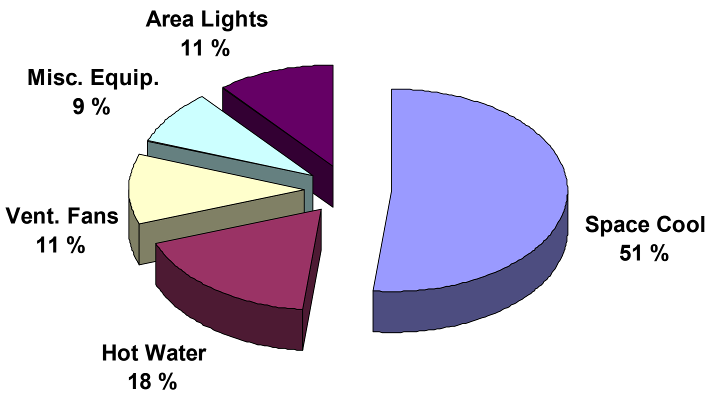


Figure 6-26 Breakdown of energy consumption in a hotel

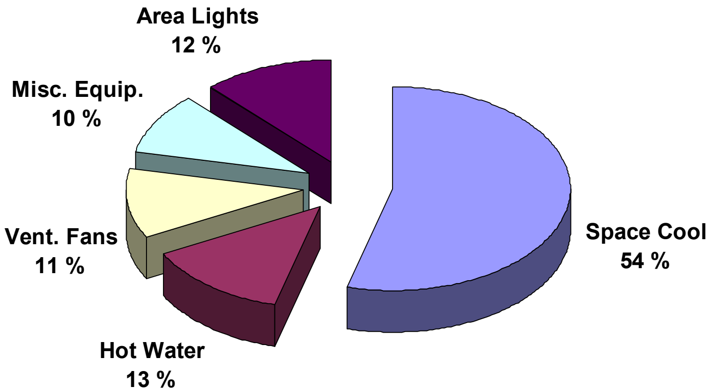


Figure 6-27 Breakdown of energy consumption in a multi family building

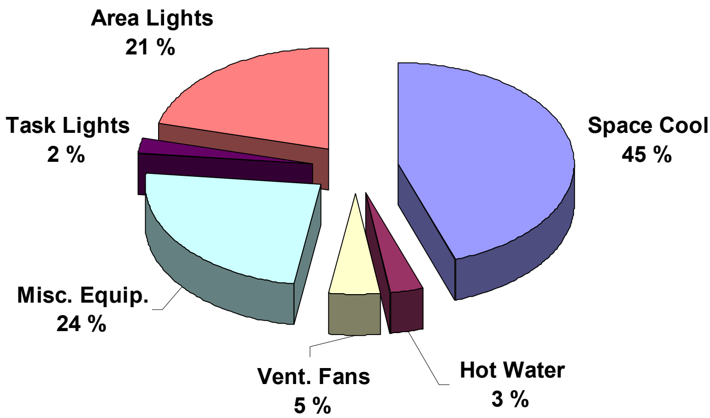


Figure 6-28 Breakdown of energy consumption in an office

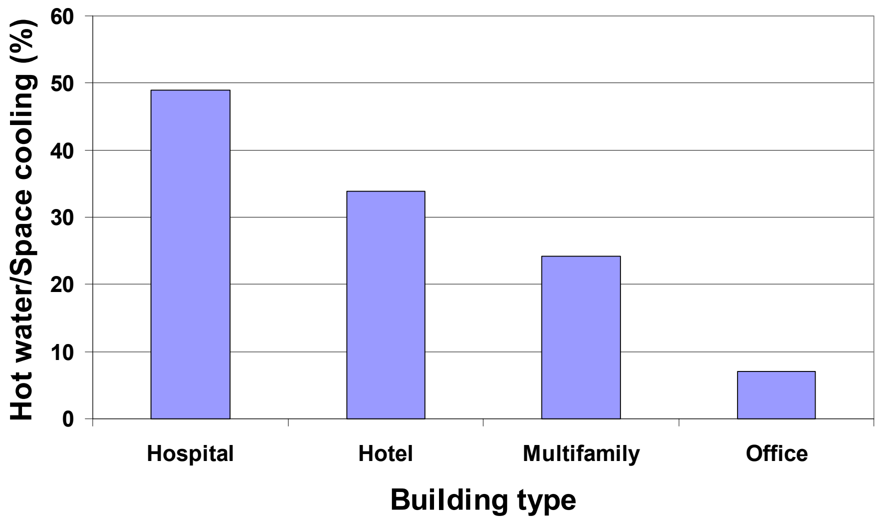


Figure 6-29 Ratio of annual hot water to space cooling energy consumption

Ratio of annual hot water to space cooling energy consumption in different type of buildings is shown in Figure 6-29. Some important aspects regarding the energy consumption of air-conditioning and water heating can be pointed out based on these estimation results.

6.10.1 Hospital

For an hospital, the energy consumption to heat water for domestic hot water purposes takes 19% while that for air-conditioning is 39% of total energy consumption. This is the potential application for applying heat recovery from the water heating side. If the gas coolers are arranged in parallel, there are also possible to improve the air-conditioning side performance. In term of load ratio defined in the previous chapter, the load ratio for hospital in this case is 0.12.

With a series configuration, as said before, the improvement of air-conditioning performance by applying heat recovery is low but gas cooler area needed to produce hot water is much smaller compared to the parallel configuration because the entire mass flow rates of CO_2 flows into the water heating heat exchanger. The problem with parallel configuration when hot storage system is applied does not exist in a series configuration because the system performance is dictated by cooling medium temperature. Moreover, CO_2 temperature entering the heat rejecting heat exchanger will always be lower then CO_2 discharge temperature so that the approach temperature will also become smaller.

Figure 6-30 shows system design for application in a hospital. The flow of refrigerant can be adjusted through valves installed in front of the gas coolers to adjust capacity of both coolers.

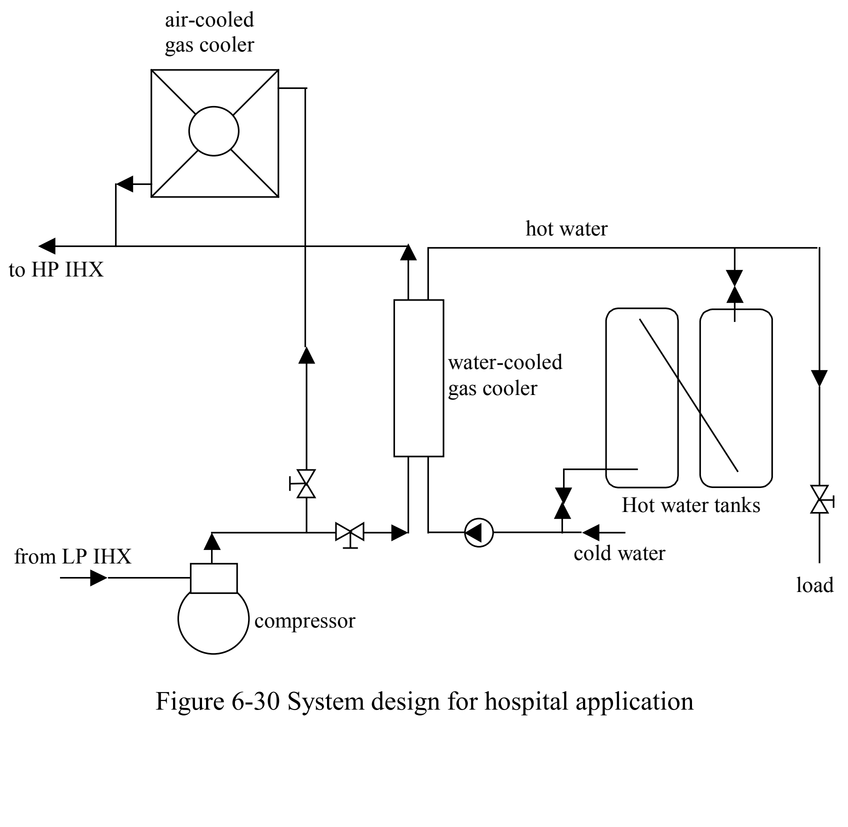


Figure 6-30 System design for hospital application

6.10.2 Hotel and Multi Family

The second promising areas for applying combined system are in hotel and multi family building in which water heating consumes 18% and 13% of total energy consumption, respectively. The heat recovery ratio in the hotel in this case is 0.08 and that in the multi family building is 0.06. This means there will be a little improvement of the air-conditioning side when parallel gas cooler configuration will be used. With the same assumption of 30°C average outdoor air temperature, the improvement of cooling-COP for hotel and multi family building are 1.5% and 1.2%, respectively.

6.10.3 Office

A small percentage of energy consumption to produce hot water in an office seems that hot water heat recovery is not an attractive option for saving energy from hot water side since the energy consumption for heating water is only 3% of total energy consumption. If heat will be recovered to heat water, the gas coolers system should be arranged in series.

Figure 6-31 shows system design for buildings with small percentage of heat recovery. Hot water pump can be run as long as temperature of cold water is lower than outdoor air temperature to avoid heat recovery process degrades the air-conditioning performance.

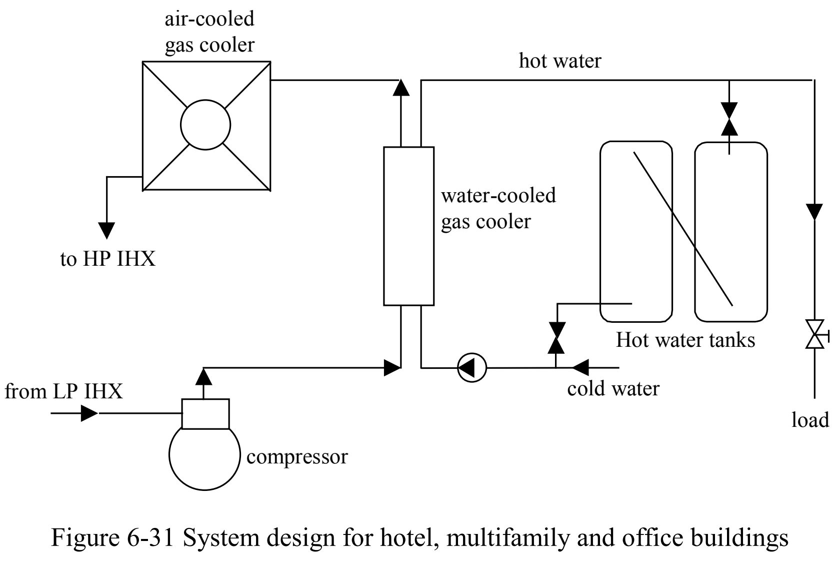


Figure 6-31 System design for hotel, multifamily and office buildings

6.11 Comparison with R22 and Standalone Water Heating System

A system comparison between transcritical CO_2 system and subcritical R22 system can be done by assuming characteristic of R22 system. The following assumptions are set to perform system comparison.

- 0°C evaporation temperature
- 27.7°C outdoor air temperature (average temperature for Jakarta)
- 22.5°C inlet water temperature (average ground water temperature for Jakarta)
- 60°C hot water temperature
- efficiency of stand-alone water heating system is 80%

For R22 air-conditioning, some assumptions are made as follows:

- 0.7 isentropic efficiency
- 2 K superheat
- 5 K subcooling
- 10 K mean temperature different in condenser
- no pressure drop in the connecting lines
-

For combined CO_2 system, some assumptions are made as follows:

- compressor efficiency of the test rig used in the simulation program are reduced by 10%, this is done because CO_2 compressor development currently achieves efficiencies in the range of 70% to 80%.
- no pressure drop in the connecting lines

The energy consumptions for space cooling and water heating obtained from the estimation program is used. Using average outdoor air temperature for Jakarta, the compressor power consumption can be calculated both for R22 and CO_2 air-conditioning systems. For separated air-conditioning and water heating system, the total energy required is energy supplied to the compressor of R22 plus energy supplied to the stand-alone water heating system. For combined CO2 air-conditioning and water heating system, the total energy required is only energy supplied to the compressor since energy to produce hot water is taken from the air-conditioning system.

Figure 6-32 shows annual energy consumption for air-conditioning and water heating system for separated R22 air-conditioning and water heating system and combined transcritical CO_2 air-conditioning and water heating. Relative to total energy consumption for air-conditioning and water heating system, a large saving of annual energy consumption (33%) can be obtained in hospital. Around 24% and 18% of annual energy consumption can be saved in hotel and multifamily, respectively. In offices, 3% saving of energy can be obtained.

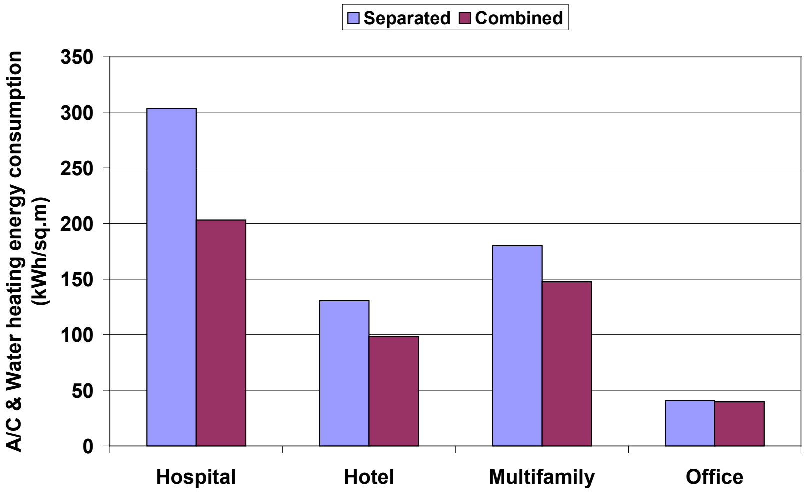


Figure 6-32 Comparison of energy consumption between separated R22 with combined CO_2 air-conditioning and water heating system system

7 Conclusion

Domestic hot water consumes significant amount of energy in buildings like hospitals, hotels, multi family buildings. By recovering part of rejected heat from an air-conditioning to heat water, the need of energy for hot water production can be eliminated. Since variation of outdoor air temperature is small in tropical countries, cooling is needed year round. This is the best condition to perform heat recovery for hot water production. Transcritical cycle using CO_2 as working fluid can be designed as a combined air-conditioning and water-heating system.

A prototype combined air-conditioning and water-heating system using CO_2 as working fluid was designed and built. Thermophysical library of CO_2 was written as computer code and a steady state computer model of the combined system was developed. Based on experimental and simulation results obtained in this work, the following conclusion can be drawn:

- As in a transcritical system, there will be an optimum condition for a combined air-conditioning and water-heating system at which the system reaches the highest cooling-COP. The optimum condition is determined by components parameters such as gas coolers configuration (series or parallel) and presence of internal heat exchanger, and by operational parameters such as cooling medium temperature, water inlet temperature, hot water temperature, evaporation temperature, and percentage of heat recovery.
- In parallel configuration, the optimum condition will be depending on the percentage of heat recovery. The performance of the air-conditioning side is determined by both inlet water temperature and cooling medium temperature. If the inlet water temperature to the water heating heat exchanger is higher than the cooling medium temperature, the air-conditioning side performance will become lower.
- The influence of heat recovery in series configuration on the performance of the air-conditioning side is insignificant and the performance of the air-conditioning side is dictated by cooling medium temperature. The location of the optimum discharge pressure in series configuration is not affected by heat recovery and a higher inlet water temperature to the water heating heat exchanger can be tolerated without degrading the performance of the air-conditioning side.
- The optimum discharge pressure is lower at a lower evaporation temperature. The variation of the optimum discharge pressure with percentage of heat recovery is similar at all evaporation temperatures ran in the experiment.
- For 20°C inlet water temperature, at all evaporation temperatures and cooling medium temperature of 30°C or higher, the cooling-COP is increased as percentage of heat recovery increases. At 25°C cooling medium temperature, the cooling-COP is decreased slightly.

- The location of the optimum discharge pressure is affected by the different between the optimum discharge pressure in air-conditioning mode and that in full recovery mode. If the different is not large there will be a minimum of optimum discharge pressure at certain percentage of heat recovery. As the different becomes larger, there will be no minimum of optimum discharge pressure and the optimum discharge pressure will vary linearly with percentage of heat recovery.
- The cooling capacity was increased at all percentage of heat recoveries and reached the highest value at full recovery mode.
- Producing hot water higher than 70°C with parallel configuration will deteriorate the performance of the air-conditioning side. At this situation, series configuration is a better option.
- Internal heat exchanger is important to improve the system performance. The optimum discharge pressure is lower and the cooling-COP is higher for the system with internal heat exchanger. The optimum cooling capacity of the system with or without internal heat exchanger is similar. The effect of heat recovery on the system with or without internal heat exchanger is also similar.
- The agreement between the experimental data and the simulation results is good. The average deviation is ±5%, which is within the average uncertainties of the measurement system.
- Exergetic efficiency of the combined system is better than exergetic efficiency of the separated R22 air-conditioning and water heating system.
- Based on the results of the energy estimation program, the most promising application of the combined air-conditioning and water-heating system is in hospitals, followed by in hotels and in multi family buildings.

The present study can be regarded as the first combined air-conditioning and water heating system using CO_2 as working fluid. Since the test rig is based on heat pump water heater and the gas coolers has been designed to cover a wide range of operation, it is necessary to design a test rig with proper sizing of gas coolers. The experimental result will be more representatives for an air-cooled air-conditioning system if the heat rejecting heat exchanger is of air-cooled gas cooler type and placed inside a climate chamber.

The steady stated modeling of the combined air-conditioning and water heating system is designed for a constant evaporation temperature. The evaporator model should be developed and integrated to the current system modeling in order to get information on the effect of heat source temperature on the system performance. To handle inlet water temperature variation, a transient model should be developed.

The energy estimation program should be verified by real energy consumption data in type of buildings studied in this work.

Bibliography

Aarlien R., Frivik P.E., 1998, *Comparison Of Practical Performance Between CO_2 And R-22 Reversible Heat Pumps For Residential Use*, Natural working fluid'98, IIR-Gustav Lorentzen Conference, Oslo, Norway

ASHRAE, 1999, *HVAC Application*, ASHRAE Handbook, SI Edition

Beaver A., Yin J., Bullard C. W., Hrnjak P. S., 1999, *Experimental And Model Study Of The Heat Pump/Air Conditioning Systems Based On Transcritical Cycle With R744,* 20th International Congress Of Refrigeration Refrigeration Into The Third Millennium, Sydney, Australia

Bong T.Y., Hawlader M.N., Mahmood W., 1988, *The Prospect Of Incorporating Desuperheaters To Room Air-Conditioners For Tropical Application*, ASHRAE Transaction 94(1): 340-349

Dorgan C.B., Linder R.J., Dorgan C.E., 1999, *Chiller Heat Recovery, Application Guide*, American Society of Heating, Refrigeration, and Air-conditioning Engineers

Dudlye K.F., Paige L.E., Dunshee K.B., 1991, *Controllable Variable Speed Heat Pump For Combined Water Heating And Space Cooling*, United State Patent No. 5050394

EQUEST, 2000, *Introductory Tutorial*, Quick Energy Simulation Tool (http://www.doe2.com)

Furuya S., Mathur G.D., 1999, *A CO_2 Refrigerant System for Vehicle Air-Conditioning*, Phoenix Alternate Refrigerant Forum, Phoenix, AZ

Gosney W.B., 1982, *Principle Of Refrigeration*, Cambridge University Press, Cambridge, UK

Hama H., Imanishi M., Funahashi Y., Kato S., 1984, *Combined Air Conditioning And Hot Water Service System*, United State Patent No. 4448037

Haugthon E., 1997, *Energy Efficiency In The Hotel Sector In Jamaica*, Roundtable on Energy Efficiency-Tourism Sector, Kingston, Jamaica

Hwang Y., Radermacher R., 1998, *Experimental Evaluation Of CO_2 Water Heater*, Natural working fluid'98, IIR-Gustav Lorentzen Conference, Oslo, Norway

Hwang Y., Radermacher R., 1998, *Theoretical Evaluation Of Carbon Dioxide Refrigeration Cycle*, HVAC&R Research, Vol. 4, No. 3

Kotas, T.J., 1995, *The Exergy Method Of Thermal Plant Analysis*, Krieger Publishing Co., Florida

Kyoto Protocol, 1997, *Kyoto Protocol To The United Nations Framework Convention On Climate Change*

Lorentzen G., 1994, *Revival of Carbon Dioxide as Refrigerant*, International Journal of Refrigeration, Vol.17, No. 3, pp. 292

Lorentzen G., 1995, *The Use Of Natural Refrigerants: A Complete Solution To The CFC/HCFC Predicament*, International Journal of Refrigeration Vol.18, No.3, pp.190-197

Montreal Protocol, 1987, *The 1987 Montreal Protocol on Substances that Deplete the Ozone Layer*, United Nations Environment Programme

Nekså P., Rekstad H., Zakeri G.R., Schiefloe P.A., 1998, *CO_2-Heat Pump Water Heater: Characteristic, System Design And Experimental Results*, International Journal of Refrigeration Vol.21, No.3, pp.172-179

Nekså P., Schiefloe P.A., Girotto S., 1998, *Commercial Refrigeration Using CO_2 As Refrigerant - System Design And Experimental Results*, Natural working fluid'98, IIR-Gustav Lorentzen Conference, Oslo, Norway

Olszewski M., 1984, *Economic Viability Of Heat Pump Desuperheaters For Supplying Domestic Hot Water*, ASHRAE Transaction 90(1B): 169-178

Pettersen J., Aarlien R.1997, *Progress In CO_2 Vapour Compression System*, Oji International Seminar, Tomakomai City, Hokkaido, Japan

Pettersen J., Hafner A., Skaugen G., 1998, *Development Of Compact Heat Exchanger For CO_2 Air-Conditioning Systems*, International Journal of Refrigeration, vol.21, No.3

Pettersen J., Vestbøstad K., 2000, *Heat Transfer And Pressure Drop For Flow Of Supercritical And Subcritical CO_2 In Microchannel Tubes*, IEA Annex 27 Workshop on: Selected Issue on CO2 as Working Fluid in Compression System, Trondheim, Norway

Pitla S.S., Robinson D.M., Groll E.A., Ramadhyani S., 1998, *Heat Transfer From Supercritical Carbon Dioxide In Tube Flow: A Critical Review*, HVAC&R Research, Vol.4, No.3

Rieberer R., Gassler M., Halozan H., 2000, *Control of CO_2 Heat pumps*, 4th IIR-Gustav Lorentzen Conference on Natural working fluids, Purdue

Robinson D.M., Groll E.A., 1998, *Heat Transfer Analysis of Air-to-Carbon Dioxide Two-Phase Heat Absorption and Supercritical Heat Rejection*, HVAC&R Research, Vol. 4, No. 4

Sakakibara H., Nishida S., Onimaru S. Uichi S., Ozaki Y., 1997, *Refrigerant System With Pressure Control Valve*, United State Patent No. 5890370

Shulters E.S., 1944, *Modern Marine Refrigeration*, Cornell Maritime Press, New York

Skaugen G., 2000, *Simulation Of Extended Surface Heat Exchanger Using CO_2 As Refrigerant*, 4th IIR-Gustav Lorentzen Conference on Natural working fluids, Purdue

Span R., Wagner W., 1996, *A New Equation Of State For Carbon Dioxide Covering The Fluid Region From The Triple-Point Temperature To 1100 K At Pressures Up To 800 MPa*, J. Phys. Chem. Ref. Data, Vol.25, No.6, pp. 1509-1596

Stoecker W.F., 1971, *A Generalized Program For Steady-State System Simulation*, ASHRAE Transaction 77(1): 140-148

Stoecker W.F., 1989, *Design Of Thermal Systems*, McGraw-Hill, Inc.

Strømmen I., Bredesen A.M., Eikevik T., Pettersen J., Aarlien R., 2000, *Refrigeration, Air-Conditioning And Heat-Pump System For The 21st Century*, Bulletin of the International Institute of Refrigeration, 2000-2

Tackett R.K., 1989, *Case Study: Office Building Uses Ice Storage, Heat Recovery, And Cold Air Distribution*, ASHRAE Transaction 95(1): 1113-1121

Tackett R.K., 1987, *Result From Operation Of A Large Membrane Stratified Cool Storage System With Heat Recovery*, ASHRAE Transaction 93(1): 728-739

VDI, 1993, *Heat Transfer In Flow Through Coils*, VDI-Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (GCV), Trans.: J.W.Fullarton

Vesovic V. et al., 1990, *The Transport Properties Of Carbon Dioxide*, J. Phys. Chem. Ref. Data, Vol.19, No.3, pp. 763-808

Wang H., 1991, *Modeling Of A Refrigeration System Coupled With A Refrigerated Room*, Ph.D. Thesis, Faculty. Of Mech. Eng., Delft University of Technology,

Yarral M.G., White S.D., Cleland D.J., Kally R.D., Hedley R.A., 1999, *Performance Of A Transcritical Cycle CO_2 Heat Pump For Simultaneous Refrigeration And Water Heating*, 20th International Congress Of Refrigeration Refrigeration Into The Third Millennium, Sydney, Australia

Ying W.M., 1989, *Performance of Room Air Conditioner Use for Cooling and Hot water heating*, ASHRAE Transaction 95(1): 441-444

Zakeri G.R., Nekså P., Rekstad H.,1999, *Work Package 11: Prototype system design and construction Final Report 01/99*, Energy Efficient And Environmental Friendly Heat Pumping Systems Using CO_2 As Working Fluid, COHEPS

Appendix

A. Library of Thermophysical Properties of CO_2

Thermophysical properties are a core of simulation program developed in this study. It was based upon extended equation of state from Span and Wagner (Span et al., 1996) for thermodynamic properties and Vesovic et al. (Vesovic et al., 1990) for transport properties.

Basic equation

In Span and Wagner equation of stated, reduced Helmholtz energy function was composed of two parts, an ideal gas part, and a residual part. This function was expressed as a function of temperature and density as follows:

$$\frac{A(\rho,T)}{RT} = \Phi(\rho,T) = \Phi^{o}(\rho,T) + \Phi^{r}(\rho,T) \qquad \text{(A.1)}$$

in which $\frac{A(\rho,T)}{RT} = \Phi(\rho,T)$ is reduced Helmholtz energy, $\Phi^{o}(\rho,T)$ is the ideal gas part and $\Phi^{r}(\rho,T)$ is the residual part.

From (A.1), all thermodynamics properties can be calculated by combining the derivatives of (A.1). Some of the properties with its definition can be seen in Table below. All derivatives of reduced Helmholtz energy can be found in the paper of Span (Span et al., 1996).

Relation of saturated properties as a function of temperature was also given in the paper and some of it is:

- Vapor pressure
- Liquid density
- Vapor density

The other saturated properties can be calculated by inserting temperature and the corresponding saturated properties. For example, to calculate saturated liquid entropy at certain temperature, saturated liquid density at this temperature must first be calculated. Then, with these temperature and saturated density, saturated liquid entropy can be calculated.

Table A-1 Relation of thermodynamic properties to reduced Helmholtz energy (Span et al., 1996)

Property and common thermodynamic definition	Relation to reduced Helmholtz energy and its derivatives
Pressure: $p(\rho,T) = -\left(\partial A/\partial v\right)_T$	$\frac{p(\delta,\tau)}{\rho RT} = 1 + \delta\Phi^r_\delta$
Entropy: $s(\rho,T) = -T\left(\partial A/\partial T\right)_v$	$\frac{s(\delta,\tau)}{R} = \tau(\Phi^o_\tau + \Phi^r_\tau) - \Phi^o - \Phi^r$
Enthalpy: $h(\rho,T) = A - \left(\partial A/\partial v\right)_T$	$\frac{h(\delta,\tau)}{RT} = 1 + \tau(\Phi^o_\tau + \Phi^r_\tau) + \delta\Phi^r_\delta$
Isobaric heat capacity: $c_p(\rho,T) = \left(\partial h/\partial T\right)_p$	$\frac{c_p(\delta,\tau)}{R} = -\tau^2(\Phi^o_{\tau\tau} + \Phi^r_{\tau\tau}) + \frac{(1+\delta\Phi^r_\delta - \delta\tau\Phi^r_{\delta\tau})^2}{1+2\delta\Phi^r_\delta + \delta^2\Phi^r_{\delta\delta}}$

Transport properties of CO_2 from Vesovic et al. are dynamic viscosity and thermal conductivity. They are also expressed as a function of temperature and density. The equation consists of three parts, contribution in the limit of zero-density, excess properties from ideal gas, and critical enhancement close to critical point. However, since the critical viscosity enhancement is very small (Vesovic et al., 1990) it is ignored here so that the viscosity equation was only consists of two parts.

$$\begin{aligned} \eta(\rho,T) &= \eta^o(T) + \frac{\Delta\eta_g}{1+\exp(-Z^*\cdot(T-T_s))} \\ &+ \frac{\Delta\eta_g}{\{1+\exp(-Z^*\cdot(T-T_s))\}\cdot\{1+\exp(-Z^*\cdot(\rho-\rho_s))\}} \\ &+ \frac{\eta_l-\eta^o}{\{1+\exp(-Z^*\cdot(T-T_s))\}\cdot\{1+\exp(-Z^*\cdot(\rho-\rho_s))\}} \end{aligned} \tag{A.2}$$

Temperature and pressure as inputs

In most of a cycle calculation, temperature and pressure are given as inputs and the other properties are calculated at this temperature and pressure. However, the equation of state uses temperature and density as its inputs. Therefore, iteration scheme must be set so that density can be calculated with temperature and pressure as its inputs. This is the critical part of calculation since iteration process will make computational time longer. Here, iteration scheme using Newthon-Raphson method was applied.

After completing iteration process in which density was obtained at given temperature and pressure, the other properties can be calculated easily as one already has temperature and density to be used as inputs. The calculation process is the same as before.

Other input combinations

Other input combinations are also needed. To determine the state of discharge gas from a compression process, entropy and pressure is given instead of temperature and pressure and the other properties must be determined with these properties as its inputs. Therefore, it is also needed to have a set of functions that use other properties but temperature and pressure. The above procedure can be applied in a similar manner to the other combination of inputs but now there will be two iteration blocks. The first block of iteration is to determine density at given pressure (or guess pressure) and guess temperature (or temperature), and the second one is to determine temperature (or pressure) at desired properties. Here, the first block of iteration acts as an inner iteration while the second one acts as an outer iteration.

Suppose entropy and pressure are given as inputs and enthalpy is need as the output. The first block of iteration must determine density at given pressure and guess temperature. With these density and guess temperature, entropy can be calculated. The second block of iteration then compares this calculated entropy with the given entropy and updated the guess temperature if the difference between the calculated entropy and the given entropy is still higher than an allowable difference. Iteration process is repeated until the difference is lower than the allowable difference.

CO_2 Library code

It is very useful to have those equations in a computer code that can accelerate process computation. With current spreadsheet program, it becomes very easy to perform cycle calculation event with thousand of refrigerant states that have to be determined. It seems impossible to do such a thing in the past when all calculation had to be done by hand. Moreover, some spreadsheet program comes with the ability to be linked with other external program, like one version of [1]Microsoft Excel™ that contains programming language [2]Visual Basic Application™.

However, if the entire code of refrigerant library was written purely in this programming language, the execution time will be extremely long since it is an interpreted programming language. The execution time can be reduced dramatically by writing the code in a compiled form and then integrates it with the spreadsheet program. The other calculation that does not need so much iteration can be coded within the spreadsheet itself.

The library of thermophysical properties of CO_2 called **CO2Lib** developed in this study was written in [3]Delphi™ language. It contains 113 functions that can be expanded easily in the future. The following table lists all the functions contained in the code. The functions format is **name_xy(x,y)** where name is the name of the function, x is the first input, and y is the second input.

List of functions contained in CO2Lib

Molw	Tcrit	Pcrit
VsatGas_t(t)	VsatLiq_t(t)	VsatGas_p(p)
H_tv(t,v)	S_tv(t,v)	Cv_tv(t,v)
UsatGas_t(t)	HsatGas_t(t)	SsatGas_t(t)
UsatLiq_t(t)	HsatLiq_t(t)	SsatLiq_t(t)
UsatGas_p(p)	HsatGas_p(p)	SsatGas_p(p)
UsatLiq_p(p)	HsatLiq_p(p)	SsatLiq_p(p)
V_tp(t,p)	U_tp(t,p)	H_tp(t,p)
S_tp(t,p)	Sos_tp(t,p)	T_sp(s,p)
Cv_sp(s,p)	Cp_sp(s,p)	Sos_sp(s,p)
U_hp(h,p)	S_hp(h,p)	Cv_hp(h,p)
V_st(s,t)	U_st(s,t)	H_st(s,t)
x_th(t,h)	x_ph(p,h)	V_tx(t,x)
Cv_tx(t,x)	Cp_tx(t,x)	Sos_tx(t,x)
S_px(p,x)	Cv_px(p,x)	Cp_px(p,x)
DvsatLiq_t(t)	DvsatGas_t(t)	DvsatLiq_p(p)
Tc_tv(t,v)	Tc_tp(t,p)	TcsatLiq_t(t)
Tc_tx(t,x)	Tc_px(p,x)	drhodp_tv(t,v)
U_ht(h,t)	S_ht(h,t)	Cv_ht(h,t)

Vcrit	Psat(t)	Tsat(p)
VsatLiq_p(p)	P_tv(t,v)	U_tv(t,v)
Cp_tv(t,v)	Cliq_tv(t,v)	Sos_tv(t,v)
CvsatGas_t(t)	CpsatGas_t(t)	SoSsatGas_t(t)
CvsatLiq_t(t)	CpsatLiq_t(t)	SoSsatLiq_t(t)
CvsatGas_p(p)	CpsatGas_p(p)	SoSsatGas_p(p)
CvsatLiq_p(p)	CpsatLiq_p(p)	SoSsatLiq_p(p)
Hg_tp(t,p)	Cp_tp(t,p)	Cv_tp(t,p)
V_sp(s,p)	U_sp(s,p)	H_sp(s,p)
T_hp(h,p)	Tg_hp(h,p)	V_hp(h,p)
Cp_hp(h,p)	Sos_hp(h,p)	P_st(s,t)
Cv_st(s,t)	Cp_st(s,t)	Sos_st(s,t)
U_tx(t,x)	H_tx(t,x)	S_tx(t,x)
V_px(p,x)	U_px(p,x)	H_px(p,x)
Sos_px(p,x)	Dv_tv(t,v)	Dv_tp(t,p)
DvsatGas_p(p)	Dv_tx(t,x)	Dv_px(p,x)
TcsatGas_t(t)	TcsatLiq_p(p)	TcsatGas_p(p)
St_t(t)	P_ht(h,t)	V_ht(h,t)
Cp_ht(h,t)	Sos_ht(h,t)	Vg_tp(t,p)

Molw	:	molar mass (g/gmol)
crit	:	critical
sat	:	saturated
Liq	:	liquid phase
Gas	:	gas phase
P,p	:	pressure (Pa)
T,t	:	temperature (°C)
V,v	:	specific volume (kg/m^3)
H,h	:	specific enthalpy (J/kg)
U	:	specific internal energy (J/kg)
S,s	:	specific entropy (J/kgK)
Cp	:	isobaric specific heat capacity (J/kgK)
Cv	:	isochoric specific heat capacity (J/kgK)
SoS	:	speed of sound (m/s)
Dv	:	dynamic viscosity (Pa.s)
Tc	:	thermal conductiviy (W/mK)
St	:	surface tension (N/m)
x	:	vapor fraction

Integration of the library with Visual Basic Application™

The code was compiled as a dynamic link library with Delphi™ 5.0 that can be called by other program like Microsoft Excel™ containing Visual Basic Application™. To be able to use one of the functions in the library, it must be linked by a special syntax depending on the program that calls the function. For Visual Basic Application™, the syntax is as follows:

Declare function **name** *lib* "**library**" (*byval* **x** *as double*, *byval* **y** *as double*) *as double*

For example, if H_tp function will be called, the syntax becomes:

Declare function **H_tp** *lib* "**CO2Lib.dll**" (*byval* **t** *as double*, *byval* **p** *as double*) *as double*

This function now can be accessed both in a spreadsheet of Microsoft Excel™ or within Visual Basic Application™ itself.
For illustration, let us calculate thermodynamic properties at 20°C and 35 bar.

The program will give:

v	kg/m^3	0.012367
u	J/kg	720340.6
h	J/kg	763621.2
s	J/kg	4069.6
cv	J/kgK	778.0
cp	J/kgK	1329.9
Sos	m/s	234.3
Dv	Pa.s	1.468E-05
Tc	W/mK	0.0188

Two figures below show an impression of how the library is used in a spreadsheet program.

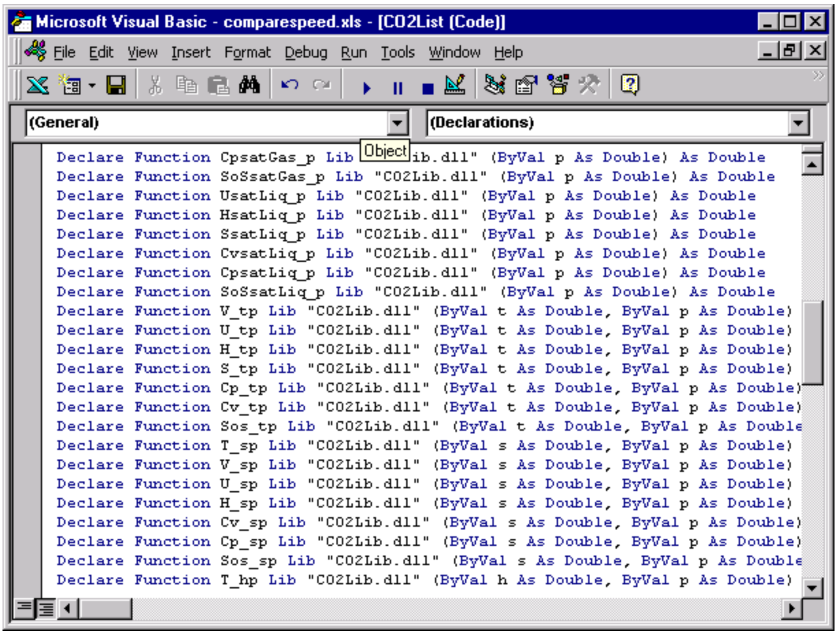


Figure 1 Declaring the library functions in Visual Basic Application™ language

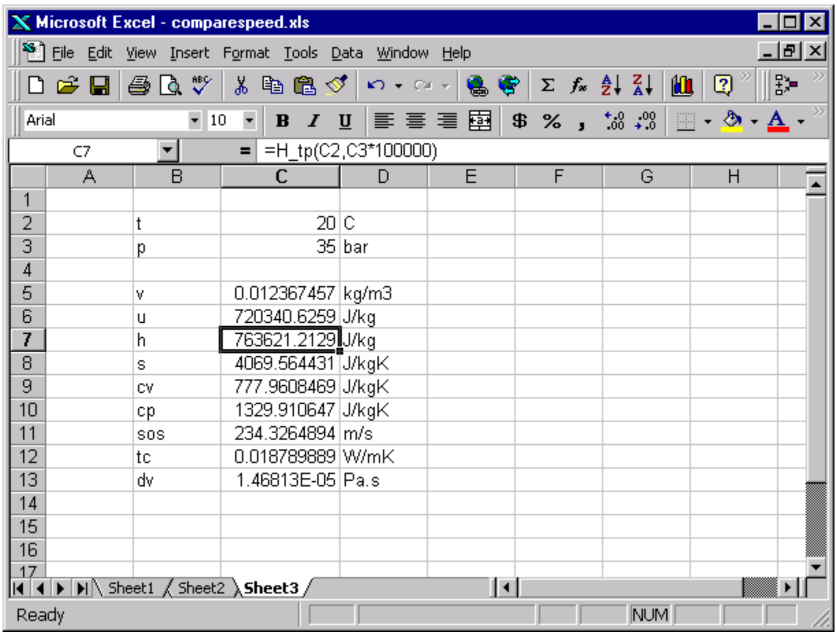


Figure 2 Using the library in Microsof Excel™ spreadsheet

[1] Microsoft Excel™ is a trademark of Microsoft Corp.
[2] Visual Basic Application™ is a trademark of Microsoft Corp.
[3] Delphi™ is a trademark of Borland Software Corp.

B. MS-Excel Simulation Program

This program has been developed as part of the study. The purpose is to have a tool date can be used to study system behavior in a broad range of operating conditions. By changing component parameters, the effect of these parameters to the system performance can be studied and evaluated. Part of the simulation program, the heat exchanger programs, allows to design a gas cooler or internal heat exchanger by providing a detail geometry of the heat exchanger being designed and giving the design condition. Only double pipe heat exchanger type that can be designed.

There are three options in the program:
1. Air-conditioning mode (air-cooled)
2. Water heating mode (water-cooled)
3. Combined Air-conditioning and water heating mode

As explained in Chapter 3, air-cooled gas cooler here has been replaced by a water-cooled gas cooler just as in the test rig. The cooling medium is water but its flow rates have been adjusted to provide the same thermal mass as if it was cooled by air.

In air-conditioning mode, only air-cooled gas cooler needs to be specified. All irrelevant options will be inactive. There will be no heat recovery in this mode so that all heat from CO2 is transferred to ambient through the air-cooled gas cooler. By filling evaporation temperature, discharge pressure, and rotational speed, calculation can be started.

In water-heating mode, water-cooled gas cooler has to be specified. All irrelevant options will be inactive. In this mode, all rejected heat is transferred to water. There are two options to control how water will be heated: fixed flow rates or fixed temperature. Fixed flow rates means the water flow rates is fixed and hot water temperature is allowed to vary. In fixed temperature option, hot water temperature is specified and the program will adjust the water flow rates to meet the target hot water temperature. By filling evaporation temperature, discharge pressure, and rotational speed, calculation can be started.

In combined mode, both gas coolers have to be specified. In addition to that, target load has also to be given. Target load here can be obtained by specifying percentage of heat recovery, for example 0.25. With rejected heat obtained in air-conditioning mode simulation at a certain discharge pressure, one can multiply this rejected by the percentage of heat recovery to get the target load. By filling evaporation temperature, discharge pressure, and rotational speed, calculation can be started.

It is also possible to choose whether to use internal heat exchanger or not.

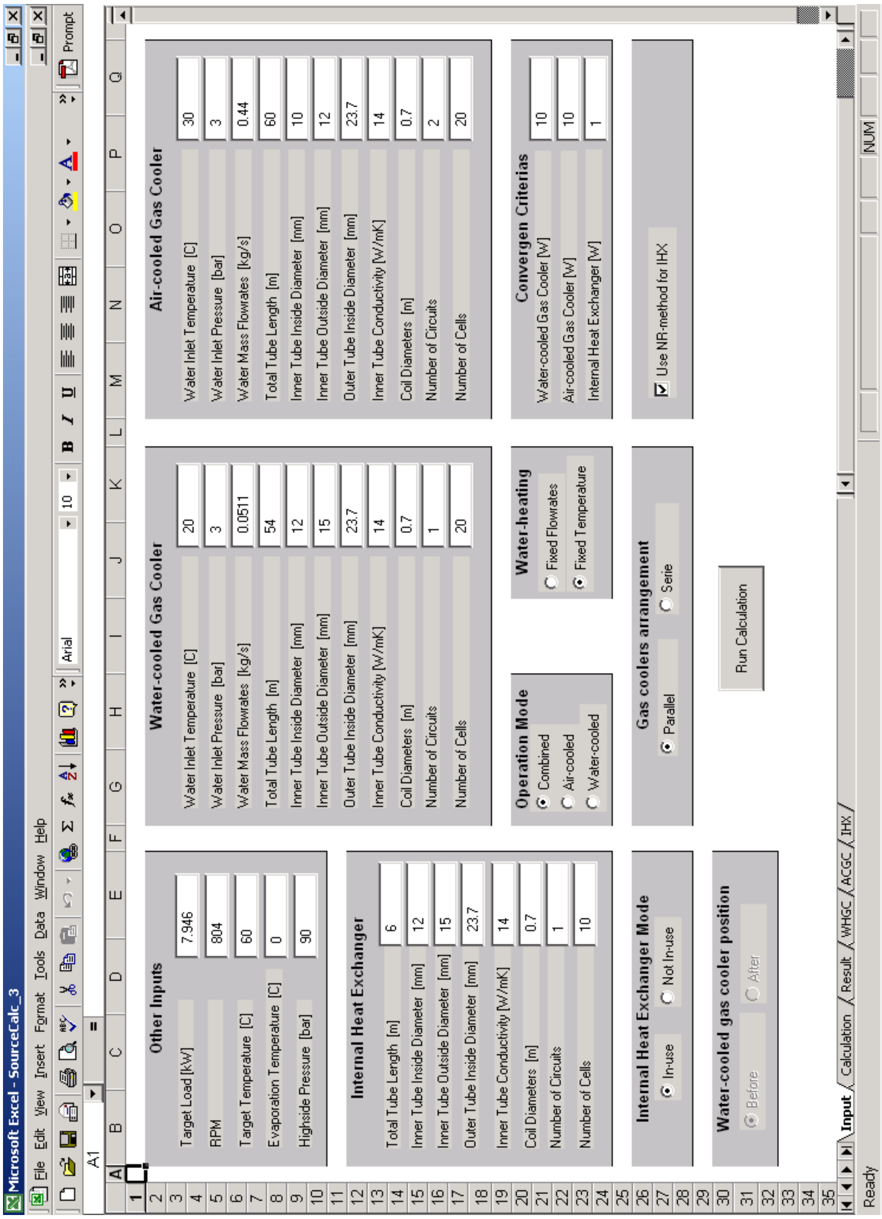


Figure 1 MS Excel workbook showing the main worksheet of the simulation program

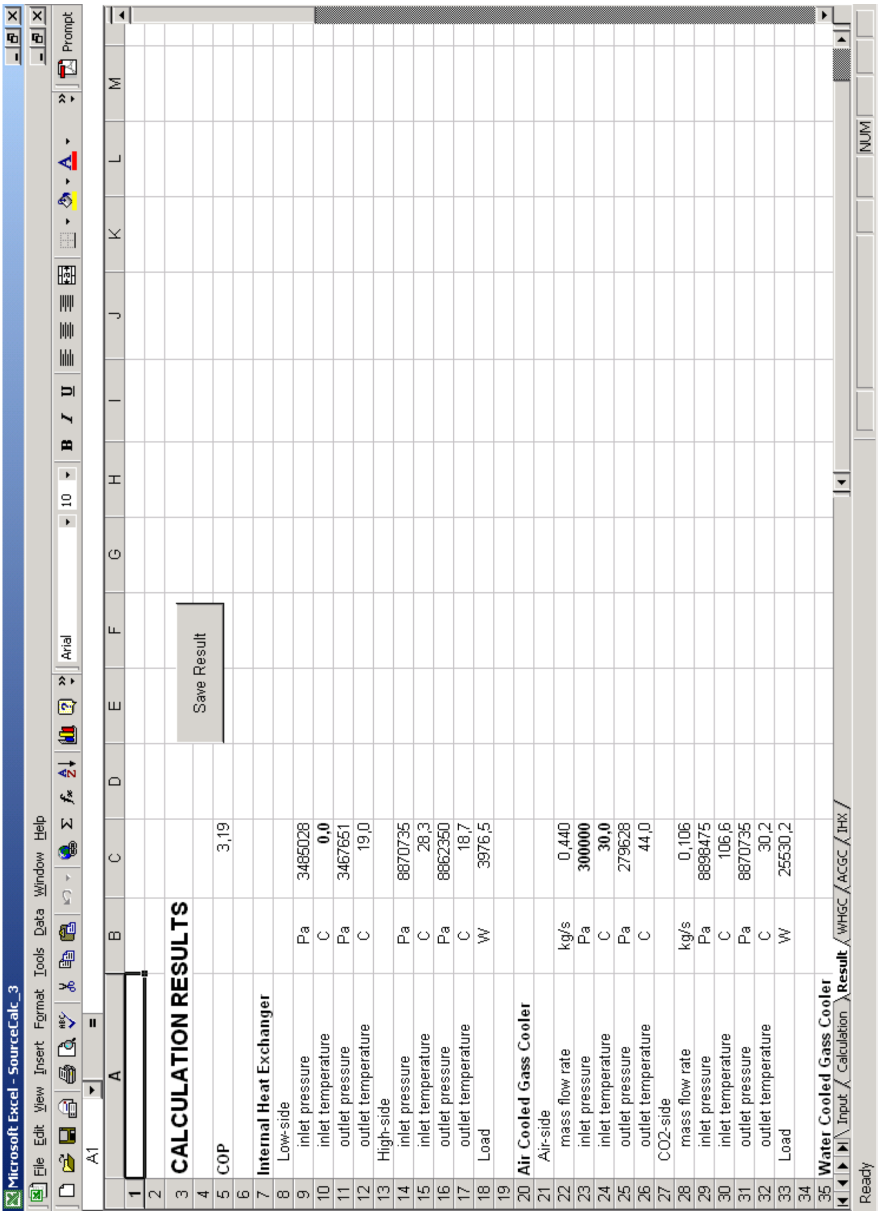


Figure 2 MS Excel workbook showing the result worksheet of the simulation program

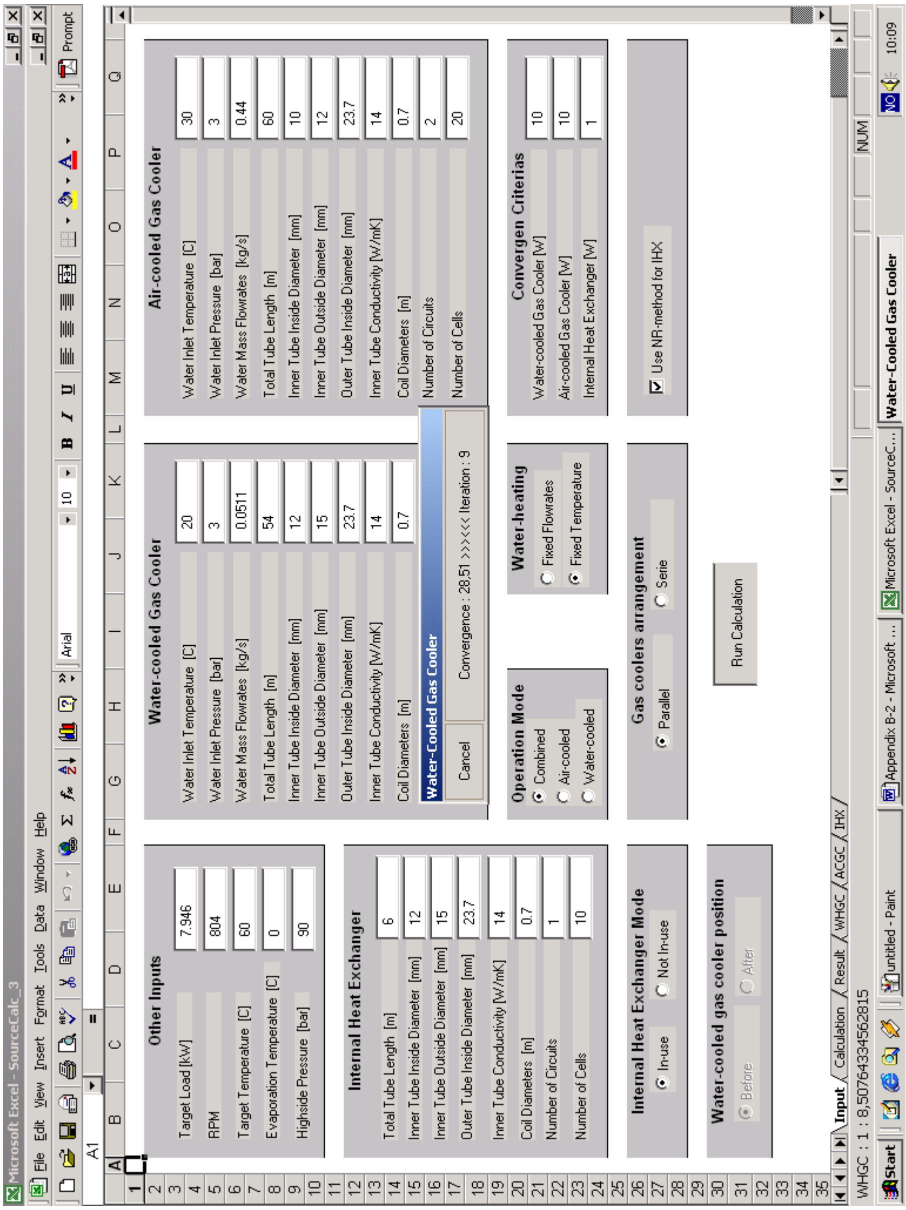


Figure 3 MS Excel workbook showing calculation in action

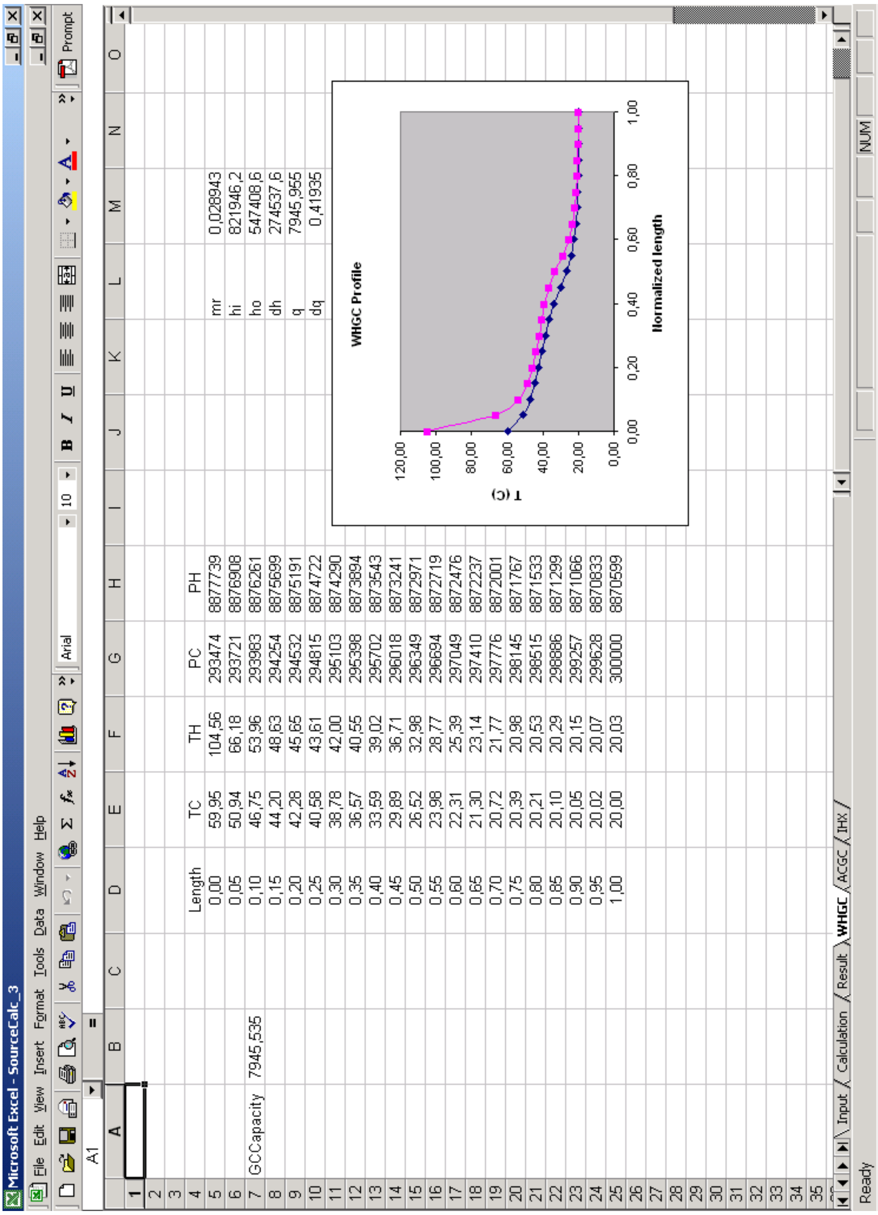
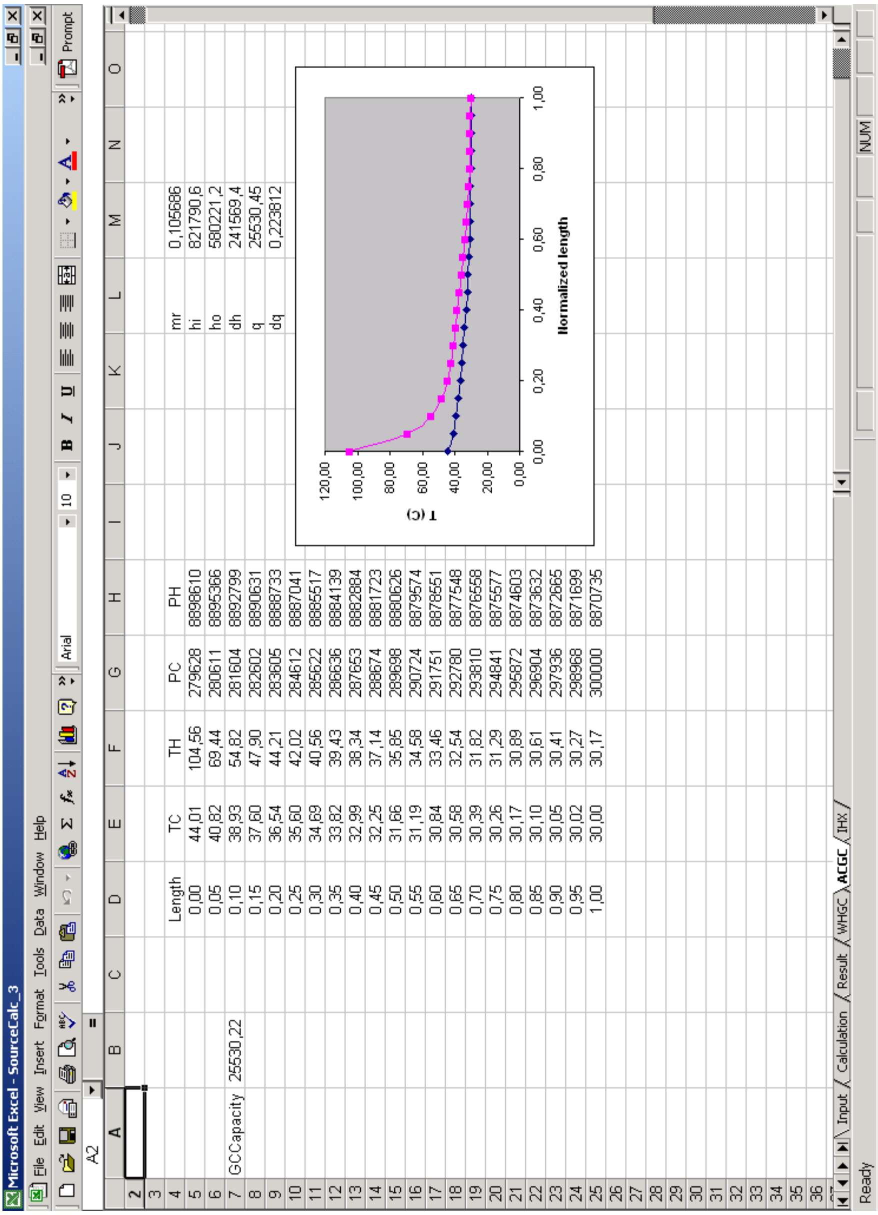


Figure 4 MS Excel workbook showing the calculation result of the water heating heat exchanger



Length	TC	TH	PC	PH
0,00	44,01	104,56	279628	8898610
0,05	40,82	69,44	280611	8895366
0,10	38,93	54,82	281604	8892799
0,15	37,60	47,90	282602	8890631
0,20	36,54	44,21	283605	8888733
0,25	35,60	42,02	284612	8887041
0,30	34,69	40,56	285622	8885517
0,35	33,82	39,43	286636	8884139
0,40	32,99	38,34	287653	8882884
0,45	32,25	37,14	288674	8881723
0,50	31,66	35,85	289698	8880626
0,55	31,19	34,58	290724	8879574
0,60	30,84	33,46	291751	8878551
0,65	30,58	32,54	292780	8877548
0,70	30,39	31,82	293810	8876558
0,75	30,26	31,29	294841	8875577
0,80	30,17	30,89	295872	8874603
0,85	30,10	30,61	296904	8873632
0,90	30,05	30,41	297936	8872665
0,95	30,02	30,27	298968	8871699
1,00	30,00	30,17	300000	8870735

mr	0,105686
hi	821790,6
ho	580221,2
dh	241569,4
q	25530,45
dq	0,223812

Figure 5 MS Excel workbook showing the calculation result of the heat rejecting heat exchanger

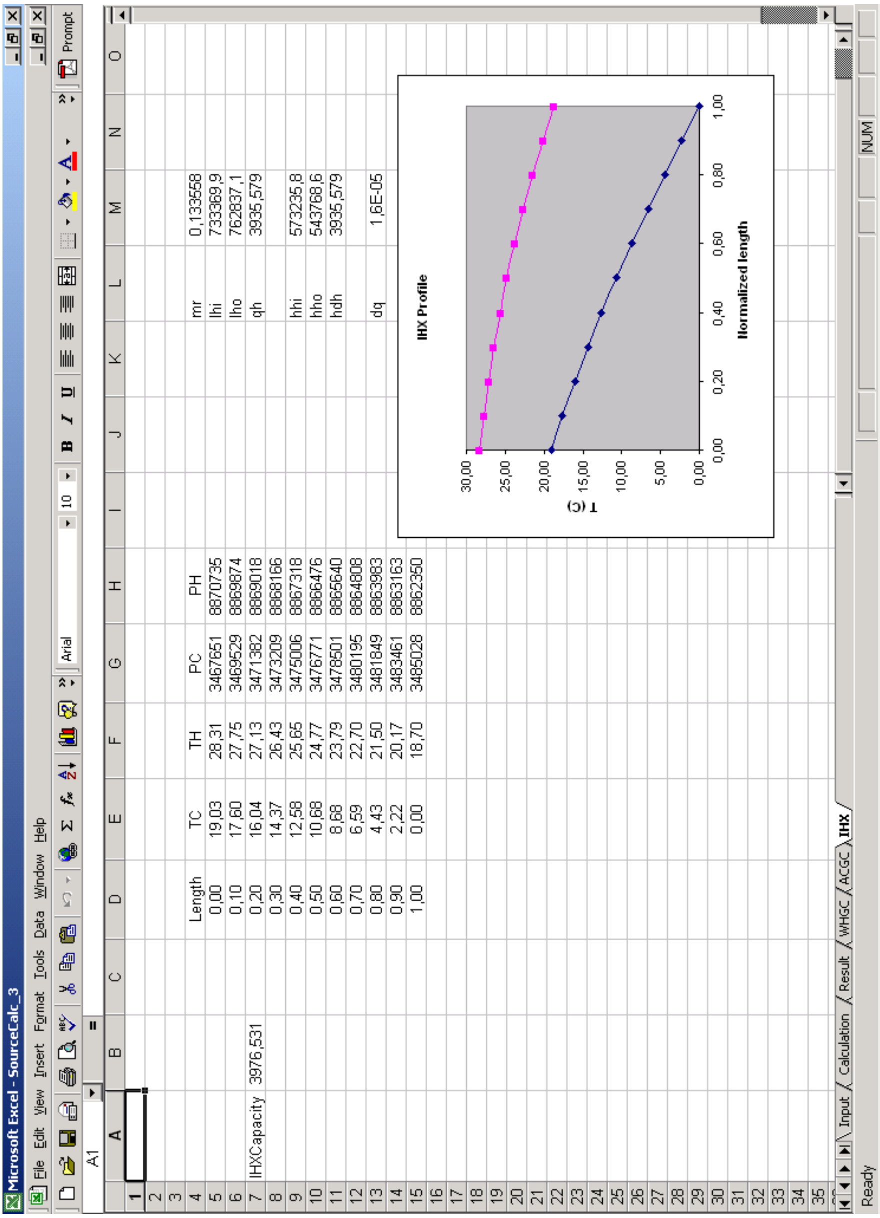


Figure 6 MS Excel workbook showing the calculation result of the internal heat exchanger

C. Some Experimental Results

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0626_1
2. Outdoor air temp.: 30,2 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	15:30:38
4. Heat Rejection Pressure: 80,7 bar	80 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	3,4 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,83	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,46	
Cooling-COP (shaft)	[-]			2,66	± 15,00 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,128	± 9,76 %
Temp. evapor, outlet (T8)	°C			-0,2	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			19,0	± 15,00 %
LMTD	K			2,71	
Overall heat transfer coeff.	W/m^2K			1309	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,128	± 9,76 %
Temp.water gascooler inlet (T14)	°C			30,2	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			45,0	± 0,50 °C
Temp. differential (T32-T14)	K			3,5	
Heating capacity	kW			25,4	± 4,76 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			5,4	
LMTD	K			14,17	
Overall heat transfer coeff. outer	W/m^2K			1352	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			6,7	
Power consumption, shaft	kW			7,1	± 0,17 %
Power input, inverter	kW			8,6	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,6	
Pressure compressor, discharge (P2)	bar			80,7	
Temp. compressor, suction (T10)	°C			27,5	± 0,50 °C
Temp. compressor, discharge (T1)	°C			105,9	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,5	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,6	
Volumetric efficiency (based on water)	[-]			0,87	± 9,77 %
Isentropic efficiency (based on water)	[-]			0,85	± 9,93 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			8,4	
Gascooler, water	m/s			0,4	
Compressor, suction	m/s			10,1	
Internal heat exchanger, HP, inlet	m/s			2,2	
Internal heat exchanger, HP, outlet	m/s			1,7	
Internal heat exchanger, LP, inlet	m/s			6,4	
Internal heat exchanger, LP, outlet	m/s			7,0	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0626_1
2. Outdoor air temp.: 30,2 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	15:30:38
4. Heat Rejection Pressure: 80,7 bar	80 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	3,4 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	Measure pnt.	Unit	Min	Mean	Max	STDEV
Temp.evap. inlet	T13	°C	4,0	4,1	4,7	0,2
Temp.differential, evap.	Td1	K	2,6	2,6	2,7	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	21,9	21,9	22,0	0,0
Temp.water gascooler, inlet 1	T14	°C	29,4	29,4	29,4	0,0
Temp.water gascooler, inlet 2	T3	°C	29,2	29,2	29,2	0,0
Temp.water gascooler, outlet	T15	°C	29,9	29,9	30,0	0,0
Temp.differential, gas cooler	Td2	K	-1,6	-0,1	1,6	1,1
Temp.water drain cock	T16	°C	29,2	29,2	29,2	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,115	0,115	0,115	0,0
Water flow, drain cock	F3	l/s	0,033	0,033	0,033	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	84,6	84,8	84,9	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	8,6	8,6	8,6	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	105,3	105,9	106,1	0,3
Temp. gascooler inlet	T2	°C	32,8	32,8	32,9	0,0
Temp. gascooler outlet	T32	°C	29,5	29,7	30,0	0,1
Temp. internal heat ex., HP outlet	T4	°C	29,6	29,8	30,0	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	29,5	29,8	29,9	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	27,3	27,5	27,7	0,1
Temp. throttle valve, inlet	T7	°C	27,3	27,5	27,7	0,1
Temp. evapor, outlet	T8	°C	-0,3	-0,2	0,3	0,1
Temp. LP-receiver, outlet	T9	°C	4,0	4,1	4,5	0,1
Temp. internal heat ex., LP inlet	T33	°C	2,7	2,8	3,1	0,1
Temp. compressor, suction	T10	°C	27,4	27,5	27,6	0,1
Pressure compressor, suction	P1	bar	32,5	32,6	32,9	0,1
Pressure compressor, discharge	P2	bar	80,5	80,7	80,9	0,1
Pressure gascoler, inlet	P002	bar	78,8	78,9	79,0	0,1
Pressure throttling valve,inlet	P3	bar	78,9	79,1	79,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,03	0,21	0,1
Pressure differential evapor LP	Pd2	bar	2,09	2,12	2,16	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,09	2,12	2,16	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,4	7,5	7,6	0,1
Temp.water gascooler, inlet 1	T001	°C	30,1	30,2	30,3	0,1
Temp.water gascooler, outlet	T002	°C	44,9	45,0	45,1	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	102,7	103,2	103,4	0,2
Temp. gascooler outlet	T004	°C	33,6	33,7	33,8	0,0
Pressure gascooler, inlet	P001	bar	79,4	79,5	79,7	0,1
Pressure differential gascooler HP	Pd001	bar	0,60	0,61	0,62	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0626_2
2. Outdoor air temp.: 30,2 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	16:00:38
4. Heat Rejection Pressure: 85,4 bar	85 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,7 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,21	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,74	
Cooling-COP (shaft)	[-]			2,98	± 4,66 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,127	± 4,35 %
Temp. evapor, outlet (T8)	°C			0,1	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			22,5	± 4,66 %
LMTD	K			3,21	
Overall heat transfer coeff.	W/m^2K			1313	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,127	± 4,35 %
Temp.water gascooler inlet (T14)	°C			30,2	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			47,2	± 0,50 °C
Temp. differential (T32-T14)	K			2,2	
Heating capacity	kW			29,2	± 4,16 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			4,2	
LMTD	K			13,81	
Overall heat transfer coeff. outer	W/m^2K			1090	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,0	
Power consumption, shaft	kW			7,6	± 0,17 %
Power input, inverter	kW			9,1	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,6	
Pressure compressor, discharge (P2)	bar			85,4	
Temp. compressor, suction (T10)	°C			24,0	± 0,50 °C
Temp. compressor, discharge (T1)	°C			107,6	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,6	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,4	
Volumetric efficiency (based on water)	[-]			0,84	± 4,37 %
Isentropic efficiency (based on water)	[-]			0,83	± 4,69 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,8	
Gascooler, water	m/s			0,4	
Compressor, suction	m/s			9,8	
Internal heat exchanger, HP, inlet	m/s			1,7	
Internal heat exchanger, HP, outlet	m/s			1,5	
Internal heat exchanger, LP, inlet	m/s			6,3	
Internal heat exchanger, LP, outlet	m/s			6,8	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0626_2
2. Outdoor air temp.: 30,2 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	16:00:38
4. Heat Rejection Pressure: 85,4 bar	85 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,7 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	4,8	4,8	4,8	0,0
Temp.differential, evap.	Td1	K	3,0	3,1	3,1	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	21,2	21,4	21,7	0,2
Temp.water gascooler, inlet 1	T14	°C	29,1	29,2	29,3	0,1
Temp.water gascooler, inlet 2	T3	°C	28,9	29,1	29,2	0,1
Temp.water gascooler, outlet	T15	°C	29,8	29,8	29,9	0,0
Temp.differential, gas cooler	Td2	K	-0,6	0,0	1,4	0,6
Temp.water drain cock	T16	°C	29,0	29,1	29,1	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,115	0,115	0,115	0,0
Water flow, drain cock	F3	l/s	0,033	0,033	0,033	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	89,8	89,9	90,0	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,1	9,1	9,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	107,4	107,6	107,7	0,1
Temp. gascooler inlet	T2	°C	33,7	33,8	33,9	0,1
Temp. gascooler outlet	T32	°C	29,5	29,8	30,3	0,3
Temp. internal heat ex., HP outlet	T4	°C	24,0	24,4	24,6	0,2
Temp. heat ex. LP-receiver, inlet	T5	°C	24,0	24,4	24,6	0,2
Temp. heat ex. LP-receiver, outlet	T6	°C	22,4	22,7	22,9	0,2
Temp. throttle valve, inlet	T7	°C	22,3	22,7	23,0	0,2
Temp. evapor, outlet	T8	°C	0,0	0,1	0,1	0,0
Temp. LP-receiver, outlet	T9	°C	4,2	4,3	4,4	0,1
Temp. internal heat ex., LP inlet	T33	°C	3,1	3,1	3,2	0,0
Temp. compressor, suction	T10	°C	23,8	24,0	24,2	0,1
Pressure compressor, suction	P1	bar	32,6	32,6	32,7	0,0
Pressure compressor, discharge	P2	bar	85,2	85,4	85,7	0,1
Pressure gascoler, inlet	P002	bar	83,7	83,8	84,0	0,1
Pressure throttling valve,inlet	P3	bar	84,0	84,1	84,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,00	0,00	0,0
Pressure differential evapor LP	Pd2	bar	2,08	2,11	2,16	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,08	2,11	2,16	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,3	7,4	7,4	0,0
Temp.water gascooler, inlet 1	T001	°C	30,0	30,2	30,3	0,1
Temp.water gascooler, outlet	T002	°C	47,2	47,2	47,3	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	104,8	104,9	104,9	0,1
Temp. gascooler outlet	T004	°C	32,1	32,3	32,5	0,1
Pressure gascooler, inlet	P001	bar	84,1	84,3	84,6	0,1
Pressure differential gascooler HP	Pd001	bar	0,51	0,54	0,56	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0626_3
2. Outdoor air temp.: 30,1 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: 0,0 °C	0 °C	Time:	16:30:34
4. Heat Rejection Pressure: 90,4 bar	90 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,4 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,23	
Cooling-COP (incl. engine, η_{eng}.= 0,90)	[-]			2,76	
Cooling-COP (shaft)	[-]			3,00	± 4,16 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,128	± 4,00 %
Temp. evapor, outlet (T8)	°C			1,2	± 0,50 °C
Evaporating temperature	°C			0,0	
Cooling capacity (glycol)	kW			24,0	± 4,16 %
LMTD	K			3,66	
Overall heat transfer coeff.	W/m^2K			1225	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,128	± 4,00 %
Temp.water gascooler inlet (T14)	°C			30,1	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			48,2	± 0,50 °C
Temp. differential (T32-T14)	K			0,6	
Heating capacity	kW			31,1	± 3,91 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,7	
LMTD	K			12,72	
Overall heat transfer coeff. outer	W/m^2K			1050	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,4	
Power consumption, shaft	kW			8,0	± 0,16 %
Power input, inverter	kW			9,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			90,4	
Temp. compressor, suction (T10)	°C			22,1	± 0,50 °C
Temp. compressor, discharge (T1)	°C			110,5	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,8	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,3	
Volumetric efficiency (based on water)	[-]			0,83	± 4,02 %
Isentropic efficiency (based on water)	[-]			0,83	± 4,35 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,5	
Gascooler, water	m/s			0,4	
Compressor, suction	m/s			9,7	
Internal heat exchanger, HP, inlet	m/s			1,6	
Internal heat exchanger, HP, outlet	m/s			1,4	
Internal heat exchanger, LP, inlet	m/s			6,4	
Internal heat exchanger, LP, outlet	m/s			6,7	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0626_3
2. Outdoor air temp.: 30,1 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: 0,0 °C	0 °C	Time:	16:30:34
4. Heat Rejection Pressure: 90,4 bar	90 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,4 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,5	5,5	5,5	0,0
Temp.differential, evap.	Td1	K	3,2	3,2	3,3	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	20,9	20,9	21,0	0,0
Temp.water gascooler, inlet 1	T14	°C	28,9	28,9	28,9	0,0
Temp.water gascooler, inlet 2	T3	°C	28,7	28,7	28,8	0,1
Temp.water gascooler, outlet	T15	°C	29,4	29,5	29,5	0,1
Temp.differential, gas cooler	Td2	K	-1,2	0,2	1,2	0,8
Temp.water drain cock	T16	°C	28,7	28,7	28,8	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,115	0,115	0,115	0,0
Water flow, drain cock	F3	l/s	0,033	0,033	0,033	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	94,8	94,9	95,1	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,6	9,7	9,7	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	110,3	110,5	110,5	0,1
Temp. gascooler inlet	T2	°C	34,3	34,5	34,6	0,1
Temp. gascooler outlet	T32	°C	28,8	29,1	29,4	0,2
Temp. internal heat ex., HP outlet	T4	°C	21,8	22,0	22,2	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	21,8	22,0	22,2	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	20,3	20,5	20,6	0,1
Temp. throttle valve, inlet	T7	°C	20,3	20,5	20,7	0,1
Temp. evapor, outlet	T8	°C	1,2	1,2	1,2	0,0
Temp. LP-receiver, outlet	T9	°C	4,8	4,9	4,9	0,0
Temp. internal heat ex., LP inlet	T33	°C	3,8	3,8	3,8	0,0
Temp. compressor, suction	T10	°C	22,0	22,1	22,3	0,1
Pressure compressor, suction	P1	bar	32,7	32,7	32,8	0,0
Pressure compressor, discharge	P2	bar	90,0	90,4	90,7	0,2
Pressure gascoler, inlet	P002	bar	88,8	88,8	88,9	0,0
Pressure throttling valve,inlet	P3	bar	89,1	89,2	89,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,07	0,36	0,1
Pressure differential evapor LP	Pd2	bar	2,07	2,09	2,15	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,07	2,09	2,15	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,3	7,4	7,4	0,0
Temp.water gascooler, inlet 1	T001	°C	29,9	30,1	30,3	0,1
Temp.water gascooler, outlet	T002	°C	48,1	48,2	48,3	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	107,7	107,7	107,8	0,1
Temp. gascooler outlet	T004	°C	30,4	30,6	30,8	0,1
Pressure gascooler, inlet	P001	bar	89,1	89,2	89,4	0,1
Pressure differential gascooler HP	Pd001	bar	0,44	0,46	0,49	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**	
1. Transcritical process	-------	Experiment no: R0626_4
2. Outdoor air temp.: 30,0 °C	30 °C	Date: 26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time: 17:00:34
4. Heat Rejection Pressure: 95,3 bar	95 bar	Operator: WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,3 %	
6. With IHX.	Comment:	SGHX was not used as for oil return.
7. With LP-receiver HX.		Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode: Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,11	
Cooling-COP (incl. engine, η_{eng}.= 0,90)	[-]			2,64	
Cooling-COP (shaft)	[-]			2,88	± 4,03 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,126	± 3,91 %
Temp. evapor, outlet (T8)	°C			1,6	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			24,2	± 4,03 %
LMTD	K			3,95	
Overall heat transfer coeff.	W/m^2K			1144	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,126	± 3,91 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			48,5	± 0,50 °C
Temp. differential (T32-T14)	K			0,2	
Heating capacity	kW			31,6	± 3,83 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,5	
LMTD	K			12,33	
Overall heat transfer coeff. outer	W/m^2K			1021	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,8	
Power consumption, shaft	kW			8,4	± 0,16 %
Power input, inverter	kW			10,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			95,3	
Temp. compressor, suction (T10)	°C			21,7	± 0,50 °C
Temp. compressor, discharge (T1)	°C			115,2	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,9	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,2	
Volumetric efficiency (based on water)	[-]			0,82	± 3,93 %
Isentropic efficiency (based on water)	[-]			0,83	± 4,23 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,1	
Gascooler, water	m/s			0,5	
Compressor, suction	m/s			9,6	
Internal heat exchanger, HP, inlet	m/s			1,5	
Internal heat exchanger, HP, outlet	m/s			1,4	
Internal heat exchanger, LP, inlet	m/s			6,3	
Internal heat exchanger, LP, outlet	m/s			6,6	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0626_4
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	26.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	17:00:34
4. Heat Rejection Pressure: 95,3 bar	95 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,3 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,6	5,7	5,7	0,0
Temp.differential, evap.	Td1	K	3,2	3,3	3,3	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	20,6	20,7	20,8	0,1
Temp.water gascooler, inlet 1	T14	°C	28,6	28,7	28,7	0,0
Temp.water gascooler, inlet 2	T3	°C	28,5	28,5	28,5	0,0
Temp.water gascooler, outlet	T15	°C	29,2	29,2	29,3	0,0
Temp.differential, gas cooler	Td2	K	-1,7	0,0	1,6	1,1
Temp.water drain cock	T16	°C	28,5	28,5	28,6	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,115	0,119	0,121	0,0
Water flow, drain cock	F3	l/s	0,033	0,033	0,033	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	99,5	99,6	99,8	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	10,1	10,2	10,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	115,0	115,2	115,3	0,1
Temp. gascooler inlet	T2	°C	34,6	34,8	35,0	0,1
Temp. gascooler outlet	T32	°C	28,0	28,1	28,4	0,2
Temp. internal heat ex., HP outlet	T4	°C	21,2	21,4	21,6	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	21,3	21,4	21,6	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	19,8	20,0	20,1	0,1
Temp. throttle valve, inlet	T7	°C	19,9	20,0	20,1	0,1
Temp. evapor, outlet	T8	°C	1,6	1,6	1,7	0,0
Temp. LP-receiver, outlet	T9	°C	5,2	5,3	5,4	0,0
Temp. internal heat ex., LP inlet	T33	°C	4,2	4,3	4,3	0,0
Temp. compressor, suction	T10	°C	21,6	21,7	21,8	0,1
Pressure compressor, suction	P1	bar	32,6	32,7	32,7	0,0
Pressure compressor, discharge	P2	bar	95,0	95,3	95,6	0,2
Pressure gascoler, inlet	P002	bar	93,7	93,8	93,9	0,1
Pressure throttling valve,inlet	P3	bar	94,0	94,2	94,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,00	0,02	0,0
Pressure differential evapor LP	Pd2	bar	2,03	2,07	2,11	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,03	2,07	2,11	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,3	7,3	7,4	0,1
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,3	0,1
Temp.water gascooler, outlet	T002	°C	48,4	48,5	48,6	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	112,2	112,4	112,5	0,1
Temp. gascooler outlet	T004	°C	30,1	30,3	30,5	0,1
Pressure gascooler, inlet	P001	bar	93,9	94,1	94,4	0,1
Pressure differential gascooler HP	Pd001	bar	0,38	0,40	0,42	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0718_0
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	18.07.2000
3. Evaporation temp.: 0,0 °C	0 °C	Time:	11:10:21
4. Heat Rejection Pressure: 100,5 bar	100 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,3 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	805	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,99	
Cooling-COP (incl. engine, η_{eng}.= 0,90)	[-]			2,52	
Cooling-COP (shaft)	[-]			2,76	± 3,96 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,125	± 3,86 %
Temp. evapor, outlet (T8)	°C			1,0	± 0,50 °C
Evaporating temperature	°C			0,0	
Cooling capacity (glycol)	kW			24,2	± 3,96 %
LMTD	K			3,18	
Overall heat transfer coeff.	W/m^2K			1424	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,125	± 3,86 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			48,7	± 0,50 °C
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			31,9	± 3,79 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,5	
LMTD	K			12,47	
Overall heat transfer coeff. outer	W/m^2K			1011	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,1	
Power consumption, shaft	kW			8,8	± 0,16 %
Power input, inverter	kW			10,7	
Number of revolutions	rpm			805	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			100,5	
Temp. compressor, suction (T10)	°C			21,3	± 0,50 °C
Temp. compressor, discharge (T1)	°C			119,5	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,1	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,1	
Volumetric efficiency (based on water)	[-]			0,81	± 3,88 %
Isentropic efficiency (based on water)	[-]			0,82	± 4,16 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			6,7	
Gascooler, water	m/s			0,5	
Compressor, suction	m/s			9,4	
Internal heat exchanger, HP, inlet	m/s			1,5	
Internal heat exchanger, HP, outlet	m/s			1,3	
Internal heat exchanger, LP, inlet	m/s			6,2	
Internal heat exchanger, LP, outlet	m/s			6,5	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0718_0
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	18.07.2000
3. Evaporation temp.: 0,0 °C	0 °C	Time:	11:10:21
4. Heat Rejection Pressure: 100,5 bar	100 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,3 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	805	Mode:	Air-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,0	5,0	5,0	0,0
Temp.differential, evap.	Td1	K	3,3	3,3	3,3	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	23,6	23,7	23,9	0,1
Temp.water gascooler, inlet 1	T14	°C	30,5	30,5	30,5	0,0
Temp.water gascooler, inlet 2	T3	°C	30,3	30,3	30,3	0,0
Temp.water gascooler, outlet	T15	°C	31,0	31,0	31,0	0,0
Temp.differential, gas cooler	Td2	K	-1,8	-0,5	1,6	1,2
Temp.water drain cock	T16	°C	26,1	26,2	26,3	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,117	0,117	0,117	0,0
Water flow, drain cock	F3	l/s	0,000	0,000	0,000	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	103,9	104,1	104,3	0,1
Number of revolutions	S1	rpm	805	805	805	0,0
Power consumption, transmitter	N1	kW	10,6	10,7	10,7	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	119,4	119,5	119,7	0,1
Temp. gascooler inlet	T2	°C	31,0	31,0	31,1	0,0
Temp. gascooler outlet	T32	°C	29,0	29,0	29,0	0,0
Temp. internal heat ex., HP outlet	T4	°C	20,6	20,7	20,8	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	20,6	20,7	20,8	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	19,1	19,2	19,3	0,1
Temp. throttle valve, inlet	T7	°C	19,1	19,2	19,3	0,1
Temp. evapor, outlet	T8	°C	1,0	1,0	1,0	0,0
Temp. LP-receiver, outlet	T9	°C	4,7	4,8	4,8	0,1
Temp. internal heat ex., LP inlet	T33	°C	3,7	3,7	3,8	0,0
Temp. compressor, suction	T10	°C	21,3	21,3	21,4	0,0
Pressure compressor, suction	P1	bar	32,7	32,8	32,9	0,0
Pressure compressor, discharge	P2	bar	100,3	100,5	100,8	0,2
Pressure gascoler, inlet	P002	bar	72,2	72,2	72,2	0,0
Pressure throttling valve,inlet	P3	bar	99,2	99,5	99,7	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,00	0,00	0,0
Pressure differential evapor LP	Pd2	bar	2,00	2,05	2,08	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,00	2,05	2,08	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,8	7,8	0,0
Temp.water gascooler, inlet 1	T001	°C	30,0	30,0	30,2	0,1
Temp.water gascooler, outlet	T002	°C	48,7	48,7	48,8	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	116,6	116,7	116,8	0,1
Temp. gascooler outlet	T004	°C	30,1	30,2	30,3	0,1
Pressure gascooler, inlet	P001	bar	99,1	99,3	99,6	0,2
Pressure differential gascooler HP	Pd001	bar	0,32	0,39	0,43	0,03

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0619_2
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	19.06.2000
3. Evaporation temp.: 0,4 °C	0 °C	Time:	17:30:37
4. Heat Rejection Pressure: 105,2 bar	105 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,7 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,89	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,41	
Cooling-COP (shaft)	[-]			2,64	± 3,91 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,125	± 3,81 %
Temp. evapor, outlet (T8)	°C			0,3	± 0,50 °C
Evaporating temperature	°C			0,4	
Cooling capacity (glycol)	kW			24,2	± 3,91 %
LMTD	K			3,41	
Overall heat transfer coeff.	W/m^2K			1326	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,125	± 3,81 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			48,8	± 0,50 °C
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			32,2	± 3,76 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			4,0	
LMTD	K			13,21	
Overall heat transfer coeff. outer	W/m^2K			1076	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,4	
Power consumption, shaft	kW			9,2	± 0,15 %
Power input, inverter	kW			11,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			33,3	
Pressure compressor, discharge (P2)	bar			105,2	
Temp. compressor, suction (T10)	°C			20,3	± 0,50 °C
Temp. compressor, discharge (T1)	°C			122,0	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,2	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,0	
Volumetric efficiency (based on water)	[-]			0,79	± 3,84 %
Isentropic efficiency (based on water)	[-]			0,80	± 4,12 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			6,4	
Gascooler, water	m/s			0,4	
Compressor, suction	m/s			9,2	
Internal heat exchanger, HP, inlet	m/s			1,5	
Internal heat exchanger, HP, outlet	m/s			1,3	
Internal heat exchanger, LP, inlet	m/s			5,9	
Internal heat exchanger, LP, outlet	m/s			6,4	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0619_2
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	19.06.2000
3. Evaporation temp.: 0,4 °C	0 °C	Time:	17:30:37
4. Heat Rejection Pressure: 105,2 bar	105 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,7 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	Measure pnt.	Unit	Min	Mean	Max	STDEV
Temp.evap. inlet	T13	°C	5,4	5,7	6,2	0,3
Temp.differential, evap.	Td1	K	3,2	3,3	3,5	0,1
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	19,6	19,7	19,7	0,0
Temp.water gascooler, inlet 1	T14	°C	24,6	24,6	24,6	0,0
Temp.water gascooler, inlet 2	T3	°C	24,5	24,5	24,5	0,0
Temp.water gascooler, outlet	T15	°C	25,2	25,2	25,2	0,0
Temp.differential, gas cooler	Td2	K	-1,1	0,4	1,9	1,0
Temp.water drain cock	T16	°C	24,5	24,5	24,5	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,117	0,117	0,117	0,0
Water flow, drain cock	F3	l/s	0,020	0,020	0,020	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	108,6	108,9	109,2	0,2
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,1	11,2	11,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	121,1	122,0	122,7	0,6
Temp. gascooler inlet	T2	°C	34,2	34,4	34,8	0,2
Temp. gascooler outlet	T32	°C	26,3	26,3	26,4	0,0
Temp. internal heat ex., HP outlet	T4	°C	18,7	18,8	19,0	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	19,4	19,4	19,6	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	13,3	13,6	14,1	0,3
Temp. throttle valve, inlet	T7	°C	13,4	13,7	14,1	0,3
Temp. evapor, outlet	T8	°C	0,1	0,3	0,8	0,3
Temp. LP-receiver, outlet	T9	°C	1,8	2,0	2,4	0,2
Temp. internal heat ex., LP inlet	T33	°C	1,3	1,5	1,8	0,2
Temp. compressor, suction	T10	°C	20,2	20,3	20,4	0,1
Pressure compressor, suction	P1	bar	33,0	33,3	33,6	0,2
Pressure compressor, discharge	P2	bar	105,0	105,2	105,4	0,1
Pressure gascoler, inlet	P002	bar	103,7	103,8	104,0	0,1
Pressure throttling valve,inlet	P3	bar	104,2	104,5	104,6	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,00	0,00	0,0
Pressure differential evapor LP	Pd2	bar	1,91	1,97	2,01	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,91	1,97	2,01	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,3	7,4	7,4	0,0
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,2	0,1
Temp.water gascooler, outlet	T002	°C	48,7	48,8	49,0	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	118,7	119,3	119,8	0,4
Temp. gascooler outlet	T004	°C	30,0	30,1	30,3	0,1
Pressure gascooler, inlet	P001	bar	103,9	104,1	104,3	0,1
Pressure differential gascooler HP	Pd001	bar	0,33	0,36	0,39	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0619_3
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	19.06.2000
3. Evaporation temp.: 0,3 °C	0 °C	Time:	17:00:28
4. Heat Rejection Pressure: 110,2 bar	110 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,5 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,79	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,30	
Cooling-COP (shaft)	[-]			2,54	± 3,88 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,124	± 3,79 %
Temp. evapor, outlet (T8)	°C			0,3	± 0,50 °C
Evaporating temperature	°C			0,3	
Cooling capacity (glycol)	kW			24,1	± 3,88 %
LMTD	K			3,43	
Overall heat transfer coeff.	W/m^2K			1317	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,124	± 3,79 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			49,0	± 0,50 °C
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			32,4	± 3,73 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,9	
LMTD	K			13,06	
Overall heat transfer coeff. outer	W/m^2K			1061	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,7	
Power consumption, shaft	kW			9,5	± 0,15 %
Power input, inverter	kW			11,6	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			33,3	
Pressure compressor, discharge (P2)	bar			110,2	
Temp. compressor, suction (T10)	°C			20,5	± 0,50 °C
Temp. compressor, discharge (T1)	°C			126,7	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,3	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			5,9	
Volumetric efficiency (based on water)	[-]			0,78	± 3,81 %
Isentropic efficiency (based on water)	[-]			0,80	± 4,07 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			6,1	
Gascooler, water	m/s			0,4	
Compressor, suction	m/s			9,1	
Internal heat exchanger, HP, inlet	m/s			1,4	
Internal heat exchanger, HP, outlet	m/s			1,3	
Internal heat exchanger, LP, inlet	m/s			5,8	
Internal heat exchanger, LP, outlet	m/s			6,3	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0619_3
2. Outdoor air temp.: 30,0 °C	30 °C	Date:	19.06.2000
3. Evaporation temp.: 0,3 °C	0 °C	Time:	17:00:28
4. Heat Rejection Pressure: 110,2 bar	110 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,5 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Air-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,6	5,6	5,7	0,0
Temp.differential, evap.	Td1	K	3,2	3,2	3,2	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	19,8	20,0	20,1	0,1
Temp.water gascooler, inlet 1	T14	°C	24,7	24,7	24,7	0,0
Temp.water gascooler, inlet 2	T3	°C	24,6	24,6	24,6	0,0
Temp.water gascooler, outlet	T15	°C	25,2	25,3	25,3	0,0
Temp.differential, gas cooler	Td2	K	-1,7	0,1	1,5	0,9
Temp.water drain cock	T16	°C	24,5	24,6	24,6	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,117	0,117	0,117	0,0
Water flow, drain cock	F3	l/s	0,020	0,020	0,020	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	112,8	113,0	113,1	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,6	11,6	11,7	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	126,6	126,7	127,0	0,1
Temp. gascooler inlet	T2	°C	34,3	34,5	34,8	0,1
Temp. gascooler outlet	T32	°C	27,1	27,2	27,3	0,1
Temp. internal heat ex., HP outlet	T4	°C	18,6	18,7	19,1	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	19,2	19,3	19,6	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	14,3	14,4	14,5	0,1
Temp. throttle valve, inlet	T7	°C	14,3	14,4	14,5	0,1
Temp. evapor, outlet	T8	°C	0,2	0,3	0,3	0,0
Temp. LP-receiver, outlet	T9	°C	2,2	2,2	2,3	0,0
Temp. internal heat ex., LP inlet	T33	°C	1,6	1,6	1,7	0,0
Temp. compressor, suction	T10	°C	20,4	20,5	20,7	0,1
Pressure compressor, suction	P1	bar	33,2	33,3	33,3	0,0
Pressure compressor, discharge	P2	bar	110,0	110,2	110,5	0,1
Pressure gascoler, inlet	P002	bar	108,5	108,8	109,0	0,1
Pressure throttling valve,inlet	P3	bar	109,3	109,4	109,7	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,01	0,05	0,0
Pressure differential evapor LP	Pd2	bar	1,86	1,90	1,94	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,86	1,90	1,94	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,4	7,4	7,5	0,0
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,5	0,1
Temp.water gascooler, outlet	T002	°C	48,9	49,0	49,3	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,41	0,41	0,41	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	123,6	123,7	123,9	0,1
Temp. gascooler outlet	T004	°C	30,0	30,2	30,5	0,1
Pressure gascooler, inlet	P001	bar	108,7	109,0	109,3	0,2
Pressure differential gascooler HP	Pd001	bar	0,33	0,36	0,39	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_1
2. Hot water temp.: 60,4 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,3 °C	30 °C	Time:	12:50:03
4. Evaporation temp.: 0,0 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 80,0 bar	80 bar		
5. % Ratio of Q_{SGHX} to Q_{Evap}	2,4 %		
7. With IHX.	Comment:	SGHX was not used as for oil return.	
8. With LP-receiver HX.		Oil return from oil separator was off.	
9. Compressor speed (RPM)	804	Mode:	Combined

Heat pump	**Unit**	**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]	3,34	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]	2,95	
Cooling-COP (shaft)	[-]	3,20	± 5,22 %
Total-COP (shaft)	[-]	5,00	± 4,73 %
Load ratio	[-]	0,50	
Evaporator:	**Unit**	**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s	0,134	± 4,85 %
Temp. evapor, outlet (T8)	°C	0,0	± 0,50 %
Evaporating temperature	°C	0,0	
Cooling capacity (glycol)	kW	22,7	± 5,21 %
LMTD	K	2,85	
Overall heat transfer coeff.	W/m^2K	1489	
Water-cooled Gascooler:	**Unit**	**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s	0,061	± 4,30 %
Temp.water gascooler inlet (T14)	°C	20,1	± 0,50 %
Temp.water gascooler outlet (T15)	°C	60,4	± 0,50 %
Temp. differential (T32-T14)	K	12,6	
Heating capacity	kW	12,7	± 2,62 %
Baseline load	kW	25,4	
Air-cooled Gascooler:	**Unit**	**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s	0,073	± 8,15 %
Temp.water gascooler inlet (T14)	°C	30,3	± 0,50 %
Temp.water gascooler outlet (T15)	°C	39,1	± 0,50 %
Temp. differential (T32-T14)	K	0,7	
Heating capacity	kW	16,4	± 8,00 %
Internal heat exchanger	**Unit**	**Value**	**Uncertainty**
Capacity ,HP	kW	5,1	
LMTD	K	14,94	
Overall heat transfer coeff. outer	W/m^2K	1217	
Compressor:	**Unit**	**Value**	**Uncertainty**
Power consumption, entalpy diff	kW	6,8	
Power consumption, shaft	kW	7,1	± 0,17 %
Power input, inverter	kW	8,6	
Number of revolutions	rpm	804	
Pressure compressor, suction (P1)	bar	32,7	
Pressure compressor, discharge (P2)	bar	80,0	
Temp. compressor, suction (T10)	°C	23,4	± 0,50 %
Temp. compressor, discharge (T1)	°C	100,5	± 0,50 %
Pressure ratio (P2/P1)	[-]	2,4	± 0,14 %
Teoretical Swept volume	m^3/h	7,6	
Swept volume (based on water)	m^3/h	6,7	
Volumetric efficiency (based on water)	[-]	0,88	± 4,86 %
Isentropic efficiency (based on water)	[-]	0,86	± 5,04 %
Velocities	**Unit**	**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s	3,9	
Water-cooled gascooler, water	m/s	0,3	
Air-cooled gascooler, CO_2 ,inlet	m/s	4,7	
Air-cooled gascooler, water	m/s	1,7	
Compressor, suction	m/s	4,7	
Internal heat exchanger, HP, inlet	m/s	0,8	
Internal heat exchanger, HP, outlet	m/s	0,7	
Internal heat exchanger, LP, inlet	m/s	3,0	
Internal heat exchanger, LP, outlet	m/s	3,2	
LP-receiver	m/s	0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_1
2. Hot water temp.: 60,4 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,3 °C	30 °C	Time:	12:50:03
4. Evaporation temp.: 0,0 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 80,0 bar	80 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	2,4 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	4,6	4,6	4,7	0,0
Temp.differential, evap.	Td1	K	3,0	3,0	3,0	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	15,9	16,2	16,3	0,1
Temp.water gascooler, inlet 1	T14	°C	20,1	20,1	20,2	0,0
Temp.water gascooler, inlet 2	T3	°C	20,0	20,1	20,1	0,0
Temp.water gascooler, outlet	T15	°C	60,4	60,4	60,4	0,0
Temp.differential, gas cooler	Td2	K	36,7	37,5	38,8	0,6
Temp.water drain cock	T16	°C	59,5	59,5	59,5	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,076	0,076	0,076	0,0
Water flow, drain cock	F3	l/s	0,062	0,062	0,062	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	84,0	84,3	84,4	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	8,5	8,6	8,6	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	100,3	100,5	100,7	0,1
Temp. gascooler inlet	T2	°C	97,8	97,9	98,0	0,1
Temp. gascooler outlet	T32	°C	32,7	32,7	32,7	0,0
Temp. internal heat ex., HP outlet	T4	°C	24,3	24,4	24,6	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	24,3	24,4	24,6	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	11,6	11,9	12,0	0,1
Temp. throttle valve, inlet	T7	°C	11,6	11,9	12,0	0,1
Temp. evapor, outlet	T8	°C	-0,1	0,0	0,1	0,0
Temp. LP-receiver, outlet	T9	°C	0,6	0,6	0,7	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,3	0,4	0,4	0,0
Temp. compressor, suction	T10	°C	23,4	23,4	23,6	0,0
Pressure compressor, suction	P1	bar	32,7	32,7	32,8	0,0
Pressure compressor, discharge	P2	bar	79,8	80,0	80,3	0,1
Pressure gascoler, inlet	P002	bar	78,9	79,1	79,3	0,1
Pressure throttling valve,inlet	P3	bar	78,8	79,0	79,2	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,57	0,73	0,2
Pressure differential evapor LP	Pd2	bar	2,14	2,17	2,22	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,14	2,17	2,22	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,8	7,9	8,0	0,1
Temp.water gascooler, inlet 1	T001	°C	30,2	30,3	30,4	0,0
Temp.water gascooler, outlet	T002	°C	39,1	39,1	39,2	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	97,6	97,7	97,8	0,1
Temp. gascooler outlet	T004	°C	30,9	31,0	31,1	0,1
Pressure gascooler, inlet	P001	bar	78,5	78,8	78,9	0,1
Pressure differential gascooler HP	Pd001	bar	0,19	0,19	0,20	0,00

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_2
2. Hot water temp.: 60,4 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,1 °C	30 °C	Time:	13:31:35
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 85,1 bar	85 bar		
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,1 %		
7. With IHX.	Comment:	SGHX was not used as for oil return.	
8. With LP-receiver HX.		Oil return from oil separator was off.	
9. Compressor speed (RPM)	804	Mode:	Combined

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,51	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			3,00	
Cooling-COP (shaft)	[-]			3,26	± 4,56 %
Total-COP (shaft)	[-]			5,11	± 4,11 %
Load ratio	[-]			0,55	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,130	± 4,49 %
Temp. evapor, outlet (T8)	°C			-0,1	± 0,50 %
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			24,7	± 4,56 %
LMTD	K			3,03	
Overall heat transfer coeff.	W/m^2K			1521	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,056	± 2,59 %
Temp.water gascooler inlet (T14)	°C			19,9	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,4	± 0,50 %
Temp. differential (T32-T14)	K			6,3	
Heating capacity	kW			14,1	± 2,48 %
Baseline load	kW			25,4	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,073	± 7,68 %
Temp.water gascooler inlet (T14)	°C			30,1	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,4	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			17,3	± 7,61 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			4,0	
LMTD	K			13,02	
Overall heat transfer coeff. outer	W/m^2K			1097	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,0	
Power consumption, shaft	kW			7,6	± 0,17 %
Power input, inverter	kW			9,1	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,6	
Pressure compressor, discharge (P2)	bar			85,1	
Temp. compressor, suction (T10)	°C			19,9	± 0,50 %
Temp. compressor, discharge (T1)	°C			102,6	± 0,50 %
Pressure ratio (P2/P1)	[-]			2,6	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,3	
Volumetric efficiency (based on water)	[-]			0,83	± 4,50 %
Isentropic efficiency (based on water)	[-]			0,82	± 4,69 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			3,4	
Water-cooled gascooler, water	m/s			0,3	
Air-cooled gascooler, CO_2 ,inlet	m/s			4,4	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,2	
Internal heat exchanger, HP, inlet	m/s			0,7	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,7	
Internal heat exchanger, LP, outlet	m/s			2,9	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0713_2
2. Hot water temp.: 60,4 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,1 °C	30 °C	Time:	13:31:35
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 85,1 bar	85 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-0,1 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	Measure pnt.	Unit	Min	Mean	Max	STDEV
Temp.evap. inlet	T13	°C	4,8	4,8	4,9	0,0
Temp.differential, evap.	Td1	K	3,2	3,3	3,3	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	14,4	14,6	14,8	0,1
Temp.water gascooler, inlet 1	T14	°C	19,7	19,9	20,0	0,1
Temp.water gascooler, inlet 2	T3	°C	19,7	19,9	20,0	0,1
Temp.water gascooler, outlet	T15	°C	60,4	60,4	60,4	0,0
Temp.differential, gas cooler	Td2	K	36,9	38,4	39,7	0,7
Temp.water drain cock	T16	°C	59,5	59,5	59,5	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,083	0,083	0,083	0,0
Water flow, drain cock	F3	l/s	0,066	0,071	0,072	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	89,8	90,0	90,1	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,1	9,1	9,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	102,5	102,6	102,9	0,1
Temp. gascooler inlet	T2	°C	100,1	100,2	100,3	0,1
Temp. gascooler outlet	T32	°C	26,1	26,3	26,4	0,1
Temp. internal heat ex., HP outlet	T4	°C	19,2	19,3	19,5	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	19,2	19,3	19,5	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	12,0	12,3	12,4	0,1
Temp. throttle valve, inlet	T7	°C	12,0	12,3	12,5	0,1
Temp. evapor, outlet	T8	°C	-0,1	-0,1	0,0	0,0
Temp. LP-receiver, outlet	T9	°C	1,1	1,2	1,2	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,7	0,7	0,7	0,0
Temp. compressor, suction	T10	°C	19,8	19,9	19,9	0,0
Pressure compressor, suction	P1	bar	32,6	32,6	32,7	0,0
Pressure compressor, discharge	P2	bar	84,9	85,1	85,4	0,1
Pressure gascoler, inlet	P002	bar	84,1	84,3	84,5	0,2
Pressure throttling valve,inlet	P3	bar	84,1	84,2	84,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,43	0,66	0,2
Pressure differential evapor LP	Pd2	bar	2,07	2,09	2,15	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,07	2,09	2,15	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,8	7,8	0,1
Temp.water gascooler, inlet 1	T001	°C	30,0	30,1	30,3	0,1
Temp.water gascooler, outlet	T002	°C	39,3	39,4	39,5	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	99,9	100,0	100,1	0,1
Temp. gascooler outlet	T004	°C	30,1	30,2	30,4	0,1
Pressure gascooler, inlet	P001	bar	83,8	83,9	84,2	0,1
Pressure differential gascooler HP	Pd001	bar	0,13	0,15	0,17	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0713_3
2. Hot water temp.: 60,1 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	14:15:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 90,4 bar	90 bar		
5. % Ratio of Q_{SGHX} to Q_{Evap}	-1,1 %		
7. With IHX.	Comment:	SGHX was not used as for oil return.	
8. With LP-receiver HX.		Oil return from oil separator was off.	
9. Compressor speed (RPM)	804	Mode:	Combined

Heat pump	Unit			Value	Uncertainty
Cooling-COP (entalpy differential)	[-]			3,48	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			3,02	
Cooling-COP (shaft)	[-]			3,28	± 4,36 %
Total-COP (shaft)	[-]			5,22	± 3,87 %
Load ratio	[-]			0,50	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,132	± 4,31 %
Temp. evapor, outlet (T8)	°C			-0,1	± 0,50 %
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			26,3	± 4,35 %
LMTD	K			3,09	
Overall heat transfer coeff.	W/m^2K			1590	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,057	± 2,43 %
Temp.water gascooler inlet (T14)	°C			19,9	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,1	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			15,6	± 2,36 %
Baseline load	kW			31,1	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,075	± 7,38 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,7	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			17,9	± 7,33 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,4	
LMTD	K			11,24	
Overall heat transfer coeff. outer	W/m^2K			1084	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,6	
Power consumption, shaft	kW			8,0	± 0,16 %
Power input, inverter	kW			9,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			90,4	
Temp. compressor, suction (T10)	°C			18,2	± 0,50 %
Temp. compressor, discharge (T1)	°C			106,4	± 0,50 %
Pressure ratio (P2/P1)	[-]			2,8	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,3	
Volumetric efficiency (based on water)	[-]			0,84	± 4,32 %
Isentropic efficiency (based on water)	[-]			0,83	± 4,49 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			3,3	
Water-cooled gascooler, water	m/s			0,4	
Air-cooled gascooler, CO_2 ,inlet	m/s			4,3	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,3	
Internal heat exchanger, HP, inlet	m/s			0,7	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,8	
Internal heat exchanger, LP, outlet	m/s			2,9	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_3
2. Hot water temp.: 60,1 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	14:15:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 90,4 bar	90 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-1,1 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,0	5,0	5,0	0,0
Temp.differential, evap.	Td1	K	3,4	3,4	3,4	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	13,3	13,6	13,7	0,1
Temp.water gascooler, inlet 1	T14	°C	19,8	19,9	19,9	0,1
Temp.water gascooler, inlet 2	T3	°C	19,7	19,8	19,9	0,0
Temp.water gascooler, outlet	T15	°C	60,0	60,1	60,1	0,0
Temp.differential, gas cooler	Td2	K	36,9	38,0	39,7	1,0
Temp.water drain cock	T16	°C	59,1	59,2	59,2	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,093	0,093	0,093	0,0
Water flow, drain cock	F3	l/s	0,066	0,066	0,066	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	95,3	95,4	95,5	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,7	9,7	9,7	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	106,3	106,4	106,7	0,1
Temp. gascooler inlet	T2	°C	103,8	103,9	104,0	0,1
Temp. gascooler outlet	T32	°C	19,9	20,0	20,1	0,1
Temp. internal heat ex., HP outlet	T4	°C	17,3	17,3	17,4	0,0
Temp. heat ex. LP-receiver, inlet	T5	°C	17,3	17,3	17,4	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	15,9	15,9	16,0	0,1
Temp. throttle valve, inlet	T7	°C	15,9	16,0	16,1	0,0
Temp. evapor, outlet	T8	°C	-0,1	-0,1	-0,1	0,0
Temp. LP-receiver, outlet	T9	°C	2,6	2,7	2,8	0,1
Temp. internal heat ex., LP inlet	T33	°C	1,9	1,9	2,0	0,0
Temp. compressor, suction	T10	°C	18,2	18,2	18,3	0,0
Pressure compressor, suction	P1	bar	32,6	32,7	32,7	0,0
Pressure compressor, discharge	P2	bar	90,2	90,4	90,5	0,1
Pressure gascoler, inlet	P002	bar	89,3	89,5	89,7	0,1
Pressure throttling valve,inlet	P3	bar	89,5	89,6	89,8	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,28	1,01	0,4
Pressure differential evapor LP	Pd2	bar	2,05	2,09	2,13	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,05	2,09	2,13	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,7	7,8	0,0
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,2	0,1
Temp.water gascooler, outlet	T002	°C	39,6	39,7	39,7	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	103,7	103,7	103,8	0,1
Temp. gascooler outlet	T004	°C	30,0	30,1	30,2	0,1
Pressure gascooler, inlet	P001	bar	89,1	89,3	89,5	0,1
Pressure differential gascooler HP	Pd001	bar	0,13	0,16	0,22	0,03

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_4
2. Hot water temp.: 60,2 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	14:45:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 95,3 bar	95 bar		
5. % Ratio of Q_{SGHX} to Q_{Evap}	-1,0 %		
7. With IHX.	Comment:	SGHX was not used as for oil return.	
8. With LP-receiver HX.		Oil return from oil separator was off.	
9. Compressor speed (RPM)	804	Mode:	Combined

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,31	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,85	
Cooling-COP (shaft)	[-]			3,11	± 4,31 %
Total-COP (shaft)	[-]			4,96	± 3,82 %
Load ratio	[-]			0,50	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,130	± 4,27 %
Temp. evapor, outlet (T8)	°C			0,2	± 0,50 %
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			26,2	± 4,31 %
LMTD	K			3,18	
Overall heat transfer coeff.	W/m^2K			1540	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,056	± 2,42 %
Temp.water gascooler inlet (T14)	°C			19,9	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,2	± 0,50 %
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			15,6	± 2,36 %
Baseline load	kW			31,6	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,073	± 7,33 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,6	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			18,0	± 7,29 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,3	
LMTD	K			11,00	
Overall heat transfer coeff. outer	W/m^2K			1055	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,9	
Power consumption, shaft	kW			8,4	± 0,16 %
Power input, inverter	kW			10,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			95,3	
Temp. compressor, suction (T10)	°C			18,1	± 0,50 %
Temp. compressor, discharge (T1)	°C			111,3	± 0,50 %
Pressure ratio (P2/P1)	[-]			2,9	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,2	
Volumetric efficiency (based on water)	[-]			0,82	± 4,28 %
Isentropic efficiency (based on water)	[-]			0,83	± 4,44 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			3,1	
Water-cooled gascooler, water	m/s			0,4	
Air-cooled gascooler, CO_2 ,inlet	m/s			4,0	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,2	
Internal heat exchanger, HP, inlet	m/s			0,7	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,8	
Internal heat exchanger, LP, outlet	m/s			2,9	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_4
2. Hot water temp.: 60,2 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	14:45:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 95,3 bar	95 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-1,0 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,0	5,0	5,1	0,0
Temp.differential, evap.	Td1	K	3,4	3,4	3,4	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	12,9	13,1	13,3	0,1
Temp.water gascooler, inlet 1	T14	°C	19,8	19,9	20,1	0,1
Temp.water gascooler, inlet 2	T3	°C	19,8	19,9	20,1	0,1
Temp.water gascooler, outlet	T15	°C	60,0	60,2	60,3	0,1
Temp.differential, gas cooler	Td2	K	36,8	38,3	39,8	1,1
Temp.water drain cock	T16	°C	59,2	59,3	59,4	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,093	0,093	0,093	0,0
Water flow, drain cock	F3	l/s	0,072	0,072	0,072	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	99,9	100,0	100,2	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	10,2	10,2	10,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	111,1	111,3	111,5	0,1
Temp. gascooler inlet	T2	°C	108,5	108,5	108,6	0,1
Temp. gascooler outlet	T32	°C	19,8	19,9	20,2	0,1
Temp. internal heat ex., HP outlet	T4	°C	17,0	17,1	17,2	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	17,1	17,1	17,2	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	15,7	15,8	15,9	0,1
Temp. throttle valve, inlet	T7	°C	15,7	15,8	15,8	0,0
Temp. evapor, outlet	T8	°C	0,2	0,2	0,2	0,0
Temp. LP-receiver, outlet	T9	°C	3,1	3,1	3,2	0,0
Temp. internal heat ex., LP inlet	T33	°C	2,3	2,3	2,4	0,0
Temp. compressor, suction	T10	°C	18,1	18,1	18,2	0,0
Pressure compressor, suction	P1	bar	32,6	32,7	32,8	0,0
Pressure compressor, discharge	P2	bar	95,0	95,3	95,7	0,2
Pressure gascoler, inlet	P002	bar	94,2	94,5	94,8	0,2
Pressure throttling valve,inlet	P3	bar	94,5	94,7	94,9	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,32	0,78	0,3
Pressure differential evapor LP	Pd2	bar	2,01	2,04	2,09	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,01	2,04	2,09	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,6	7,7	7,7	0,0
Temp.water gascooler, inlet 1	T001	°C	29,8	30,0	30,1	0,1
Temp.water gascooler, outlet	T002	°C	39,6	39,6	39,7	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	108,3	108,5	108,6	0,1
Temp. gascooler outlet	T004	°C	29,9	30,0	30,2	0,1
Pressure gascooler, inlet	P001	bar	94,0	94,3	94,5	0,1
Pressure differential gascooler HP	Pd001	bar	0,09	0,13	0,16	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_5
2. Hot water temp.: 60,6 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	15:15:34
4. Evaporation temp.: 0,0 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 100,7 bar	100 bar		
5. % Ratio of Q_{SGHX} to Q_{Evap}	-1,0 %		
7. With IHX.	Comment:	SGHX was not used as for oil return.	
8. With LP-receiver HX.		Oil return from oil separator was off.	
9. Compressor speed (RPM)	804	Mode:	Combined

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,18	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,70	
Cooling-COP (shaft)	[-]			2,96	± 4,28 %
Total-COP (shaft)	[-]			4,74	± 3,77 %
Load ratio	[-]			0,49	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,128	± 4,24 %
Temp. evapor, outlet (T8)	°C			0,8	± 0,50 %
Evaporating temperature	°C			0,0	
Cooling capacity (glycol)	kW			26,0	± 4,28 %
LMTD	K			3,24	
Overall heat transfer coeff.	W/m^2K			1498	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,056	± 2,41 %
Temp.water gascooler inlet (T14)	°C			20,1	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,6	± 0,50 %
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			15,7	± 2,35 %
Baseline load	kW			31,9	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,072	± 7,32 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,7	± 0,50 %
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			18,0	± 7,28 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,1	
LMTD	K			10,64	
Overall heat transfer coeff. outer	W/m^2K			1041	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,2	
Power consumption, shaft	kW			8,8	± 0,16 %
Power input, inverter	kW			10,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			100,7	
Temp. compressor, suction (T10)	°C			18,2	± 0,50 %
Temp. compressor, discharge (T1)	°C			116,2	± 0,50 %
Pressure ratio (P2/P1)	[-]			3,1	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,1	
Volumetric efficiency (based on water)	[-]			0,81	± 4,25 %
Isentropic efficiency (based on water)	[-]			0,82	± 4,40 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			2,9	
Water-cooled gascooler, water	m/s			0,4	
Air-cooled gascooler, CO_2 ,inlet	m/s			3,8	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,1	
Internal heat exchanger, HP, inlet	m/s			0,7	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,8	
Internal heat exchanger, LP, outlet	m/s			2,9	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_5
2. Hot water temp.: 60,6 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	15:15:34
4. Evaporation temp.: 0,0 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 100,7 bar	100 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-1,0 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,2	5,2	5,3	0,0
Temp.differential, evap.	Td1	K	3,4	3,4	3,4	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	12,7	12,8	13,0	0,1
Temp.water gascooler, inlet 1	T14	°C	20,0	20,1	20,2	0,1
Temp.water gascooler, inlet 2	T3	°C	20,0	20,1	20,2	0,1
Temp.water gascooler, outlet	T15	°C	60,4	60,6	60,9	0,1
Temp.differential, gas cooler	Td2	K	36,7	38,3	39,9	1,3
Temp.water drain cock	T16	°C	59,6	59,7	60,0	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,093	0,093	0,093	0,0
Water flow, drain cock	F3	l/s	0,072	0,072	0,072	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	104,3	104,5	104,7	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	10,7	10,7	10,8	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	116,0	116,2	116,3	0,1
Temp. gascooler inlet	T2	°C	113,3	113,4	113,5	0,1
Temp. gascooler outlet	T32	°C	20,0	20,1	20,2	0,1
Temp. internal heat ex., HP outlet	T4	°C	17,0	17,0	17,1	0,0
Temp. heat ex. LP-receiver, inlet	T5	°C	17,0	17,1	17,2	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	15,7	15,8	15,9	0,1
Temp. throttle valve, inlet	T7	°C	15,7	15,8	15,8	0,0
Temp. evapor, outlet	T8	°C	0,8	0,8	0,8	0,0
Temp. LP-receiver, outlet	T9	°C	3,6	3,7	3,8	0,0
Temp. internal heat ex., LP inlet	T33	°C	2,9	2,9	2,9	0,0
Temp. compressor, suction	T10	°C	18,2	18,2	18,3	0,0
Pressure compressor, suction	P1	bar	32,8	32,8	32,9	0,0
Pressure compressor, discharge	P2	bar	100,4	100,7	101,0	0,2
Pressure gascoler, inlet	P002	bar	99,6	100,0	100,1	0,1
Pressure throttling valve,inlet	P3	bar	100,0	100,2	100,5	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,26	0,51	0,2
Pressure differential evapor LP	Pd2	bar	1,98	2,03	2,06	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,98	2,03	2,06	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,7	7,8	0,0
Temp.water gascooler, inlet 1	T001	°C	29,8	30,0	30,1	0,1
Temp.water gascooler, outlet	T002	°C	39,6	39,7	39,8	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	113,2	113,3	113,4	0,1
Temp. gascooler outlet	T004	°C	29,9	30,0	30,2	0,1
Pressure gascooler, inlet	P001	bar	99,4	99,8	100,2	0,2
Pressure differential gascooler HP	Pd001	bar	0,05	0,08	0,11	0,02

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**	
1. Transcritical process	-------	Experiment no: R0713_6
2. Hot water temp.: 60,7 °C	60 °C	Date: 13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time: 15:45:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator: WA
5. Heat Rejection Pressure: 105,8 bar	105 bar	
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,9 %	
7. With IHX.	Comment:	SGHX was not used as for oil return.
8. With LP-receiver HX.		Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode: Combined

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,03	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,54	
Cooling-COP (shaft)	[-]			2,79	± 4,27 %
Total-COP (shaft)	[-]			4,51	± 3,74 %
Load ratio	[-]			0,49	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,126	± 4,23 %
Temp. evapor, outlet (T8)	°C			1,1	± 0,50 %
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			25,7	± 4,27 %
LMTD	K			3,42	
Overall heat transfer coeff.	W/m^2K			1406	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,055	± 2,40 %
Temp.water gascooler inlet (T14)	°C			20,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,7	± 0,50 %
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			15,8	± 2,35 %
Baseline load	kW			32,2	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,070	± 7,34 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,6	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			18,0	± 7,31 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,0	
LMTD	K			10,43	
Overall heat transfer coeff. outer	W/m^2K			1017	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,5	
Power consumption, shaft	kW			9,2	± 0,15 %
Power input, inverter	kW			11,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			105,8	
Temp. compressor, suction (T10)	°C			18,2	± 0,50 %
Temp. compressor, discharge (T1)	°C			121,4	± 0,50 %
Pressure ratio (P2/P1)	[-]			3,2	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,0	
Volumetric efficiency (based on water)	[-]			0,79	± 4,24 %
Isentropic efficiency (based on water)	[-]			0,81	± 4,37 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			2,8	
Water-cooled gascooler, water	m/s			0,4	
Air-cooled gascooler, CO_2 ,inlet	m/s			3,5	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,1	
Internal heat exchanger, HP, inlet	m/s			0,6	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,7	
Internal heat exchanger, LP, outlet	m/s			2,8	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_6
2. Hot water temp.: 60,7 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	15:45:34
4. Evaporation temp.: -0,1 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 105,8 bar	105 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-0,9 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,2	5,3	5,3	0,0
Temp.differential, evap.	Td1	K	3,3	3,4	3,4	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	12,4	12,6	12,7	0,1
Temp.water gascooler, inlet 1	T14	°C	19,8	20,0	20,2	0,1
Temp.water gascooler, inlet 2	T3	°C	19,8	19,9	20,2	0,1
Temp.water gascooler, outlet	T15	°C	60,6	60,7	60,8	0,1
Temp.differential, gas cooler	Td2	K	36,8	38,5	40,4	1,2
Temp.water drain cock	T16	°C	59,7	59,9	59,9	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,093	0,093	0,093	0,0
Water flow, drain cock	F3	l/s	0,072	0,072	0,072	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	109,1	109,3	109,6	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,2	11,2	11,3	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	121,1	121,4	121,5	0,1
Temp. gascooler inlet	T2	°C	118,5	118,5	118,7	0,1
Temp. gascooler outlet	T32	°C	19,8	20,0	20,2	0,1
Temp. internal heat ex., HP outlet	T4	°C	16,9	17,0	17,0	0,0
Temp. heat ex. LP-receiver, inlet	T5	°C	16,9	17,0	17,0	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	15,7	15,7	15,7	0,0
Temp. throttle valve, inlet	T7	°C	15,6	15,7	15,7	0,0
Temp. evapor, outlet	T8	°C	1,0	1,1	1,1	0,0
Temp. LP-receiver, outlet	T9	°C	4,0	4,0	4,1	0,0
Temp. internal heat ex., LP inlet	T33	°C	3,2	3,2	3,3	0,0
Temp. compressor, suction	T10	°C	18,2	18,2	18,2	0,0
Pressure compressor, suction	P1	bar	32,7	32,8	32,9	0,0
Pressure compressor, discharge	P2	bar	105,5	105,8	106,1	0,2
Pressure gascoler, inlet	P002	bar	104,7	104,9	105,1	0,1
Pressure throttling valve,inlet	P3	bar	105,0	105,2	105,6	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,16	0,72	0,2
Pressure differential evapor LP	Pd2	bar	1,93	1,96	1,98	0,01
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,93	1,96	1,98	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,7	7,8	0,1
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,1	0,1
Temp.water gascooler, outlet	T002	°C	39,6	39,6	39,7	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	118,2	118,3	118,4	0,1
Temp. gascooler outlet	T004	°C	30,0	30,1	30,2	0,1
Pressure gascooler, inlet	P001	bar	104,5	104,7	105,1	0,2
Pressure differential gascooler HP	Pd001	bar	0,09	0,11	0,12	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured): **Nominal:**

1. Transcritical process ------- Experiment no: R0713_7
2. Hot water temp.: 60,3 °C 60 °C Date: 13.07.2000
3. Outdoor air temp.: 30,0 °C 30 °C Time: 16:15:34
4. Evaporation temp.: -0,2 °C 0 °C Operator: WA
5. Heat Rejection Pressure: 110,8 bar 110 bar
5. % Ratio of Q_{SGHX} to Q_{Evap} -0,9 %
7. With IHX. Comment: SGHX was not used as for oil return.
8. With LP-receiver HX. Oil return from oil separator was off.
9. Compressor speed (RPM) 804 Mode: Combined

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,90	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,43	
Cooling-COP (shaft)	[-]			2,68	± 4,24 %
Total-COP (shaft)	[-]			4,32	± 3,72 %
Load ratio	[-]			0,49	
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,124	± 4,21 %
Temp. evapor, outlet (T8)	°C			1,3	± 0,50 %
Evaporating temperature	°C			-0,2	
Cooling capacity (glycol)	kW			25,5	± 4,24 %
LMTD	K			3,56	
Overall heat transfer coeff.	W/m^2K			1340	
Water-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,054	± 2,41 %
Temp.water gascooler inlet (T14)	°C			19,8	± 0,50 %
Temp.water gascooler outlet (T15)	°C			60,3	± 0,50 %
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			15,7	± 2,36 %
Baseline load	kW			32,4	
Air-cooled Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,070	± 7,25 %
Temp.water gascooler inlet (T14)	°C			30,0	± 0,50 %
Temp.water gascooler outlet (T15)	°C			39,8	± 0,50 %
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			18,2	± 7,23 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			2,9	
LMTD	K			10,22	
Overall heat transfer coeff. outer	W/m^2K			1006	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,8	
Power consumption, shaft	kW			9,5	± 0,15 %
Power input, inverter	kW			11,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			110,8	
Temp. compressor, suction (T10)	°C			18,3	± 0,50 %
Temp. compressor, discharge (T1)	°C			126,3	± 0,50 %
Pressure ratio (P2/P1)	[-]			3,4	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			5,9	
Volumetric efficiency (based on water)	[-]			0,78	± 4,22 %
Isentropic efficiency (based on water)	[-]			0,81	± 4,34 %
Velocities	**Unit**			**Value**	**Uncertainty**
Water-cooled gascooler, CO_2 ,inlet	m/s			2,7	
Water-cooled gascooler, water	m/s			0,4	
Air-cooled gascooler, CO_2 ,inlet	m/s			3,4	
Air-cooled gascooler, water	m/s			1,7	
Compressor, suction	m/s			4,0	
Internal heat exchanger, HP, inlet	m/s			0,6	
Internal heat exchanger, HP, outlet	m/s			0,6	
Internal heat exchanger, LP, inlet	m/s			2,7	
Internal heat exchanger, LP, outlet	m/s			2,8	
LP-receiver	m/s			0,0	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0713_7
2. Hot water temp.: 60,3 °C	60 °C	Date:	13.07.2000
3. Outdoor air temp.: 30,0 °C	30 °C	Time:	16:15:34
4. Evaporation temp.: -0,2 °C	0 °C	Operator:	WA
5. Heat Rejection Pressure: 110,8 bar	110 bar		
6. % Ratio of Q_{SGHX} to Q_{Evap}	-0,9 %		
7. With IHX.	Comment:		SGHX was not used as for oil return.
8. With LP-receiver HX.			Oil return from oil separator was off.
9. Compressor speed (RPM)	804	Mode:	Combined

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,3	5,3	5,3	0,0
Temp.differential, evap.	Td1	K	3,3	3,3	3,3	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	12,2	12,3	12,5	0,1
Temp.water gascooler, inlet 1	T14	°C	19,7	19,8	19,9	0,0
Temp.water gascooler, inlet 2	T3	°C	19,7	19,8	19,8	0,0
Temp.water gascooler, outlet	T15	°C	60,3	60,3	60,3	0,0
Temp.differential, gas cooler	Td2	K	36,5	37,8	39,6	1,0
Temp.water drain cock	T16	°C	59,4	59,4	59,4	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,093	0,093	0,093	0,0
Water flow, drain cock	F3	l/s	0,072	0,072	0,072	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	113,2	113,3	113,5	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,7	11,7	11,8	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	126,1	126,3	126,4	0,1
Temp. gascooler inlet	T2	°C	123,0	123,2	123,3	0,1
Temp. gascooler outlet	T32	°C	19,7	19,8	19,9	0,0
Temp. internal heat ex., HP outlet	T4	°C	16,8	16,9	16,9	0,0
Temp. heat ex. LP-receiver, inlet	T5	°C	16,9	16,9	16,9	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	15,6	15,6	15,7	0,0
Temp. throttle valve, inlet	T7	°C	15,6	15,6	15,7	0,0
Temp. evapor, outlet	T8	°C	1,2	1,3	1,3	0,0
Temp. LP-receiver, outlet	T9	°C	4,2	4,3	4,3	0,0
Temp. internal heat ex., LP inlet	T33	°C	3,5	3,5	3,5	0,0
Temp. compressor, suction	T10	°C	18,3	18,3	18,3	0,0
Pressure compressor, suction	P1	bar	32,7	32,8	32,8	0,0
Pressure compressor, discharge	P2	bar	110,5	110,8	111,0	0,2
Pressure gascoler, inlet	P002	bar	109,7	109,9	110,3	0,2
Pressure throttling valve,inlet	P3	bar	110,0	110,2	110,4	0,1
Pressure differential gascooler HP	Pd1	bar	0,00	0,35	0,86	0,4
Pressure differential evapor LP	Pd2	bar	1,88	1,92	1,96	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,88	1,92	1,96	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	7,7	7,8	7,8	0,0
Temp.water gascooler, inlet 1	T001	°C	29,9	30,0	30,1	0,1
Temp.water gascooler, outlet	T002	°C	39,7	39,8	40,0	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,45	0,45	0,45	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	122,9	123,0	123,1	0,1
Temp. gascooler outlet	T004	°C	30,0	30,1	30,3	0,1
Pressure gascooler, inlet	P001	bar	109,5	109,7	109,9	0,1
Pressure differential gascooler HP	Pd001	bar	0,09	0,10	0,12	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_1
2. Hot water temp.: 60,7 °C	60 °C	Date:	20.06.2000
3. Evaporation temp.: 0,3 °C	0 °C	Time:	10:33:37
4. Heat Rejection Pressure: 80,1 bar	80 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	1,1 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			2,93	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,62	
Cooling-COP (shaft)	[-]			2,83	± 2,64 %
Total-COP (shaft)	[-]			6,58	± 3,36 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,135	± 4,26 %
Temp. evapor, outlet (T8)	°C			0,3	± 0,50 °C
Evaporating temperature	°C			0,3	
Cooling capacity (glycol)	kW			20,1	± 7,35 %
LMTD	K			2,50	
Overall heat transfer coeff.	W/m^2K			1508	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,135	± 4,26 %
Temp.water gascooler inlet (T14)	°C			20,5	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,7	± 0,50 °C
Temp. differential (T32-T14)	K			12,2	
Heating capacity	kW			26,7	± 1,99 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			5,6	
LMTD	K			14,96	
Overall heat transfer coeff. outer	W/m^2K			1343	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			6,9	
Power consumption, shaft	kW			7,1	± 0,17 %
Power input, inverter	kW			8,5	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			33,0	
Pressure compressor, discharge (P2)	bar			80,1	
Temp. compressor, suction (T10)	°C			26,2	± 0,50 °C
Temp. compressor, discharge (T1)	°C			102,7	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,4	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,8	
Volumetric efficiency (based on water)	[-]			0,90	± 4,28 %
Isentropic efficiency (based on water)	[-]			0,87	± 4,66 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			8,8	
Gascooler, water	m/s			0,6	
Compressor, suction	m/s			10,5	
Internal heat exchanger, HP, inlet	m/s			2,3	
Internal heat exchanger, HP, outlet	m/s			1,7	
Internal heat exchanger, LP, inlet	m/s			6,5	
Internal heat exchanger, LP, outlet	m/s			7,3	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_1
2. Hot water temp.: 60,7 °C	60 °C	Date:	20.06.2000
3. Evaporation temp.: 0,3 °C	0 °C	Time:	10:33:37
4. Heat Rejection Pressure: 80,1 bar	80 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	1,1 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	4,3	4,4	4,5	0,0
Temp.differential, evap.	Td1	K	2,5	2,6	2,7	0,1
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	14,0	14,3	14,5	0,1
Temp.water gascooler, inlet 1	T14	°C	20,1	20,5	21,2	0,4
Temp.water gascooler, inlet 2	T3	°C	20,1	20,4	21,1	0,4
Temp.water gascooler, outlet	T15	°C	60,4	60,7	61,0	0,1
Temp.differential, gas cooler	Td2	K	36,3	38,2	39,7	0,9
Temp.water drain cock	T16	°C	59,5	59,8	59,9	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,159	0,159	0,159	0,0
Water flow, drain cock	F3	l/s	0,057	0,066	0,070	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	84,3	84,4	84,5	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	8,5	8,5	8,6	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	102,6	102,7	102,8	0,1
Temp. gascooler inlet	T2	°C	100,0	100,2	100,2	0,1
Temp. gascooler outlet	T32	°C	32,6	32,7	32,8	0,1
Temp. internal heat ex., HP outlet	T4	°C	28,6	29,0	29,5	0,4
Temp. heat ex. LP-receiver, inlet	T5	°C	28,7	29,0	29,5	0,3
Temp. heat ex. LP-receiver, outlet	T6	°C	12,1	12,3	12,7	0,2
Temp. throttle valve, inlet	T7	°C	12,1	12,3	12,7	0,2
Temp. evapor, outlet	T8	°C	0,3	0,3	0,4	0,0
Temp. LP-receiver, outlet	T9	°C	0,4	0,4	0,4	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,3	0,4	0,4	0,0
Temp. compressor, suction	T10	°C	26,0	26,2	26,4	0,2
Pressure compressor, suction	P1	bar	32,9	33,0	33,1	0,1
Pressure compressor, discharge	P2	bar	79,8	80,1	80,3	0,1
Pressure gascoler, inlet	P002	bar	78,9	79,0	79,1	0,0
Pressure throttling valve,inlet	P3	bar	77,2	77,3	77,5	0,1
Pressure differential gascooler HP	Pd1	bar	1,43	1,94	2,26	0,3
Pressure differential evapor LP	Pd2	bar	2,14	2,18	2,22	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,14	2,18	2,22	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	22,9	23,0	23,0	0,0
Temp.water gascooler, inlet 1	T001	°C	25,6	25,7	25,8	0,1
Temp.water gascooler, outlet	T002	°C	23,7	23,7	23,8	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	31,3	31,4	31,4	0,0
Temp. gascooler outlet	T004	°C	27,5	27,7	27,9	0,2
Pressure gascooler, inlet	P001	bar	76,8	76,9	76,9	0,0
Pressure differential gascooler HP	Pd001	bar	0,00	0,00	0,00	0,00

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**	
1. Transcritical process	-------	Experiment no: R0620_2
2. Hot water temp.: 60,3 °C	60 °C	Date: 20.06.2000
3. Evaporation temp.: 0,1 °C	0 °C	Time: 14:39:55
4. Heat Rejection Pressure: 85,4 bar	85 bar	Operator: WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,2 %	
6. Without IHX.	Comment:	SGHX was not used as for oil return.
7. With LP-receiver HX.		Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode: Water-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,38	
Cooling-COP (incl. engine, η_{eng}.= 0,90)	[-]			3,02	
Cooling-COP (shaft)	[-]			3,28	± 2,49 %
Total-COP (shaft)	[-]			7,47	± 1,41 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,136	± 1,96 %
Temp. evapor, outlet (T8)	°C			0,1	± 0,50 °C
Evaporating temperature	°C			0,1	
Cooling capacity (glycol)	kW			24,9	± 2,00 %
LMTD	K			3,27	
Overall heat transfer coeff.	W/m^2K			1420	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,136	± 1,96 %
Temp.water gascooler inlet (T14)	°C			20,5	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,3	± 0,50 °C
Temp. differential (T32-T14)	K			9,9	
Heating capacity	kW			31,9	± 1,94 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			4,6	
LMTD	K			14,32	
Overall heat transfer coeff. outer	W/m^2K			1144	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,4	
Power consumption, shaft	kW			7,6	± 0,17 %
Power input, inverter	kW			9,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			85,4	
Temp. compressor, suction (T10)	°C			21,1	± 0,50 °C
Temp. compressor, discharge (T1)	°C			103,7	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,6	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,6	
Volumetric efficiency (based on water)	[-]			0,88	± 2,00 %
Isentropic efficiency (based on water)	[-]			0,87	± 2,69 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			8,2	
Gascooler, water	m/s			0,7	
Compressor, suction	m/s			10,2	
Internal heat exchanger, HP, inlet	m/s			1,7	
Internal heat exchanger, HP, outlet	m/s			1,5	
Internal heat exchanger, LP, inlet	m/s			6,6	
Internal heat exchanger, LP, outlet	m/s			7,1	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_2
2. Hot water temp.: 60,3 °C	60 °C	Date:	20.06.2000
3. Evaporation temp.: 0,1 °C	0 °C	Time:	14:39:55
4. Heat Rejection Pressure: 85,4 bar	85 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	0,2 %		
6. Without IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,2	5,2	5,3	0,0
Temp.differential, evap.	Td1	K	3,2	3,2	3,2	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	13,2	13,4	13,5	0,1
Temp.water gascooler, inlet 1	T14	°C	20,4	20,5	20,5	0,1
Temp.water gascooler, inlet 2	T3	°C	20,3	20,4	20,5	0,1
Temp.water gascooler, outlet	T15	°C	60,1	60,3	60,6	0,2
Temp.differential, gas cooler	Td2	K	36,0	37,4	39,1	1,1
Temp.water drain cock	T16	°C	59,2	59,3	59,7	0,2
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,192	0,192	0,192	0,0
Water flow, drain cock	F3	l/s	0,073	0,073	0,073	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	90,0	90,1	90,3	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,1	9,2	9,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	103,6	103,7	103,8	0,1
Temp. gascooler inlet	T2	°C	101,1	101,2	101,2	0,0
Temp. gascooler outlet	T32	°C	30,3	30,4	30,6	0,1
Temp. internal heat ex., HP outlet	T4	°C	21,1	21,2	21,4	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	21,1	21,2	21,4	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	9,9	10,0	10,1	0,1
Temp. throttle valve, inlet	T7	°C	9,9	10,0	10,1	0,1
Temp. evapor, outlet	T8	°C	0,1	0,1	0,2	0,0
Temp. LP-receiver, outlet	T9	°C	0,5	0,6	0,6	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,4	0,4	0,4	0,0
Temp. compressor, suction	T10	°C	21,0	21,1	21,2	0,1
Pressure compressor, suction	P1	bar	32,8	32,8	32,9	0,0
Pressure compressor, discharge	P2	bar	85,3	85,4	85,9	0,1
Pressure gascoler, inlet	P002	bar	84,3	84,4	84,5	0,1
Pressure throttling valve,inlet	P3	bar	83,0	83,1	83,3	0,1
Pressure differential gascooler HP	Pd1	bar	1,22	1,66	2,04	0,3
Pressure differential evapor LP	Pd2	bar	2,11	2,15	2,19	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,11	2,15	2,19	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	23,6	23,6	23,7	0,1
Temp.water gascooler, inlet 1	T001	°C	26,1	26,1	26,2	0,0
Temp.water gascooler, outlet	T002	°C	24,8	24,9	24,9	0,1
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	30,0	30,4	30,7	0,2
Temp. gascooler outlet	T004	°C	25,6	25,7	25,8	0,1
Pressure gascooler, inlet	P001	bar	82,5	82,6	82,7	0,1
Pressure differential gascooler HP	Pd001	bar	-0,01	0,00	0,01	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	Nominal:		
1. Transcritical process	-------	Experiment no:	R0620_3
2. Hot water temp.: 60,5 °C	60 °C	Date:	20.06.1990
3. Evaporation temp.: 0,0 °C	0 °C	Time:	15:05:23
4. Heat Rejection Pressure: 90,2 bar	90 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,1 %		
6. Without IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	Unit			Value	Uncertainty
Cooling-COP (entalpy differential)	[-]			3,59	
Cooling-COP (incl. engine, η_{eng}.= 0,90)	[-]			3,21	
Cooling-COP (shaft)	[-]			3,49	± 2,39 %
Total-COP (shaft)	[-]			7,92	± 1,36 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,91 %
Temp. evapor, outlet (T8)	°C			0,0	± 0,50 °C
Evaporating temperature	°C			0,0	
Cooling capacity (glycol)	kW			28,0	± 1,94 %
LMTD	K			3,58	
Overall heat transfer coeff.	W/m^2K			1463	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,91 %
Temp.water gascooler inlet (T14)	°C			20,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,5	± 0,50 °C
Temp. differential (T32-T14)	K			3,8	
Heating capacity	kW			35,5	± 1,88 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,5	
LMTD	K			11,52	
Overall heat transfer coeff. outer	W/m^2K			1081	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			7,8	
Power consumption, shaft	kW			8,0	± 0,16 %
Power input, inverter	kW			9,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			90,2	
Temp. compressor, suction (T10)	°C			14,8	± 0,50 °C
Temp. compressor, discharge (T1)	°C			102,3	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,8	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,5	
Volumetric efficiency (based on water)	[-]			0,86	± 1,95 %
Isentropic efficiency (based on water)	[-]			0,85	± 2,70 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,7	
Gascooler, water	m/s			0,8	
Compressor, suction	m/s			10,0	
Internal heat exchanger, HP, inlet	m/s			1,6	
Internal heat exchanger, HP, outlet	m/s			1,4	
Internal heat exchanger, LP, inlet	m/s			6,7	
Internal heat exchanger, LP, outlet	m/s			6,9	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_3
2. Hot water temp.: 60,5 °C	60 °C	Date:	20.06.1990
3. Evaporation temp.: 0,0 °C	0 °C	Time:	15:05:23
4. Heat Rejection Pressure: 90,2 bar	90 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,1 %		
6. Without IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,6	5,7	5,8	0,0
Temp.differential, evap.	Td1	K	3,5	3,6	3,6	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	12,6	12,9	13,1	0,1
Temp.water gascooler, inlet 1	T14	°C	19,9	20,0	20,2	0,1
Temp.water gascooler, inlet 2	T3	°C	19,9	20,0	20,2	0,1
Temp.water gascooler, outlet	T15	°C	60,5	60,5	60,6	0,0
Temp.differential, gas cooler	Td2	K	36,8	38,4	39,7	1,0
Temp.water drain cock	T16	°C	59,5	59,6	59,7	0,0
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,210	0,210	0,210	0,0
Water flow, drain cock	F3	l/s	0,092	0,092	0,092	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	95,1	95,2	95,4	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	9,7	9,7	9,7	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	102,1	102,3	102,5	0,1
Temp. gascooler inlet	T2	°C	99,9	100,0	100,1	0,1
Temp. gascooler outlet	T32	°C	23,5	23,8	24,2	0,2
Temp. internal heat ex., HP outlet	T4	°C	14,6	14,8	15,1	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	14,6	14,8	15,1	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	7,3	7,4	7,5	0,0
Temp. throttle valve, inlet	T7	°C	7,4	7,4	7,5	0,0
Temp. evapor, outlet	T8	°C	0,0	0,0	0,1	0,0
Temp. LP-receiver, outlet	T9	°C	0,5	0,5	0,5	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,3	0,3	0,4	0,0
Temp. compressor, suction	T10	°C	14,6	14,8	15,1	0,1
Pressure compressor, suction	P1	bar	32,6	32,7	32,8	0,1
Pressure compressor, discharge	P2	bar	90,0	90,2	90,5	0,2
Pressure gascoler, inlet	P002	bar	89,0	89,2	89,3	0,1
Pressure throttling valve,inlet	P3	bar	88,0	88,2	88,3	0,1
Pressure differential gascooler HP	Pd1	bar	1,24	1,45	1,69	0,2
Pressure differential evapor LP	Pd2	bar	2,14	2,18	2,23	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,14	2,18	2,23	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	23,8	23,9	23,9	0,1
Temp.water gascooler, inlet 1	T001	°C	26,3	26,4	26,5	0,1
Temp.water gascooler, outlet	T002	°C	25,0	25,1	25,1	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	30,2	30,4	30,9	0,2
Temp. gascooler outlet	T004	°C	26,6	27,0	27,4	0,2
Pressure gascooler, inlet	P001	bar	87,5	87,6	87,7	0,1
Pressure differential gascooler HP	Pd001	bar	-0,01	0,00	0,01	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_4
2. Hot water temp.: 60,6 °C	60 °C	Date:	20.06.2000
3. Evaporation temp.: -0,2 °C	0 °C	Time:	17:04:01
4. Heat Rejection Pressure: 95,5 bar	95 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,5 %		
6. Without IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	Unit			Value	Uncertainty
Cooling-COP (entalpy differential)	[-]			3,53	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			3,16	
Cooling-COP (shaft)	[-]			3,44	± 2,40 %
Total-COP (shaft)	[-]			7,80	± 1,37 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,91 %
Temp. evapor, outlet (T8)	°C			-0,2	± 0,50 °C
Evaporating temperature	°C			-0,2	
Cooling capacity (glycol)	kW			29,0	± 1,94 %
LMTD	K			3,80	
Overall heat transfer coeff.	W/m^2K			1426	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,91 %
Temp.water gascooler inlet (T14)	°C			20,5	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,6	± 0,50 °C
Temp. differential (T32-T14)	K			0,6	
Heating capacity	kW			36,8	± 1,88 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			3,0	
LMTD	K			9,99	
Overall heat transfer coeff. outer	W/m^2K			1084	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,2	
Power consumption, shaft	kW			8,4	± 0,16 %
Power input, inverter	kW			10,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,6	
Pressure compressor, discharge (P2)	bar			95,5	
Temp. compressor, suction (T10)	°C			13,1	± 0,50 °C
Temp. compressor, discharge (T1)	°C			105,8	± 0,50 °C
Pressure ratio (P2/P1)	[-]			2,9	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,4	
Volumetric efficiency (based on water)	[-]			0,85	± 1,96 %
Isentropic efficiency (based on water)	[-]			0,85	± 2,65 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,3	
Gascooler, water	m/s			0,8	
Compressor, suction	m/s			9,9	
Internal heat exchanger, HP, inlet	m/s			1,5	
Internal heat exchanger, HP, outlet	m/s			1,4	
Internal heat exchanger, LP, inlet	m/s			6,8	
Internal heat exchanger, LP, outlet	m/s			6,9	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0620_4
2. Hot water temp.: 60,6 °C	60 °C	Date:	20.06.2000
3. Evaporation temp.: -0,2 °C	0 °C	Time:	17:04:01
4. Heat Rejection Pressure: 95,5 bar	95 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,5 %		
6. Without IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,7	5,7	5,8	0,0
Temp.differential, evap.	Td1	K	3,6	3,6	3,6	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	9,1	9,2	9,3	0,1
Temp.water gascooler, inlet 1	T14	°C	20,1	20,5	20,8	0,2
Temp.water gascooler, inlet 2	T3	°C	20,1	20,4	20,7	0,2
Temp.water gascooler, outlet	T15	°C	60,6	60,6	60,8	0,1
Temp.differential, gas cooler	Td2	K	36,6	37,9	39,9	0,9
Temp.water drain cock	T16	°C	59,6	59,7	59,9	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,216	0,220	0,221	0,0
Water flow, drain cock	F3	l/s	0,163	0,163	0,163	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	99,9	100,2	100,4	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	10,2	10,2	10,2	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	105,3	105,8	105,9	0,2
Temp. gascooler inlet	T2	°C	102,7	103,3	103,5	0,2
Temp. gascooler outlet	T32	°C	21,0	21,1	21,3	0,1
Temp. internal heat ex., HP outlet	T4	°C	12,5	12,7	12,8	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	12,6	12,7	12,8	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	7,5	7,7	7,8	0,1
Temp. throttle valve, inlet	T7	°C	7,5	7,7	7,9	0,1
Temp. evapor, outlet	T8	°C	-0,2	-0,2	-0,1	0,0
Temp. LP-receiver, outlet	T9	°C	0,6	0,6	0,7	0,0
Temp. internal heat ex., LP inlet	T33	°C	0,3	0,3	0,4	0,0
Temp. compressor, suction	T10	°C	13,1	13,1	13,2	0,1
Pressure compressor, suction	P1	bar	32,5	32,6	32,8	0,1
Pressure compressor, discharge	P2	bar	95,1	95,5	95,8	0,2
Pressure gascoler, inlet	P002	bar	93,9	94,4	94,7	0,2
Pressure throttling valve,inlet	P3	bar	93,4	93,7	94,0	0,2
Pressure differential gascooler HP	Pd1	bar	0,56	1,22	1,84	0,5
Pressure differential evapor LP	Pd2	bar	2,09	2,12	2,16	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,09	2,12	2,16	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	24,3	24,3	24,3	0,0
Temp.water gascooler, inlet 1	T001	°C	27,0	27,0	27,0	0,0
Temp.water gascooler, outlet	T002	°C	25,2	25,2	25,2	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	29,6	29,7	29,9	0,1
Temp. gascooler outlet	T004	°C	25,7	25,7	25,7	0,0
Pressure gascooler, inlet	P001	bar	92,6	93,1	93,3	0,2
Pressure differential gascooler HP	Pd001	bar	-0,01	0,00	0,01	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0621_1
2. Hot water temp.: 60,6 °C	60 °C	Date:	21.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	15:29:54
4. Heat Rejection Pressure: 100,4 bar	100 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,3 %		
6. Without IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	Unit			Value	Uncertainty
Cooling-COP (entalpy differential)	[-]			3,40	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			3,03	
Cooling-COP (shaft)	[-]			3,32	± 2,40 %
Total-COP (shaft)	[-]			7,58	± 1,36 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,89 %
Temp. evapor, outlet (T8)	°C			-0,1	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			29,2	± 1,92 %
LMTD	K			3,63	
Overall heat transfer coeff.	W/m^2K			1504	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,138	± 1,89 %
Temp.water gascooler inlet (T14)	°C			20,1	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,6	± 0,50 °C
Temp. differential (T32-T14)	K			0,2	
Heating capacity	kW			37,4	± 1,87 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			2,8	
LMTD	K			9,52	
Overall heat transfer coeff. outer	W/m^2K			1041	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,6	
Power consumption, shaft	kW			8,8	± 0,16 %
Power input, inverter	kW			10,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,6	
Pressure compressor, discharge (P2)	bar			100,4	
Temp. compressor, suction (T10)	°C			12,3	± 0,50 °C
Temp. compressor, discharge (T1)	°C			109,6	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,1	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,4	
Volumetric efficiency (based on water)	[-]			0,84	± 1,94 %
Isentropic efficiency (based on water)	[-]			0,85	± 2,60 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			7,0	
Gascooler, water	m/s			0,9	
Compressor, suction	m/s			9,8	
Internal heat exchanger, HP, inlet	m/s			1,5	
Internal heat exchanger, HP, outlet	m/s			1,4	
Internal heat exchanger, LP, inlet	m/s			6,8	
Internal heat exchanger, LP, outlet	m/s			6,8	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**					
1. Transcritical process	-------			Experiment no:		R0621_1
2. Hot water temp.: 60,6 °C	60 °C			Date:		21.06.2000
3. Evaporation temp.: -0,1 °C	0 °C			Time:		15:29:54
4. Heat Rejection Pressure: 100,4 bar	100 bar			Operator:		WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,3 %					
6. Without IHX.	Comment:					SGHX was not used as for oil return.
7. With LP-receiver HX.						Oil return from oil separator was off.
8. Compressor speed (RPM)	804			Mode:		Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,5	5,7	5,9	0,1
Temp.differential, evap.	Td1	K	3,6	3,8	3,9	0,1
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	11,1	11,4	11,8	0,3
Temp.water gascooler, inlet 1	T14	°C	19,8	20,1	20,5	0,2
Temp.water gascooler, inlet 2	T3	°C	19,7	20,1	20,5	0,2
Temp.water gascooler, outlet	T15	°C	60,5	60,6	60,7	0,1
Temp.differential, gas cooler	Td2	K	36,5	38,5	40,3	1,2
Temp.water drain cock	T16	°C	59,6	59,7	59,8	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,222	0,222	0,222	0,0
Water flow, drain cock	F3	l/s	0,173	0,173	0,173	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	104,2	104,5	104,7	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	10,7	10,7	10,8	0,0
CO_2-circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	109,4	109,6	109,8	0,1
Temp. gascooler inlet	T2	°C	107,1	107,2	107,3	0,1
Temp. gascooler outlet	T32	°C	19,9	20,3	20,6	0,2
Temp. internal heat ex., HP outlet	T4	°C	12,0	12,3	12,5	0,2
Temp. heat ex. LP-receiver, inlet	T5	°C	12,1	12,4	12,6	0,2
Temp. heat ex. LP-receiver, outlet	T6	°C	10,6	11,0	11,2	0,2
Temp. throttle valve, inlet	T7	°C	10,7	11,0	11,3	0,1
Temp. evapor, outlet	T8	°C	-0,2	-0,1	0,0	0,1
Temp. LP-receiver, outlet	T9	°C	1,5	1,6	1,7	0,1
Temp. internal heat ex., LP inlet	T33	°C	1,0	1,0	1,1	0,0
Temp. compressor, suction	T10	°C	12,1	12,3	12,5	0,1
Pressure compressor, suction	P1	bar	32,5	32,6	32,7	0,0
Pressure compressor, discharge	P2	bar	100,1	100,4	100,6	0,2
Pressure gascoler, inlet	P002	bar	99,2	99,4	99,6	0,1
Pressure throttling valve,inlet	P3	bar	98,7	98,8	99,0	0,1
Pressure differential gascooler HP	Pd1	bar	0,74	0,99	1,38	0,2
Pressure differential evapor LP	Pd2	bar	2,11	2,15	2,19	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	2,11	2,15	2,19	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	25	25,0	25,0	0,0
Temp.water gascooler, inlet 1	T001	°C	26,6	26,7	26,7	0,0
Temp.water gascooler, outlet	T002	°C	26,5	26,6	26,6	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2-circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	29,8	30,0	30,2	0,1
Temp. gascooler outlet	T004	°C	27,4	27,6	27,8	0,1
Pressure gascooler, inlet	P001	bar	98,0	98,2	98,4	0,1
Pressure differential gascooler HP	Pd001	bar	-0,01	0,00	0,01	0,01

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0622_1
2. Hot water temp.: 60,2 °C	60 °C	Date:	22.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	11:14:25
4. Heat Rejection Pressure: 105,6 bar	105 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,7 %		
6. Without IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,24	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,84	
Cooling-COP (shaft)	[-]			3,13	± 2,45 %
Total-COP (shaft)	[-]			7,19	± 1,36 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,134	± 1,90 %
Temp. evapor, outlet (T8)	°C			0,1	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			28,7	± 1,93 %
LMTD	K			3,61	
Overall heat transfer coeff.	W/m^2K			1486	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,134	± 1,90 %
Temp.water gascooler inlet (T14)	°C			20,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,2	± 0,50 °C
Temp. differential (T32-T14)	K			0,1	
Heating capacity	kW			37,2	± 1,88 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			2,5	
LMTD	K			8,43	
Overall heat transfer coeff. outer	W/m^2K			1074	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			8,9	
Power consumption, shaft	kW			9,2	± 0,15 %
Power input, inverter	kW			11,2	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,7	
Pressure compressor, discharge (P2)	bar			105,6	
Temp. compressor, suction (T10)	°C			13,5	± 0,50 °C
Temp. compressor, discharge (T1)	°C			115,8	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,2	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,2	
Volumetric efficiency (based on water)	[-]			0,82	± 1,95 %
Isentropic efficiency (based on water)	[-]			0,84	± 2,54 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			6,6	
Gascooler, water	m/s			0,9	
Compressor, suction	m/s			9,6	
Internal heat exchanger, HP, inlet	m/s			1,4	
Internal heat exchanger, HP, outlet	m/s			1,3	
Internal heat exchanger, LP, inlet	m/s			6,6	
Internal heat exchanger, LP, outlet	m/s			6,6	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0622_1
2. Hot water temp.: 60,2 °C	60 °C	Date:	22.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	11:14:25
4. Heat Rejection Pressure: 105,6 bar	105 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,7 %		
6. Without IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,6	5,6	5,7	0,0
Temp.differential, evap.	Td1	K	3,6	3,6	3,6	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	13,6	14,7	15,9	0,8
Temp.water gascooler, inlet 1	T14	°C	19,9	20,0	20,3	0,1
Temp.water gascooler, inlet 2	T3	°C	19,8	20,0	20,2	0,1
Temp.water gascooler, outlet	T15	°C	59,9	60,2	60,3	0,1
Temp.differential, gas cooler	Td2	K	35,7	38,0	39,4	1,0
Temp.water drain cock	T16	°C	59,1	59,3	59,4	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,222	0,223	0,227	0,0
Water flow, drain cock	F3	l/s	0,156	0,165	0,169	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	108,9	109,0	109,2	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,2	11,2	11,3	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	115,5	115,8	116,5	0,3
Temp. gascooler inlet	T2	°C	113,0	113,5	114,4	0,4
Temp. gascooler outlet	T32	°C	20,0	20,1	20,3	0,1
Temp. internal heat ex., HP outlet	T4	°C	12,3	12,5	12,6	0,1
Temp. heat ex. LP-receiver, inlet	T5	°C	12,4	12,5	12,7	0,1
Temp. heat ex. LP-receiver, outlet	T6	°C	11,5	11,6	11,8	0,1
Temp. throttle valve, inlet	T7	°C	11,5	11,6	11,8	0,1
Temp. evapor, outlet	T8	°C	0,1	0,1	0,1	0,0
Temp. LP-receiver, outlet	T9	°C	2,5	2,6	2,8	0,1
Temp. internal heat ex., LP inlet	T33	°C	1,9	1,9	2,1	0,1
Temp. compressor, suction	T10	°C	13,3	13,5	13,7	0,1
Pressure compressor, suction	P1	bar	32,7	32,7	32,8	0,0
Pressure compressor, discharge	P2	bar	105,4	105,6	105,8	0,1
Pressure gascoler, inlet	P002	bar	104,4	104,6	104,8	0,1
Pressure throttling valve,inlet	P3	bar	104,1	104,3	104,5	0,1
Pressure differential gascooler HP	Pd1	bar	0,67	0,97	1,17	0,2
Pressure differential evapor LP	Pd2	bar	1,97	2,02	2,06	0,03
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,97	2,02	2,06	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	24,1	24,3	24,5	0,1
Temp.water gascooler, inlet 1	T001	°C	26,0	26,2	26,4	0,1
Temp.water gascooler, outlet	T002	°C	25,3	25,3	25,4	0,0
Temp.differential, gas cooler	DT001	K	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F001	l/s	0,00	0,00	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	29,8	30,0	30,2	0,1
Temp. gascooler outlet	T004	°C	27,2	27,5	27,9	0,2
Pressure gascooler, inlet	P001	bar	103,4	103,6	103,7	0,1
Pressure differential gascooler HP	Pd001	bar	-0,01	0,00	0,00	0,00

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0622_2
2. Hot water temp.: 60,2 °C	60 °C	Date:	22.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	11:44:34
4. Heat Rejection Pressure: 111,0 bar	110 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,7 %		
6. With IHX.	Comment:	SGHX was not used as for oil return.	
7. With LP-receiver HX.		Oil return from oil separator was off.	
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Heat pump	**Unit**			**Value**	**Uncertainty**
Cooling-COP (entalpy differential)	[-]			3,13	
Cooling-COP (incl. engine, $\eta_{eng.}$= 0,90)	[-]			2,70	
Cooling-COP (shaft)	[-]			2,98	± 2,47 %
Total-COP (shaft)	[-]			6,88	± 1,40 %
Evaporator:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,132	± 1,96 %
Temp. evapor, outlet (T8)	°C			0,4	± 0,50 °C
Evaporating temperature	°C			-0,1	
Cooling capacity (glycol)	kW			28,4	± 2,07 %
LMTD	K			3,67	
Overall heat transfer coeff.	W/m^2K			1447	
Gascooler:	**Unit**			**Value**	**Uncertainty**
Refrigerant, flow rate (calculated)	kg/s			0,132	± 1,96 %
Temp.water gascooler inlet (T14)	°C			20,0	± 0,50 °C
Temp.water gascooler outlet (T15)	°C			60,2	± 0,50 °C
Temp. differential (T32-T14)	K			0,0	
Heating capacity	kW			37,2	± 1,88 %
Internal heat exchanger	**Unit**			**Value**	**Uncertainty**
Capacity ,HP	kW			2,5	
LMTD	K			8,22	
Overall heat transfer coeff. outer	W/m^2K			1075	
Compressor:	**Unit**			**Value**	**Uncertainty**
Power consumption, entalpy diff	kW			9,1	
Power consumption, shaft	kW			9,5	± 0,15 %
Power input, inverter	kW			11,7	
Number of revolutions	rpm			804	
Pressure compressor, suction (P1)	bar			32,8	
Pressure compressor, discharge (P2)	bar			111,0	
Temp. compressor, suction (T10)	°C			13,5	± 0,50 °C
Temp. compressor, discharge (T1)	°C			120,2	± 0,50 °C
Pressure ratio (P2/P1)	[-]			3,4	± 0,14 %
Teoretical Swept volume	m^3/h			7,6	
Swept volume (based on water)	m^3/h			6,1	
Volumetric efficiency (based on water)	[-]			0,80	± 2,00 %
Isentropic efficiency (based on water)	[-]			0,83	± 2,53 %
Velocities	**Unit**			**Value**	**Uncertainty**
Gascooler, CO_2 ,inlet	m/s			6,3	
Gascooler, water	m/s			0,9	
Compressor, suction	m/s			9,4	
Internal heat exchanger, HP, inlet	m/s			1,4	
Internal heat exchanger, HP, outlet	m/s			1,3	
Internal heat exchanger, LP, inlet	m/s			6,4	
Internal heat exchanger, LP, outlet	m/s			6,5	
LP-receiver	m/s			0,1	

Heat pump using CO_2 as refrigerant

Conditions (Measured):	**Nominal:**		
1. Transcritical process	-------	Experiment no:	R0622_2
2. Hot water temp.: 60,2 °C	60 °C	Date:	22.06.2000
3. Evaporation temp.: -0,1 °C	0 °C	Time:	11:44:34
4. Heat Rejection Pressure: 111,0 bar	110 bar	Operator:	WA
5. % Ratio of Q_{SGHX} to Q_{Evap}	-0,7 %		
6. With IHX.	Comment:		SGHX was not used as for oil return.
7. With LP-receiver HX.			Oil return from oil separator was off.
8. Compressor speed (RPM)	804	Mode:	Water-cooled

Glycolcircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.evap. inlet	T13	°C	5,7	5,7	5,7	0,0
Temp.differential, evap.	Td1	K	3,6	3,6	3,6	0,0
Glycol flow rate	F1	l/s	0,00	0,00	0,00	0,0
Watercircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T36	°C	11,8	12,3	12,7	0,2
Temp.water gascooler, inlet 1	T14	°C	19,8	20,0	20,1	0,1
Temp.water gascooler, inlet 2	T3	°C	19,8	19,9	20,1	0,1
Temp.water gascooler, outlet	T15	°C	60,1	60,2	60,3	0,1
Temp.differential, gas cooler	Td2	K	36,3	38,2	40,0	1,1
Temp.water drain cock	T16	°C	59,3	59,4	59,4	0,1
Power consumption, heat element 3	N3	W	0,0	0,0	0,0	0,0
Water flow gascooler, inlet	F2	l/s	0,222	0,222	0,222	0,0
Water flow, drain cock	F3	l/s	0,143	0,147	0,148	0,0
Compressordata:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Torque	W1	Nm	113,2	113,3	113,4	0,1
Number of revolutions	S1	rpm	804	804	804	0,0
Power consumption, transmitter	N1	kW	11,7	11,7	11,8	0,0
CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. compressor, discharge	T1	°C	120,1	120,2	120,2	0,0
Temp. gascooler inlet	T2	°C	117,3	117,5	117,6	0,1
Temp. gascooler outlet	T32	°C	19,8	20,0	20,1	0,1
Temp. internal heat ex., HP outlet	T4	°C	12,2	12,3	12,4	0,0
Temp. heat ex. LP-receiver, inlet	T5	°C	12,3	12,4	12,4	0,0
Temp. heat ex. LP-receiver, outlet	T6	°C	11,3	11,4	11,5	0,0
Temp. throttle valve, inlet	T7	°C	11,3	11,4	11,5	0,1
Temp. evapor, outlet	T8	°C	0,4	0,4	0,5	0,0
Temp. LP-receiver, outlet	T9	°C	2,7	2,7	2,7	0,0
Temp. internal heat ex., LP inlet	T33	°C	2,0	2,1	2,1	0,0
Temp. compressor, suction	T10	°C	13,4	13,5	13,5	0,0
Pressure compressor, suction	P1	bar	32,7	32,8	32,9	0,1
Pressure compressor, discharge	P2	bar	110,7	111,0	111,1	0,1
Pressure gascoler, inlet	P002	bar	108,9	109,0	109,1	0,1
Pressure throttling valve,inlet	P3	bar	109,7	109,8	109,9	0,1
Pressure differential gascooler HP	Pd1	bar	0,63	0,91	1,13	0,2
Pressure differential evapor LP	Pd2	bar	1,94	1,98	2,02	0,02
Pressure diff. outlet evap, inlet compr.	Pd3	bar	1,94	1,98	2,02	0,0
New WaterCircuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp.freshwater, inlet	T005	°C	26,8	27,1	27,5	0,2
Temp.water gascooler, inlet 1	T001	°C	0,0	0,0	0,0	0,0
Temp.water gascooler, outlet	T002	°C	26,6	26,7	26,9	0,1
Temp.differential, gas cooler	DT001	K	24,3	24,4	24,5	0,1
Water flow gascooler, inlet	F001	l/s	-0,02	-0,01	0,00	0,00
New CO_2 -circuit:	**Measure pnt.**	**Unit**	**Min**	**Mean**	**Max**	**STDEV**
Temp. gascooler inlet	T003	°C	25,4	25,4	25,5	0,1
Temp. gascooler outlet	T004	°C	29,9	29,9	30,0	0,0
Pressure gascooler, inlet	P001	bar	0,0	0,0	0,0	0,0
Pressure differential gascooler HP	Pd001	bar	1,10	1,10	1,10	0,00