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En kunskapssammanställning för att stimulera användandet av naturliga köldmedier inom elektronikkylning

Utvärdering av systemlösningar med naturliga köldmedier i serverhallar/datacenters

- Presentation av resultaten som hölls vid konferensen Thermal Management of Electronics Systems i Stockholm 7 september 2017 med titeln: ” Natural Refrigerants in Data Centre Cooling”
- Skriftlig rapport över studien som helhet: ”Natural Refrigerants in Data Center Cooling with Thermosiphon Application”

Jörgen Rogstam

17 januari 2018

Energi & Kylanalys AB I Prästgårdsgränd 4 I 125 44 Älvsjö I Tel. 08-550 102 10 I www.ekanalys.se

Natural Refrigerants in Data Centre Cooling

@ Thermal Magement of Electronic Systems

Jörgen Rogstam

Stockholm

7 September 2017

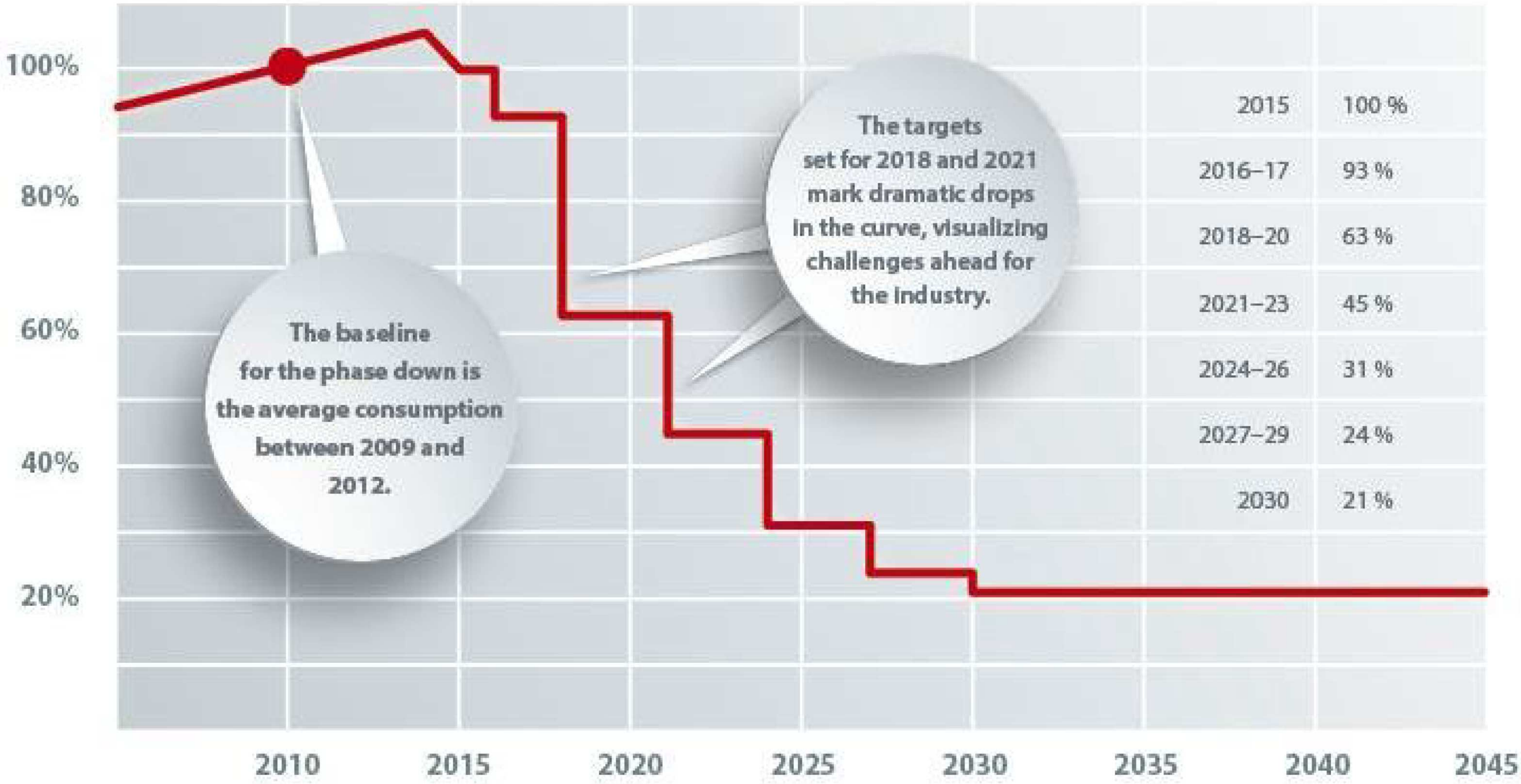
Agenda

- Introduction
  - Environmental aspects
- Refrigerants
  - Selection
- Case study on Data Centre Cooling
  - System solutions
  - Results
- Field applications with CO_2
  - “AQUILON”
- Q&A

Environmental aspects

- F-gases are the most commonly used refrigerants today
- Fluorinated hydrocarbons (F-gases) are so called greenhouse gases
- Greenhouse gases need to be reduced by 80-95% by 2050 to “limit” the Global Warming to 2^{o}C (IPCC)
- F-gases need to be reduced by 70-78%
- This is the reason to look at natural refrigerants!

F-gas phase out in Europe



Similar propsed actions in the rest of the world!

EKA

Alternative refrigerants - short list

Refrigerant	ODP	GWP	Type	Classification
R12	1	10900	CFC	A1
R22	0,05	1810	HCFC	A1
R134a	0	1430	HFC	A1
R1234yf	0	<1	HFO	A2L
R1234ze	0	<1	HFO	A2L
R170 (Ethane)	0	6	Natural (Hydrocarbon)	A3
R290 (Propane)	0	3	Natural (Hydrocarbon)	A3
R600a (Isobutane)	0	3	Natural (Hydrocarbon)	A3
R717 (Ammonia)	0	0	Natural	B2L
R744 (CO_2)	0	1	Natural	A1

To be replaced

Indirect system

Direct or indirect system

Natural

Non flammable

- History
- Market today
- Environment
- How does it work?

CO2 REFRIGERATION SYSTEMS

CO2 systems globally in 2017

Reference: Shecco 2017

EKA

Case study in 2016 – Data Centre cooling

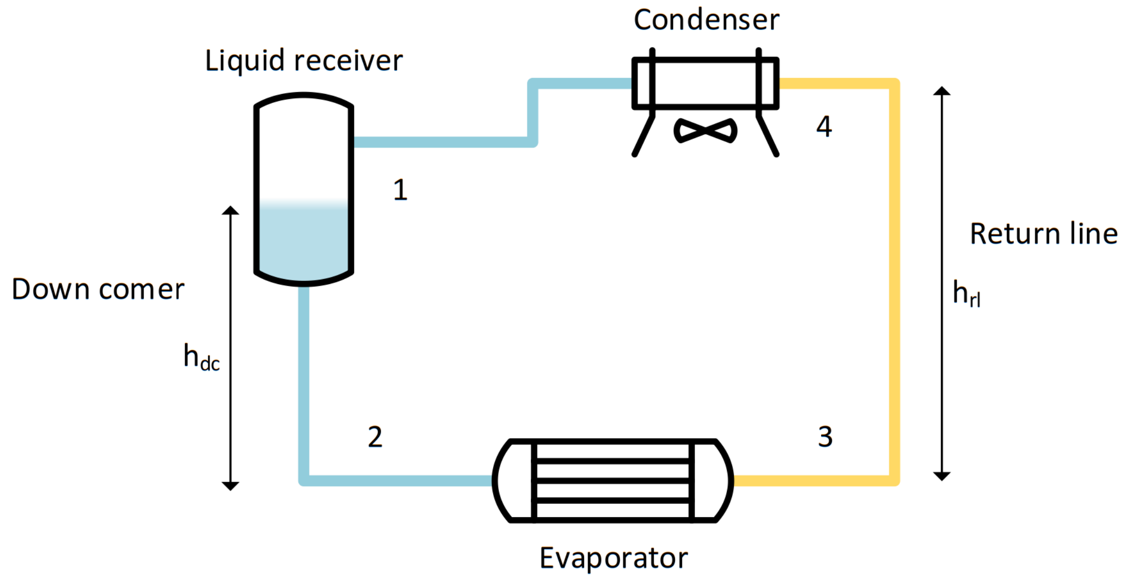
Natural Refrigerants in Data Center Cooling with Thermosiphon Application

ANDRÉ SAHLSTEN
VICTOR HEINERUD

KTH Industrial Engineering and Management

Thermosiphon model

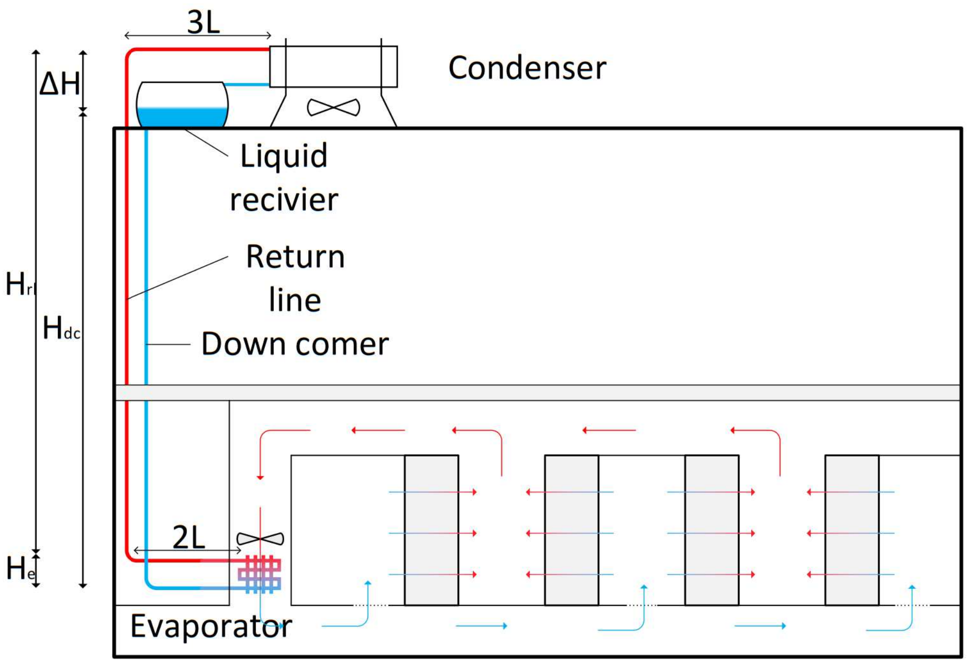
- Head pressure
  - Down comer
- Pressure drops
  - Down comer
  - Evaporator
  - Return line



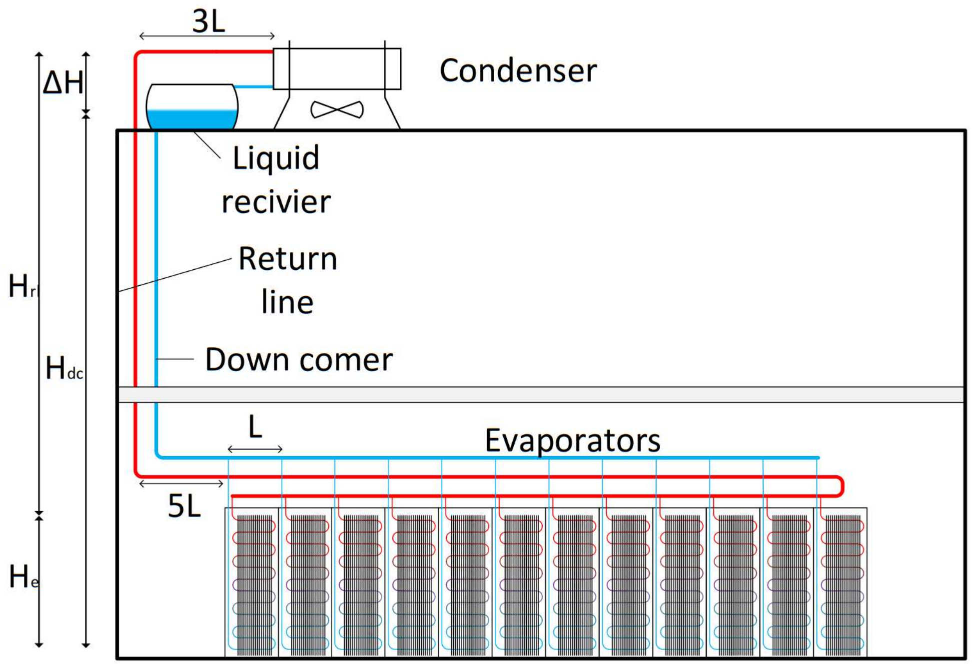
- Steady state - Head pressure balances the pressure drops

Schematic thermosiphon model

Central cooling unit

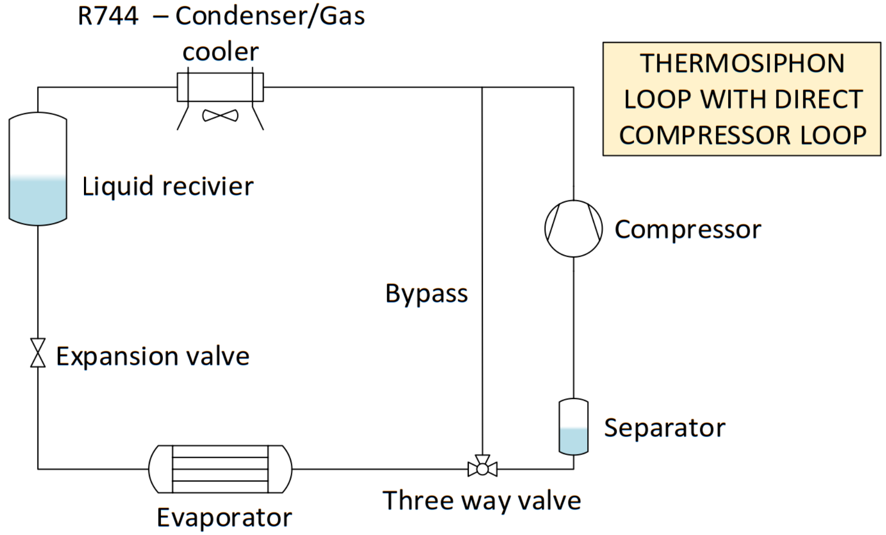


Local cooling units

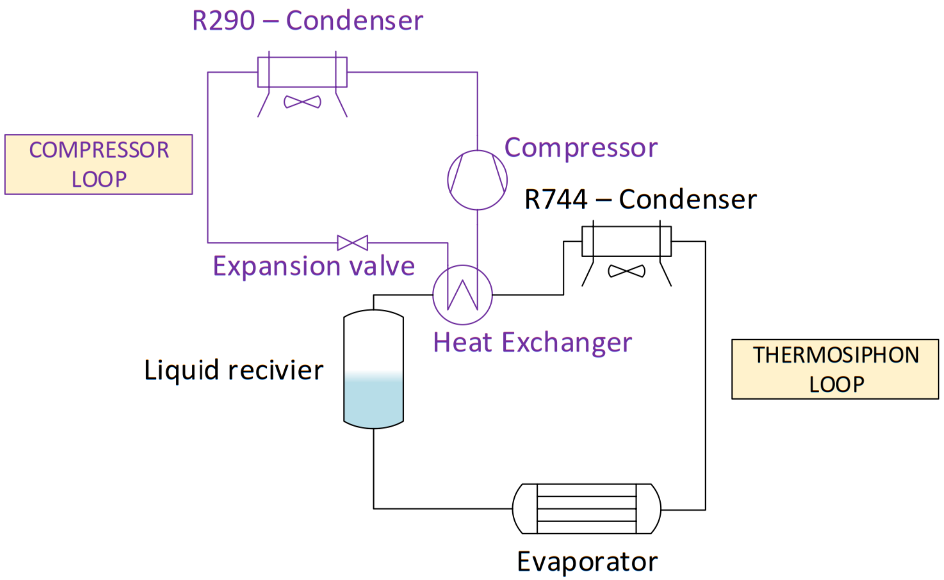


Auxiliary cooling

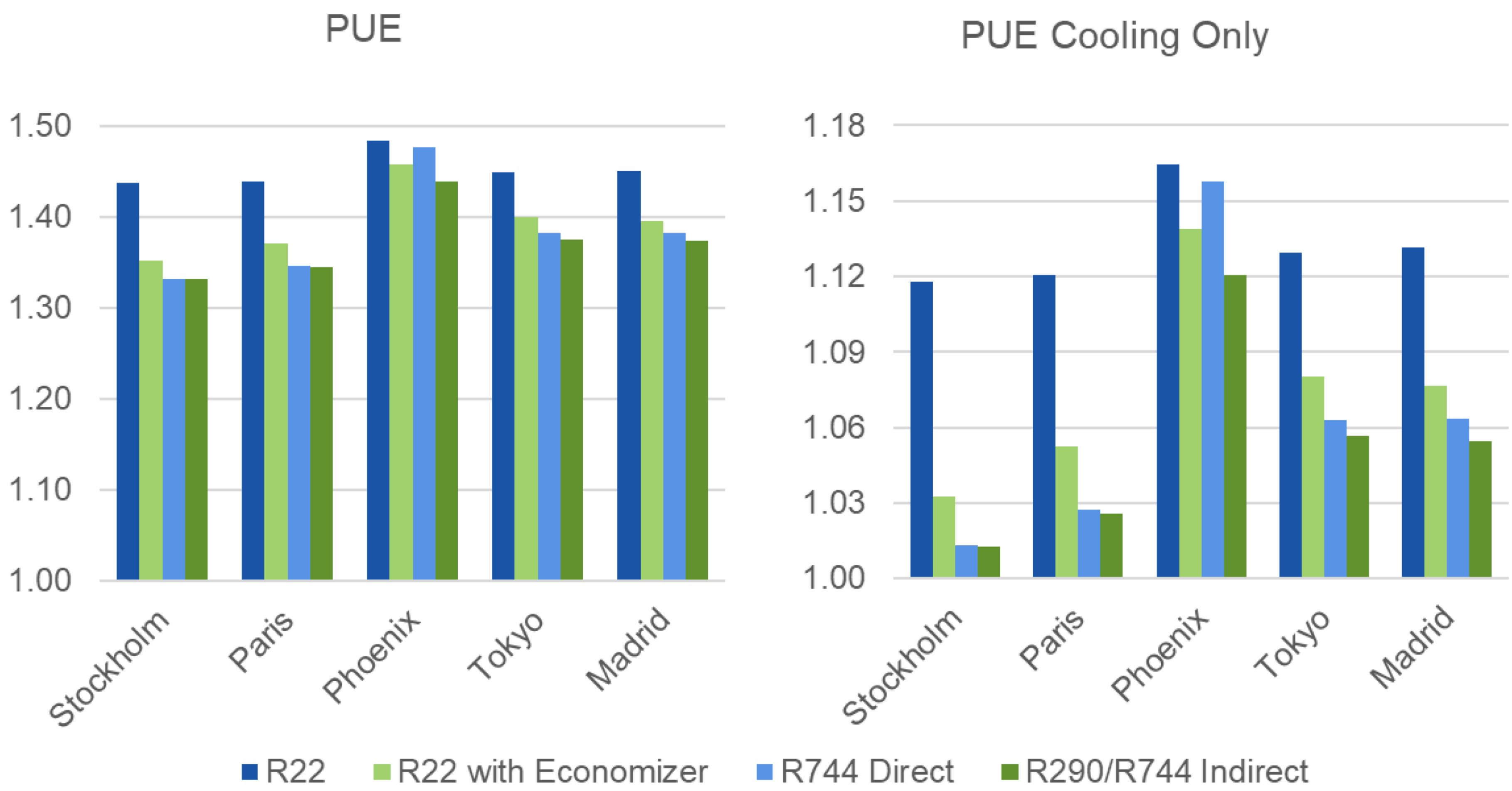
- Direct compressor cycle R744
    - Evaporation at 20˚C
    - Min PR 1.35



- Indirect compressor cycle R290/R744
    - Evaporation at 12.5˚C
    - Min PR 1.51



Results – PUE - energy usage



PUE - Power Usage Effectiveness

$$PUE = \frac{\text{Total energy}}{\text{IT energy}},$$

Economizer = ”free cooling”

Carnot Refrigeration - Aquilon

/Vendor profile/

From left: Marc-Andre Lesmerises, Ann-Sophie Hamel-Boisvert, and Thierry Vasseur, Carnot Refrigeration, at Data Center World

CO_2 FOR SERVER ROOMS

At Data Center World, Carnot's Marc-André Lesmerises explained the advantages of CO_2 cooling units for server rooms, including free cooling

– By Michael Garry

"We get free cooling 80%-85% of the time in Montreal."

Since its founding in 2008, Quebec-based Carnot Refrigeration has been one of the most versatile OEMs in the HVAC&R industry when it comes to marketing transcritical CO_2 refrigeration systems.

Carnot's greatest success has been as a provider of transcritical CO_2 systems to Sobeys, Canada's second largest food retailer, and the North American leader in transcritical installations, with more than 80 stores using the technology, including over 50 Carnot units. But Carnot has not stopped there, building transcritical units for industrial warehouses, ice rinks and data centers as well.

It was the data-center market that Carnot's founder, CEO and president, Marc-André Lesmerises, was targeting in March in a presentation at Data Center World in Las Vegas. The explosive growth of server farms and cloud computing in the IT and telecommunications sectors to support the digital economy makes this one of the most potential-rich marketplaces for advanced cooling systems like Carnot's

Thank you for your attention!

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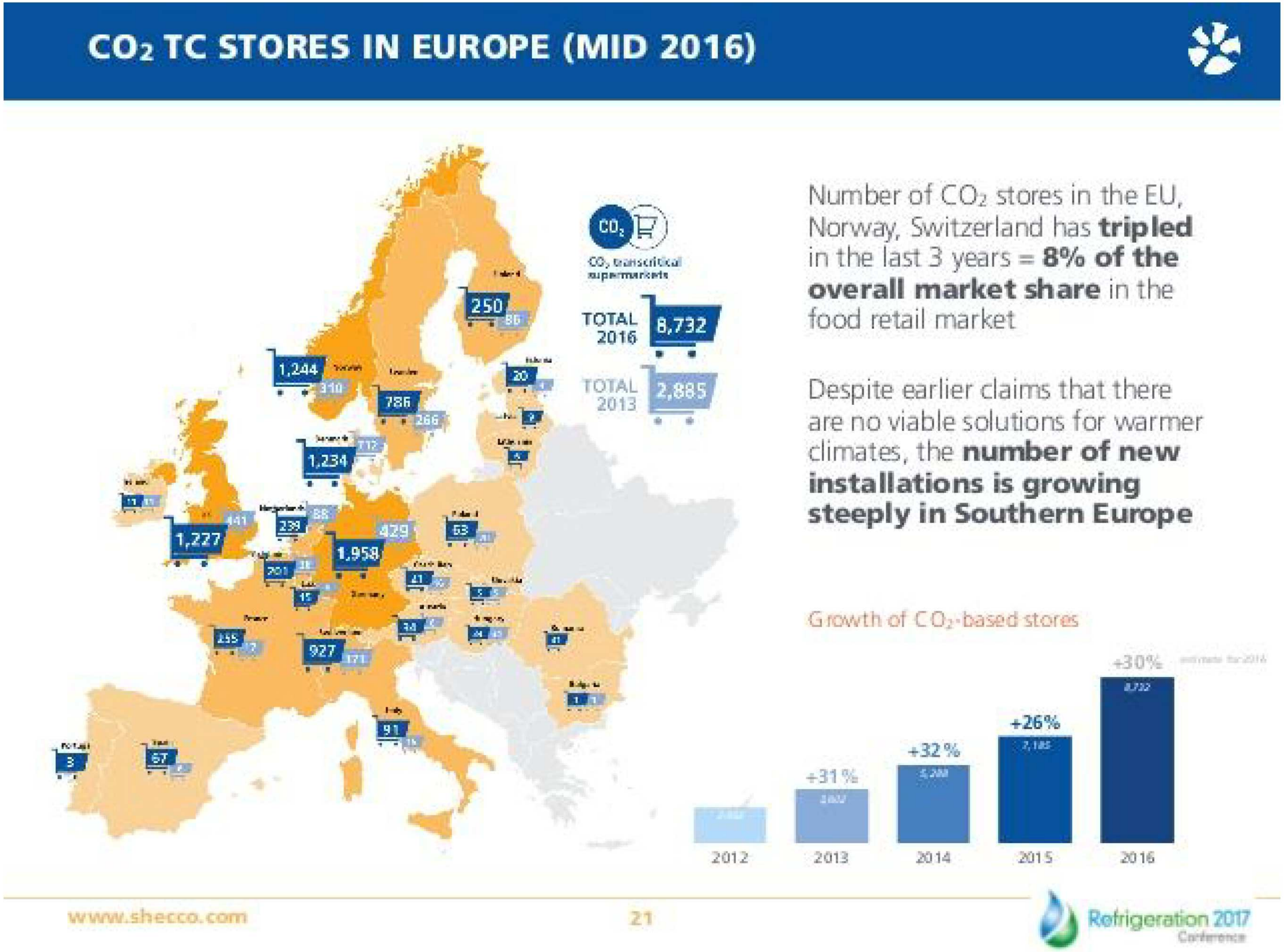
Jörgen Rogstam
(M. Sc./Eng. Lic.)
Managing Director

The CO2 market in Europe 2016



Referens: Shecco 2017

Natural Refrigerants in Data Center Cooling with Thermosiphon Application

ANDRÉ SAHLSTEN
VICTOR HEINERUD

KTH Industrial Engineering and Management

Master of Science Thesis
Stockholm, Sweden 2016

Natural Refrigerants in Data Center Cooling with Thermosiphon Application

André Sahlsten

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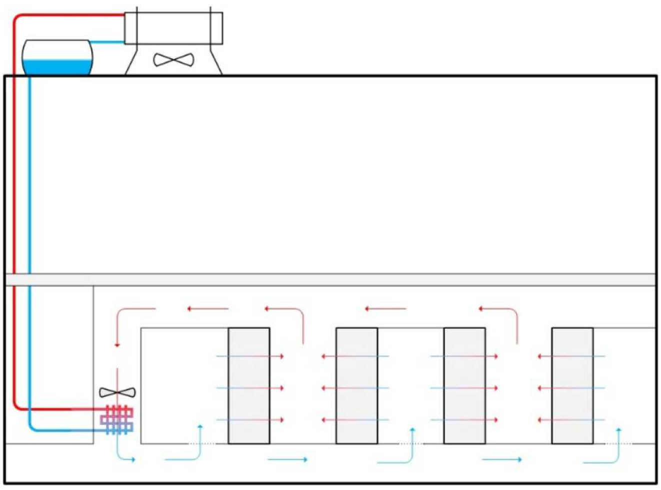

Master of Science Thesis
KTH School of Industrial Engineering and Management
Energy Technology EGI-2010- EGI_2016-042 MSC
Division of Applied Thermodynamics
SE-100 44 STOCKHOLM

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KTH Industrial Engineering and Management

Natural Refrigerants in Data Center Cooling with Thermosiphon Application

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Approved	Examiner	Supervisor
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Supervisor at industry	Commissioner	Contact person
Jörgen Rogstam		

Abstract

Ever since the computer was invented, there has been a need of data storage and the demand has strictly grown since. This has resulted in a huge amount of data centers and the trend has shown no signs of changing. The data centers are powered by electricity and in 2010 the electricity consumption for data centers stood for 1.3% of the world's electricity usage. The most energy consuming part of a data center is the servers themselves, but the second largest energy consuming part is the cooling system which, in a normal data center, stands for two fifths of the energy usage. Besides the energy consumption, the cooling systems are in most cases a cooling machine using HCFC and HFC refrigerants. These refrigerants are all bad for the environment since HCFCs have high ODP and GWP values and HFCs have high GWP values.

The purposes of this work is:

A) Find a way to make the cooling systems more efficient. Previous work has shown that using free cooling from the ambient air is an effective method of reducing the yearly electricity demand. Further the systems use a two-phase thermosiphon to move heat from the servers to the ambient, which means that there is no need of pumping power.
B) Find solutions using natural refrigerants that have no ODP and very low or zero GWP.
C) Evaluated if there is a possibility to recover the waste heat from the data center to e.g. an office building.

This work contains two systems being mathematically modeled with the software Engineering Equation Solver: a direct R744 system and an indirect system running with R290 and R744. Both systems are using a thermosiphon application, connected to a condenser, to use free cooling up to a certain set point temperature and the rest is covered with a vapor compression cycle. These systems are then matched to temperature profiles for five cities, Stockholm, Paris, Phoenix, Tokyo and Madrid, to see how many hours of the year are covered by free cooling. The systems are then evaluated considering both energy consumption and cost. To be able to compare these systems to a present cooling system, a reference system is modeled which uses R22 as refrigerant, that is the most commonly used refrigerant in the world today for the data center cooling application.

The results show that a direct R744 system or an indirect system with R290/R744 with a thermosiphon application have both energy and economical savings compared to the reference system. The energy savings are up to 88% and the total annual cost savings are up to 69%. The Power Usage Effectiveness is reduced with up to 6% and up to 8% if only cooling is considered. These savings are for an optimized condenser with a 2000 m2 fin area and 6 fans with a set point temperature of 22°C.

The indirect R290/R744 system is the best in all cities considering energy efficiency. Both systems are also well suited for use with heat recovery. The Seasonal Performance Factor for the heat recovery is between 8.3 and 15.2, which is a consequence of the high evaporation temperature and low supply temperature to the heating system.

Sammanfattning

Ända sedan datorn uppfanns har det funnits ett behov av datalagring, ett behov som ökat stadigt. Detta har resulterat i en stor mängd datacenter och det finns inget som tyder på att trenden kommer ändras. Datacenter drivs av el och under 2010 var elförbrukningen för datacenter 1.3% av världens totala elanvändning. Den mest energikrävande delen av ett datacenter är de faktiska servrarna och den näst största energikrävande delen är kylsystemet, vilket i ett normalt datacenter står för två femtedelar av energianvändningen. Förutom energiförbrukningen, är kylsystemen i de flesta fall, en kylmaskin med HCFC- och HFC-köldmedier. Dessa köldmedier är dåliga för miljön eftersom HCFC har högt ODP- och GWP-värden och HFC har höga GWP-värden.

Syftet med detta arbete är:

A) Hitta ett sätt att göra kylsystem effektivare. Tidigare arbeten har visat att användning av frikyla från den omgivande luften är en effektiv metod för att minska det årliga elbehovet. Det finns även system som använder en två-fas termosifon för att flytta värme från servrar till den omgivande luften, vilket innebär att det inte behövs några pumpar.
B) Hitta systemlösningar med naturliga köldmedier som har noll ODP och mycket låg eller noll GWP.
C) Utvärdera om det finns möjlighet att återvinna spillvärme från ett datacenter till exempelvis en kontorsbyggnad.

Detta arbete innehåller två system vilka modelleras matematiskt med hjälp av programvaran Engineering Equation Solver: ett direkt R744-system och ett indirekt system som använder R290 och R744. Båda systemen använder en termosifonslinga som är ansluten till en kondensor för att kunna använda frikyla upp till en viss brytpunktstemperatur och det resterande behovet täcks av en kylmaskin. Dessa system matchas sedan mot temperaturprofiler för fem städer, Stockholm, Paris, Phoenix, Tokyo och Madrid, för att se hur många timmar av året som frikylakan användas. Systemen utvärderas sedan utifrån både energiförbrukning och kostnad. För att kunna jämföra dessa system mot ett befintligt kylsystem modelleras ett referenssystem med R22 som kylmedel, vilket är det vanligaste köldmediet i världen idag för kylning av datacenter.

Resultaten visar att ett direkt R744-system eller ett indirekt system med R290/R744, båda med en termosifonslinga, är både energieffektivare och ekonomiskt fördelaktigare jämfört med referenssystemet. Energibesparingen uppgår till 88% och de totala årliga kostnadsbesparingarna uppgår till 69%. Power Usage Effectiveness värdet reduceras med upp till 6% och om enbart hänsyn tas till nedkylning, upp till 8%. Dessa besparingar är för en optimerad kondensor med en flänsyta på 2000 m^2 samt 6 stycken fläktar då kondensatorn har en brytpunktstemperatur på 22° C.

Det indirekta R290/R744-systemet är det bästa i alla städer vad gäller energieffektivitet. Båda systemen är också väl lämpade för användning med värmeåtervinning. Årsvärmefaktorn för värmeåtervinningen är mellan 8.3 och 15.2, vilket är en följd av den höga förångningstemperaturen och den låga framledningstemperaturen till värmesystemet.

Acknowledgements

This report is the result of a master thesis project during 20 weeks in the spring 2016 at Kungliga Tekniska Högskolan in cooperation with the company Energi och Kylanalys AB (EKA). The work is the last part of the civil engineering program at Kungliga Tekniska Högskolan.

We want to give a special thanks to our company supervisor Jörgen Rogstam from Energi och Kylanalys that helped us a lot through our difficulties and getting us on the right path in the work. Also Simon Bolteau at EKA has brought good inputs to our work.

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Stockholm 20th June 2016

André Sahlsten Victor Heinerud

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Nomenclature

A = Area

a = Annuity factor

c_p = Specific heat

D = Diameter

$\dot{E}$ = Electrical power

f_F = Fanning friction factor

Fr = Froude number

g = Gravitational acceleration

h = Height

I = Investment cost

L = Length

$\dot{m}$ = Mass flow

p = Pressure

$\dot{Q}$ = Heat

Re = Reynolds number

s = Entropy

T = Temperature (dry bulb)

t = Temperature (dry bulb)

u = Velocity

U = Overall heat transfer coefficient

$\dot{V}$ = Volume flow

We = Weber number

x = Vapor quality

z = Height

Greek

α = Fraction

η = Efficiency

μ = Dynamic viscosity

ρ = Density

τ = Time

ϕ = Relative humidity

Φ = Phase multiplier

Subscripts

a = Acceleration

amb = Ambient

b = Bend

c = Condenser

dc = Down comer

e = Evaporator

el = Electrical

f = Friction

g = Gas

HR = Heat Recovery

k = Compressor

l = Liquid

m = Mean

rt = Rising tube

tot = Total

TP = Two-Phase

wb = Wet bulb

Abbreviations

CFC – Chlorofluorocarbon

CO_2 – Carbon dioxide

COP – Coefficient of Performance

CRAC – Computer Room Air Condition unit

CRAH – Computer Room Air Handler

ERE – Energy Reuse Effectiveness

GWP – Global Warming Potential

HCFC – Hydrochlorofluorocarbon

HFC - Hydrofluorocarbon

HFO – Hydrofluoroolefin

HVAC – Heating, ventilating and air-conditioning

HX – Heat exchanger

LEL – Lower Explosion Limit

LMTD – Logarithmic Mean Temperature Difference

MAC – Mobile Air Conditioning

ODP – Ozone Depletion Potential

PDU – Power Distribution Unit

PUE – Power Usage Effectiveness

R12 – Dichloridfluoromethane

R22 – Chloridfluoromethane

R290 – Propane

R717 – Ammonia

R718 – Water

R744 – Carbon dioxide

SPF – Seasonal Performance Factor

UPS – Uninterruptable Power Supply

1 Introduction

Ever since the beginning of computers the need of a location to hold the physical data storage has been of major concern. Today's widely usage of computers and their mutually communication, i.e. the internet, along with smart phones and other "smart" devices results in the need for a huge amount of servers. In addition, the increase in cloud-based storage also contributes to the need for servers as storage units (Thoresson 2011).

Not only produces these servers heat that needs to be cooled off, the cooling itself consumes energy. In 2010 data centers stood for 1.3% of the world total electricity consumption which corresponds to 235.5 TWh (Renzenbrink 2011). The consumed energy is estimated to increase in accordance with Figure 1 to 1,400 TWh in 2025 (Fishman 2014).

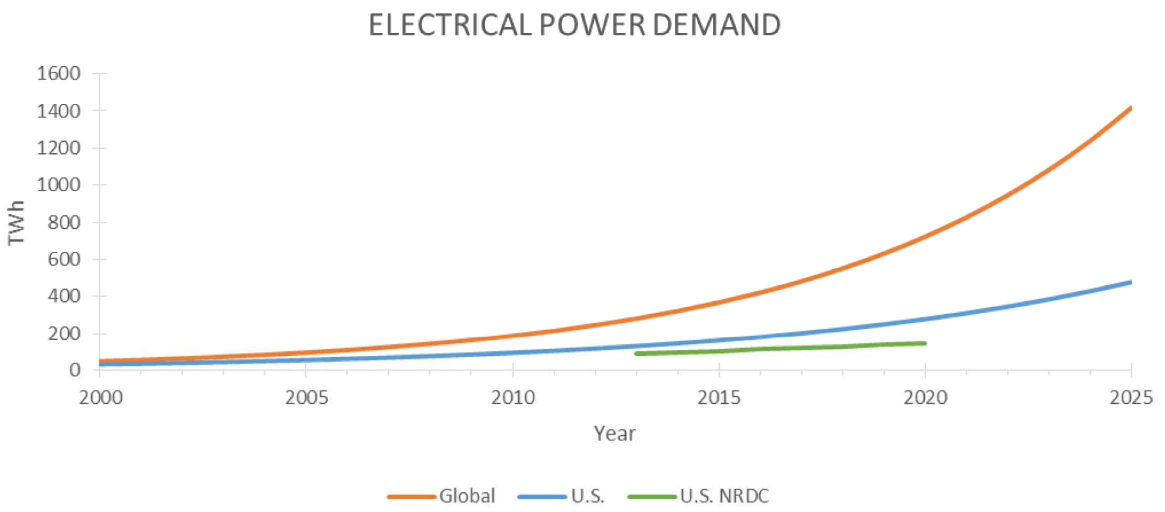

Figure 1 *- Estimated increase in electrical power demand from 2000 to 2025*

According to U.S. Natural Resources Defense Council (NRDC) the nation's data centers consumed 91 TWh of electricity in 2013 which is twice the electricity demand for households in New York City (Whitney and Delforge 2014). NRDC's estimation for the electricity consumption in 2020 is only half of estimation done by Fishman. Although an increase is to expect, the magnitude is harder to predict. A compilation done in an article in *Computerworld* (Table 1) shows the estimated increase in electricity consumption, according to NRDC, and the consequential increases in electrical bills, amount of needed power plants and CO_2-equvivalent emissions (Thibodeau 2014).

Table 1 - Comparison between 2013 and estimations done for 2020 in data center electrical consumption, electrical bills, power plants to supply the demand and CO_2-eqvivalent emissions

Year	**Electrical consumption (TWh)**	**Electrical bills ($B)**	**Power plants (at 500 MW each)**	**CO_2-eqvivalent emissions (metric tons)**
2013	91	$9.0	34	97
2020	139	$13.7	51	147
Increase	47	$4.7	17	50

As seen data centers consumes a lot of electrical energy. For a typical data center, the electricity consumed by the servers is barely half of the total electricity consumed by the whole center. The electricity used only for cooling stands for about two fifths of the total consumption (Rasmussen 2011). One way to lower the electricity demand for data centers is to use the surroundings with lower temperature for cooling purposes. This is called "free cooling" (Potts 2011). This implies, in order for heat transferring, that the temperature of the surroundings needs to be lower than the exhaust from the server room.

Another factor that is impacting the electricity consumption is the allowed temperatures in data centers. The current recommended range, which has been increased during the past, lies between 18°C and 27°C with an allowable range spanning 15°C to 32°C, but in some sort of applications this temperature range even extends from 5°C to 45°C (Strutt 2012).

The traditional configuration of data center cooling is to simply circulate cooled air through the servers and a *Computer Room Air Condition unit* (CRAC) or *Computer Room Air Handler* (CRAH) (Sasser 2014). These central cooling unit systems is cooled in several ways e.g. by air, water, brine or refrigerant (Evans, The Different Technologies for Cooling Data Centers 2012).

Currently most of the data centers are cooled by refrigerants from the *hydrofluorocarbon* and *hydrochlorofluorocarbon* family (HFC and HCFC) (Evans, Fundamental principles of Air Conditioners for Information Technology 2014). HCFC-refrigerants, along with *chlorofluorocarbons* (CFC), are being phased put since 1987 as a result of the Montreal-protocol (Nationalencyklopedin 2016) because of their negative impact on the ozone layer (ozone depletion) (Evans, Fundamental principles of Air Conditioners for Information Technology 2014). As a substitute for CFCs and HCFCs the HFCs, also known as F-gases, were developed.
Figure 2 further explains the development of refrigerants starting from the use of CFCs with examples of refrigerants and application areas for the different types of refrigerants.

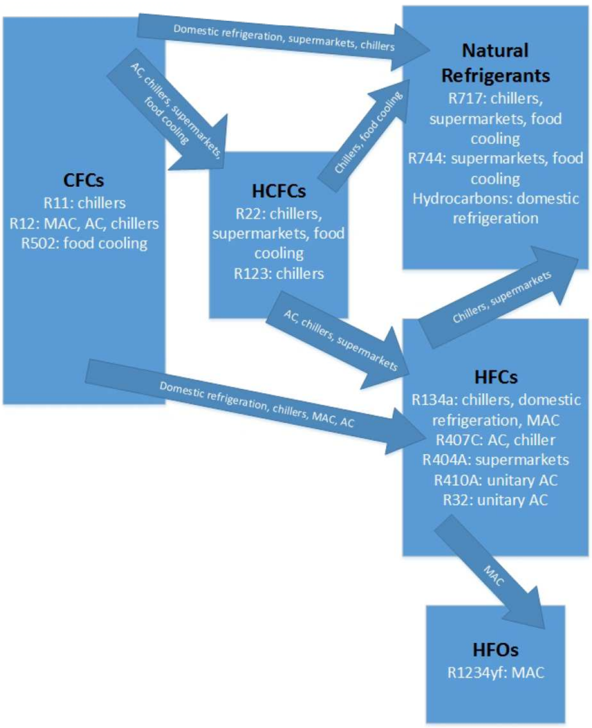

Figure 2 *- Development of refrigerants since the use of CFCs*

The downside of HFCs are that they are greenhouse gases, i.e. high GWP, and in 2014 the European Union adopted a new *F-gas regulation* to reduce the use of HFCs. The regulation started the 1st of January 2015 and the outcome is hoped to be that the F-gas emissions in 2030 is one-third of 2014's levels (European Union 2016).
This development of refrigerants can be divided into four generations. Figure 3shows a retrospect view of the use of refrigerants since 1830 (Makhnatch, Refrigerants and refrigerant mixtures 2015).

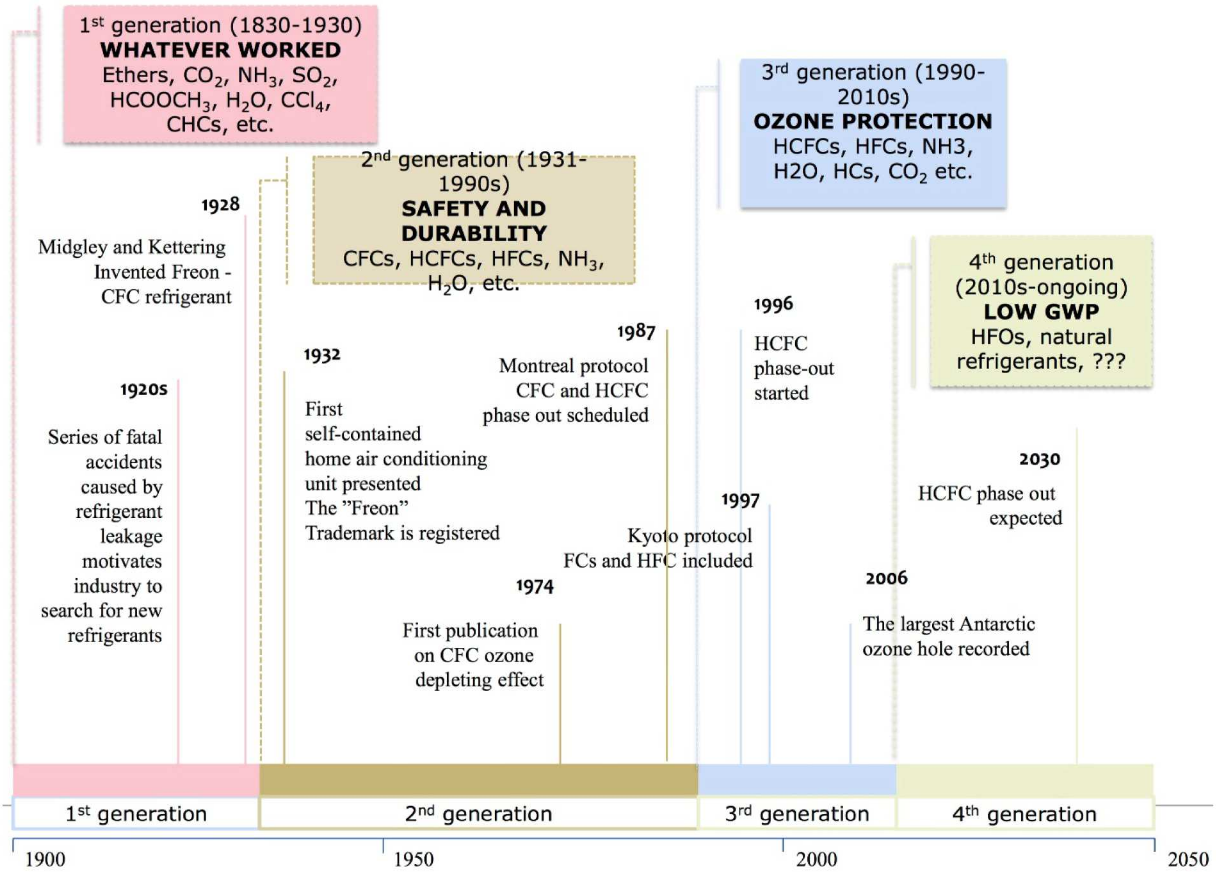

Figure 3 *- Development of refrigerants since 1830 with future expectations*

Although the *F-gas regulation* only concerns the European Union. Other parts of the world are working for phasing out HFCs as well (Makhnatch, Khodabandeh and Palm, Utvecklingen på köldmediefronten under året som gått 2015) (Doniger 2011). The phase out schedule for HCFCs and HFCs is presented in Figure 4 (Danfoss 2014).

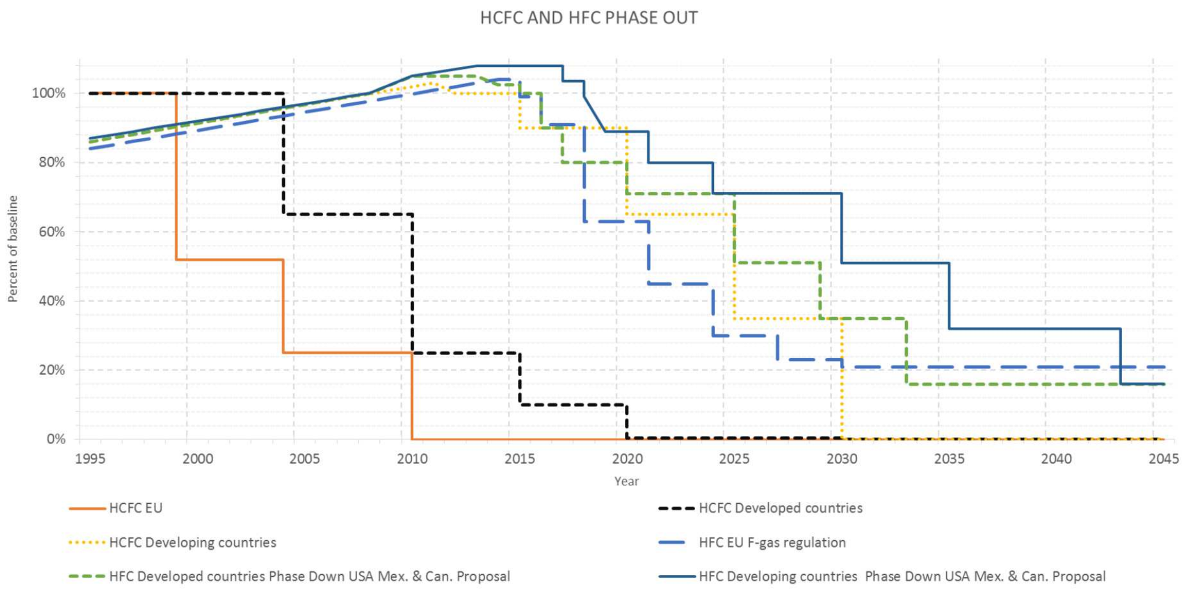

***Figure 4** - Phase out schedules for HCFCs and HFCs according to EU's F-gas regulation and USA Mex. and Can.-proposal*

There are several alternatives for HCFC's and HFC's, both natural and synthetic, with low GWP (Global warming potential) and ODP (Ozone depletion potential). They all have different properties which makes them suitable for different applications. Natural refrigerants which may be suitable for data center cooling application are R717, R744, R718 and hydrocarbons (Department of Energy Technology KTH 2013). Synthetic refrigerants, hydrofluoroolefins (HFO), with low ODP and GWP could also be of use but with respect to that they, as well as some natural refrigerants, are considered as flammable (Department of Energy Technology KTH 2015). Further are these refrigerants a mixture of the same components used in HFC's but in an unsaturated state, i.e. the molecule holds at least one double binding, which make them more reactive (Tomczyk 2014).

Finally, the heat removed from the data centers must be released somewhere. Often the heat is released to the ambient, such as the outside air, the ground or water (i.e. lake, sea or bigger watercourses) (Sasser 2014). For bigger data center there is a considerable amount of dissipated heat. To improve energy efficiency this (waste) heat could instead be used for heating purposes in adjacent areas, known as heat recovery (Data Center Knowledge 2016)

1.1 Objectives

The underlying objectives are:

- How to replace HCFCs and HFCs as refrigerants in data center cooling applications?
- How to make the data center cooling systems more energy efficient?

Concretized this is investigated through:

- Which is the most suitable natural refrigerant for substituting HCFC and HFC?
- Is it possible to have a solution using a thermosiphon i.e. eliminate the need of pump power?
- How do these systems perform worldwide?
- Is it possible to recover the exhaust heat?

1.2 Methodology

This work is purely theoretical and is a comparison between different solutions which operates in similar conditions. The results give the systems mutual relationship although the absolute values do not necessarily reflect reality.

First off, a literature review is conducted giving information of what technology that currently is used for cooling data centers. This serves as basis for developing new cooling systems. During the literature review key parameters are identified as these later is used for measuring the performance of the developed systems.

To measure the system performance in different climates, climate profiles are derived from data from *Meteonorm* for five cities around the world. The systems total energy demand is calculated from the power demand for a certain temperature and the amount of hours for that temperature during one year.

The work is focusing on two main parts, one calculating thermosiphon: balancing the head pressure to the pressure drops. The second part is the energy demand and economical calculations to find the threshold for switching operation mode. The key object in the second part is the condenser since its size impacts both prize and possible temperature difference between refrigerant and ambient.

The model is primary mathematical where schematic system structure models are used in finding the necessary calculating points. Most of the equations is taken from the literature but, where there is needed, necessary equations are derived from given data. The main calculating is conducted with *Microsoft Excel* and *Engineering Equation Solver* (*EES*) with support from *IMST-ART* and *AlfaSelect Air* for minor parts. The post-processing and refining of the results is done with *Microsoft Excel.*

The report is written in *Microsoft Word* using *Math Type* for equations.

1.3 Scope and Limitations

This study is focused on air-cooled medium sized data centers, about 200 kW and 12 server racks, with constant cooling demand which primary purposes are (data) storage and communication, i.e. low energy density. Consequently, only air-cooling refrigeration is considered. Further the study is investigating how to maximize the use of free cooling in combination with an environmental friendly refrigerant that also is not harmful nor causing any hazards to the IT-systems.

In a conventional cooling system there are three major electricity consuming devices (Petschke 2008). This study main focus is on reducing the electricity demand for compressors and circulation pumps, leaving the air circulating fans as a necessary evil. Electricity consumptions of auxiliary equipment are left untreated.

The systems are seen as always running at full capacity and thereby are no efficiency measurement for partial loads conducted.

2 Data Center Cooling

A data center is usually a room where server racks are aligned in rows, as in Figure 5. Depending on the size of the data center the amount of servers and rows differs.

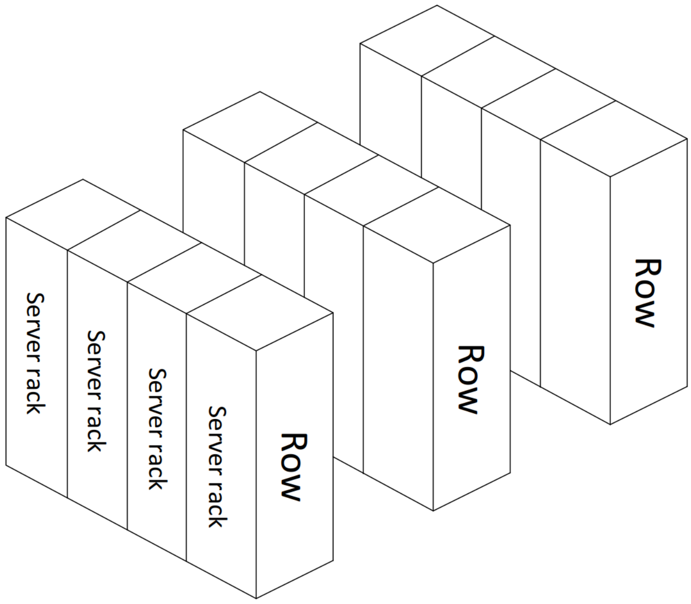

Figure 5 - *Schematic drawing of server rack rows*

Data center performance is typically measured in their *Power Usage Effectiveness*, PUE-value, which is a quantity telling how much auxiliary energy, additional energy than required by the servers, that is needed to operate the data center. The distribution of electrical energy consumption in a typical data center is shown in Figure 6 (Rasmussen 2011).

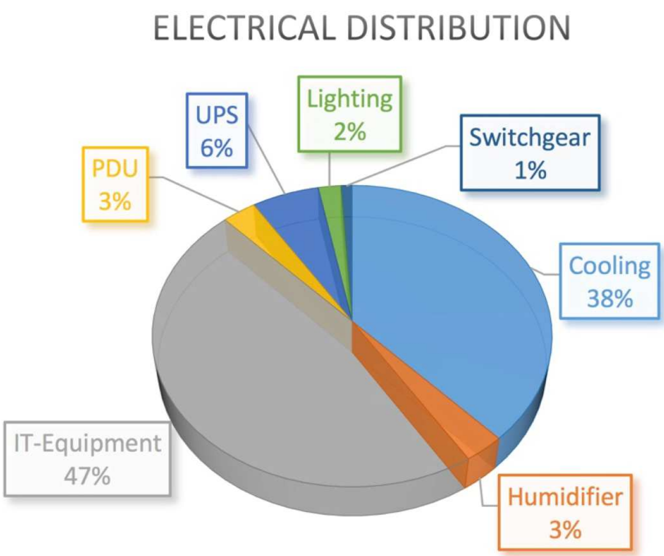

Figure 6 - *Energy distribution in a typical data center*

As seen in Figure 6 the second largest electricity consuming part is cooling, standing for 38% of the total demand.

The many types of present data centers in the world there are also many different cooling technologies. There are three standard principles of technologies that are the most common setups today (Evans, The

Different Technologies for Cooling Data Centers 2012), but also a few new highly developed systems that are newer to the market. Figure 7 shows an overview of the three technologies.

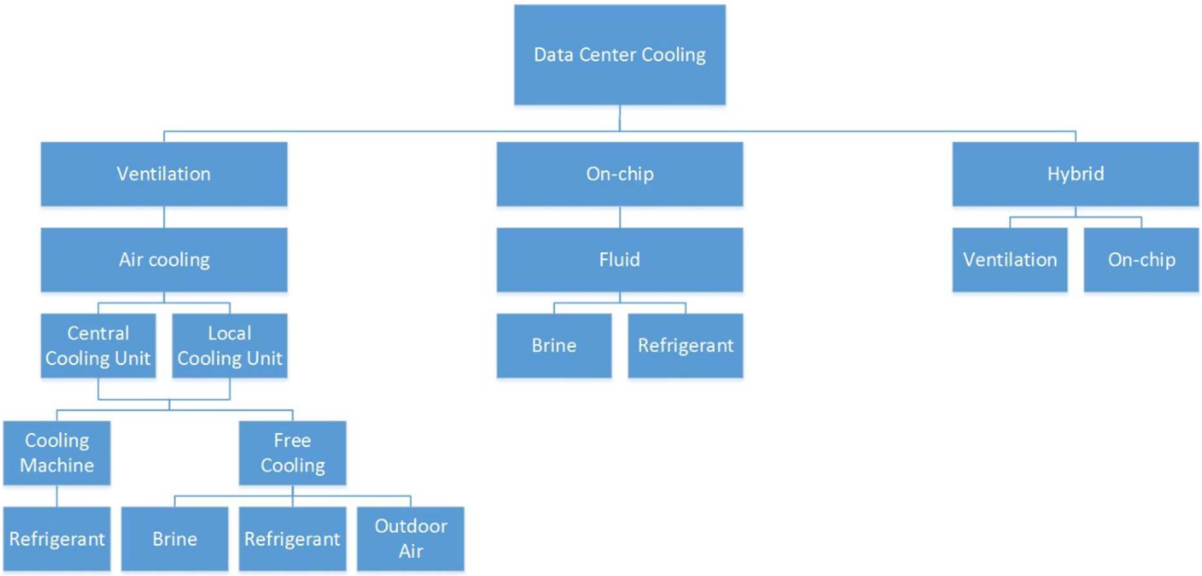

***Figure 7** - Overview of data center cooling techniques*

Data center cooling is divided into three branches with different types of cooling where either the data centers is cooled by ventilating the air, direct cooling on the electronics or a combination of them both. The choice of technique is dependent on the power density; ventilation cooling is suitable for low power densities whereas on-chip cooling is suitable for high power densities. The most used technique is the ventilation cooling method which in turn is divided into central cooling, one unit handling all the air in the room, or local cooling, one unit for each server rack. For both central and local cooling, the air is cooled by a fluid from either a cooling machine or free cooling (2.3.5) (Evans, The Different Technologies for Cooling Data Centers 2012)..

2.1 Power Density

Depending on the purpose of the servers they have different power densities. The power density also varies among the electronic components in the server. The most heat generating components in the servers, i.e. processors and memory sticks, has its own built-in cooling devices, e.g. heat sinks (Intel 2009). Studies has shown that adding extra cooling directly on these components, called on-chip cooling, decreases the electricity consumption while maintaining the cooling capacity (Sorell, et al. 2015). The threshold for when to use on-chip cooling is for components dissipating at least 1 kW/cm^2 (Ebrahimi, Jones and Fleischer 2014).

Figure 8 shows the distribution of consumed electricity in server racks in 2014 and the assumed values two years later (Data Center Users' Group 2014).

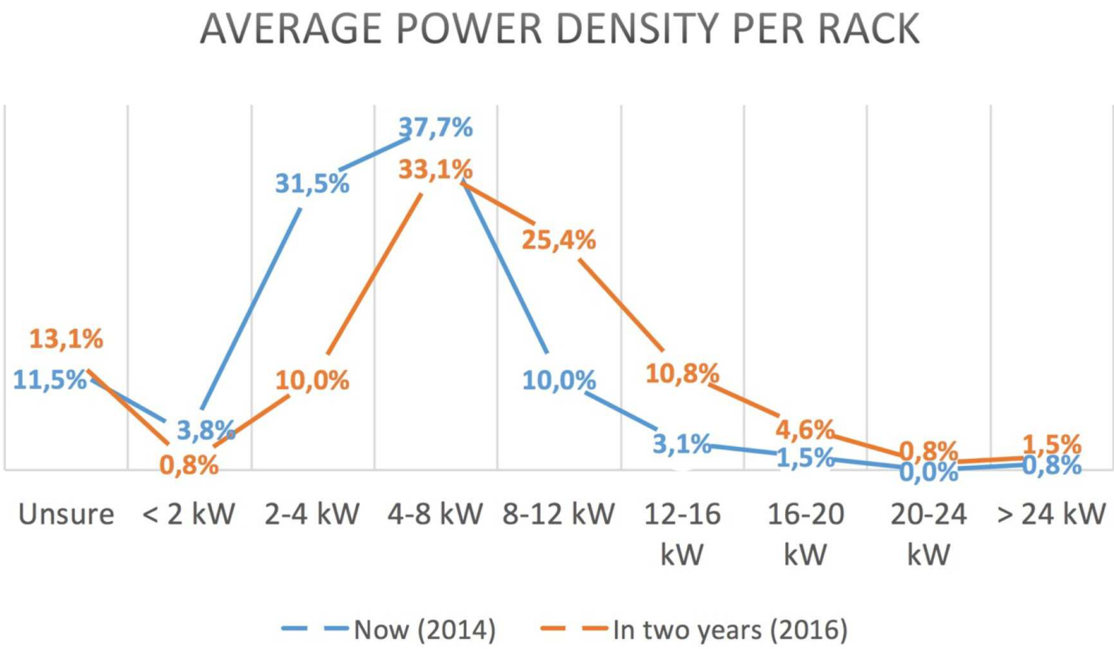

Figure 8 - *Average electrical power consumed by a server rack*

As seen, more than two-thirds of the server racks consumes less than 8 kW on an average basis. If the peak load distribution, Figure 9 (Data Center Users' Group 2014), also is taken into consideration, three-fifths of the electricity consumption never exceeds 16 kW and on-chip cooling would not be necessary for server racks where the area of the high heat generating components exceeds 16 cm². The increase of rack power density in Figure 8 and Figure 9 reinforces the prediction of increased power demand.

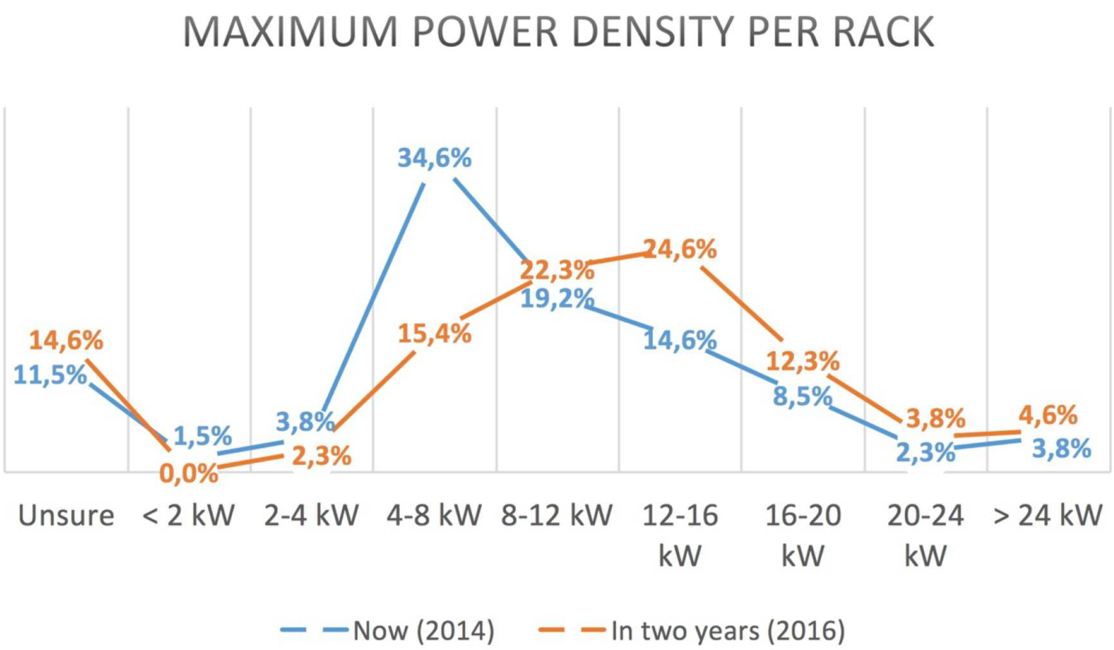

Figure 9 - *Maximum electrical power consumed by a server rack*

2.2 System Efficiency Comparison

To compare data centers energy consumption, the *Power Usage Effectiveness* (PUE), is often used, which is defined as

$$\text{PUE}=\frac{\text{Total energy}}{\text{IT energy}}, \tag{1}$$

where the total energy is the energy used for IT-equipment, humidifier, cooling, switchgear, lighting, UPS (Uninterruptable Power Supply) and PDU (Power Distribution Unit) and IT energy is the energy for the IT-equipment. In this work the energy demands are divided into three groups: IT-equipment, cooling and auxiliary equipment which contains things as UPS, PDU, lighting etc.

The PUE-value does not consider any recovered energy such as heat recovery. Therefore, the *Energy Reuse Effectiveness* (ERE) also is used. It is possible for a system can have a high PUE but still have a low ERE if e.g. much heat is recovered. The ERE is defined as

$$\text{ERE}=\frac{\text{Total energy - Recovered energy}}{\text{IT energy}}. \tag{2}$$

The ERE-value gives a truer measurement of the whole energy efficiency, but both methods is useful in system comparison (Grid 2011).

2.3 Current Cooling Technologies

There are both old and inefficient systems as well as there are new systems with better energy efficiency on the market right now. Regardless of system configuration, all cooling units in a data center need a cool medium which in turn cools the data center. Either this is done by a cooling machine or by free cooling (2.3.5) (Evans, The Different Technologies for Cooling Data Centers 2012).

2.3.1 Current Efficiencies

To find possibilities to reduce the energy consumption, and also the PUE and ERE values, the energy usage in at a typical present data center serves as a basis.

According to (1) the PUE for the system in Figure 6 is 2.1, which means that there are more energy used for subsidiary purposes than running the IT equipment. The market average for PUE-values are 1.5 and the systems with the currently lowest has PUE-values of 1.07-1.10 (Alfa Laval 2015) (Carnot Refrigeration 2016) (e3computing 2015).

As the heat is released to the ambient, the temperature level that the heat is released at, is dependent on the ambient temperature. This affects the PUE and the efficiency is therefore dependent on where in the world the data center is placed. The change in PUE-value as of ambient temperature is shown for a typical data center in Figure 10 (Rasmussen 2011) with the assumption of an IT-load at 30 % of full capacity.

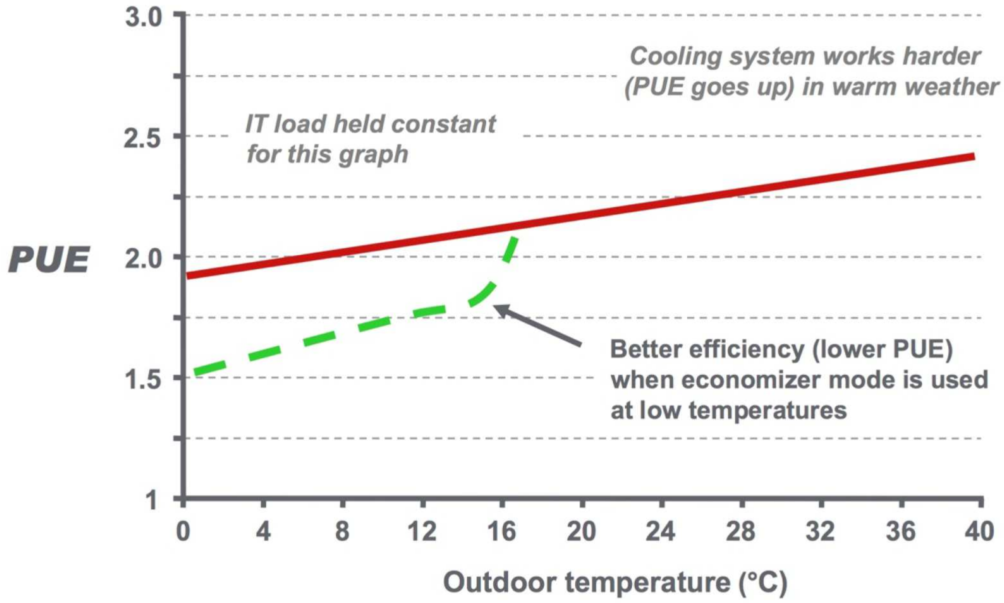

Figure 10 - The PUE as function of the outdoor temperature

The green dotted line represents a system using economizer mode which is a system with free cooling (further explained in chapter 2.3.5). The PUE is also depending on the IT-load, which for a typical data center is shown in Figure 11 (Rasmussen 2011).

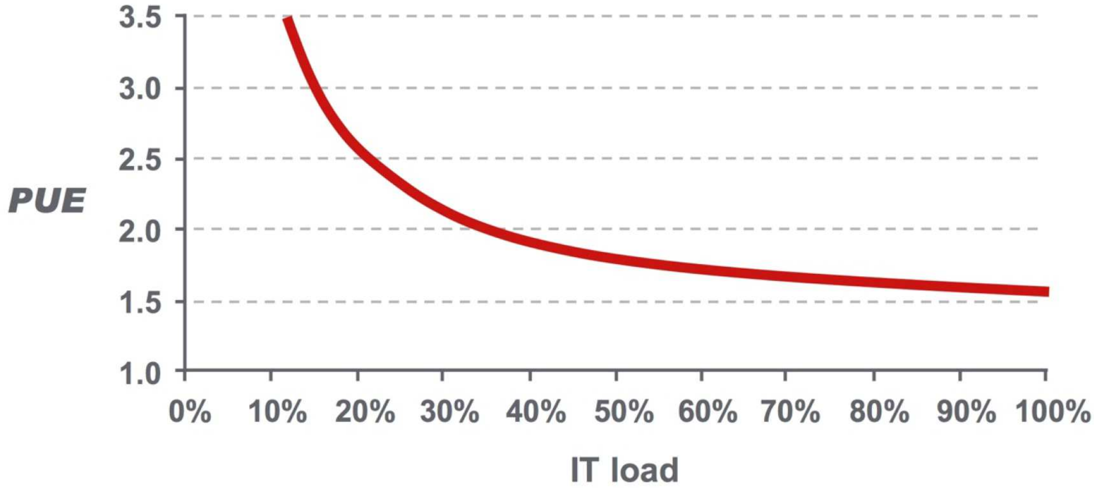

Figure 11 - PUE as function of the IT Load

According to Figure 11 it is important to dimension the system right to have an efficient data center. If the IT-load often is low the other parts of the system will have too high capacity and work inefficient (Rasmussen 2011). In this work the IT-load is assumed to be constant at 100%.

2.3.2 Central Cooling Unit

Central cooling configuration is the most common setup in older data centers is a system. The air is cooled in a *Computer Room Air Conditioner* (CRAC) or *Computer Room Air Handler* (CRAH) and then led through a raised floor and further through the server racks as in Figure 12 (Sasser 2014) .

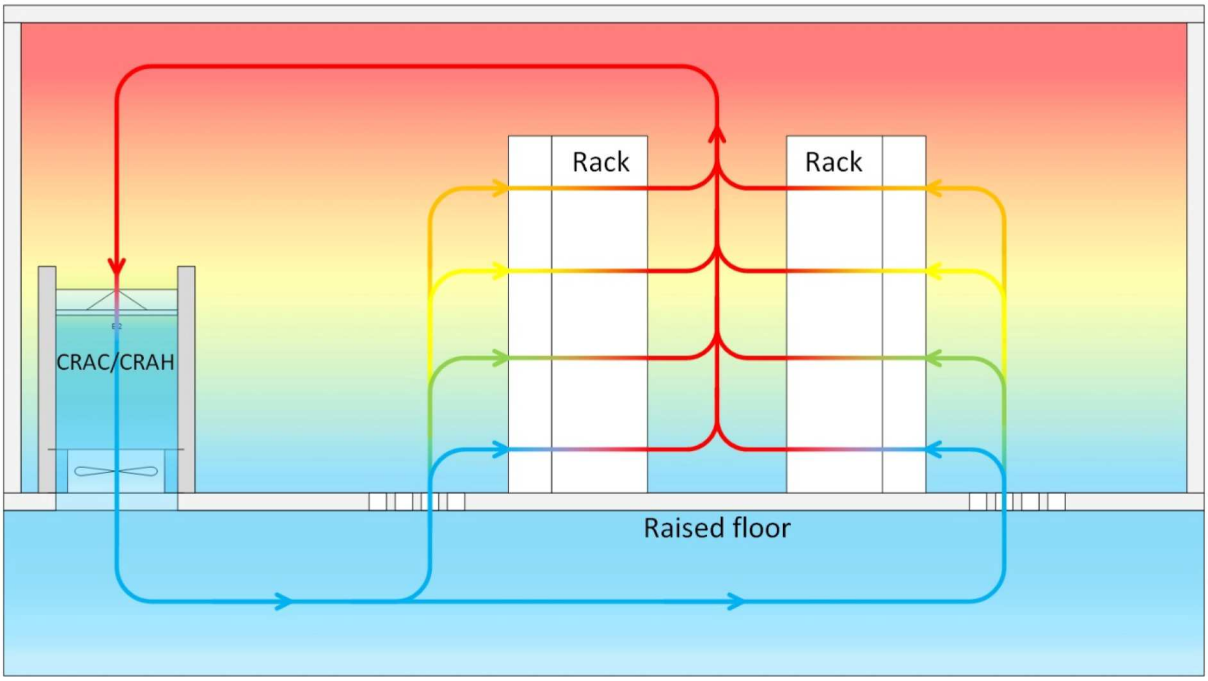

Figure 12 - Old system design for cooling

The downside of this system, as the temperature scale in Figure 12 shows, is that the uneven temperature distribution resulting in that the servers placed in the top section of the rack is not getting as cold air as the bottom section. This means that this system needs more cooling capacity to keep the right temperature in the top section resulting in a more power consuming, and thus inefficient, system. A solution to this problem is to make two separated aisles, one cool and one hot. This system layout is shown in Figure 13 (Sasser 2014).

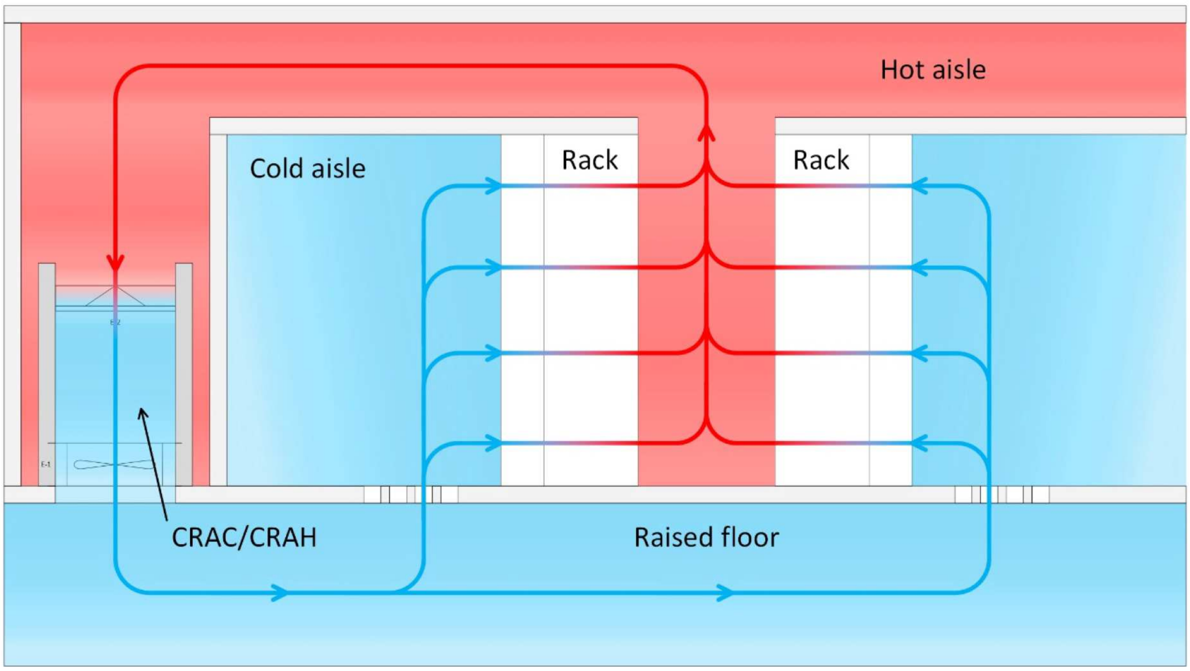

Figure 13 - Separated aisles with cool and hot air

As seen in Figure 13 this system has a more even distributed temperature over the rack as the hot and cold air never mixes.

2.3.3 Local Cooling Unit

In addition to use central cooling for cooling the server room, several parallel evaporators is mounted on the rear doors of the server racks as in Figure 14, referred to as local cooling. This reduces the needed amount of air flow in the server room (Geoffrey and Bell 2010). The air flow in Figure 14 is created either by separate fans or fans on the evaporators.

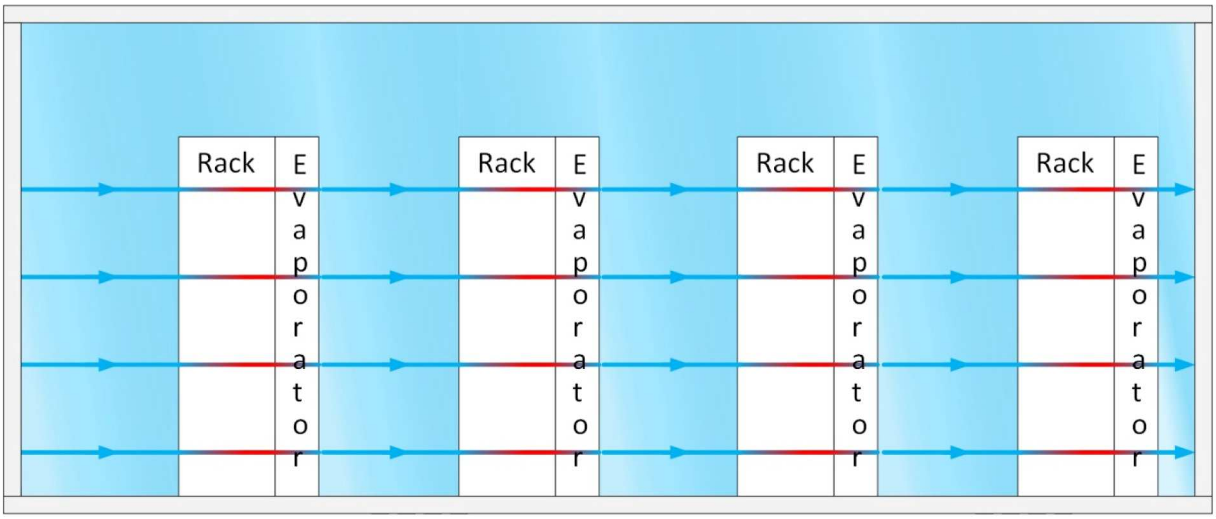

Figure 14 - Server room with local cooling

2.3.4 Refrigerants in Air Cooling Units

Most of the refrigeration machines uses a nontoxic nor flammable refrigerant but which often is harmful to the environment, if released into the atmosphere, due to their high GWP-value (The Linde Group 2016). Older central cooling unit chillers in data centers have a leakage rate of 10-15% while newer chillers leak about 1% of their charge (Alger 2009). In data center cooling applications, the most common refrigerant is R22 (HCFC) but in older applications even R12 (CFC) is used. Both refrigerants are banned from further manufacturing and are replaced by R134a (HFC) in the near future (Evans, Fundamental principles of Air Conditioners for Information Technology 2014). Neither R134a is a sustainable solution since it is a HFC and actions are taken to replace it too. These refrigerants are used worldwide in other applications than data center cooling and the share used in data centers are not revealed in the literature. Figure 15 shows the total yearly production of R12, R22 and R134a between 1980 and 2007 (AFEAS 2016).

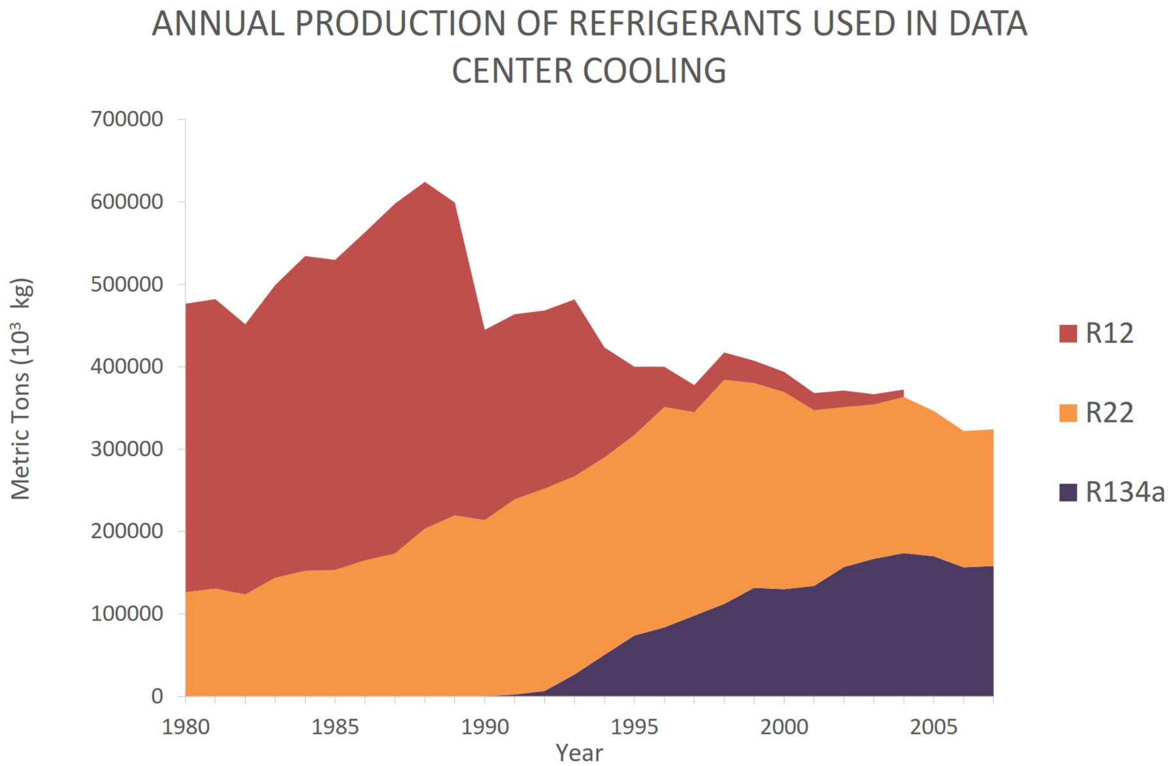

Figure 15 - *Amount of production for refrigerants used in data center cooling*

The production of R12 has drastically decreased since the late 80's due to the Montreal Protocol, to definitely end in 2004. From Figure 15 it also seen that, in 2007, the production of R22 and R134a are still several 100,000 of metric tons.

2.3.5 Free Cooling

To reduce the amount of mechanical work, natural cool surroundings e.g. outdoor air, the ground or a water source can be used, referred to as free cooling. A simplified system layout for free cooling is shown in Figure 16.

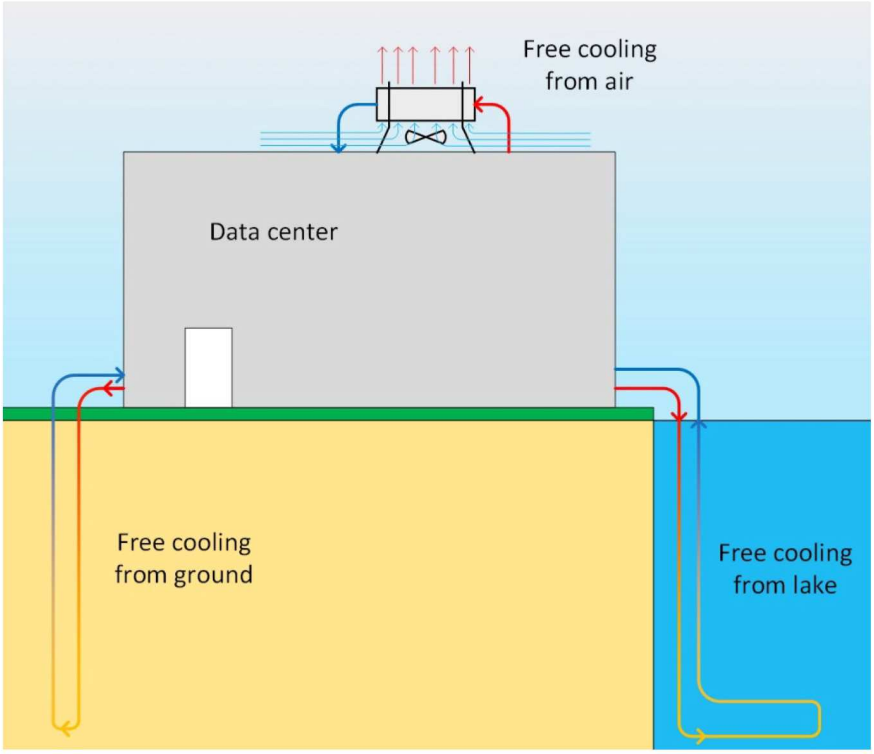

Figure 16 *- Free cooling utilized from air, ground and water source*

Sometimes the term economizer mode is used which is an operation mode synonymous to free cooling (Niemann, Bean and Avelar 2011). There are several different setups to utilize free cooling and it can be divided into two branches: air economizer mode and water economizer mode. Air economizer mode uses (cool) outdoor air directly into the server environment compared to water economizer mode which supplies cooling to a CRAH- or CRAC-unit which in turn conditions the server environment (Potts 2011) (Niemann, Bean and Avelar 2011).

The term "free" refers to that no further energy has to be added to the system since the ambient cools the medium for "free". This is not completely true since some (pump) work often has to be added to circulate the medium (Potts 2011) and also power is needed for the air circulating fans. Often the pump and fan power are 5-10 % of the cooling effect (Evans, Fundamental principles of Air Conditioners for Information Technology 2014).

Since the free cooling is directly depending on the ambient temperature the system often needs to be complimented with mechanical refrigeration, e.g. a vapor compression cycle (Nortek Air Solutions 2015). But with increased allowed temperatures in the server environment, the amount of days, for when the use of free cooling is possible, increases (Strutt 2012).

2.3.6 Applications Using Free Cooling

One data center that uses free cooling is The GreenCube located in Darmstadt, Germany. In this data center the heat exchangers are mounted directly on the rear-doors on each server rack. In the heat exchangers water is circulated and is cooled by the primary loop in which turn is cooled by the outdoor air. The system also uses evaporative cooling explained in 2.3.7. The PUE for this data center is below 1.07 corresponding to an IT-Equipment share of about 93.5% (e3computing 2015). Figure 17 shows the system layout of The GreenCube (Kollegger 2015). As seen, the only two things that needs electrical power are the circulating pumps and the cooling fans.

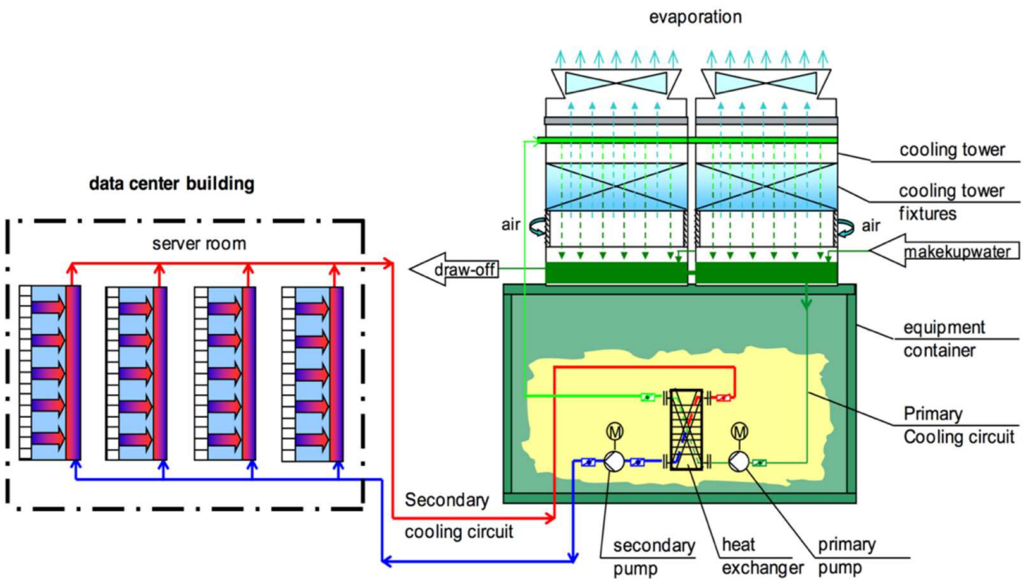

Figure 17 - *System layout of The GreenCube data center*

Carnot Refrigeration, has developed a cooling system for data centers, claiming a PUE below 1.10, using a thermosiphon and R744 (Carnot Refrigeration 2016).

For larger applications a system using the ambient air directly in data center and exhausting the hot air back to the ambient, called a direct air evaporative cooling system, is used. The system principle is shown in Figure 18 (Evans, The Different Technologies for Cooling Data Centers 2012).

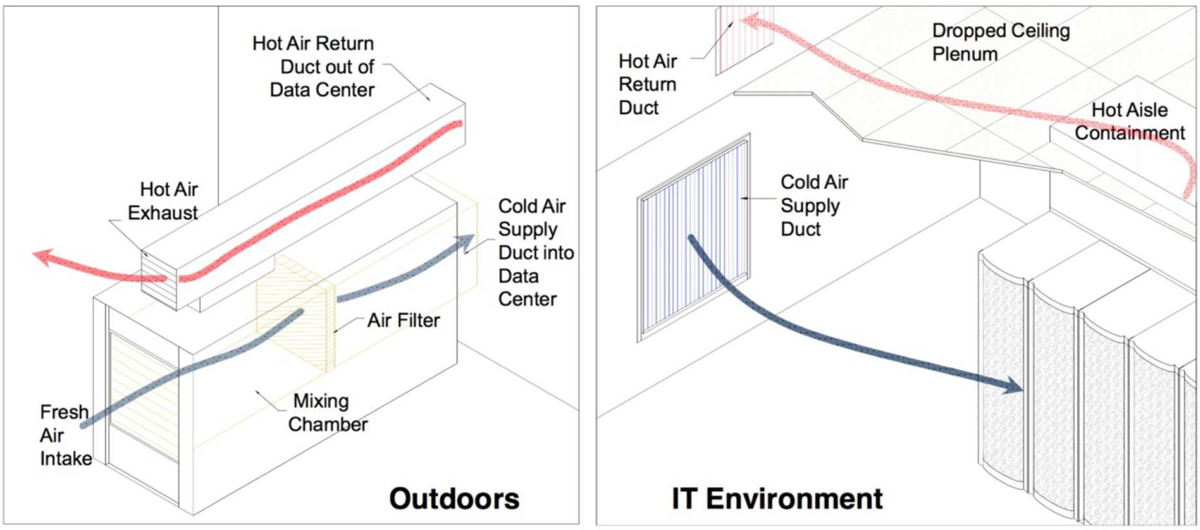

Figure 18 - *Principle of direct air evaporative cooling system*

The system solution in Figure 18 is often used where the cooling power of the data center exceeds 1000 kW and where the power density is high. This solution is economic and saves much energy but the disadvantages are that it can be hard to retrofit in existing systems, it will need frequent filter changes if the outside air is not clean and the evaporative cooling can cause humidity inside the data center (Evans, The Different Technologies for Cooling Data Centers 2012).

2.3.7 Evaporative Cooling

One method to cool refrigerants at higher temperatures is to use evaporative cooling. Water is sprayed over the heat exchanger, as in Figure 17, and uses latent heat from the refrigerant to vaporize resulting in a higher heat transfer. For mechanical refrigeration cycles this implies a reduction in compressor work hence the ability to have a lower condensing temperature. The amount of water that can evaporate is given by the relative humidity, the lower relative humidity there is, the more water can be vaporized (Alliance for Water Efficiency 2016).

Evaporative cooling is as most useful in hot dry climates where also the risks for drought and limited access for water are higher. It is also not possible for ambient temperatures below 0°C due to freezing. (Kozlowicz 2014). Corrosion of the construction material is a possible drawback caused by impurities in the water leading to refrigerant leakage. To avoid fouling on the heat exchanger, and by that lower the heat transfer possibility, the heat exchanger has to be regularly cleaned (H&C Heat Transfer Solutions 2014) (Hindelang, et al. 2014).

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3 Improvements

Due to the limitations in today's systems, several improvements need to be implemented to meet increasing demand and governmental policies. From Figure 6 it is seen that cooling is the single one most energy consuming sector except from the servers themselves. The servers are on the other hand the cause for the need of cooling by the Joule heating effect (COMSOL 2016). To further increase the energy efficiency, the dissipated heat could be recovered and used elsewhere. Since electricity is not free there also is an economical aspect that is considered with lower energy demand for cooling and recovering of waste heat (Strutt 2012).

The three most electricity consuming parts of a conventional cooling application in a data center are the compressor in the cooling machine, the circulating pumps and the air circulating fans (Petschke 2008). The U.S Federal Management Program states that removing the need of compressors contributes the most to reducing the electricity demand (Tschudi 2013). Although there are ways to also eliminate the fan work (Eklund 2016). Another drawback of today's data center cooling systems is the use of refrigerants with high GWP in their cooling process.

3.1 Alternative Refrigerants

As there are many data centers with outdated solutions using HCFC, CFC and HFC refrigerants there is need for new solutions with other refrigerants. To find a suitable refrigerant some different alternatives is compared by their ODP, GWP and classification.

The system design depends inter alia on the choice of refrigerant and the application. Some refrigerants can be used in a direct application whilst some must be used indirectly, depending on the refrigerants properties. Data centers need a high reliability and hazardous substances should be used with caution and in an as low extant as possible. For instance, a flammable refrigerant can ignite by the electronics or a corrosive medium can cause short circuits in case of leakage. One way to minimize the charge is using an indirect system (Clodic, Le Pellec och Darbord 1998) but with that follows more heat exchangers and thereby increased losses. Another considerable aspect for choosing refrigerant is the size of the data center and its cooling demand.

3.1.1 Definitions

The ODP is defined as the ozone depletion potential for the refrigerant R11 where the value for R11 act as reference value set to 1. The GWP is defined as the global warming potential for R744 that has the reference value of one (The Linde Group 2016).

The classification of a refrigerant is a letter and a number. The letters, A and B, stand for the toxicity level and the numbers 1, 2 and 3 stands for its flammability. The letters are defined as:

- **Group A** - LC50 ≥ 10 000 ppm
- **Group B** - LC50 < 10 000 ppm

where LC50 is the concentration that will cause death to half the experimental population when being exposed to it for four hours. The numbers instead are defined as:

- **Group 1** - Non flammable
- **Group 2** - LEL ≥ 3,5 %Vol
- **Group 3** - LEL < 3.5 %Vol

where LEL stands for the lower explosion limit. The definition is the minimum concentration of a refrigerant in a homogenous mixture with air, at 21°C and 101 kPa, to propagate a flame (AIRAH 2013). For group 2 refrigerants with a maximum burning velocity at less than 10 cm/s the letter "L" is added (ASHRAE 2010).

3.1.2 Comparison

The GWP, ODP and classification are displayed for some of the most common natural refrigerants in Table 2 with values for R12, R22 and R134a as references for comparison (The Linde Group 2016) (ASHRAE 2013) (Department of Energy Technology 2015) (Gaved 2013) (Honeywell 2015).

Table 2 - Comparison between some of the most common refrigerants

Refrigerant	ODP	GWP	Type	Classification
R12	1	10900	CFC	A1
R22	0,05	1810	HCFC	A1
R134a	0	1430	HFC	A1
R1234yf	0	<1*	HFO	A2L
R1234ze	0	<1*	HFO	A2L
R170 (Ethane)	0	6	Natural (Hydrocarbon)	A3
R290 (Propane)	0	3	Natural (Hydrocarbon)	A3
R600a (Isobutane)	0	3	Natural (Hydrocarbon)	A3
R717 (Ammonia)	0	0	Natural	B2L
R744 (CO_2)	0	1	Natural	A1

All the natural refrigerants and HFOs have zero ODP and low or zero GWP. R290 is suitable for areas with high ambient temperatures and also, R290, has similar thermodynamic properties as R22 (Jürgensen 2016). But, as all hydrocarbons, it is classified as highly flammable. R717 is classified as flammable in high concentrations and also as poisonous. Another aspect with R717 is that it is corrosive which means that there is a risk of harming electronic components in case of leakage (Depertment of Health 2004). The corrosiveness of R717 also impacts the pipe fittings and copper windings in the compressor making a use of open compressor necessary. To overcome the problems with hydrocarbons and R717, i.e. lower the charge and keep away from critical components, they are used in indirect systems as primary refrigerants. For applications over 3°C water is a suitable secondary working fluid. Other secondary fluids may be an alcohol or a saline solution (Melinder 2007).

HFOs are unsaturated HFCs meaning that at least one of their atomic bindings is a double binding (Tomczyk 2014). This double bond makes HFOs unstable and reactive to the ambient (Higashi 2010). Also the chemical components of HFCs and HFOs are the same (Department of Energy Technology 2015) but unlike HFCs they have a low GWP-value.

3.2 Thermosiphon Application

One way to reduce the electricity demand and costs is to eliminate the pump work (Mikielewicz 2011). By using a thermosiphon, which is a self-circulating heat moving unit that uses density and height difference to circulate the working medium, this is accomplished. There are many applications for thermosiphons e.g. solar heaters and electronic component cooling (Khodabandeh and Palm 2003) (Berber 2011). Further Carnot Refrigeration has developed a data center cooling system using a thermosiphon, as mentioned in 2.3.6.

* According to older calculations GWP for R1234yf and R1234ze is 4 and 6 respectively (The Linde Group 2016)

Earlier projects have used a thermosiphon in combination with a vapor compression cycle with refrigerants R134a (Lee, Kang and Kim 2009) and R22 (Han, et al. 2013) (Zhang, et al. 2015). The cooling capacity for these systems has been 1.8 kW – 12.5 kW.

3.2.1 Principle

The principle of a thermosiphon is to move heat from a heat source to a heat sink driven by the density difference from the corresponding temperature difference between the heat source and the heat sink. This density difference, combined with gravity, creates buoyancy forces that creates natural circulation (Mikielewicz 2011). Thermosiphons is divided into two sections,

Single- and Two-phase flow system

Open and closed system

A single-phase flow system works with a medium that does not change phase and the density difference is only from the temperature difference. A two-phase flow system has a phase change in the evaporator as the refrigerant evaporates and the density difference is both from the temperature difference and different phases. An open system can only work in single phase as the liquid cannot evaporate as it will evaporate to the ambient. A closed system is able to operate in both single- and two-phase flow applications (KTH Energy Department 2016).

3.2.2 Single- and Two-phase Flow

There are some things to account for when choosing which mode to use. The advantages and disadvantages are presented in Table 3 (Berber 2011) (KTH Energy Department 2016).

Table 3 - Comparison between the use of single- and two-phase

Single-phase		**Two-phase**	
+	-	+	-
Water can be used as heat carrier	Low efficiency	High efficiency	More advanced
Simple	High temperature gradient	Low temperature gradient	
	Increased weight	High heat fluxes due to latent heat	
	More volume to fill	Reduced weight	
		Less volume to fill	

From Table 3 there is seen that a two-phase system has a lot of benefits and this comes much from the use of latent heat i.e. the heat that is needed to evaporate and condensate the working media (KTH Energy Department 2016). This gives a low temperature gradient, high heat flux and high efficiency. The reduced weight is because the area that is needed for heat transfer can be smaller in a two-phase system. As the volume is smaller, less amount of working medium is needed. This also reduces the weight of the total equipment (Berber 2011).

3.2.3 Open and Closed Systems

Closed systems are divided into pipe thermosiphon, simple loop thermosiphon and advanced loop thermosiphon system (KTH Energy Department 2016). These different systems are illustrated shown in Figure 19.

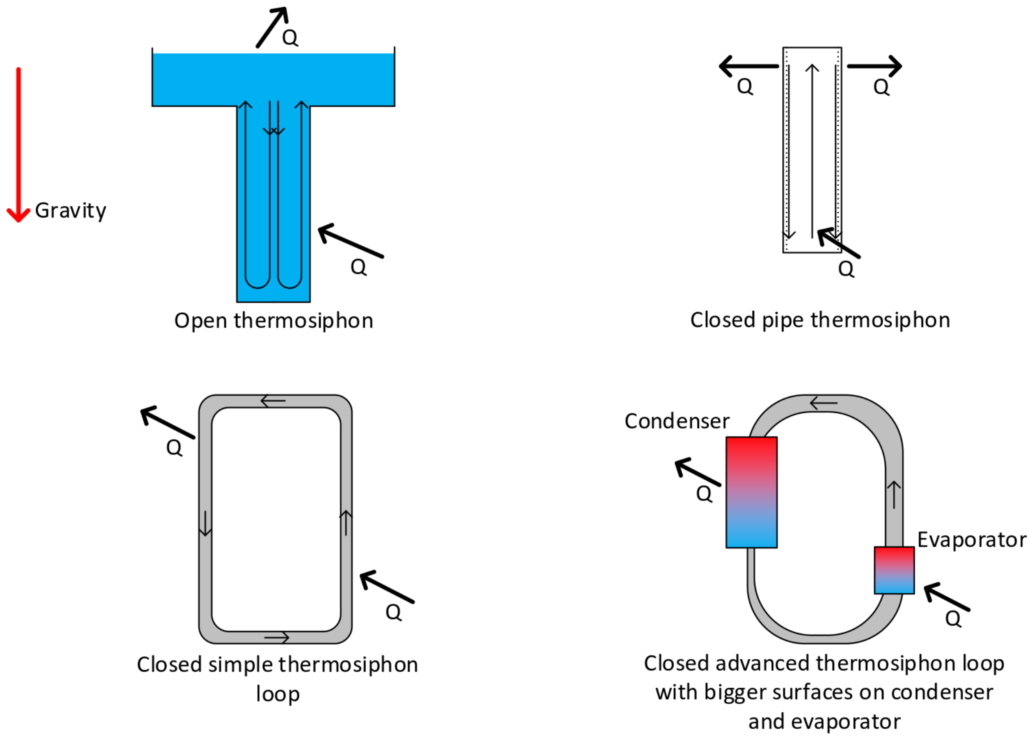

Figure 19 - Different thermosiphon solutions

The open single-phase system shows how it collects heat in the outer surface of the pipe going down and how it starts moving up to the surface where it releases heat. In all the closed systems there will be a part of the thermosiphon where it collects heat and one where heat is released. If it is a two phase thermosiphon the medium will start to boil in the heat collecting part and be condensed in the heat releasing part. The advanced closed two-phase thermosiphon loop differs from the others. In this system the loop will be connected to an evaporator and a condenser to increase the heat transfer surface. The similarities between all these solutions are that the heat releasing points always need to be at a higher point than the heat collecting point as it is driven by gravity (Mikielewicz 2011).

The major difference between these systems is the many advantages the advanced loop has compared to the other systems. The advantages is that the possibility to dissipate higher heat fluxes and still have a low temperature difference between the evaporator wall and the medium but also that it is easier to get a close contact to components when using an evaporator instead of a closed pipe or a simple loop (KTH Energy Department 2016).

3.2.4 Ability for Use of Free Cooling

A thermosiphon is compatible with the use of free cooling and even allows the compressor to be completely switched off thus eliminating the compressor work. This is also a drawback compared to other free cooling configurations since the compressor is not able to run in partial mode when the thermosiphon is operating (Niemann, Bean and Avelar 2011). When running in compressor mode additional heat from the compressor is added to the system that also needs to be cooled off (Tschudi 2013). With less heat to be dissipated from the system the temperature difference between ambient and exhaust can be smaller implying a tolerance for higher ambient temperature for which free cooling is still possible. Combined with the rise of recommended temperatures (Strutt 2012) the possibilities for using free cooling around the year increases and thereby enables to fully eliminate the need a compressor. Logically a climate with a lower yearly average temperature can utilize more hours of free cooling. Figure 20 is a geographical map over Europe showing the amount of hours were the ambient temperature is in the temperatures range for utilizing free cooling (Strutt 2012).

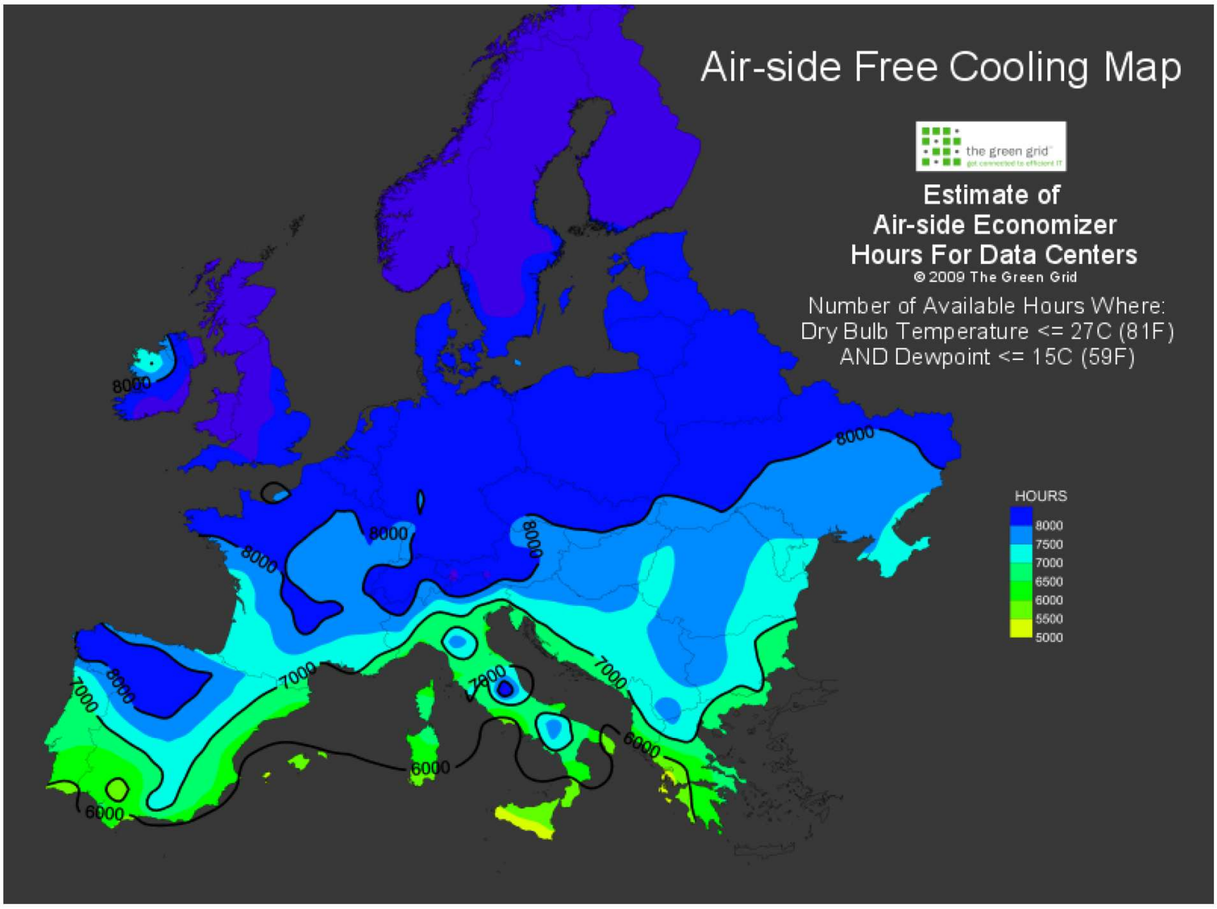

Figure 20 *- Free cooling geography of Europe*

Figure 21 shows a corresponding map for North America (Strutt 2012).

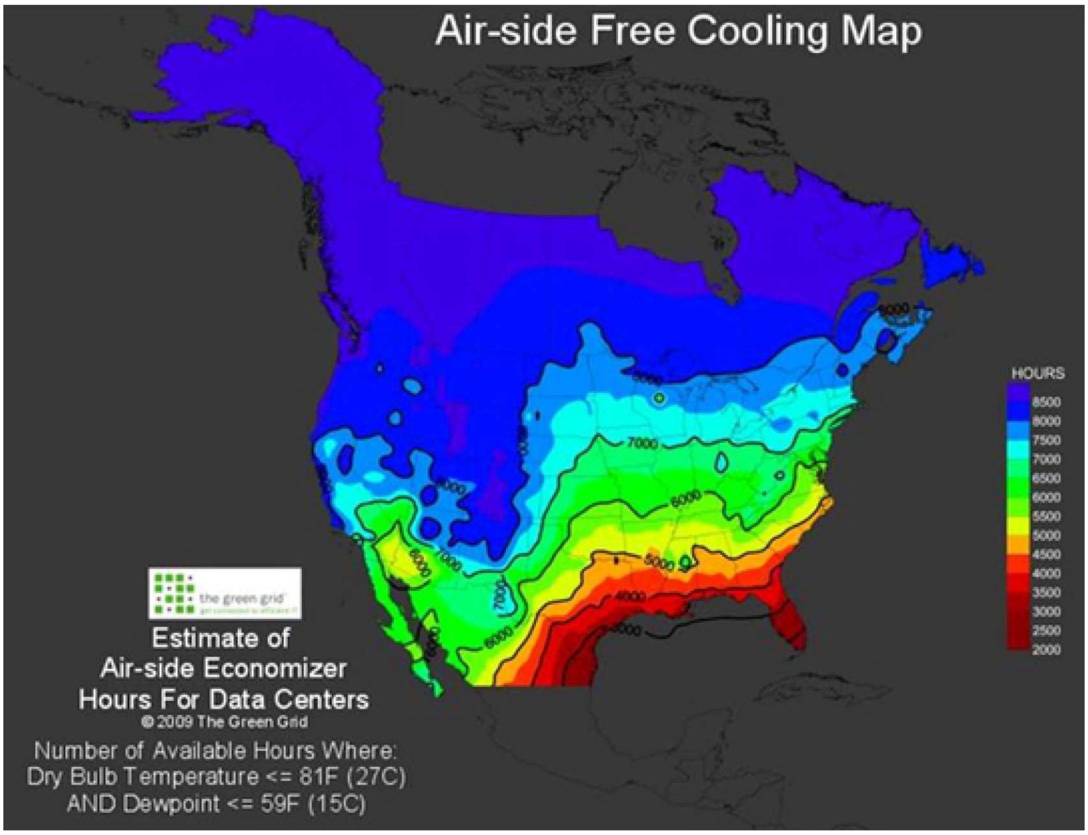

Figure 21 *- Free cooling geography of North America*

In Figure 17 a heat exchanger is installed between the server room-circuit and the outdoor-circuit. In a thermosiphon application with a refrigerant, the heat exchanger can be excluded and the following heat losses it causes. The used working medium should not be harmful for the servers in the case of leakage.

3.3 Heat Recovery

Heat recovery is when heat, instead of dissipated to the ambient, is used for heating purposes in the building, neighborhood etc. Depending on the heating demand for the application area different supply temperatures are needed. Table 4 list temperature demands for some heating applications.

Table 4 - Temperature demand temperatures for some heating application areas

Application area	**Hot water production**	**Space heating (radiators)**	**District heating**	**Preheating other medium**	**Heat source**
Supply temperature	50-60°C (BRE's Water Center 2016)	37°C (floor heating) 55°C (radiators) (Energijägarna 2012) (Havtun and Bohdanowicz 2014)	65-115°C (Finsk energiindustri 2007)	Hotter than the medium that is to be pre-heated	Enough to evaporate the refrigerant in the heat pump

The air temperature after the servers, i.e. in the hot aisle, is approximately 46-49°C (Granmar 2013) (Miller 2011). This temperature range means that the waste heat is low-graded and thus generally less economical beneficial to recover (ASHRAE 2012). For instance, the required temperatures in Table 4 gives that floor heating is the only application possible to use directly. Hot water production and radiator space heating require higher temperatures than the given from the server exhaust. One method to enable utilizing of the waste heat is to lift the temperature to the desired temperature, e.g. by a heat pump, where the air is used as a heat source (Fortum 2016). Other ways to utilize this low-grade heat are absorption cooling, direct power generation, indirect power generation, biomass co-location and desalination where absorption cooling and indirect power generation were found to be most efficient (Ebrahimi, Jones and Fleischer 2014). All the applications where the temperature has to be increased consume additional power with the sequent increase in cost.

One example of a solution that works with heat recovery is a trans-critical booster R744 system that is new to the market for supermarket refrigeration. A possible system layout is shown in Figure 22.

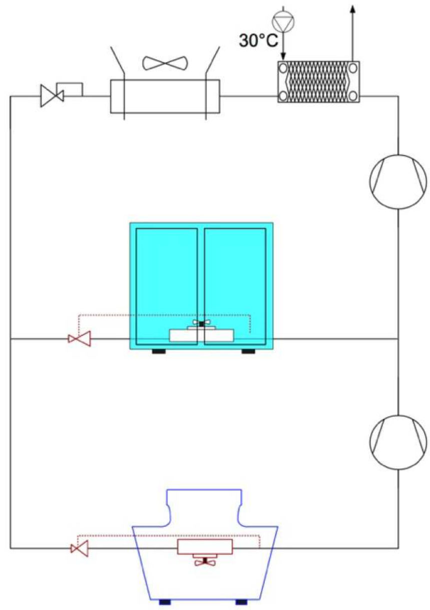

Figure 22 *- Schematic of a trans-critical booster R744 system*

In Figure 22 there are two temperature levels, one for the refrigerators and one for the freezers. In the top right corner is a de-super heater where heat is recovered instead of being released to the ambient in the gas cooler. This system compared to conventional systems, with R404A as refrigerant, works efficient, at least in cold climates, even though R744 is not the most effective refrigerant (Sawalha, Investigation of heat recovery in CO2 trans-critical solution for supermarket refrigeration 2012) (Trane 2011).

4 System Modelling

The choice of main refrigerant is R744 since it is non-flammable nor toxic nor corrosive and therefore possible to use in direct system. Additionally, R744, in case of leakage, vaporizes in the server room temperature and pressure conditions making R744 harmless to the electrical components. To evaluate the system performance three different systems are compared, a direct R744 system, an indirect R744 system with R290 as primary refrigerant and two reference systems working with R22 since it is the present most common one. The choice of R290 is based on that is it commonly used for applications with similar cooling effect compared to e.g. R717 which is used in bigger systems (BOC 2012).

The direct R744 system and the secondary circuit in the indirect R744 system uses a thermosiphon as main operation mode described in 4.2. The R22-systems are both using brine in a secondary circuit and are divided into one pure mechanical solution and one running with economizer.

The simplified thermosiphon base system layout, serving as a basis for the calculations, is shown in Figure 23.

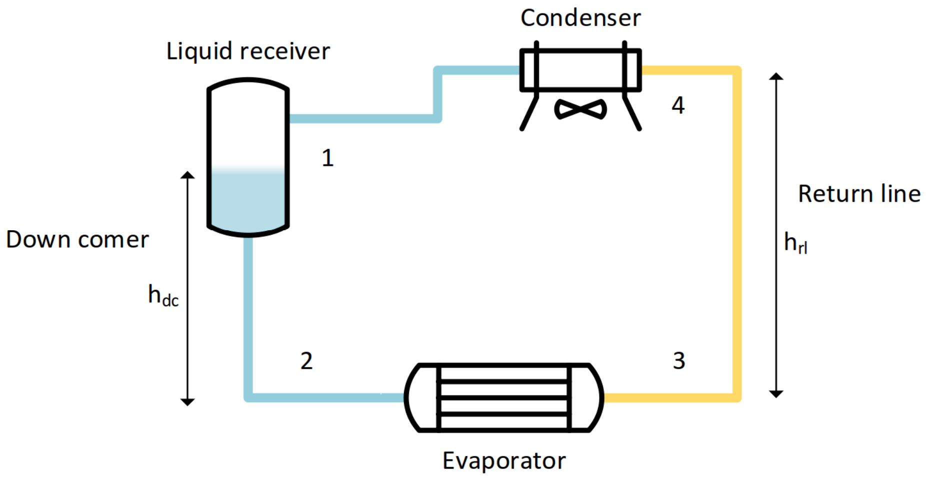

Figure 23 *- Base thermosiphon system layout with four calculation points*

In Figure 23 the points 1, 2, 3 and 4 are points where the different states of the refrigerant are calculated.

To assure that there only is liquid in the down comer a liquid receiver is used. The driving force in the thermosiphon is the density difference between down comer and return line. For the system to achieve a steady state condition the driving head pressure must balance the pressure losses in the system. The governing geometrical design parameters for the thermosiphon are marked with an "x" in Table 5. Some of the design parameters have geometrical relationships which are presented later in this chapter.

Table 5 - Needed design parameters for the thermosiphon

Part	**Height**	**Length**	**Width**	**Diameter**	**Bends**
Down comer	x	x	-	x	x
Return line	x	x	-	x	x
Evaporator	x	x	x	x	x

From the literature some of the other design parameters, shown in **Table 6**, are given.

Table 6 - Cooling and temperature design parameters

Cooling demand	Evaporating temperature	Evaporator air inlet temperature	Evaporator air outlet temperature
200 kW	26°C	46°C	27°C

In order to find out how the system would fit in an existing building several parameters are varied. The parameters are:

- Down comer height
- Down comer pipe diameter
- Return line pipe diameter
- Evaporator pipe diameter
- Amount of evaporator bends
- Amount of evaporator circuits

To find the amount of heat the system is able to absorb in the evaporator, the measured quantity is the quality out from the evaporator. This quality is referred to as the evaporator outlet quality.

4.1 Critical Assumptions and Limitations

As the system that is modelled is not real and does not exist, there are several assumptions that are made. All the assumptions and limitations are discussed in this part.

4.1.1 System in General

From Figure 8 and Figure 9 it is given that a data center with power demand of 17 kW per server rack covers over 80 % of the used data centers. A power demand of 17 kW per server rack corresponds to a system with low server rack power density. The system is approximated to have 12 server racks making the total power demand approximately 200 kW, equivalent to a medium sized data center. Since all energy supplied to the servers is assumed to become heat the cooling demand for the system is as well 200 kW. In all heat exchangers, except from the evaporator and condenser, the approach temperature is set to 2˚C.

In the switch between thermosiphon mode and mechanical refrigeration the system is transient until it is stabilized (steady state). This process is neglected and the system is assumed to achieve steady state conditions immediately. Further, all changes in and to the system are assumed to be discrete and therefore happen stepwise e.g. change in ambient temperature.

4.1.2 Down comer and Return line

In the down comer there is only liquid and in the return line a two-phase state. In a real system there will probably be a small heat exchange to ambient but in the model both down comer and return line will be seen as adiabatic i.e. no heat exchange to ambient. The density varies a little bit as the pressure is different at the top compared to the bottom of both down comer and return line. This density difference is very small and are therefore neglected.

To simplify the calculations, the vapor fraction is assumed to be constant in the return line as the flow will be seen as adiabatic and the pressure difference is neglected.

4.1.3 Evaporator and Condenser

The evaporator is modelled in two configurations (Figure 24 and Figure 25), in the local cooling configuration the evaporator is assumed to have the same dimensions as the rear door. The condenser is modelled after an existing unit from Alfa Laval, Alfa VXD (Alfa Laval 2015). From the literature it is given that the refrigerant pressure drop in condenser is usually neglected and therefore as well is this work.

To simplify the calculations of the two-phase density, the quality is set to linearly increase to the quality at the evaporator outlet. This assumption is valid as long as the refrigerant is not subcooled at the evaporator inlet (Tengblad 1996). The means that the quality will be calculated as

$$x_m = \frac{x_{out}}{2}. \tag{3}$$

In air pressure loss calculations in the condenser two areas are used. The given fan area and an assumed face area. The ratio between theses is expressed as

$$\alpha_{area} = \frac{A_{face}}{A_{fan}}. \tag{4}$$

The value of α_{area} is determined by finding the equilibrium between the calculated fan power and the given from Alfa Laval (Alfa Laval 2011).

The friction loss through the finned coils in the condenser is found from the software IMST-ART (University of Valencia 2015). The pressure drop for several velocities is calculated and then a correlation of a 2nd degree polynomial (57) is derived, knowing that the frictional pressure loss is proportional to the squared velocity.

4.2 Thermosiphon Pressure Drops

The base system is a thermosiphon application complemented with an auxiliary cooling device to cover the whole demand. For each part the governing equations used in the modelling is presented (Tengblad 1996) (Skaugen, Vist och Sveinsson 1999).

4.2.1 Down comer

In the down comer the fluid is in liquid phase. Due to gravity, a head pressure occurs at the bottom of the down comer, causing a circulation of the fluid in the system which in turn causes pressure losses due to friction. The head pressure due to gravity is given by

$$\frac{dp}{dz} = \rho_l \cdot g \tag{5}$$

where ρ_l is the saturated liquid density and g is the gravitational acceleration. (5) is integrated over the down comer height to

$$\Delta p_g = \rho_l \cdot g \cdot h_{dc} \tag{6}$$

as the density is assumed constant along the length of the down comer.

The friction pressure loss in the down comer is calculated using a single phase flow model. The one that is used is the Fanning equation defined as

$$\Delta p_F = -f_F \cdot \frac{2 \cdot \dot{m}_{flux}^2 \cdot L_{dc}}{\rho_l \cdot D}, \tag{7}$$

where f_F is the Fanning friction factor, $\dot{m}_{flux}$ is the mass flux in the pipe and D is the hydraulic diameter of the pipe. The Fanning friction factor is calculated differently depending on the type of flow i.e. laminar or turbulent flow.

The equations are:

$$f_F = \frac{16}{\text{Re}} \text{ for Re} < 2400 \tag{8}$$

and

$$f_F = 0.079 \cdot \text{Re}^{-0.25} \text{ for Re} \geq 2400, \tag{9}$$

where Re is calculated as

$$\text{Re} = \frac{4 \cdot \dot{m}}{D \cdot \pi \cdot \mu}, \tag{10}$$

where μ is the dynamic viscosity.

The pressure drops from bends in the down comer are calculated according to 4.2.4.

4.2.2 Return line

The return line is different from the down comer as the fluid flows in two phase instead of single phase. The return line has both gravitational and frictional pressure drops. Due to the two-phase flow the pressure drop is more complicated to calculate. According to Tengblad (Tengblad 1996) the CESNEF-2 correlation is the most accurate when calculating friction losses for two-phase flow, but according to Skaugen et. al. (Skaugen, Vist och Sveinsson 1999) the Friedel correlation is more accurate when the viscosity ratio is below 1000. The viscosity ratio for R744 is around 10 (Skaugen, Vist och Sveinsson 1999). The used correlation is therefore the Friedel correlation.

The gravitational pressure drop is calculated by

$$\frac{dp}{dz} = -\rho_{TP} \cdot g, \tag{11}$$

where ρ_{TP} is the two phase density and g is the gravitational acceleration. (11) is integrated over the down comer height to

$$\Delta p_g = -\rho_{TP} \cdot g \cdot h_{rl} \tag{12}$$

as the density is assumed constant along the length of the return line.

The two phase density is given by

$$\rho_{TP} = \rho_l \cdot (1 - \alpha) + \rho_g \cdot \alpha \tag{13}$$

where ρ_l and ρ_g is the density at the evaporator temperature of saturated liquid and vapor, respectively. α is the void fraction given by

$$\alpha = \frac{x \cdot \rho_l}{x \cdot \rho_l + (1 - x) \cdot \rho_g} \tag{14}$$

where x is the quality of the refrigerant at evaporator outlet.

The frictional pressure drop is given by the Friedel correlation:

$$\left(\frac{dp}{dl}\right)_{TP} = \Phi_l^2 \cdot \left(\frac{dp}{dl}\right)_l \tag{15}$$

where Φ_l^2 is the two phase multiplier and $\left(\frac{dp}{dl}\right)_l$ is the pressure drop for saturated liquid only, which is calculated according to the Fanning equation (7) with the length of the return line L_{rl}.

The two phase multiplier have two different definitions depending on if the flow direction is up-or downwards. In this case the flow is upwards and the two phase multiplier Φ_l^2 is then calculated as

$$\Phi_l^2 = E + \frac{3.24 \cdot F \cdot H}{\mathrm{Fr}^{0.0454} \cdot \mathrm{We}^{0.035}}. \tag{16}$$

The factors in (16) are calculated as:

$$E = (1-x)^2 + x^2 \cdot \left(\frac{\rho_l \cdot f_g}{\rho_g \cdot f_l}\right) \tag{17}$$

$$F = x^{0.78} \cdot (1-x)^{0.0224} \tag{18}$$

$$H = \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \cdot \left(\frac{\mu_g}{\mu_l}\right)^{0.19} \cdot \left(1 - \frac{\mu_g}{\mu_l}\right)^{0.7} \tag{19}$$

$$\mathrm{Fr} = \frac{\dot{m}_{flux}^2}{g \cdot D \cdot \rho_{TP}^2} \tag{20}$$

$$\mathrm{We} = \frac{\dot{m}_{flux}^2 \cdot D}{\rho_{TP} \cdot \sigma} \tag{21}$$

The pressure drops from bends in the return line are calculated according to 4.2.4.

4.2.3 Evaporator

The evaporator is made in two different configurations. The first one has the evaporator in a central cooling unit whereas the other configuration has evaporator located on every server rack rear door, local cooling. In the second configuration the cooling demand is assumed to be evenly distributed among the server racks. In modelling the evaporator there are three kinds of pressure drops: gravitational pressure drop, frictional pressure drop and acceleration pressure drop and also heat transfer.

Figure 24 shows a schematic drawing over a system configured with one evaporator in a central cooling unit. From Figure 24 a relationship between the heights is

$$H_E + H_{rl} = H_{dc} + \Delta H. \tag{22}$$

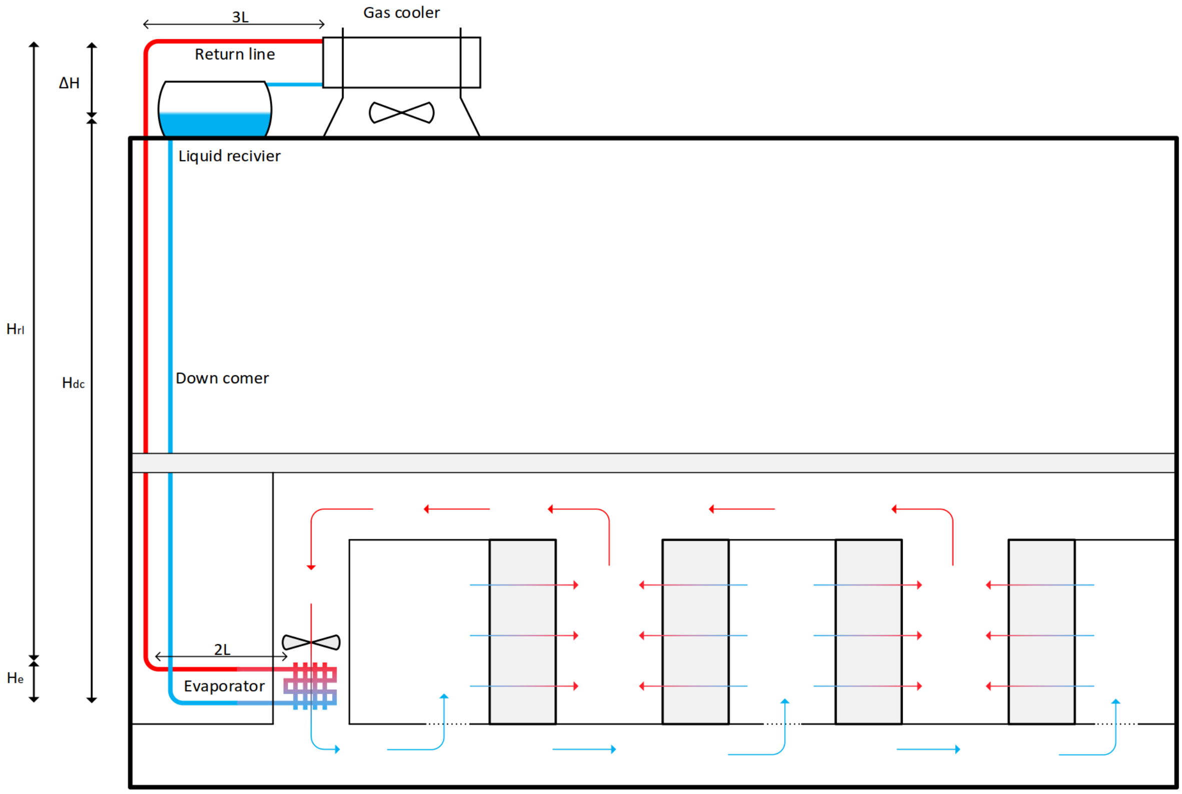

Figure 24 - Schematic drawing a system with evaporator in a central cooling unit

The evaporator is horizontal and has a face area that is perpendicular to the air flow. **Table 7** holds the dimensions for the system in Figure 25.

Table 7 - Dimensions for modelling baseline scenario of the system with central cooling unit

H_e	H_{dc}	H_{rl}	ΔH	L	d_e	d_{dc}	d_{rl}	**Bends**
0.2 m	8.8 m	7 m	2 m	0.6 m	24 mm	42 mm	42 mm	3

In Figure 25 the system is configured with local cooling. The relationship in (22) is still applicable.

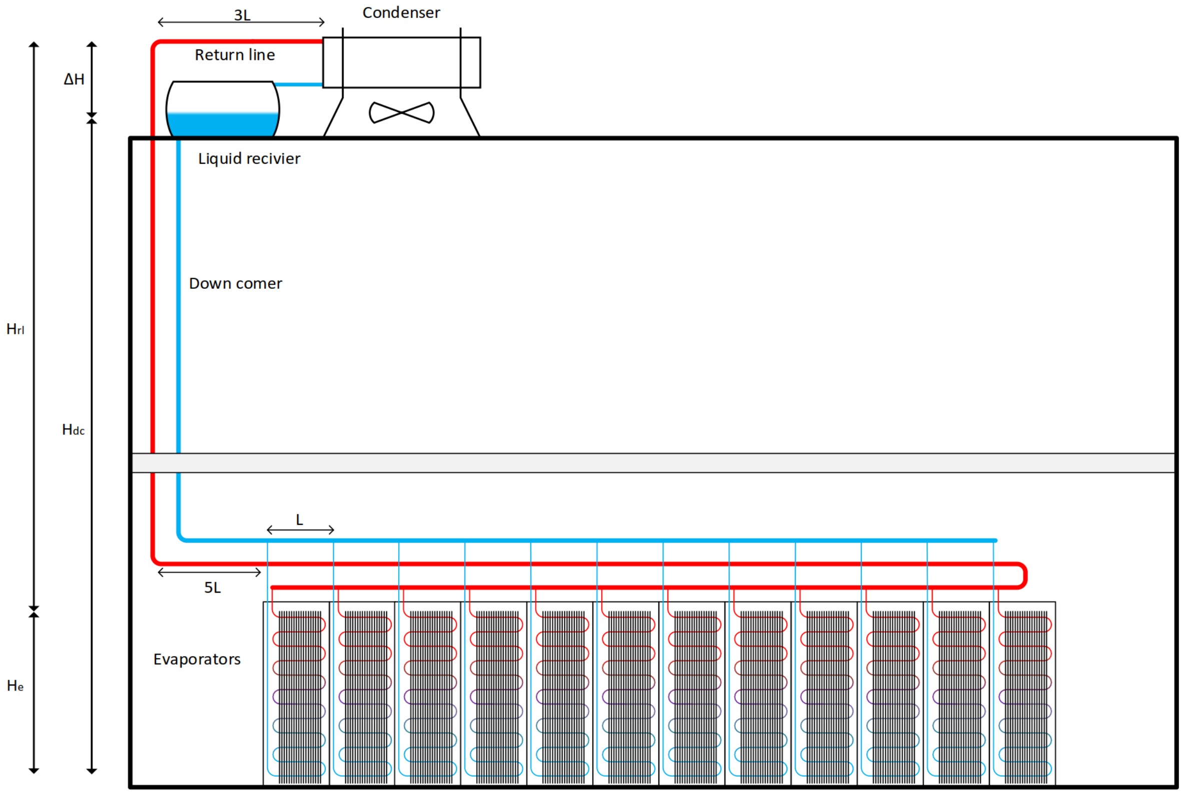

Figure 25 - Schematic drawing over system with local cooling

Table 8 holds the dimensions used in modelling the baseline scenario of the system (Sun Microsystems Inc. 2004) (Dell Inc. 2012). Since the circuits are parallel, the pressure drop is assumed to be equal for all circuits.

Table 8 - Dimensions for modelling baseline scenario of the system with local cooling units

H_e	H_{dc}	H_{rl}	ΔH	L	d_e	d_{dc}	d_{rl}	**Bends**
2 m	7 m	7 m	2 m	0.6 m	24 mm	42 mm	42 mm	6

There are system solutions where the condenser cannot be placed on the roof but has to be in the level as the server room, as shown in Figure 26.

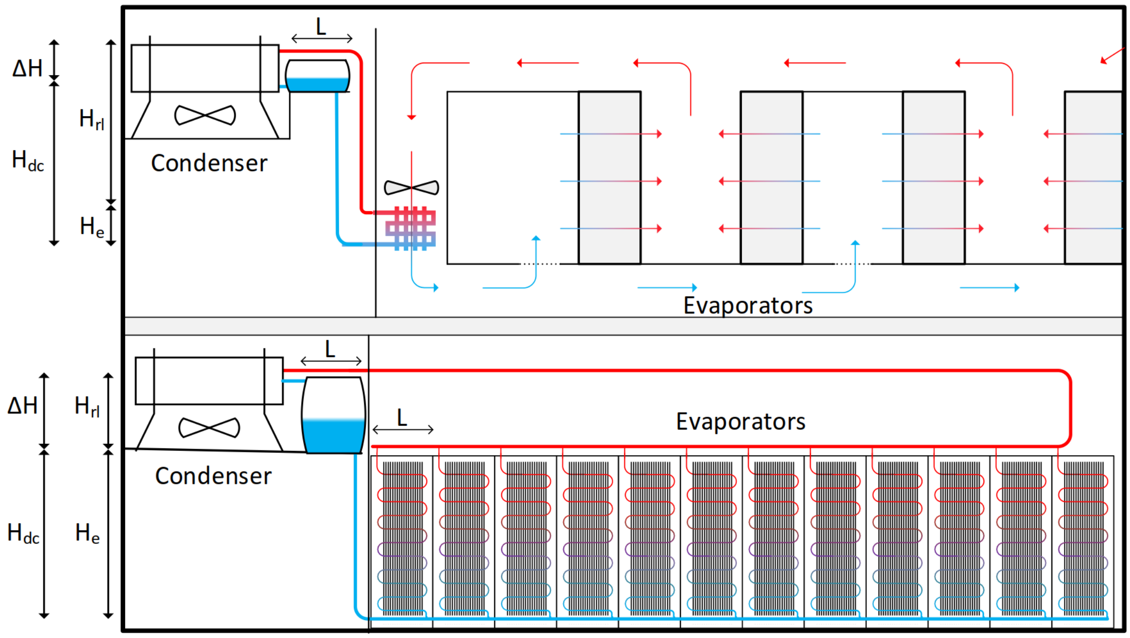

Figure 26 - Central and local cooling unit systems with the condenser in the same level as the server room

In Figure 26 the condenser is separated from the server room to keep the temperature difference. The condenser is as well cooled by the ambient. The base line scenario parameters are then according to *Table 9*.

Table 9 - Dimensions for modelling baseline scenario of the system with the condenser in the same level as the server room

	H_e	H_{dc}	H_{rl}	ΔH	L	d_e	d_{dc}	d_{rl}	**Bends**
Central cooling	0.2 m	2 m	2 m	0.2 m	0.6 m	24 mm	42 mm	42 mm	3
Local cooling	0.2 m	2 m	2 m	0.2 m	0.6 m	24 mm	42 mm	42 mm	3

To lower the frictional pressure drop over the evaporator the refrigerant is divided into two or more parallel circuits. Table 10 shows the baseline configuration of the two different evaporator configurations.

Table 10 - Evaporator tube configuration

Central cooling		**Rear-door**	
Bends	Circuits	Bends	Circuits
3	10	11	2

The gravitational pressure drop is calculated according to (12) with the height h_e and quality from (3). The frictional pressure drop is calculated according to Friedels correlation shown in (15).

The acceleration pressure drop is a reaction to the change in velocity due to the change in density. The pressure drop in this case can be described as

$$\Delta p_a = \frac{\dot{m}^2}{A^2} \cdot \left(\frac{1}{\rho_{TP}} - \frac{1}{\rho_l} \right), \tag{23}$$

where A is the area of the evaporator channel in this case.

The tubes in the evaporator has several bends which causes pressure drops. These are calculated according to 4.2.4 and a 180° bend is seen as two consecutive 90° bends.

4.2.4 Pressure drop in Bends

The pressure drop in bends is calculated as

$$\Delta p_b = \Delta p_l \cdot \left(1 + C \cdot \left(\frac{\Delta p_g}{\Delta p_l} \right)^{0.5} + \frac{\Delta p_g}{\Delta p_l} \right) \tag{24}$$

where

$$C = \left[k + \left(C_2 \right) \cdot \left[\frac{v_g - v_l}{v_g} \right]^{0.5} \right] \cdot \left[\left(\frac{v_g}{v_l} \right)^{0,5} + \left(\frac{v_l}{v_g} \right)^{0.5} \right] \tag{25}$$

where $k = 1$ and $C_2 = 1 + 20 \cdot \left(\frac{d}{L} \right)$ for 180°, with $\frac{d}{L} = \frac{1}{50}$, and 90°, with $\frac{d}{L} = \frac{1}{30}$ bends (Native Dynamics 2016).

The pressure differences for the fluid in liquid phase is

$$\Delta p_l = \frac{\zeta \cdot m_{flux}^2 \cdot (1-x)^2}{2 \cdot \rho_l} \tag{26}$$

and the pressure difference for the fluid in gaseous phase

$$\Delta p_g = \frac{\zeta \cdot m_{flux}^2 \cdot x^2}{2 \cdot \rho_g} . \tag{27}$$

ζ is an empirical constant (Ekroth and Granryd 2006).

4.3 Fan power

The needed mechanical fan power is found from the volume flow and the pressure difference and assuming incompressible flow

$$P_{fan} = \dot{V} \Delta p , \tag{28}$$

where the pressure difference is found through the Bernoulli equation and the friction loss through the finned coils

$$\Delta p = \frac{u^2 \rho}{2} + \Delta p_f , \tag{29}$$

where Δp_f is the friction pressure loss from (57).

Introducing a fan efficiency η_{fan} the electrical power needed for the fan can be expressed as

$$P_{fan} = \frac{\dot{V}}{\eta_{fan}} \left(\frac{\rho u_{fan}^{2}}{2} + \Delta p_f \right) \tag{30}$$

An efficiency of 65% can be assumed for estimating purposes (LEESON Electric 2016).

The volume flow is found from the heat balance in the condenser

$$Q = \dot{m} c_{p,air} \left(T_{c,out} - T_{amb} \right). \tag{31}$$

The volume flow is the mass flow divided by the density:

$$\dot{V} = \frac{\dot{m}}{\rho}. \tag{32}$$

The velocity is found through the volume flow divided by their respective area:

$$u = \frac{\dot{V}}{A}. \tag{33}$$

The temperature difference in (31) is found through a heat balance for the dissipated heat from the gas cooler:

$$Q = U \cdot A_{fin} \cdot LMTD, \tag{34}$$

where U is the overall heat transfer coefficient, A_{fin} is the fin area and $LMTD$ is the logarithmic mean temperature:

$$LMTD = \frac{T_c - T_{air,exhaust} - T_c + T_{amb}}{\ln\left(\frac{T_c - T_{air,exhaust}}{T_c - T_{amb}} \right)} = \frac{T_{amb} - T_{air,exhaust}}{\ln\left(\frac{T_c - T_{air,exhaust}}{T_c - T_{amb}} \right)}. \tag{35}$$

The heat transfer coefficient depends on inter alia the flow velocity. An empirical correlation (Engineering Toolbox 2016), valid for flow velocities between 2 and 20 m.s^{-1} is

$$U = 10.45 - u + 10u^{0.5}. \tag{36}$$

(31)-(36) are now used to solve the volume flow and calculate the fan power. When the fan power is known the PUE-value is calculated according to (1). The fans are set to have a maximum air mass flow of 9.1 kg.s^{-1}, which corresponds to a fan power of 2.7 kW per fan.

4.4 Compressor work

For the reference systems and the auxiliary additional compressor work is needed. The working cycle looks as in Figure 27.

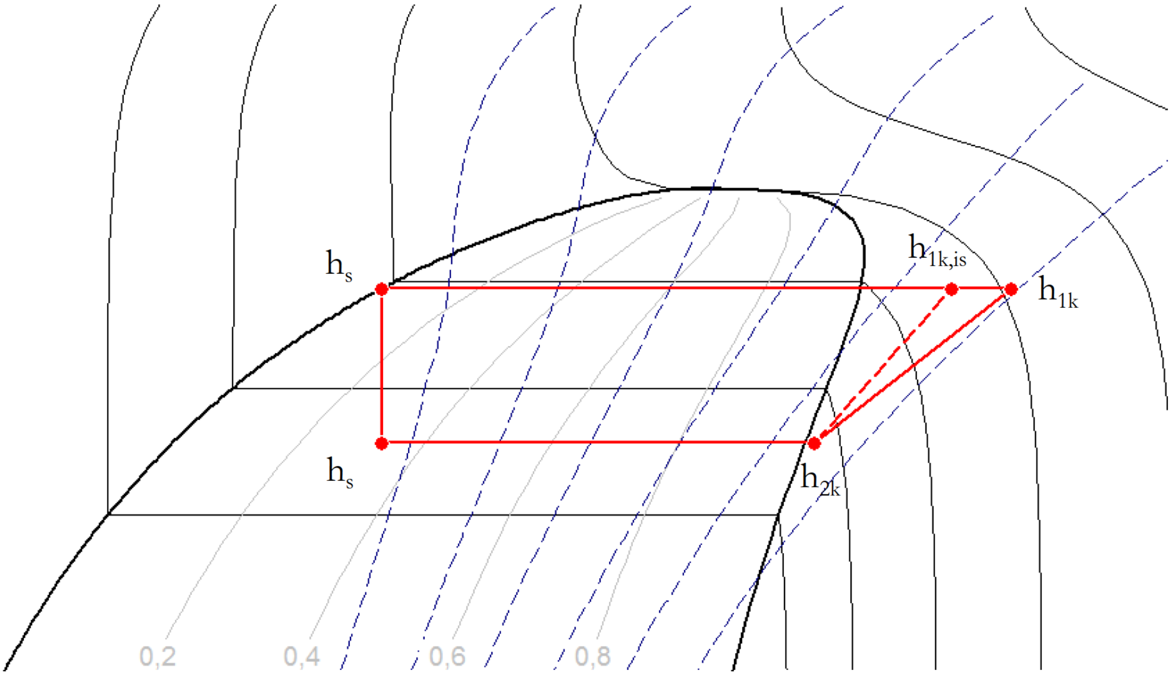

Figure 27 - Compressor working cycle

The compressor work is found from the enthalpy difference and the mass flow

$$E_{el} = \frac{\dot{m}}{\eta_{is}}\left(h_{1k,is} - h_{2k}\right). \tag{37}$$

where η_{is} is the isentropic efficiency of the compressor set to 0.7 (Baumann 2001) and the enthalpies is found through

$$h_{1k,is} = f\left(p_c, s_{2k}\right) \tag{38}$$

and

$$h_{2k} = f\left(p_e, \mathrm{t}_{2k}\right) \tag{39}$$

where

$$s_{2k} = f\left(p_e, \mathrm{t}_{2k}\right) \tag{40}$$

and

$$t_{2k} = T_e + \Delta t_{sh}. \tag{41}$$

where Δt_{sh} is the superheat set to 5K.

4.5 Reference System

To compare the performance of the new system solutions a reference system is designed out of the information from the literature review. The reference system is an indirect system with R22, as refrigerant as it is the most common today, and a 30% ethylene glycol/water solution in the secondary loop. The brine in the secondary loop is circulated with a pump with a specific pumping power $\dot{E}_{pump}$. The temperature difference in the secondary brine loop is set to be 4°C which is used to calculate the flow of the brine.

The reference system has two setups where the first one is only a running vapor compression cycle and the second system is using an economizer up to 16°C ambient temperatures. In this case the vapor compression cycle does not run at all up to 16°C i.e. only free cooling is used. The compressor work is found from 0. The evaporation temperature in this system is set to 12.5°C as it is the highest possible for today's R22-compressors (BITZER 2016). The systems are shown in Figure 28.

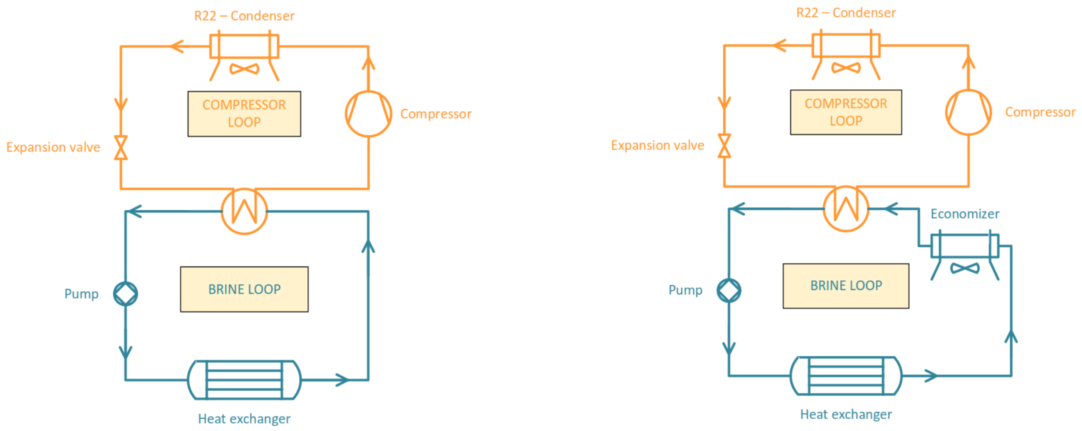

Figure 28 - *Reference system layout with and without economizer*

The pumping power $\dot{E}_{pump}$ is taken from Grundfos (Grundfos 2016) and the choice of pump is dependent of the flow of the brine which is calculated as

$$\dot{V}_{brine} = \frac{\dot{Q}_2}{\rho_{brine} \cdot c_{p,brine} \cdot (t_{brine;in} - t_{brine;out})}. \tag{42}$$

The pump is assumed to circulate the brine at constant load in the calculation model and work to a height at 10 m which corresponds to a pressure drop of 100 kPa.

The condenser size in both systems is set to 2000 m² fin area and using 6 fans. The economizer size is calculated in Alfa Laval's software (Alfa Laval 2011) with brine inlet temperature at 30°C and outlet temperature at 26°C.

4.6 Climate Profiles

One aim for the system is the possibility to be placed in different climates. For instance, areas with lower annual temperatures has more potential to be suitable for free cooling and areas with lower relative humidity is suitable for free cooling using evaporative cooling. Therefore, the performance is tested as if the system were located in the following cities:

- Stockholm
- Madrid
- Paris
- Tokyo
- Phoenix

Besides the temperature profiles the relative humidity of the air affects the cooling possibilities. A lower relative humidity is equivalent to a higher amount of water vapor that the air can absorb using evaporative cooling. The temperature and relative humidity profiles for the different cities are based on meteorological data from *Meteonorm* (Meteonorm 2016).

To determine if both the temperature and the relative humidity are of suitable values, the quantity of wet bulb temperature is used. The wet bulb temperature is dependent on the temperature and relative humidity. Stull (Stull 2011) gives the correlation

$$T_{wb} = T\arctan(0.151977(\phi + 8.313659)^{1/2}) + \arctan(T + \phi) - \arctan(\phi - 1.676331) + 0.00391828\phi^{3/2}\arctan(0.023101\phi) - 4.686035 \tag{43}$$

The investigated quantity is the amount of hours for which the five cities are at certain climate states. For non-evaporative cooling it is the ambient temperature and for evaporative cooling: the ambient temperature and relative humidity.

4.7 Auxiliary Cooling

Since the thermosiphon system is not able to cover the whole cooling demand by itself, for higher ambient temperatures, it must be complemented with an auxiliary cooling device. The governing temperature for when to use auxiliary cooling is presented in 5.2.

As auxiliary cooling a vapor compression cycle is used in two different setups. A direct system is presented in 4.7.2 and in 4.7.3 an indirect variant. The electrical power needed for running the systems is found through 4.7.1.

4.7.1 Compressor Cycle

In the compressor cycle the condensing temperature is governed by the ambient temperature. Although if the condensing temperature is dependent of only the ambient temperature it, for some cases, would be below the evaporation temperature. To avoid this obstacle, the system is regulated such that the condensing temperature never falls below a certain value. The value of the minimum condensing temperature is governed by the minimum pressure ratio and maximum evaporation temperature of the compressor. These are given in Table 11 (BITZER 2016).

Table 11 - Minimum pressure ratios for R744, R290 and R22

Refrigerant	**R744**	**R290/R22**
Minimum Pressure Ratio	1,35	1,51
Maximum Evaporation Temperature (°C)	20	12,5
*Minimum Condensing Temperature (°C)**	31	28

The corresponding temperature profiles for the evaporator and condenser is shown in Figure 29.

* Based on minimum pressure ratio and maximum evaporation temperature

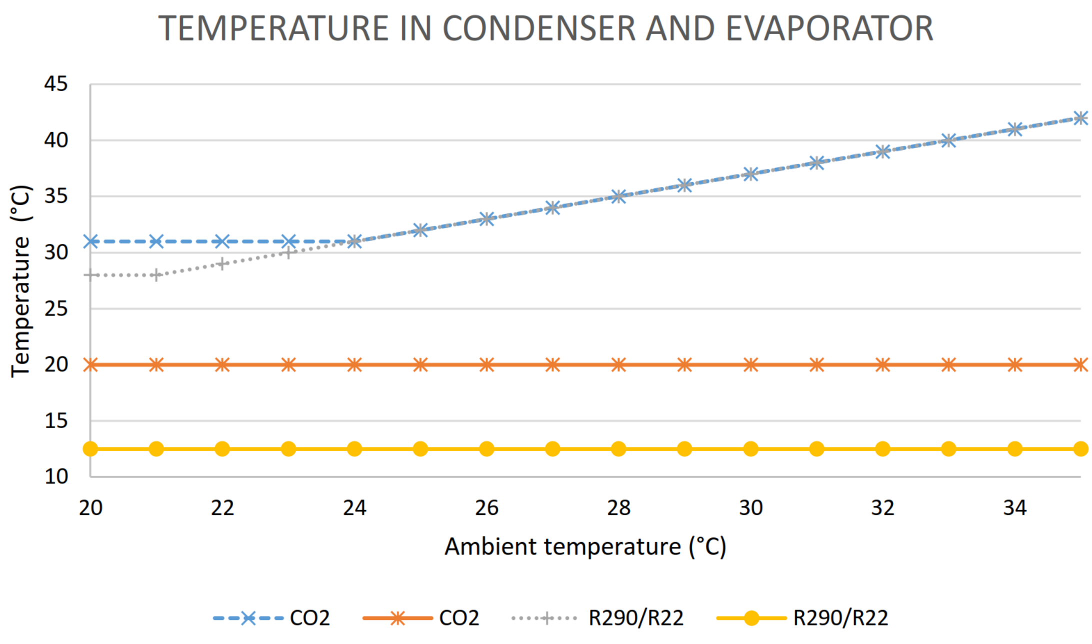

Figure 29 - *Condensing and evaporation temperatures in compressor cycle*

As the system sometimes operates in the supercritical region for R744, the refrigerant no longer condensates, it is only cooled. This is referred to as gas cooling and the condenser is in this operating mode referred to as gas cooler. Since temperature and pressure are independent, the optimum pressure is calculated from the temperature in the gas cooler. Sawalha (Sawalha, CO2 and Transcritical Cycle - node.do 2015) gives the equation

$$p_{c,optimum} = 2,7 \cdot T_c - 6,1 \,. \tag{44}$$

The other refrigerants always work in subcritical mode. The compressor work is found from 0.

4.7.2 Direct System

The proposed direct system running with R744 is presented in Figure 30.

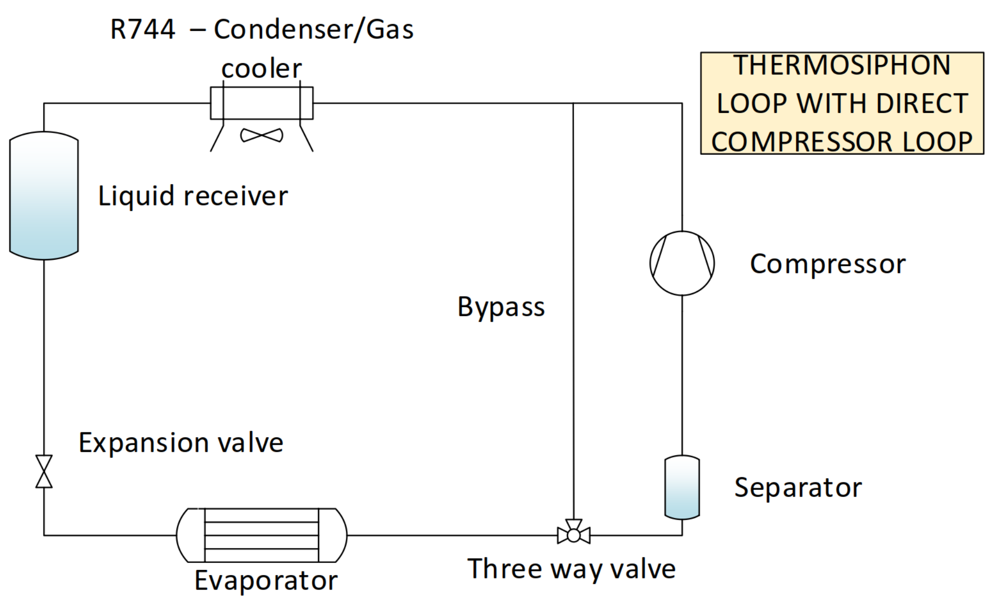

Figure 30 - *Thermosiphon loop with direct vapor compression system*

The system operates in two different modes: thermosiphon mode and vapor compression mode. In the thermosiphon mode, the expansion valve is then fully open and the three-way valve is directing the flow through the bypass. In vapor compression mode the expansion valve is set to achieve the right amount of superheat in the evaporator.

4.7.3 Indirect System

A simplified layout of the indirect system is shown in Figure 31.

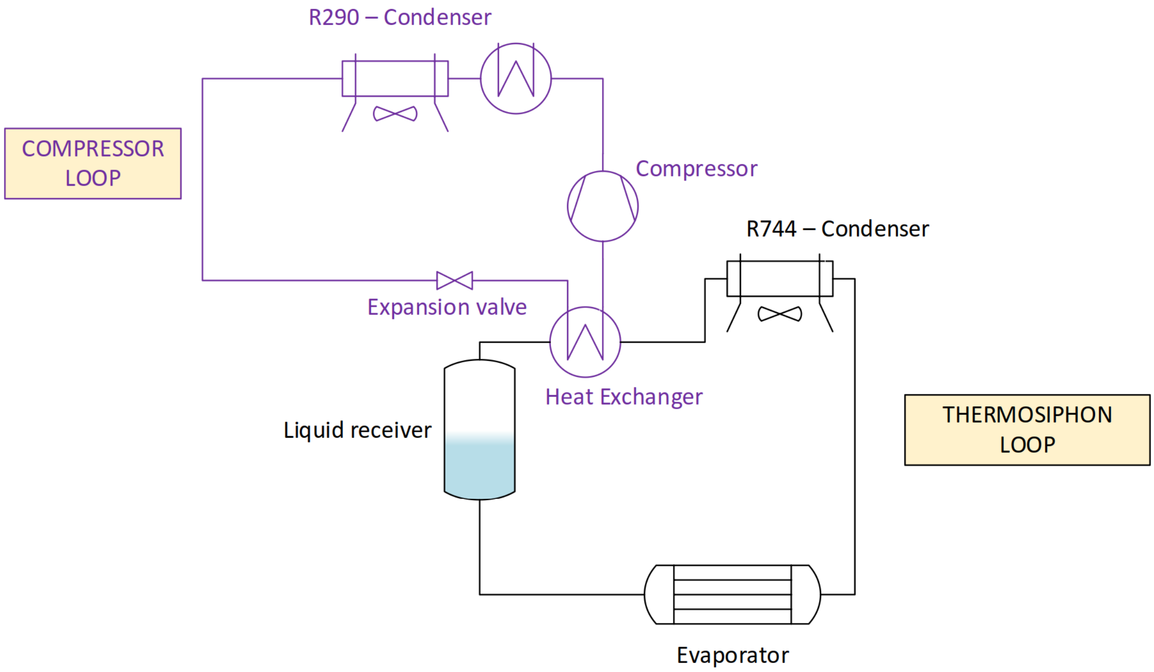

Figure 31 *- Thermosiphon loop with an indirect vapor compression system*

In the indirect system the thermosiphon loop always run as a thermosiphon. For higher ambient temperatures, the compressor loop, using R290 as refrigerant, is used for cooling the thermosiphon loop in the heat exchanger. The R744 Condenser is then bypassed.

4.8 Auxiliary Equipment

In the calculations for the key parameters the energy demand from the auxiliary equipment are taken from Figure 6. The IT-equipment demand share is equal to 200 kW as all electricity is assumed to become heat. This gives the annual energy demands for the auxiliary equipment as in Table 12.

Table 12 - Annual energy demand for auxiliary equipment

UPS	PDU	Lighting	Switchgear	Dehumidifier
223,7 MWh	111,8 MWh	74,6 MWh	37,3 MWh	111,8 MWh

4.9 Heat Recovery

As both system solutions is running with a high evaporation temperature it is possible to recover heat from the data center since the temperature lift is small. The R744 system is running in the supercritical region which means that it recovers sensible heat at a high temperature. The R290 system never reaches above its critical point which means that the temperature level is lower and that latent heat is extracted. Both the transcritical R744 system and the subcritical system is shown in Figure 32.

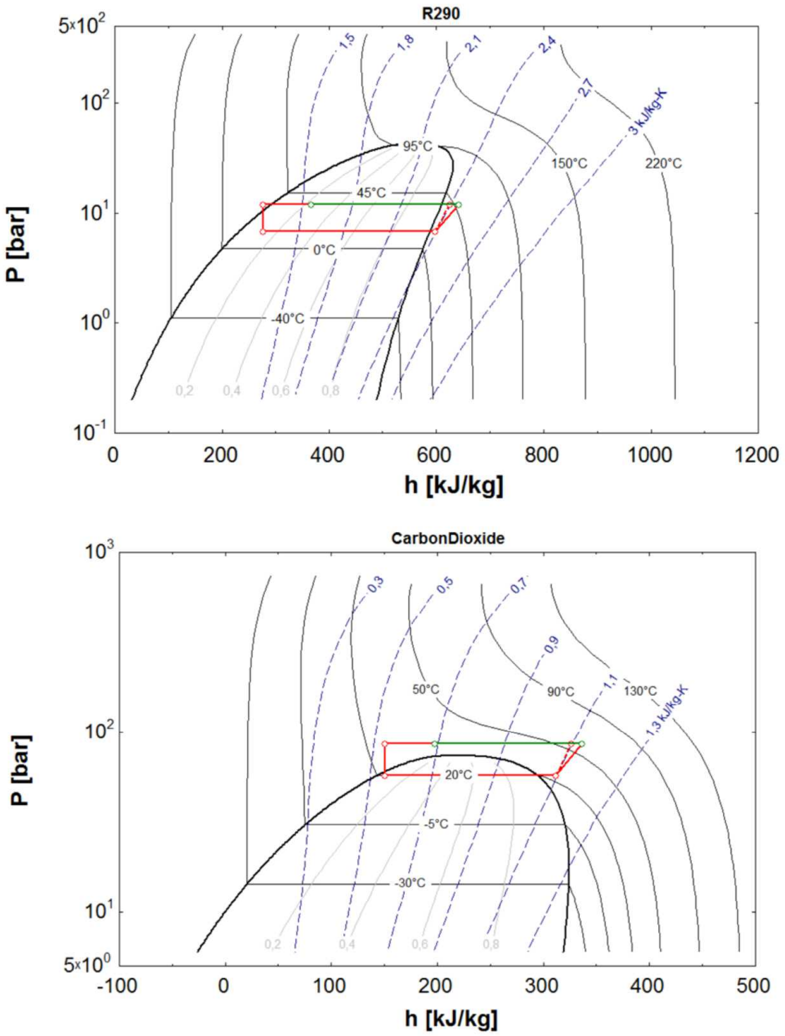

Figure 32 *- P-h diagram for subcritical cycle with R290 and transcritical cycle with R744*

The cycles in Figure 32 are shown in red with one part that is green. The green part is where the heat exchanger for heat recovery is placed i.e. before the condenser. The dotted line is for the fully ideal cycle, but as the compressor have an isentropic efficiency the solid line, which shows the real case, ends up with a higher enthalpy.

The comparison with the two systems is made only in one city, Stockholm, and the condenser size is set to 2000 m^2 fin area. To make the systems comparable the R744 system changes outlet temperature from the heat recovery exchanger according to the supply temperature to the heating system. The R290 system is controlled so that the condensing temperature is changing with the needed temperature for the heat recovery heat exchanger.

To get reasonable estimation of the needed heating demand for the model, an office at 8000 m^2 and a dimensioning heating demand of 10 W/m^2 (Boverket 2012). The dimensioning heating demand is for Stockholm at its annual temperature of 6.6˚C (Havtun och Bohdanowicz, Sustainable Energy Utilisation 2014). The total heating demand is therefore 80 kW at the annual temperature and this is to hold an indoor temperature of 21˚C (Boverket 2007). To get the heat needed at other temperatures, the total heat transfer coefficient, UA-value, of the house is calculated as

$$UA = \frac{\dot{Q}_{heating;dim}}{t_{indoor} - t_{annual}} . \tag{45}$$

The heating demand is then calculated as

$$\dot{Q}_{heating} = UA \cdot (t_{indoor} - t_{amb}) . \tag{46}$$

The heat available for heat recovery is calculated by

$$\dot{Q}_{1,HR} = \dot{m}_{refrig} \cdot (h_{1k} - h_{ahr}) , \tag{47}$$

where h_{1k} is the enthalpy after the compressor and h_{ahr} is the enthalpy after the heat recovery heat exchanger. The enthalpy after the heat exchanger is taken from refrigerant properties with the temperature t_{ahr} that is decided from a low temperature supply temperature curve as

$$t_{ahr} = t_{supply} + \Delta t_{app} , \tag{48}$$

where t_{supply} is the supply temperature to the heating system and Δt_{app} is the approach temperature. The supply temperature is determined according to Figure 33, which is a temperature curve for a floor heating system (Energijägarna 2012).

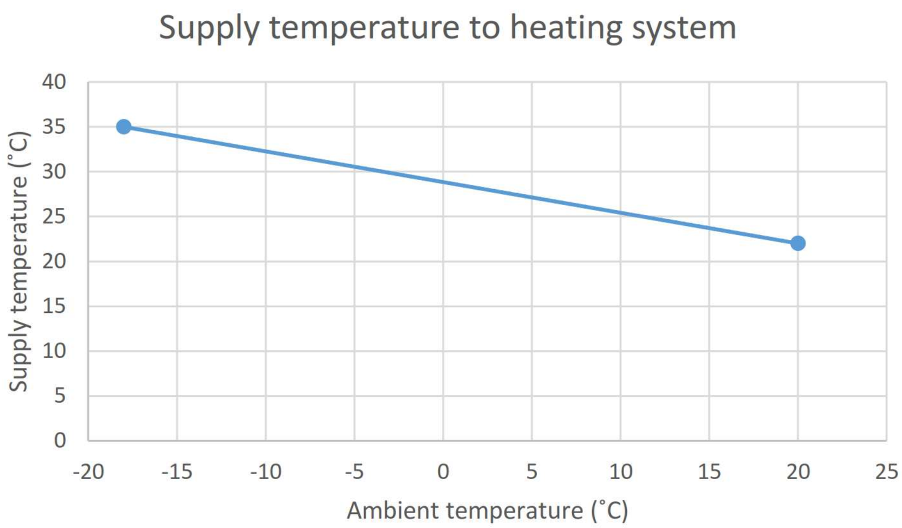

Figure 33 *- Supply temperature curve*

The curve in Figure 33 is linear approximation between two data points: supply temperature of 35˚C at an outdoor temperature of -18˚C and supply temperature of 22˚C at an outdoor temperature of 20˚C (Energijägarna 2012). The function is used in the simulations and described as

$$t_{supply} = 0.3421 \cdot t_{amb} + 28.842 . \tag{49}$$

To measure the efficiency of the system the *Seasonal Performance Factor* (SPF) is calculated as

$$SPF = \frac{\text{Annual total heat}}{\text{Annual total electricity consumption}} . \tag{50}$$

To compare the performance of the system to not using heat recovery, the SPF is calculated only for the extra electricity added to cover the heat demand for the office as in

$$SPF_{HR} = \frac{\text{Total heat}}{\text{Total electricty consumption - Total electricty consumption for cooling}} \tag{51}$$

The ERE-value is calculated according to (2)as well as the PUE-value according to (1)

Furthermore, the *Coefficient of Performance* (COP) is calculated at different temperatures and is described as

$$COP_{1,HR} = \frac{\dot{Q}_1}{\dot{E}} \tag{52}$$

5 Energy Demand and Economical Modelling

It is not only the systems physically performance that is modelled, its economic feasibility is also tested. The key parameters in the economical comparison is the total annual energy demand, the set point temperature and the condenser area.

5.1 Total Annual Energy Demand

The total energy demand is strictly dependent on the set point temperature since free cooling needs less electrical power than mechanical cooling. The total electricity demand is calculated as the sum of hours of needed electrical power for each ambient temperature

$$E_{tot} = \sum_{i=t_{\min}}^{t_{\max}} \dot{E}_i \cdot \tau_i \ . \tag{53}$$

5.2 Set Point Temperature and Condenser Area

One of the most critical limitations in the thermosiphon system is the heat transfer in the condenser. Since the ambient temperature sometimes is close to, and also higher than, the condenser temperature the system must switch between two different operating modes. This is regulated by a *set point temperature* which is the ambient temperature for when the system switches operation mode. The choice of set point temperature is directly dependent on the temperature difference in the condenser, with a smaller delta temperature the set point temperature can be higher. In this application a high set point temperature equals more hours of free cooling, but a low delta temperature in the condenser often comes with a higher investment cost. The optimum economical delta temperature is where the savings in energy bills meets the extra investment costs.

In finding the set point temperature two costs are considered. These are the annual investment cost and the annual running cost. The total annual cost is then the sum of these two and is calculated as

$$C = a \cdot I + E_{tot} \cdot \chi_E \ , \tag{54}$$

where a is the annuity factor, I is the investment cost, τ is hours running at a specific electrical power E_{tot} and χ_E is the electricity price, which is set to 0.1 € /kWh in the model. The annuity factor a is calculated as

$$a = \frac{i}{1-(1+i)^{-n}} \ , \tag{55}$$

where i is the interest rate and n is years of depreciation.

The investment cost of the condenser is a function of fin area, where a larger area has a higher investment cost. To create the function data from Alfa Laval's software *Alfaselect AIR Cas 2000* (Alfa Laval 2011) is used and from these data points, a linear fit is conducted.

When comparing the different system solutions, the evaporators, compressors and installation cost is seen as equal, which means the only investment cost that varies is the condenser depending on its size. The size of the condenser is key parameter in determining the set point temperature.

To find the economical size of the condenser, which also decides the set point temperature, the total cost is plotted for fin areas of 1500, 2000, 3000, 4000 and 5500 m^2. The set point temperature for the different condensers is determined according to chapter 6.4.

6 Results

In this part of the report the results from according to the modeling chapter 5 is presented. It includes all the partial results e.g. temperature profiles and also the final results where the performance of the new systems is presented compared to the reference system.

Where it is not written specifically the standard fin area is 2000 m² with 6 fans. This is calculated from the condenser optimization in 6.4.

6.1 Condenser Pressure Drop

To calculate the condenser pressure drop the air velocity through the condenser finned coils needs to be calculated. This is empirically derived through finding the equilibrium between the calculated fan power and the given fan power from Alfa Laval. The equilibrium was found when the ratio α_{area} between the fan area and condenser face area was equal to 1.8.

α_{area} is used to empirically derive a correlation between the air velocity and the air pressure drop. Figure 34 shows the frictional pressure drop over the finned coils in the condenser for different air velocities. A curve fit was examined with a 2nd degree polynomial to calculate pressure drop as a function of air velocity.

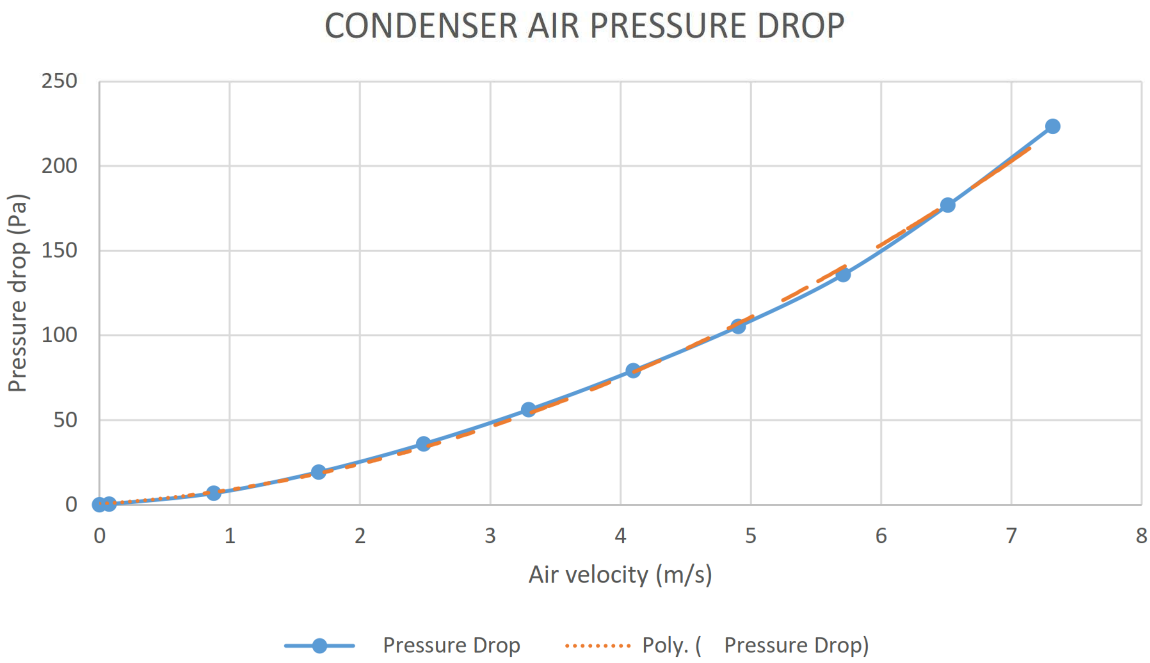

Figure 34 *- Frictional pressure drop in condenser with curve fit*

The derived equation for the curve fit is

$$\Delta p_f = 3,4256u^2 + 4,9003u + 0,6436 . \tag{56}$$

Since the pressure drop is flow dependent a velocity at zero should have a pressure drop at zero. With only 11 data points, an accuracy of 5 significant figures are unlikely. The used correlation was then truncated to

$$\Delta p_f = 3.4u^2 + 4.9u . \tag{57}$$

The difference in pressure drop between (56) and (57) is at maximum 2 Pa or 0.9 % which will have a low impact on the final result.

6.2 Evaporator Outlet Quality

To investigate the influence of the design parameters on the evaporator outlet quality the following key parameters were variated:

- Down comer height
- Down comer pipe diameter
- Return line pipe diameter
- Evaporator pipe diameter
- Amount of evaporator bends
- Amount of evaporator circuits

Figure 35 shows the evaporator outlet quality as a function of pipe diameter for different down comer heights. An evaporator outlet quality closer to 1 equals a maximization of absorbed latent heat.

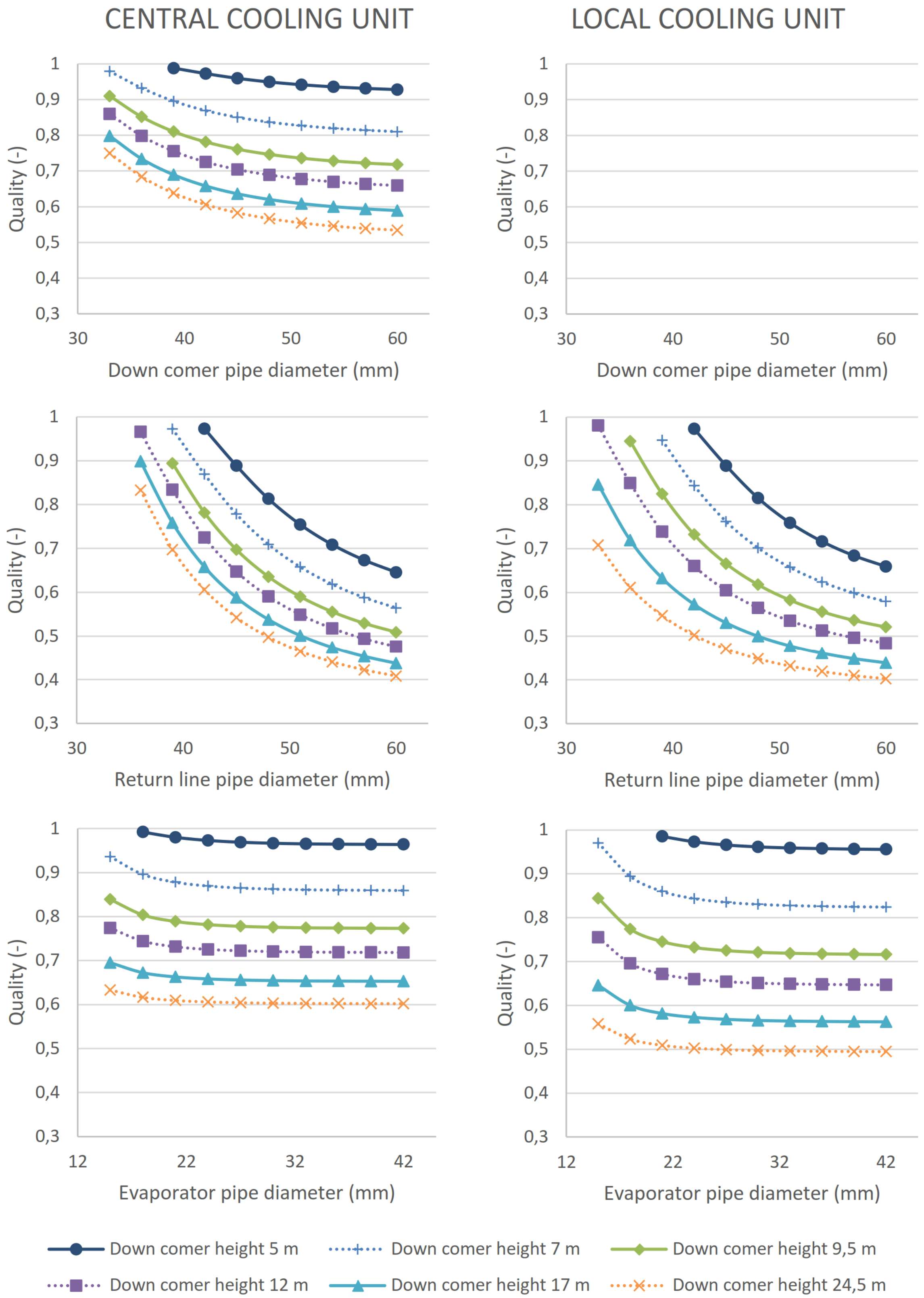

Figure 35 *- Influence on quality of pipe diameter and down comer height*

It is seen that the pipe diameter impacts the quality more for narrower diameters. In the evaporator the quality lines straighten out indicating an independency of pipe diameter for wider diameters. With increased down comer height this occurs at narrower pipe diameters. For the down comer and the return line, the same behavior is spotted but, for the given range, not as clearly. From Figure 35 it is also seen that the diameter of the return line influences the evaporator outlet quality the most. Generally, the local cooling configuration has somewhat lower quality for the same pipe diameter and the evaporator outlet quality decreases with increased down comer height.

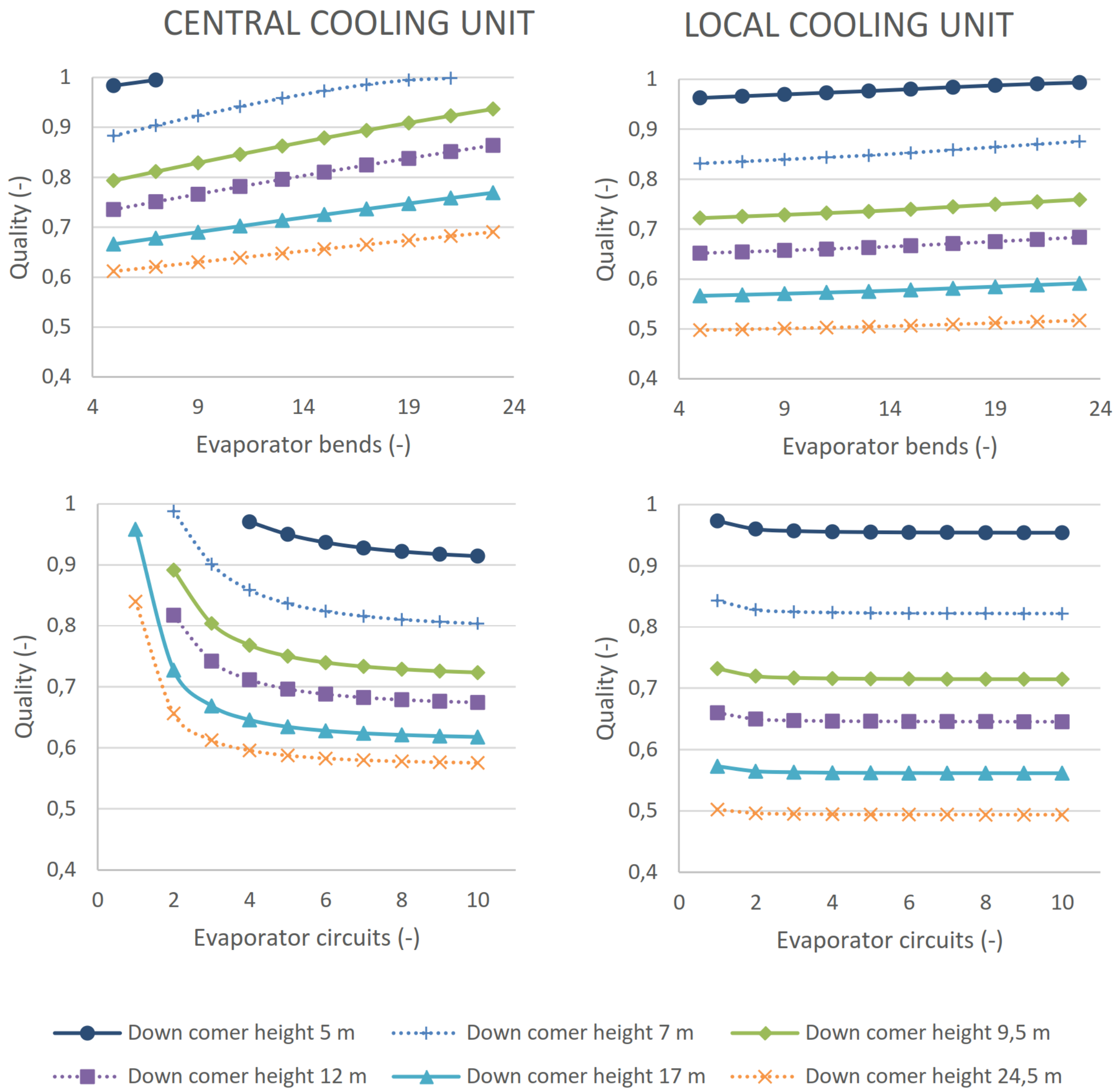

Figure 36 *- Influence on quality of amount of evaporator bends and circuits*

The same trend is seen in Figure 36 where an increase in evaporator circuits decreases the frictional pressure drop since the tube length shortens. An increase in amount of evaporator bends linearly increases the pressure drop in bends and the frictional pressure drop since the evaporator tube length is extended. The variance in evaporator outlet quality caused by the variated parameters in Figure 36 are comparatively small to the pressure drop related to the down comer and rising tube diameter.

When placing the whole system in the server room the evaporator outlet quality is as Figure 37.

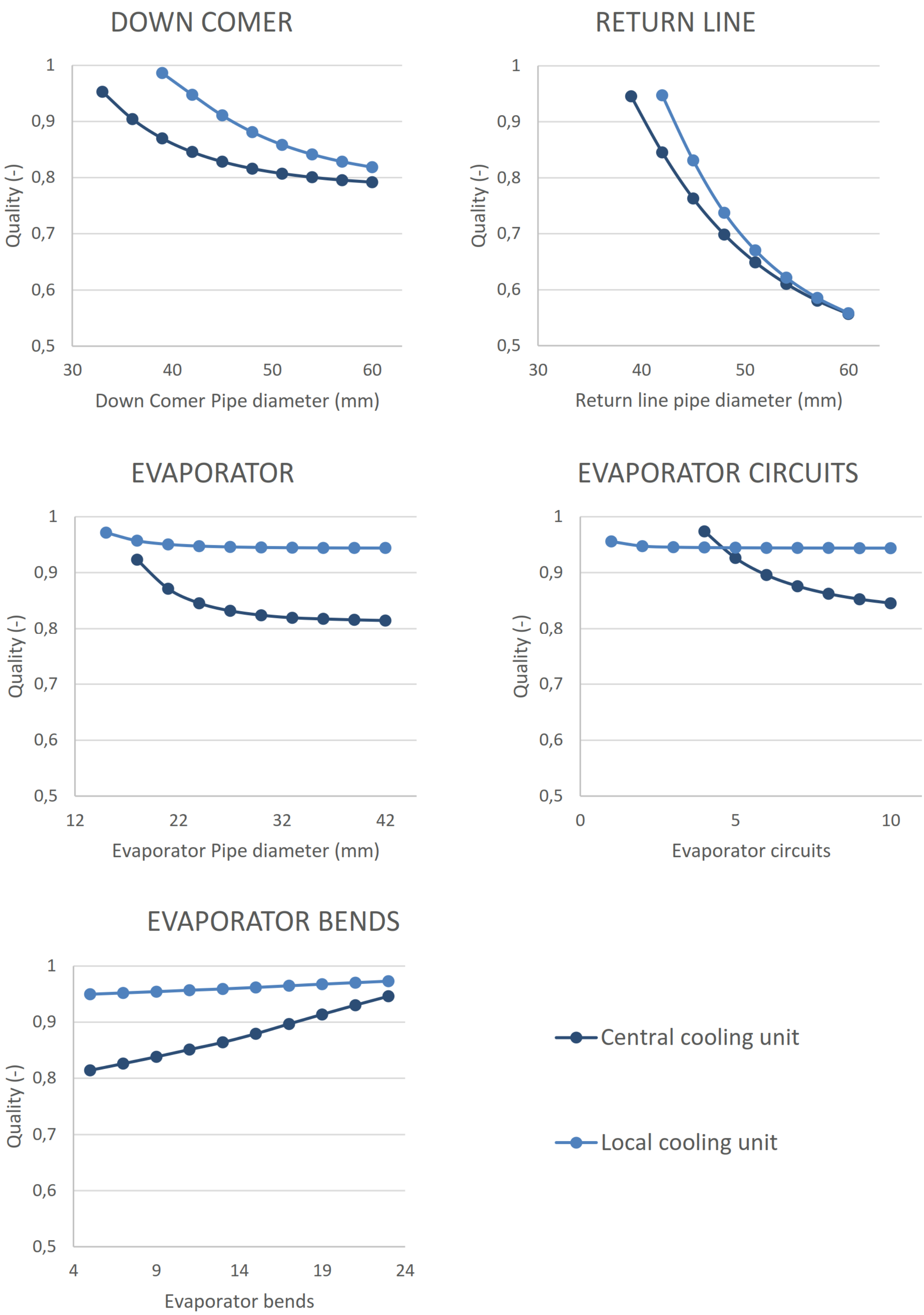

Figure 37 - *Evaporator outlet quality for entire system placed in server room*

The notable thing in Figure 37 is that a system with local cooling units is less affected of the evaporator design. As in Figure 35 and Figure 36 the return line and down comer diameter impacts the outlet quality the most. Further it should be noticed that the absolute values between the systems with the condenser on the roof and the systems with the condenser on the same level are not comparable since they do not have the same piping layout as seen in Figure 35, Figure 36 and Figure 37.

6.3 Climate Profiles

In Figure 38 the amount of hours for certain temperatures is presented.

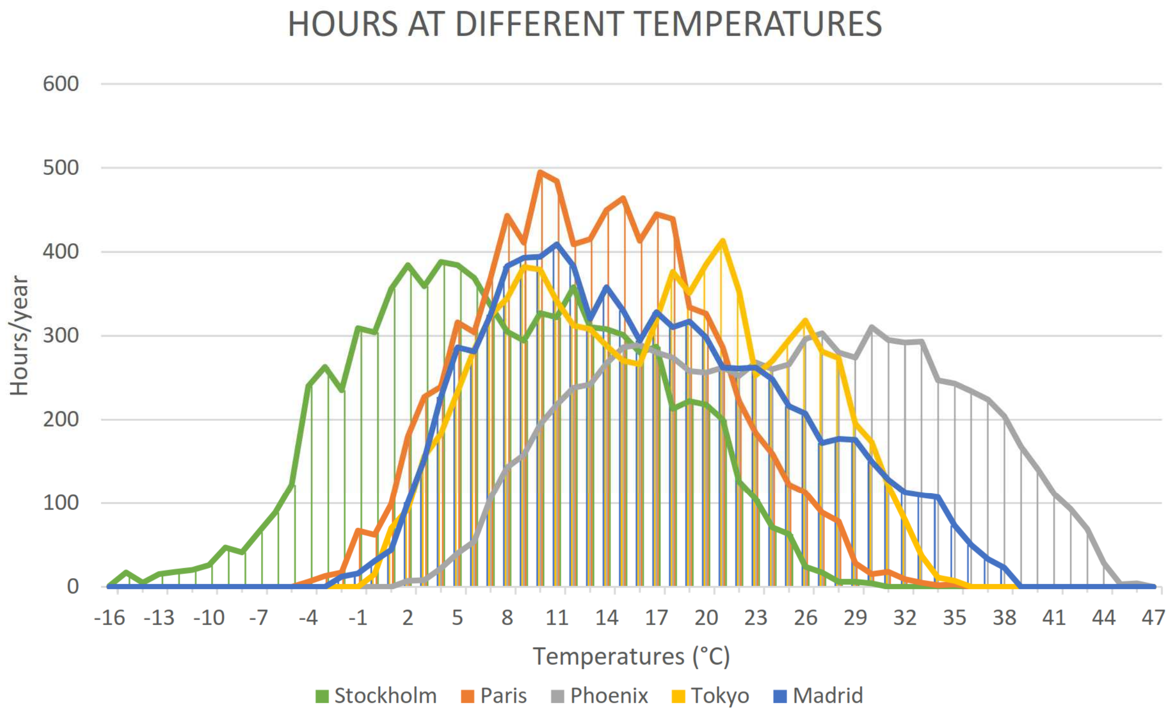

***Figure 38** - Temperature profiles for Stockholm, Paris, Phoenix, Tokyo and Madrid*

Paris and Madrid peaks around the same temperature with somewhat longer plateau for Paris with steeper sides, i.e. a shorter temperature range, than Madrid which has a flatter decrease from its peak value. Tokyo has two significant peaks at 9-10°C and 21°C and its sides has inclinations similar to Paris. Stockholm has all through lower temperatures than the other cities and has, compared to them, its curve offset to the left. Phoenix has the highest temperatures and also the widest temperature range. A wider temperature range requires a more complex design, e.g. different operation modes, since it must cover more operational conditions.

Figure 39 shows the relative humidity profiles for the cities.

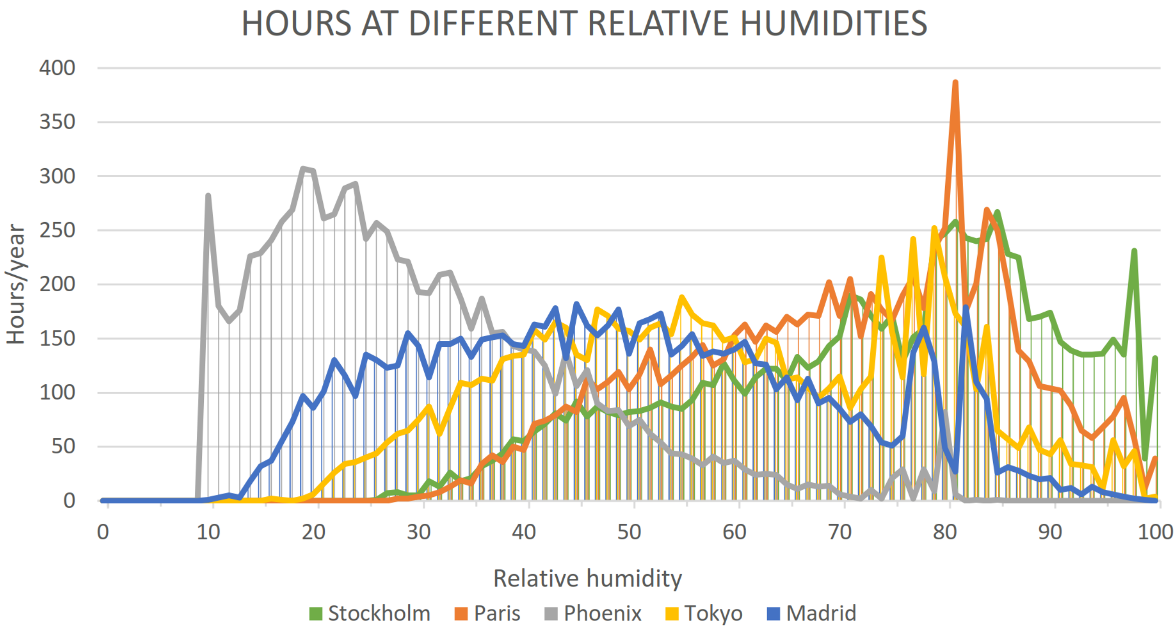

Figure 39 *- Relative humidity profiles for Stockholm, Paris, Phoenix, Tokyo and Madrid*

Both Paris and Stockholm have their peak at higher relative humidity compared to Phoenix which has an opposite curve shape with the hour density at lower relative humidity. Tokyo and Madrid has similar shapes with a longer range of frequently appearing relative humidity although Madrid has lower values of relative humidity with a steeper inclination and a longer plateau than Tokyo. Both cities have additional peaks at higher relative humidity.

The wet bulb temperatures for the five cities is presented in Figure 40.

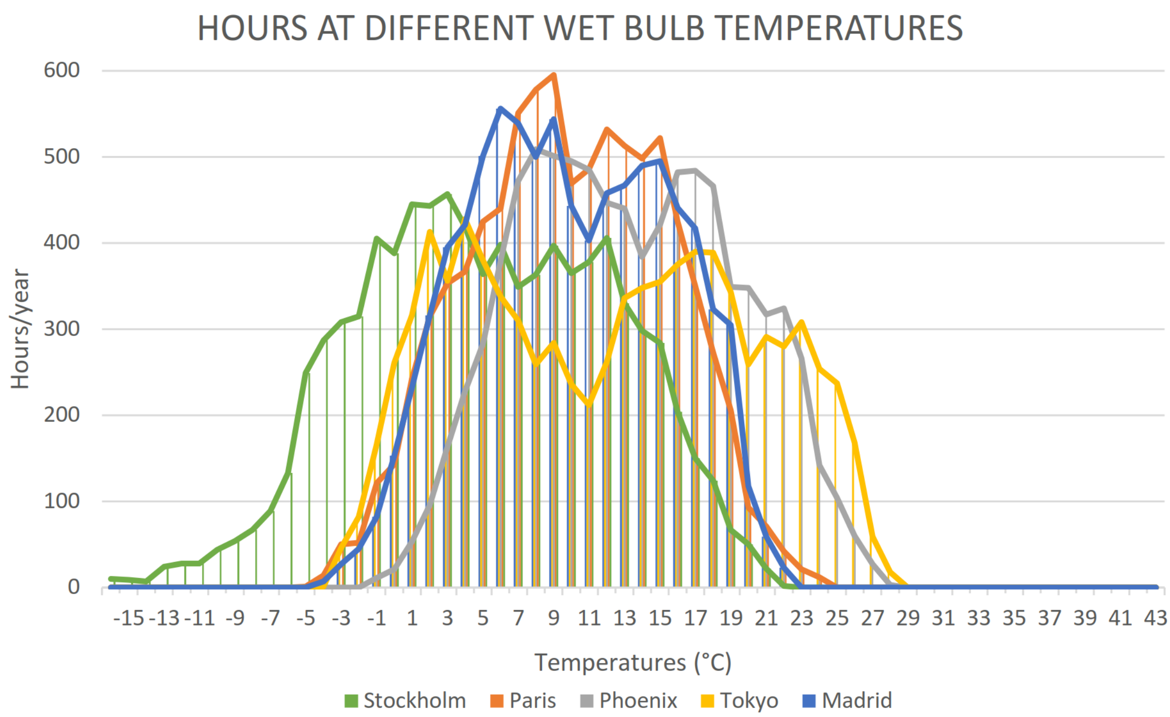

Figure 40 *- Wet bulb temperature profiles for Stockholm, Paris, Phoenix, Tokyo and Madrid*

The thermosiphon application has a set point temperature for which it switches from utilizing free cooling to mechanical cooling. The higher the set point temperature is; the more hours of free cooling are available. If the system uses evaporative cooling the wet bulb temperatures is the governing temperature for the regulation of the different operation modes. In Figure 41 the share of hours per year above and below three different set point temperatures is shown. 100 % percent is equal to 8760 hours.

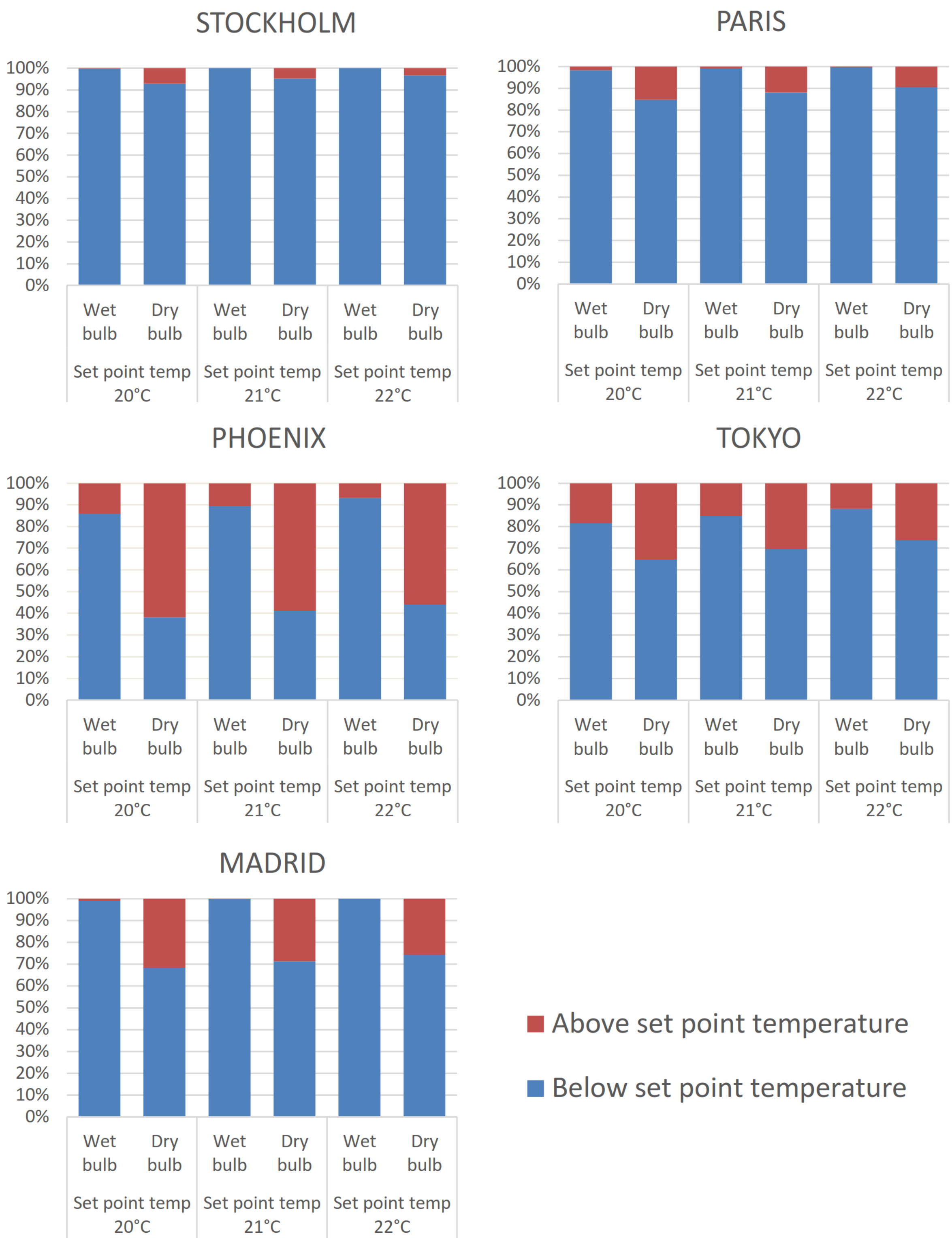

Figure 41 - Share of available free cooling hours

The values are given in Appendix A – Available free cooling hours for dry and wet bulb temperatures

To further visualize the difference in hours, Figure 42 shows the difference in hours per year for the cities for the three set point temperatures.

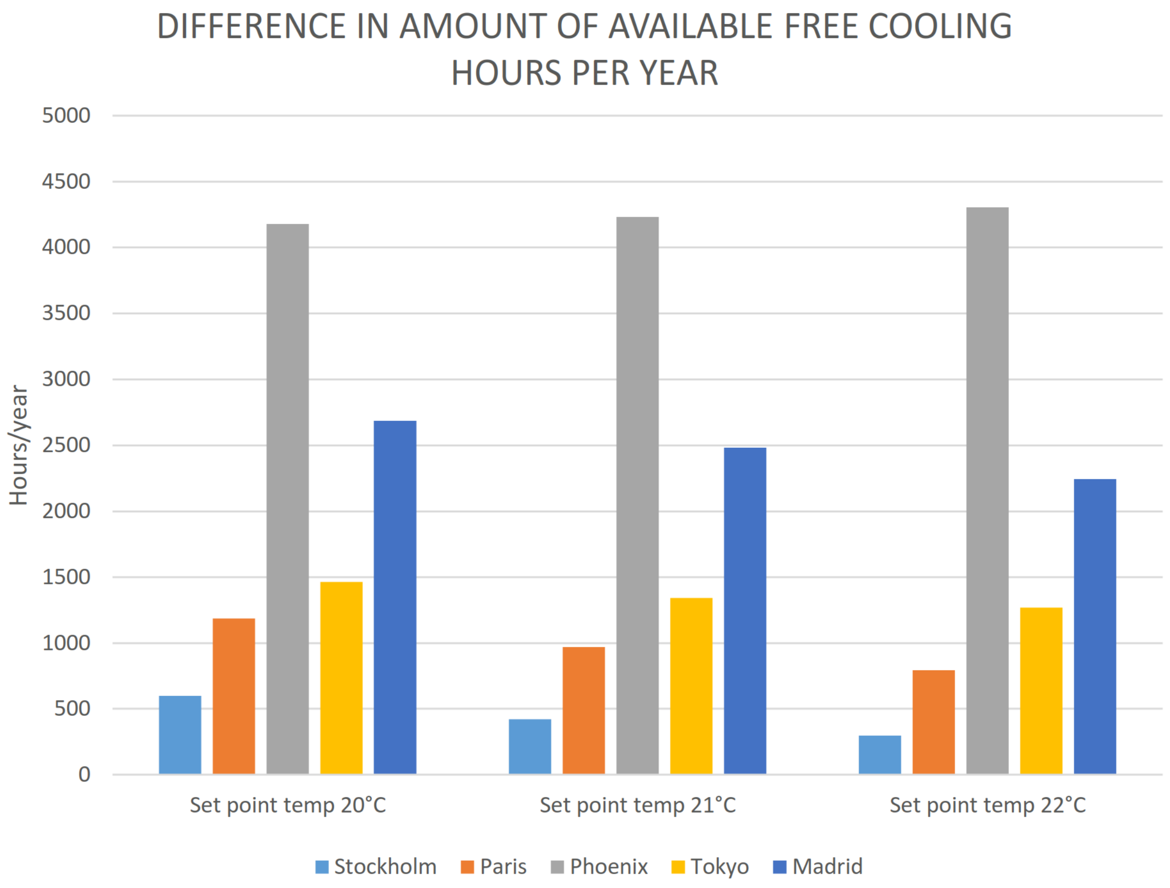

Figure 42 *- Difference in amount of available cooling hours/year between evaporative and non-evaporative cooling*

From Figure 42 it is obtained that Phoenix and Madrid would benefit from evaporative cooling while it would have a lower impact for Tokyo, Stockholm and Paris. The trends are that for Phoenix, Tokyo and Madrid the difference increases along with increased set point temperature but for Stockholm and Paris there are the opposite way. The trends is further identified when looking at a longer range of set point temperatures as shown in Figure 43.

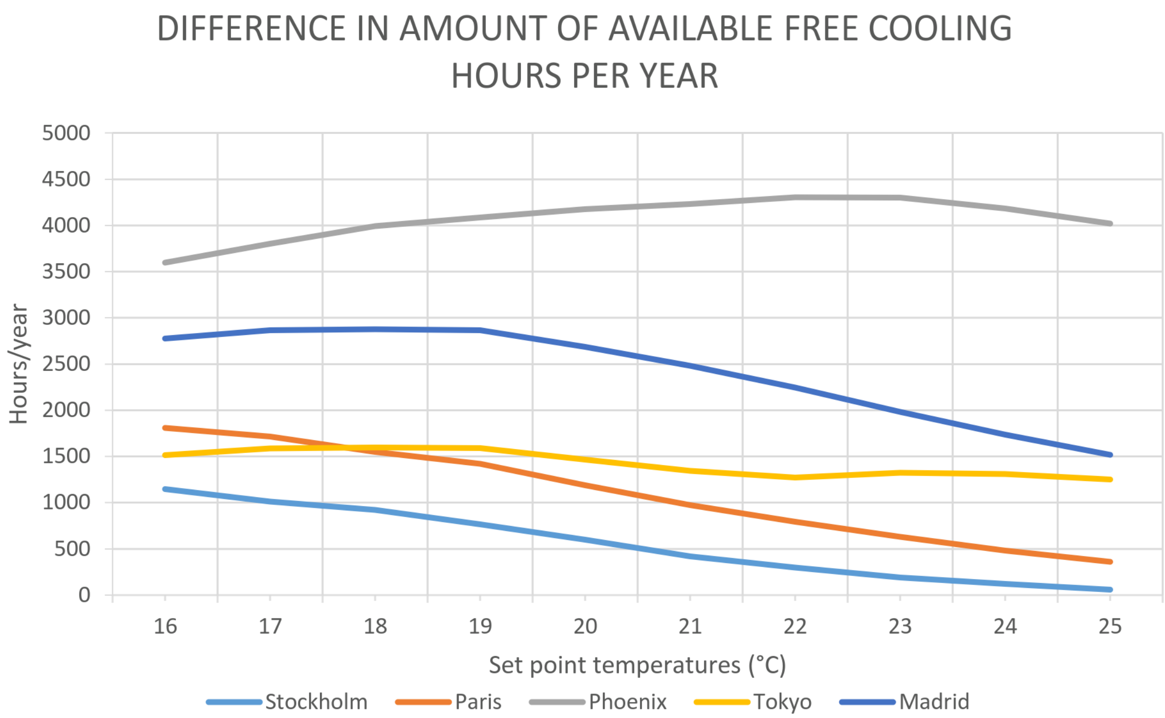

Figure 43 - Available free cooling hours per year as a function of set point temperature

For the chosen temperature range Phoenix, Madrid and Tokyo have their peak values contained while Paris and Stockholm are strictly decreasing. The set point temperature for a peak value is the set point temperature for when evaporative cooling will have the most effect compared to non-evaporative cooling.

6.4 Energy Demand and Economical Findings

The investment cost function for several fin areas is shown in Figure 44 with a linear interpolation from the data points according to Figure 44.

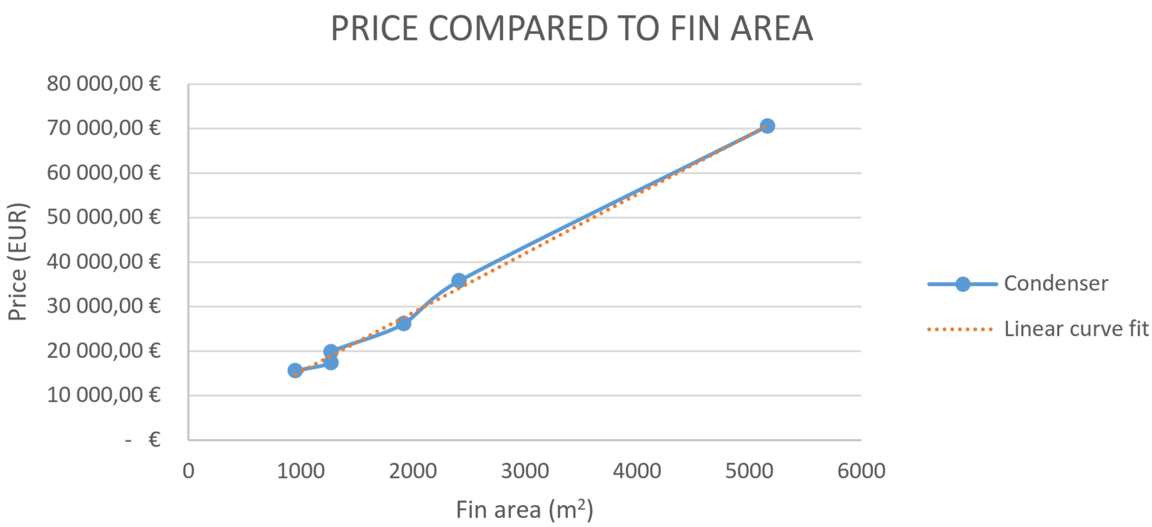

Figure 44 - Price compared to size of the condenser

The function of the linear interpolation in Figure 44 is

$$I = 13.289 \cdot A_{fins} + 2070.2 \tag{58}$$

where A_{fins} is the area of the fins that is exposed to the flowing air through the condenser. The function (58) is used in the simulation program to calculate the price of the condenser.

To find the set point temperature of the different sizes of condensers, the fan power along with the total power for the systems, are plotted in Figure 45.

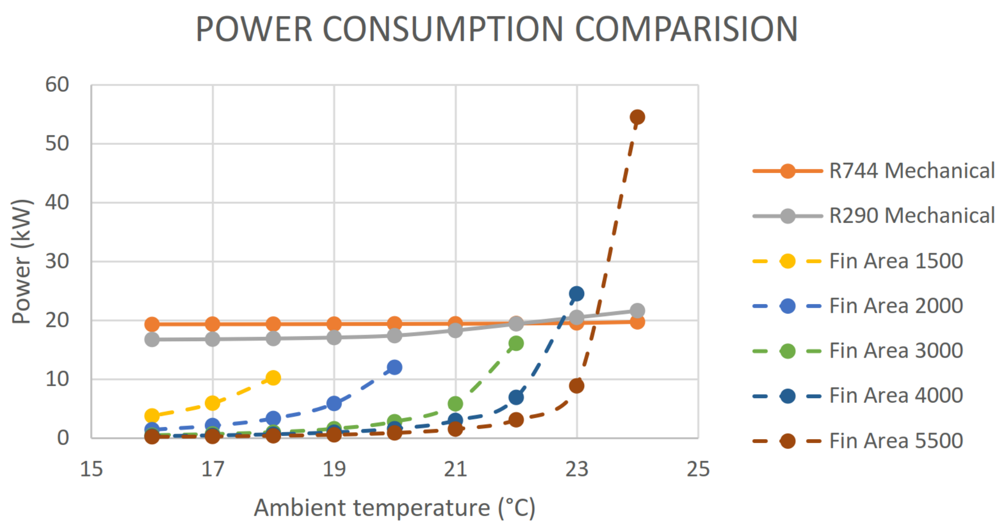

Figure 45 - Fan power consumptions at different ambient temperatures compared to total power of compressor system with a 2000 m2 condenser¨

Because of the limitations in air mass flow, maximum 9.1 kg/s, the curves for 1500, 2000 and 3000 m² does not go any further even though the fan power is lower than the compressor work for the R744 system. The set point temperatures that is used in the cost analysis are found in Table 13.

Table 13 - Set point temperatures for the two systems with a thermosiphon

Fin area	**R744 Direct system**	**R290 Indirect system**	**Number of fans**
1500 m²	18°C	18°C	4 pcs
2000 m²	20°C	20°C	6 pcs
3000 m²	22°C	22°C	10 pcs
4000 m²	22°C	22°C	12 pcs
5500 m²	23°C	23°C	16 pcs

The electricity demand, using these set point temperatures, for the three system is shown in Figure 46.

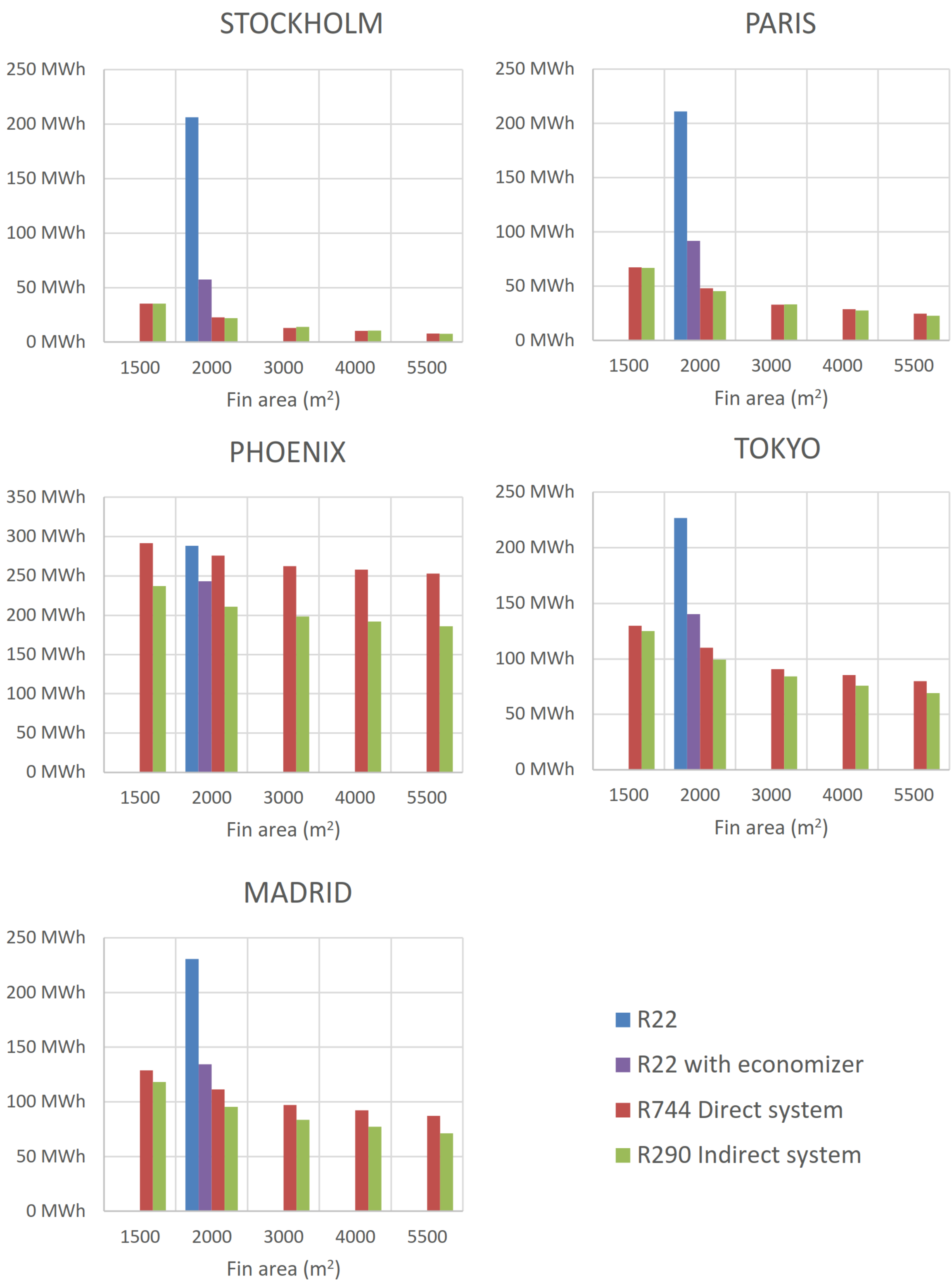

Figure 46 *- Electricity demand for the systems in different cities*

The energy demands are compared in Table 14.

Table 14 - Percentage of decreased energy demand for the R744 and R290/R744 systems compared to the R22 systems

City	**R744 system compared to R22 system**	**R290/R744 system compared to R22 system**	**R744 system compared to R22 system with economizer**	**R290/R744 system compared to R22 system with economizer**
Stockholm	88%	88%	61%	62%
Paris	77%	77%	49%	50%
Phoenix	5%	26%	-11%	13%
Tokyo	51%	55%	23%	29%
Madrid	52%	57%	20%	29%

The total annual cost for the different systems and temperature profiles are shown in Figure 47.

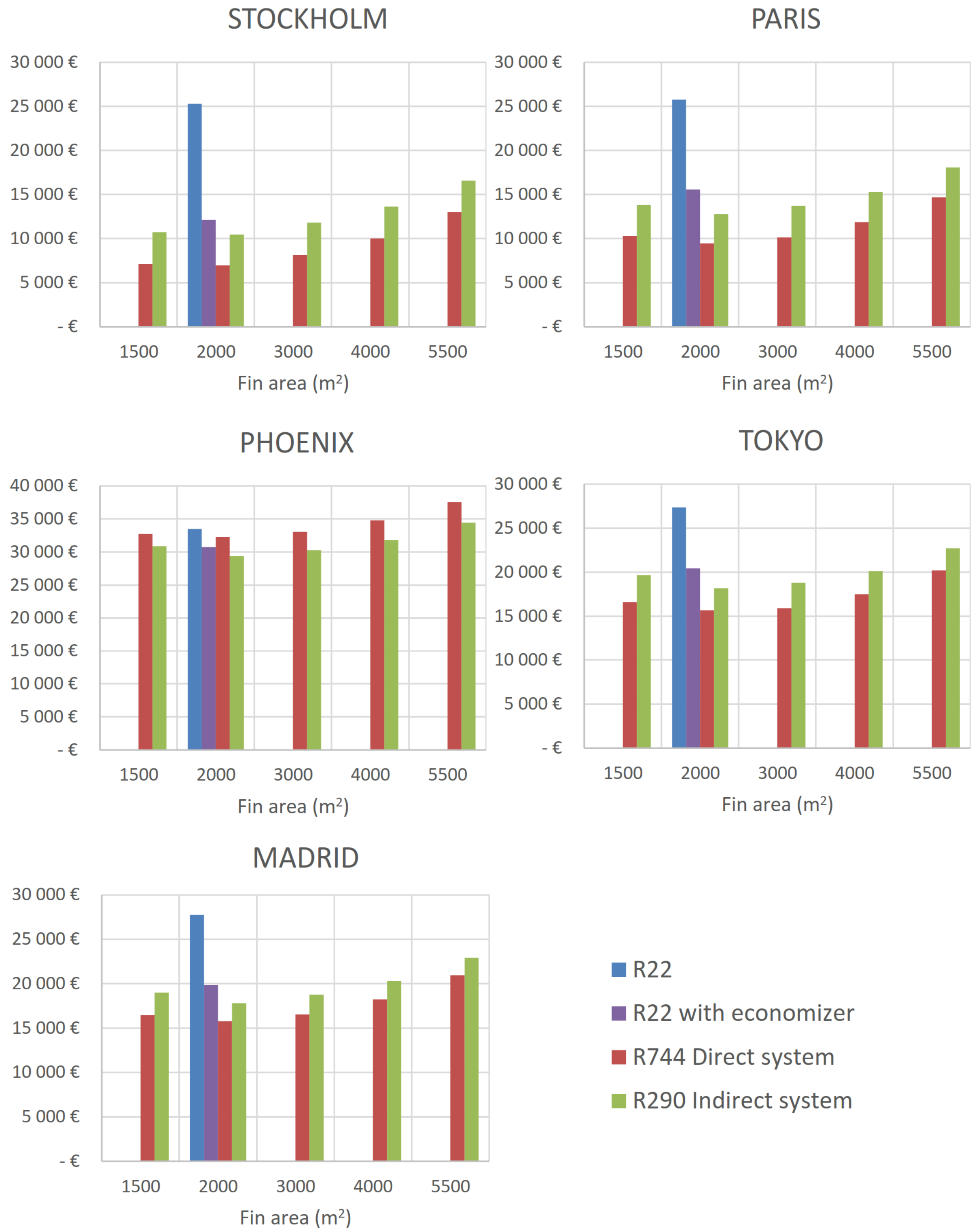

Figure 47 - Total annual cost for the cities for the three different system solutions

The plots in Figure 47 are calculated with an interest rate of 10% and 10 years of deprecation. It should be observed that the R290 needs two condensers and therefore has a higher investment cost which makes the system less cost effective. In Figure 46 and Figure 47 the reference systems are only calculated for a fin area of 2000 m^2. The total annual costs are compared in Table 15.

Table 15 - Percentage of decrease in total annual costs for the R744 and R290/R744 systems compared to the R22 systems

City	**R744 system compared to R22 system**	**R290/R744 system compared to R22 system**	**R744 system compared to R22 system with economizer**	**R290/R744 system compared to R22 system with economizer**
Stockholm	69%	52%	42%	11%
Paris	60%	44%	39%	14%
Phoenix	5%	8%	-2%	2%
Tokyo	41%	28%	25%	8%
Madrid	41%	30%	22%	8%

In Phoenix the R744 and R290/R744 systems performs worse compared to other cities. This I s due to the many hours of high ambient temperatures. To be able to utilize more hours of free cooling, evaporative cooling is used. The reduction in annual energy demand and total annual costs are show in Table 16. The annual energy demand and the total annual costs for the five cities are fund in Appendix A – Available free cooling hours for dry and wet bulb temperatures

Table 16 - Percentage of decrease in annual energy demand and total annual costs for Phoenix using evaporative cooling

	R744 system compared to R22 system	**R290/R744 system compared to R22 system**	**R744 system compared to R22 system with economizer**	**R290/R744 system compared to R22 system with economizer**
Annual energy demand decrease	79%	79%	53%	53%
Total annual cost decrease	61%	44%	41%	15%

6.5 Power Usage Effectiveness

The PUE-values for the different systems are presented in Figure 48. The PUE was calculated both for the system in total and only considering power for cooling and IT-Equipment. This due to that the cooling demand was the treated part but also that the energy demand for the other parts is constant for all locations.

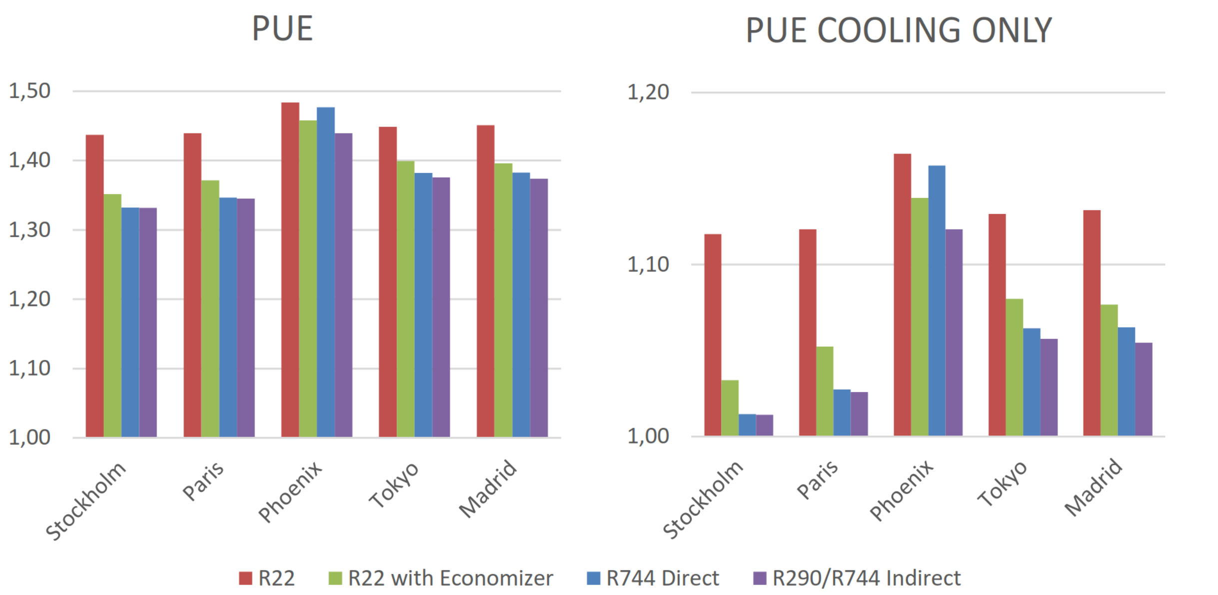

Figure 48 *- PUE values for the different systems at various locations*

The PUE-values from Figure 48 in table form are presented in Appendix C - Energy distribution. The PUE-values are compared in Table 17.

Table 17 - Percentage of decrease in PUE-value for the R744 and R290/R744 systems compared to the R22 systems

City	**R744 system compared to R22 system**	**R290/R744 system compared to R22 system**	**R744 system compared to R22 system with economizer**	**R290/R744 system compared to R22 system with economizer**
Stockholm	6%	6%	1%	1%
Paris	5%	5%	2%	2%
Phoenix	0%	2%	-1%	1%
Tokyo	4%	4%	1%	1%
Madrid	4%	4%	1%	1%

The PUE-values when only cooling is consider is compared in Table 18.

Table 18 - Percentage of decrease in PUE-values when only cooling is considered for the R744 and R290/R744 systems compared to R22 systems

City	**R744 system compared to R22 system**	**R290/R744 system compared to R22 system**	**R744 system compared to R22 system with economizer**	**R290/R744 system compared to R22 system with economizer**
Stockholm	8%	8%	2%	2%
Paris	7%	7%	2%	2%
Phoenix	1%	3%	-1%	1%

Tokyo	5%	5%	1%	2%
Madrid	5%	5%	1%	2%

The energy distributions for the systems placed in Stockholm are shown in ***Figure 49***. The other cities has similar shape and are found in Appendix C - Energy distribution Compared to Figure 6 the cooling share is much smaller, even for the R22-reference systems.

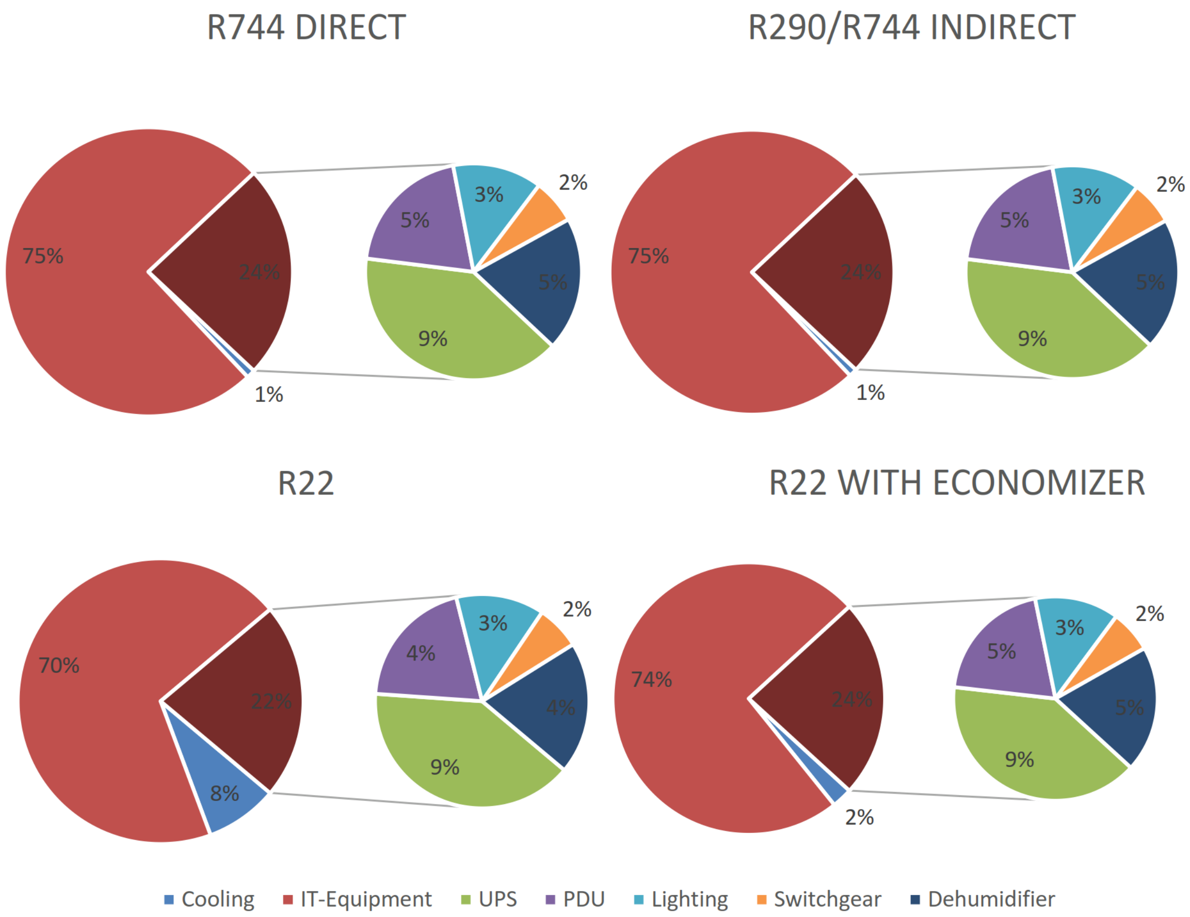

Figure 49 - *Energy distribution for the four systems if they were to be placed in Stockholm*

6.6 Heat Recovery

The available amount of heat recovery in both systems is shown in Figure 50.

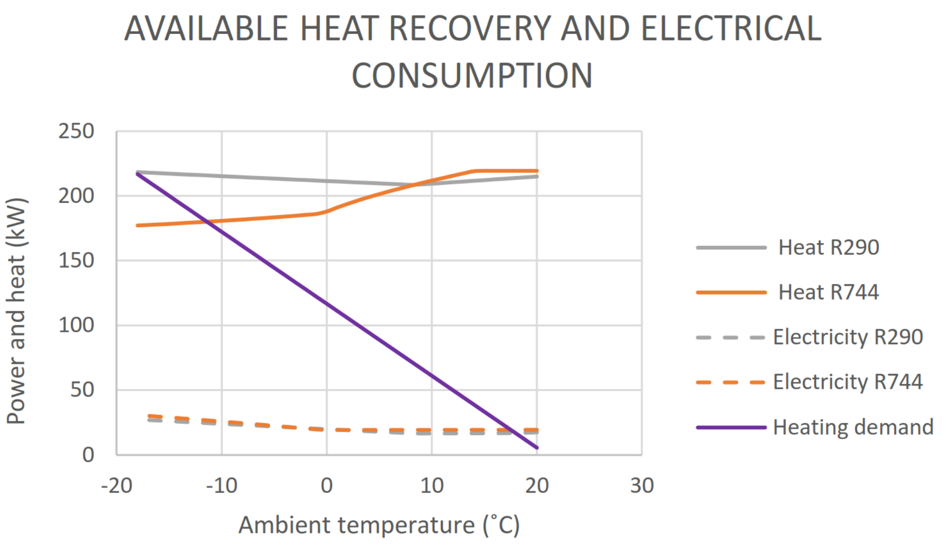

Figure 50 - Available heat recovery

As seen in Figure 50 the available heat in the systems differ from each other caused by the different operational modes, i.e. subcritical and transcritical.

Both system in these configurations is working with a high evaporation temperature which makes the vapor compression cycle efficient. $COP_{1,HR}$ for the systems are plotted in Figure 51 as function of the ambient temperature.

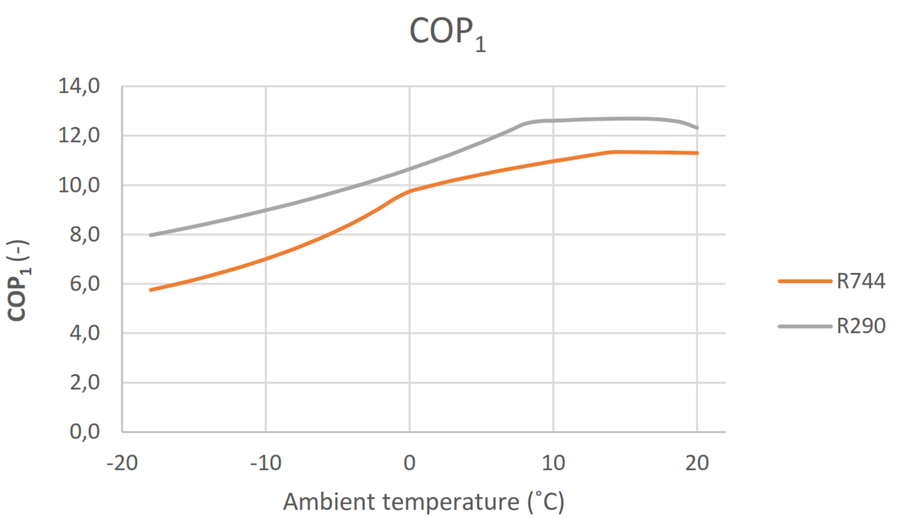

Figure 51 - Heating coefficient of performance for the two system solutions

In Figure 51 it is seen that the R290 solution works better at all temperatures due to its better refrigerant properties.

The maximum possible area of the office that can be heated is shown in Figure 52, both with a 100% and 70 % peak power coverage.

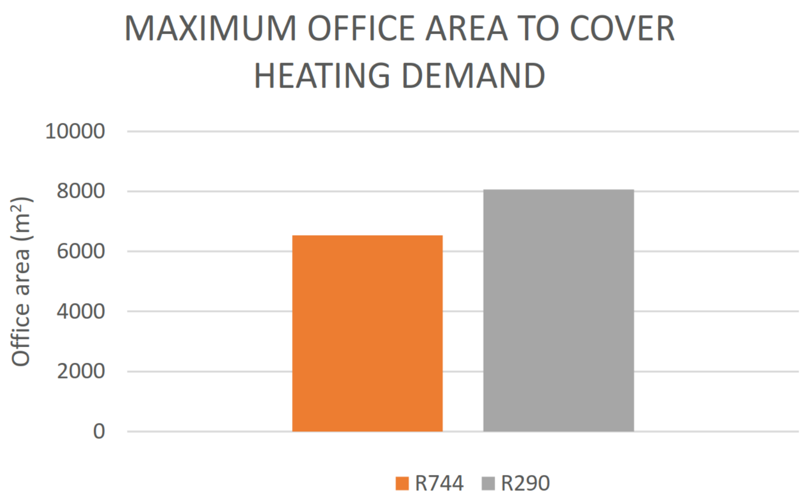

***Figure 52** - Maximum possible heating area to cover 100% of the power demand and 70% of power demand*

As seen in Figure 52 the maximum size of an office the two system can cover differs as the R290 system works more efficient. The R744 system cannot cover the whole demand for an office of 8000 m^2 that was used in the model, and the rest has to be covered with pure electricity. The results of the performance are shown in Table 19. The use of heat recovery forces the systems to consume additional amount of energy. The heating system only demands this additional energy since the operation of cooling system already requires the rest of the energy of the total energy demand. To determine the SPF of the systems when only consider this additional amount of energy the SPF_{HR} is used. The PUE and ERE-values are calculated with a condenser fin area of 2000 m^2.

Table 19 - Seasonal performance factors with and without heat recovery and ERE- and PUE-values

	R744	**R290**
SPF	8.3	8.7
SPF_{HR}	11.0	15.2
ERE	0.677	0.675
PUE	1.044	1.042

7 Discussion

In this work only air free cooling applications where examined. The choice of using refrigerant to transport heat from the data center to the ambient, and not use outdoor air directly, was primarily based on: 1) possible retrofit in older systems and 2) applications using direct ambient air have higher cooling demand than the modelled system, above 1000 kW compared to 200 kW.

In the calculations evaporative cooling is treated as non-energy consuming. In practice this is not completely true since there is need of pumping power to transport water to and spray it over the condenser fins. The heat transfer also increases with evaporative cooling and the results may therefore show a too high energy demand and annual cost.

There is also a water demand that has to be considered, often the areas that benefits most from evaporative cooling have limited access to water. Governmental policies etc. may have restrictions or fees that makes the usage too expensive. Another drawback with evaporative cooling is the increased maintenance need, impurities in the water can cause corrosion and fouling. The first one leads to replacement of components and possible system downtimes whereas the second decreases the heat transfer coefficient. The use of water also makes the system dependent on Mother Nature since evaporative cooling demands temperatures above 0°C to prevent freezing damage and are therefore unsuitable at places where the temperature may fall quickly below or that varies around 0°C.

Several correlations have been used in order to simplify the calculations but preserve a dynamic model. Since the final system design is unknown and then having a too detailed model could have a misleading effect. Some correlations (the U-value for the condenser) are taken from other sources where they are claimed to be good for approximation purposes. Some correlations (condenser pressure drop and cost by fin area) are derived through curve fitting from given data points.

The expression for cost by fin area are a linear function which is reasonable since the condenser is purchased in modules, where two modules has twice the area and capacity as one. From Figure 44 a doubling in fin area corresponds to a doubling in price.

One major purpose was to find alternatives for commonly used refrigerants. All modern refrigerants have zero ODP and low GWP making the application the governing quantity. From Table 2 it is given that most of the natural refrigerants and HFO's are more or less flammable. The less flammable R717 has other drawbacks as it is poisonous and corrosive. The only one considered as not flammable, poisonous or corrosive, is R744. Since data centers requires high reliability, R744 is the only refrigerant that could be used in a direct system. The other natural refrigerants need to be used together with a secondary system e.g. a brine loop that provides cooling of the air in the server room. Besides that, a leakage of the secondary working fluid may be hazardous to the servers.

A secondary brine loop also brings a temperature loss as there will be a temperature difference in the heat exchanger, resulting in a lower evaporation temperature for the vapor compression cycle. This can result in a less efficient vapor compression cycle depending on the refrigerant. When comparing the proposed systems, the direct system with R744 as refrigerant will not have these kinds of temperature losses, but the refrigerant properties are not very good if looking at the efficiency. This means that the indirect system with R290 will work even more efficient even though the evaporation temperature is lower, caused by the temperature difference in the heat exchanger. In both cases the free cooling will be used to the same set point temperature. If only looking at the energy use the R290 system is the most efficient for all temperature profiles, but if looking at the cost the R744 system is the cheapest for all temperature profiles except Phoenix where the climate is very hot.

Due to the higher pressure in R744 applications, fittings etc. must have higher reliability to prevent the increased risk of leakages. In case of leakage of R744 the electronic equipment would not take any harm since the fluid would vaporize immediately in the surrounding conditions (atmospheric pressure and room temperature). Although, if there is a leakage, the personnel working in the environment could suffer from asphyxia and therefore a proper warning and ventilation system needs to be installed as well.

A higher evaporator outlet quality means that more latent heat has been absorbed by the refrigerant, but also that the system is closer to its maximum latent cooling capacity. A heat load beyond that will have an additional sensitive heating resulting in a rise in temperature and consequently superheating the gas.

The decrease in evaporator outlet quality in Figure 35 and Figure 36 comes from the decrease in pressure drop for wider pipe dimensions since the inertia of the flow dominates the friction forces. A reduction in pressure drop allows the refrigerant to circulate faster which gives a smaller enthalpy increase in the evaporator. Since the quality is dependent on temperature and enthalpy and the temperature is constant for all cases the reduction in enthalpy gives a direct reduction in evaporator outlet quality.

The design parameters can be designed in several ways, with some limitations. Generally narrower pipe diameters cause a too big pressure drop since the viscous forces then overrules the dynamic forces. For a local cooling unit system, the minimum down comer height is 2 m since lower heights requires a reduction of evaporator height as well, i.e. decreasing the height of the server racks. A system with a central cooling unit can have a lower height than the server racks since the evaporator height is not related to the server racks. For low down comer heights the head pressure gets too low in order to balance the pressure drops in the system, giving a too slow mass flow. As a consequence, the refrigerant would not only be heated by latent heat but sensitive heat as well, i.e. superheating. Since the thermosiphon works at almost constant temperature and pressure levels the system would get unbalanced. One way to overcome this problem is to generally use wider pipe diameters and specifically use wider diameters for the return line pipe compared to down comer pipe, since the frictional pressure drop decreases with wider pipe diameters. Looking at Figure 37 a wider return line diameter than down comer diameter is suggested since this is the only case where the refrigerant is not superheated. From Figure 37 it is also seen that the demand of wider return line pipe diameter diminish with increased pipe diameter as a consequence of the decreased frictional pressure drop.

Another method is to lower the heating load in the evaporator. Data centers with heating loads with magnitudes of hundreds of kW could benefit from using several semi-central air cooling units instead of having a local cooling unit for each rack. This could also be combined with the use of several thermosiphons, dividing the cooling system into a number of subsystems. More systems will of course need more material etc. making them more expensive. This cost should be compared against the cost for one bigger system since the subsystem will be smaller.

In the vapor compression cycles the evaporation temperature is lower than in the thermosiphon. This since the calculations are based on currently existing parts and currently does not exists any compressor that are able to work at such high conditions for the treated refrigerants. The highest possible evaporation temperature for R744 today is 20°C and for R22 and R290 it is 12.5°C. Since the compressors must start on a "too low" pressure level than necessary it consumes excessive amount of energy. The R22 and R290 cycles has the lowest maximum evaporation temperature and therefore has a higher extra energy demand. If these restrictions would not be considering, the vapor compression cycles would run more efficient in general but the other system would look less inefficient compared to the R744 system. The minimum pressure ratio in the compressors also causes additional compressor work. If the minimum pressure ratio is neglected the vapor compression cycle will be even more effective.

The temperature profiles that are used are based on how many hours, of a certain temperature level, occurs for a whole year. This means that there is no consideration of when the temperature occurs during the day. For example, there might be possible to store energy in accumulator tanks if there is a low temperature during the night and then use that (stored) energy during the day. This could reduce the energy use in the systems even more and enabling more use of free cooling.

The calculated PUE-values are in the same range as literature although the PUE for the R744 and R290-system are rather high compared to the modern high efficiency systems e.g. from Carnot Refrigeration or Alfa Laval. One reason for that could be the impact from the auxiliary equipment, which are untreated and also assumed to be equal for all systems. Compared to Figure 6 the cooling share was more than twice the total share of the electrical consumption from the auxiliary equipment. From ***Figure 49*** the cooling share is considerably less than the total share of other electrical consuming parts. Although the reduction in PUE from Figure 6 to Figure 48 shows that lowering the electricity demand for cooling has a major impact on the PUE.

On the contrary the PUE-values for the two reference R22-systems are somewhat lower than what is given from the literature. This is probably caused by that the systems in the literature are running on partial load which increases the PUE-value compared to the modelled systems that are always running at full capacity. Further are the reference systems modelled to work in the same conditions as the developed ones. In the real world it is not unlikely that the vapor compression cycle works at a lower evaporation temperature and a higher pressure ratio although the literature does not reveal any such data.

The share of cooling electricity demand for the reference systems is also much lower than given from the literature which further indicates that the electrical consumption from the auxiliary equipment impacts the results in such way that total electricity demand gets too high. Looking at the energy distribution the cooling share is vanishingly small but to achieve a PUE at Alfa Laval's or Carnot Refrigeration's levels the auxiliary equipment electricity share should only stand for about 6-7%.

Looking on the solution with heat recovery it can be seen that both system will work well with low temperature heat recovery to cover a heating demand of an office that has 8000 m^2. The problem with the heat recovery for this system is that the heating demand is highest when the thermosiphon works as most efficient. Also when the systems need to use mechanical refrigeration the ambient temperatures are as highest i.e. low heating demand. But as there are a lot of waste heat from data centers at a relatively high temperature, this is a really good heat source that should be used if it is possible. Somehow that office needs heat anyway and if this solution is implemented where it is possible to use heat recovery it should be used.

8 Conclusions

- Refrigerant choice

R744 is a suitable refrigerant in data center cooling application with a thermosiphon. Due to its non-flammability nor toxicity in corporation with that it is not corrosive enables the use of R744 directly in the server room. Additionally, the fact that R744, in case of leakage, vaporizes in the server room temperature and pressure conditions makes R744 harmless to the electrical components.
R744 is not a very well suited refrigerant for higher condensing temperatures due to its low critical point. From Figure 46 and Figure 47 it is shown that an indirect system with R290 is more suitable for climates with general higher ambient temperatures. The choice of R290 as primary refrigerant is based on its good performance characteristics in hotter climates and the ability to have a small system with low charge, placed preferably on the roof, with a long physical distance from the servers. Therefore: for a thermosiphon operating at mostly low ambient temperatures a direct R744 system is the most suitable one. For higher ambient temperatures it is preferred to accompany the thermosiphon with a R290 vapor compression cycle making an indirect system.

- Thermosiphon application

The results show that there is possible to use a thermosiphon in data center cooling application with reasonable system sizing e.g. pipe lengths and diameters. This is supported by the literature although most of the previous work is done for smaller applications (less cooling demand).
Since the thermosiphon is operational only to a theoretical ambient temperature of slightly below 26°C, if the evaporation temperature is set to be 26°C, and a technical temperature of 20°C, auxiliary cooling is needed for higher ambient temperatures.

- Energy and economics

If looking at the systems and only consider the electricity demand that is shown in Figure 46, the indirect thermosiphon system is the most efficient in all the cities. This is caused by a very energy efficient R744 thermosiphon that cover a big part of the energy demand and the rest is covered by an efficient vapor compression cycle running with R290 as refrigerant. The drawback with this system is then showed in Figure 47 as the cost is very high for this system as it will need two condensers, one for the thermosiphon loop and one for the vapor compression cycle. Figure 47 also shows that either the indirect R290 system or the direct R744 is more efficient than the reference system running on R22 in all cities except Phoenix where the temperature often is over the set point temperature. The system that fits best in each city according to the annual cost and interest rate at 10% and 10 years of deprecation is shown in Table 20.

Table 20 - Best solution for different cities according to lowest total annual cost

	System	**Set point temperature**	**Condenser fin area**
Stockholm	Direct R744	20°C	2000 m^2
Paris	Direct R744	20°C	2000 m^2
Phoenix	Indirect R290	20°C	2000 m^2
Tokyo	Direct R744	20°C	2000 m^2
Madrid	Direct R744	20°C	2000 m^2

- Influence of design parameters

The down comer and return line designs shows to have the greatest impact on the evaporator outlet quality. When designing a real system theses parameters should therefore be taken the most care to. Generally, the evaporator design t influences the system less, specifically wider evaporator pipe diameters. Therefore, it is more legitimate to have higher uncertainties there.

- Evaporator outlet quality

If the cooling systems is designed to meet a maximum cooling demand at 200 kW the design parameters should be chosen such as that the evaporator outlet quality is maximized. Although this will make the cooling system sensitive to unexpected higher peak loads.
The results show that the system works for different down comer heights hence the system is possible to be designed for different buildings. It does not show any limitations for highering the down comer height but with lower down comer height the pipe diameter might have to be widered to avoid superheating of the refrigerant. A configuration with the condenser in the server room, i.e. low down comer height, requires wider diameters for the return line pipe than the down comer pipe, if using local cooling units. The results shows that a central cooling unit are able to operate under the given design conditions without the demand of a wider return line pipe diameter.

- Economical aspect

The calculations show that a thermosiphon system is economical defendable compared to conventional cooling although the numbers should be more of indication purposes to be used for comparison. In order to get more exact values, the system design needs to be further defined.

- Evaporative cooling

Although the efficient reduction in energy demand, evaporative cooling has its drawbacks. Before using it things as annual ambient temperatures and relative humidity, water supply and additional cost for maintenance and pumping should be considered.

- Uncertainty analysis

Generally, most of the values have a considerable uncertainty since many of the input parameters rely on assumptions. Since the comparisons between the systems are based on the same assumptions, their individual relationship is a truer representation of the results than the absolute values.

- PUE

It is not possible to determine the effectiveness of the system with unknown electricity consumption from other parts of the data center than from cooling and IT-equipment. The assumption of a constant electricity demand from other additional parts than cooling probably gives them a too big share compared to system with similar electricity demand for cooling. Although if only comparing the electricity demand for cooling the thermosiphon systems have significantly lower PUE. The lowering of the cooling demand is a major key to lower the PUE but to achieve even lower values other electricity consuming things, such as UPS, lighting, humidifier etc., should be considered.

- Heat recovery

The data center is a good heat source and in combination with a heat pump the temperature can be lifted with a small amount of supplied electricity. This can be implemented in both system solutions with good results according to the theoretical analysis.

9 Future work

The next step is to build an experimental setup for both systems in order to measure the real performance. As there are a lot of assumptions and simplifications in the model used in the report there would be interesting to see how the systems work during real conditions. Then the system is likely to run at partial IT-loads as well, wherefore a performance evaluation for partial IT-loads operations is of interest. Besides IT-loads, further evaluations could also be conducted on more refrigerants.
The controlling of the systems is not optimized in this work. To further increase the system efficiency the system regulation should be evaluated and then also considering the transient process for when the system switches operation mode from thermosiphon to mechanical refrigeration and vice versa.
Additionally, the possibility of shorter time heat storage during low temperatures i.e. at night time, could be investigated. This could increase the hours that is covered with free cooling. Evaporative cooling is another good theoretical solution for extending the amount of hours of free cooling. A future work could consider the effects of using evaporative cooling for hot dry areas (where it is as most effective) regarding e.g. water consumption and maintenance need.
Finally, the use of heat recovery could be investigated for such subject as other heating demands such as hot water or radiators. Looking into combining a data center heat recovery heating demand from adjacent facilities such as a public bath could also be of interest.

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Appendix A – Available free cooling hours for dry and wet bulb temperatures

Table 21 - Hours above and below different set point temperatures

City	**Set point temp 20°C**				**Set point temp 21°C**				**Set point temp 22°C**			
	Hours below set point		Hours above set point		Hours below set point		Hours above set point		Hours below set point		Hours above set point	
	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb
Stockholm	8736	8138	24	622	8758	8338	2	422	8760	8464	0	296
Paris	8614	7427	146	1333	8685	7713	75	1047	8727	7935	33	825
Phoenix	7517	3340	1243	5420	7834	3602	926	5158	8158	3854	602	4906
Tokyo	7145	5680	1615	3080	7436	6093	1324	2667	7716	6446	1044	2314
Madrid	8678	5992	82	2768	8737	6254	23	2506	8760	6515	0	2245

Appendix B - Energy demand and Annual cost for Evaporative Cooling

Figure 53 shows the needed fan power for five different condenser fin diameter and the power needed for mechanical refrigeration at the same ambient temperatures.

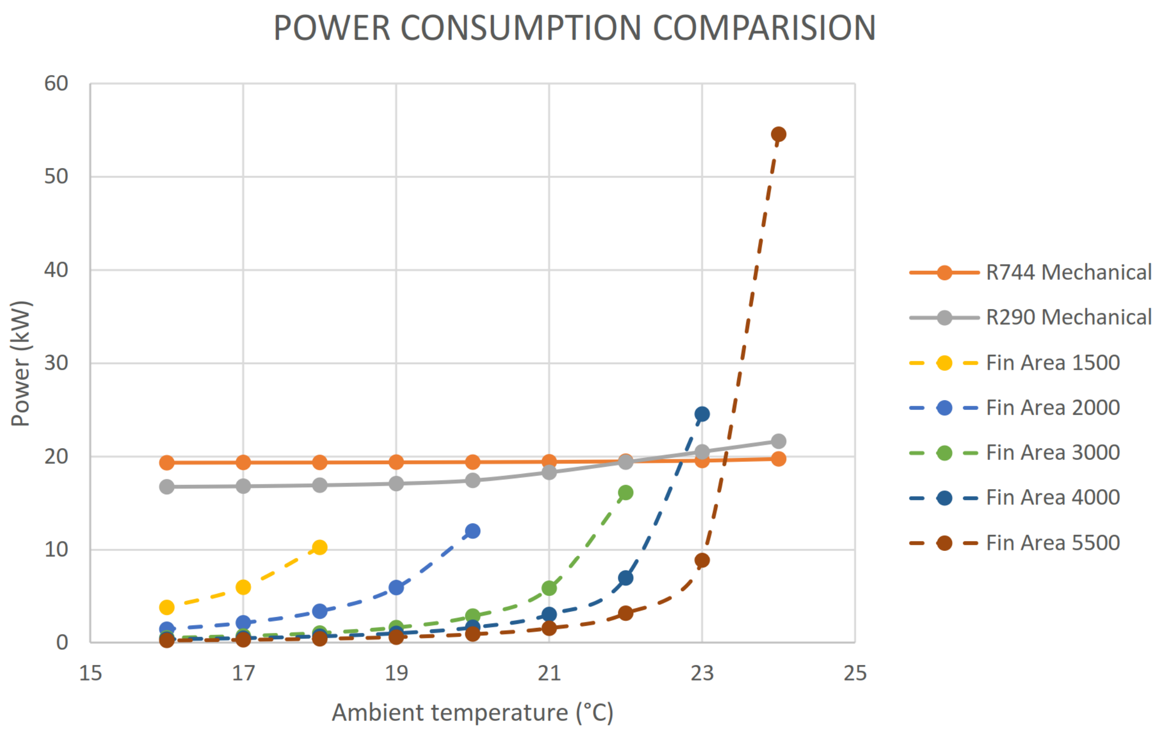

Figure 53 - Power consumption for the system with different fin areas with evaporative cooling

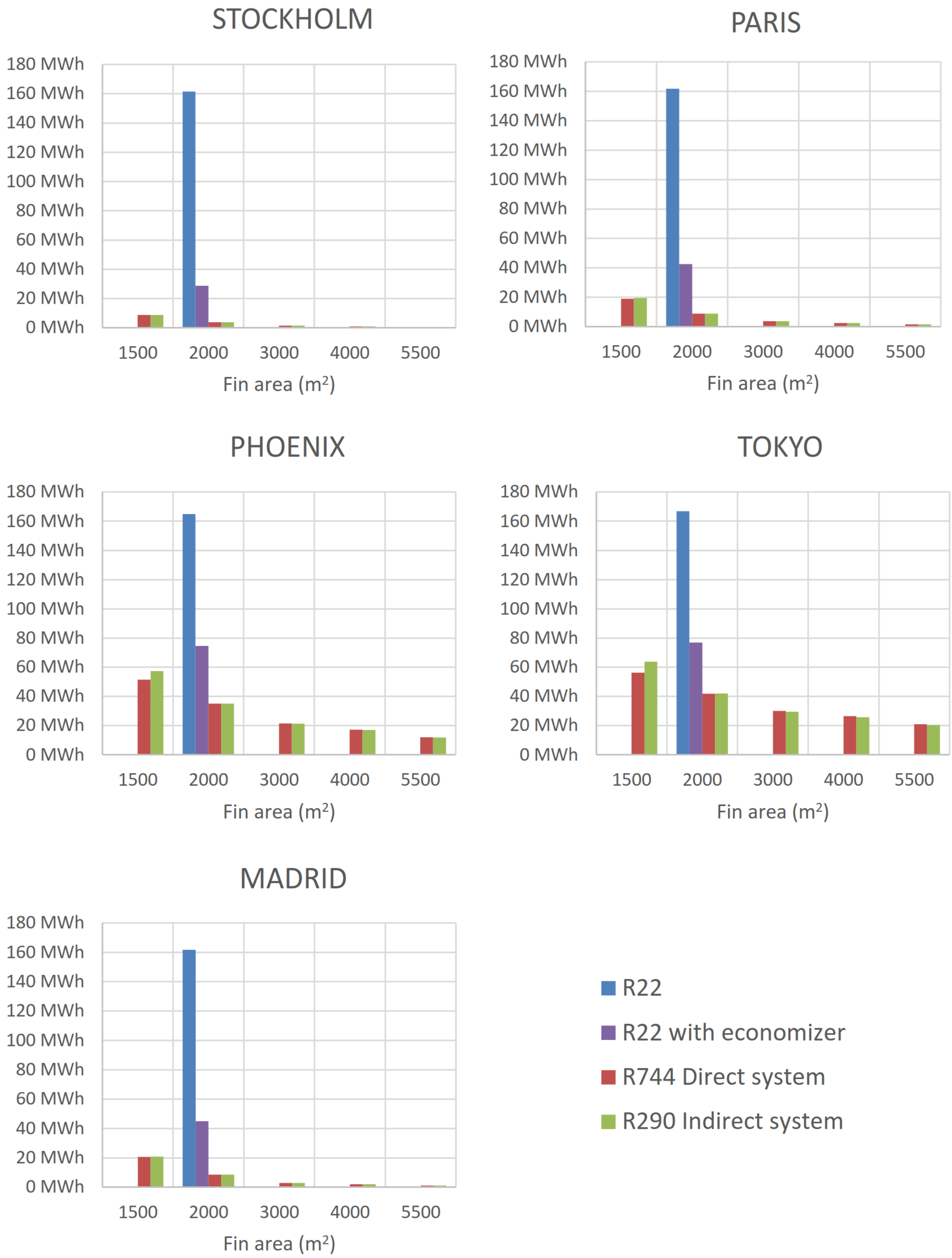

Figure 54 - *Electricity demand for the systems in different cities with evaporative cooling*

Figure 54 shows the annual energy demand for evaporative cooling for the five cities with varying condenser area. Figure 55 shows the corresponding cost.

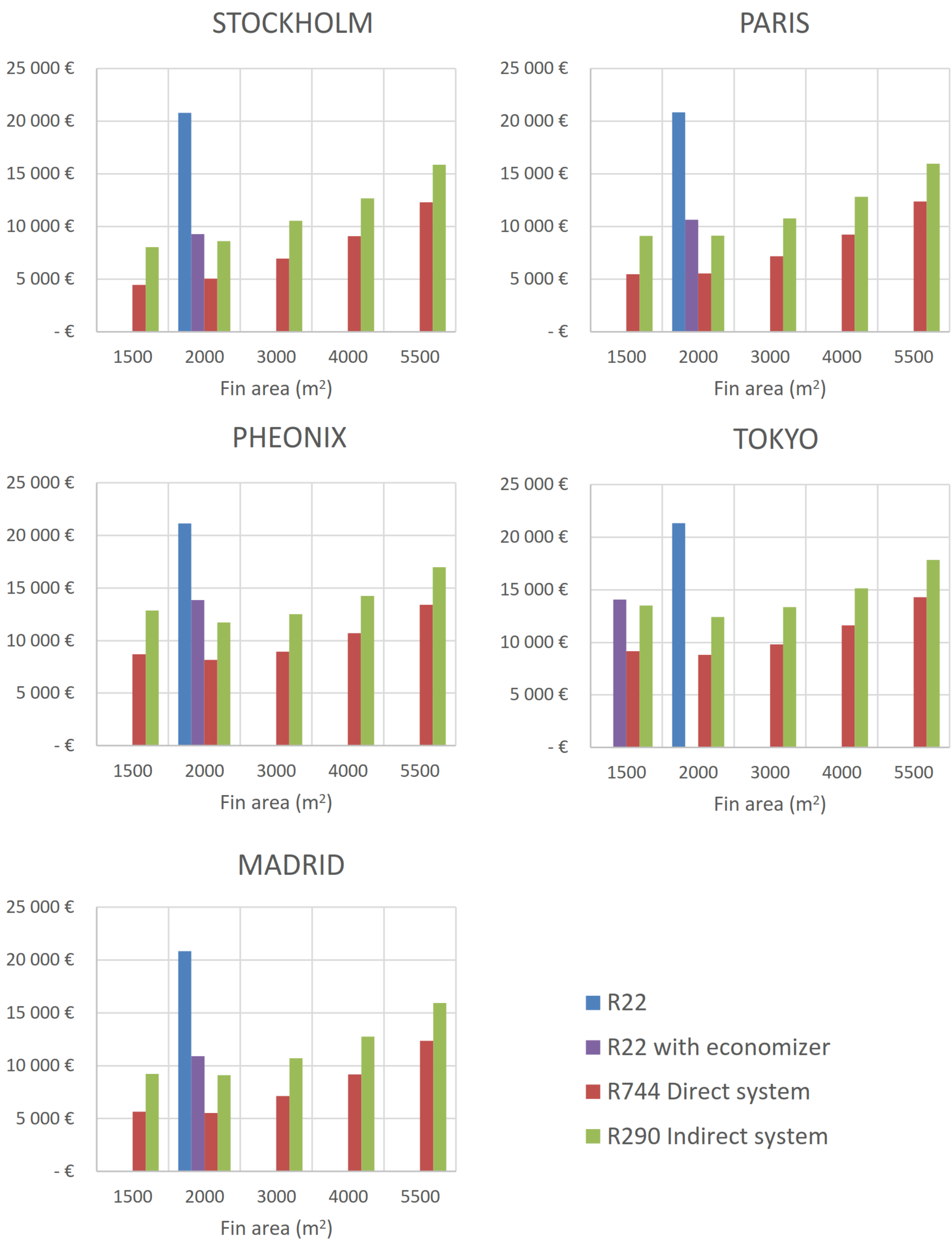

Figure 55 *- Total annual cost for the cities for the three different system solutions for evaporative cooling*

Appendix C - Energy distribution and PUE

The PUE-values for the different systems at different locations is shown in Table 22.

Table 22 - PUE-values for the evaluated systems

Key Parameter	**City**	**Direct R744**	**Indirect R290/R744**	**Reference: R22**	**Reference: R22 with economizer**
PUE	Stockholm	1,332	1,437	1,352	1,332
	Paris	1,347	1,440	1,371	1,345
	Phoenix	1,477	1,484	1,458	1,440
	Tokyo	1,382	1,449	1,399	1,376
	Madrid	1,383	1,451	1,396	1,374
PUE Cooling only	Stockholm	1,013	1,118	1,033	1,013
	Paris	1,027	1,120	1,052	1,026
	Phoenix	1,157	1,165	1,139	1,120
	Tokyo	1,063	1,129	1,080	1,057
	Madrid	1,064	1,132	1,077	1,055

The energy distribution for the direct R744 system is shown in Figure 56.

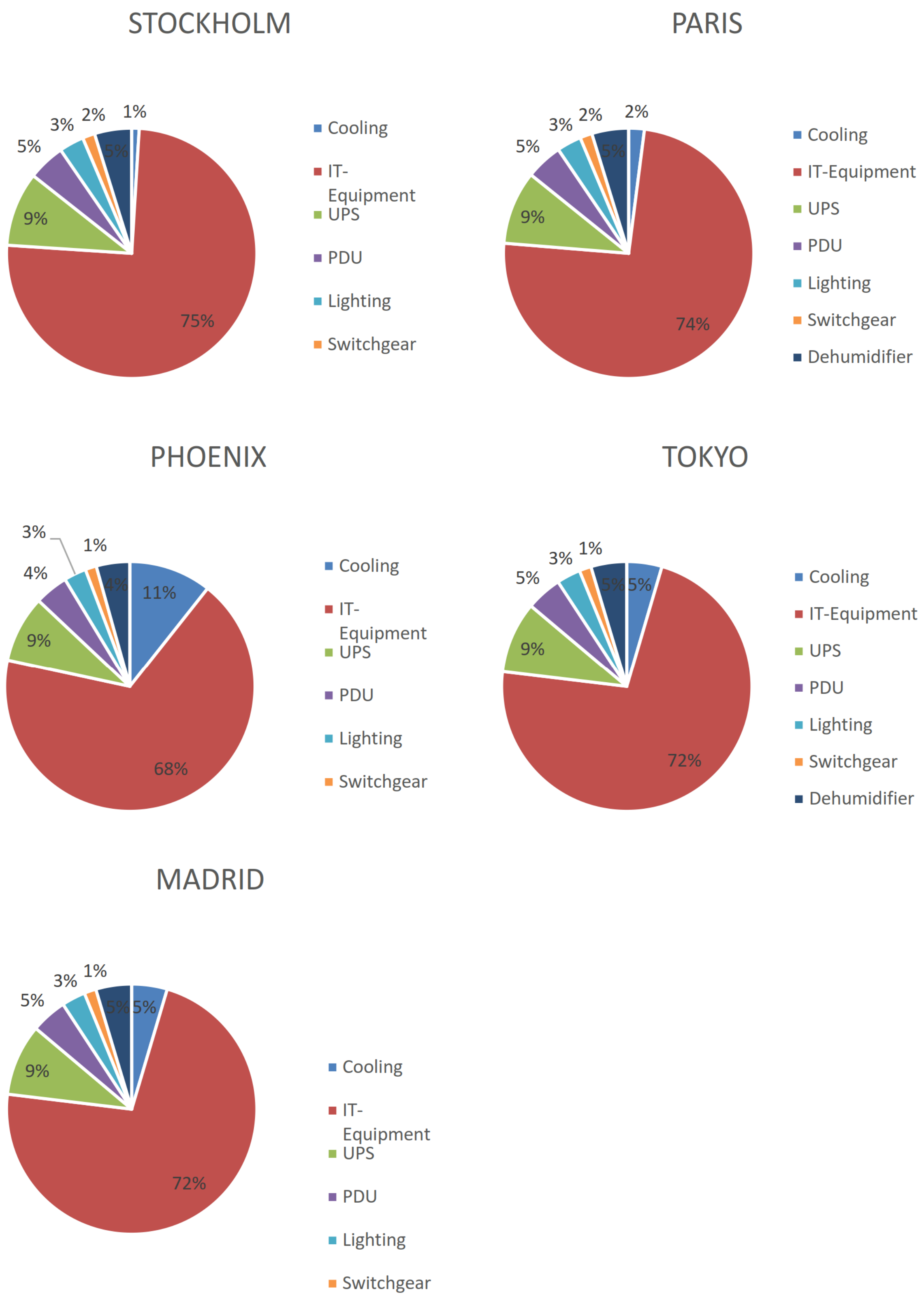

Figure 56 *- Energy distribution in the direct R744 system*

The energy distribution for the indirect R290-system is shown in Figure 57.

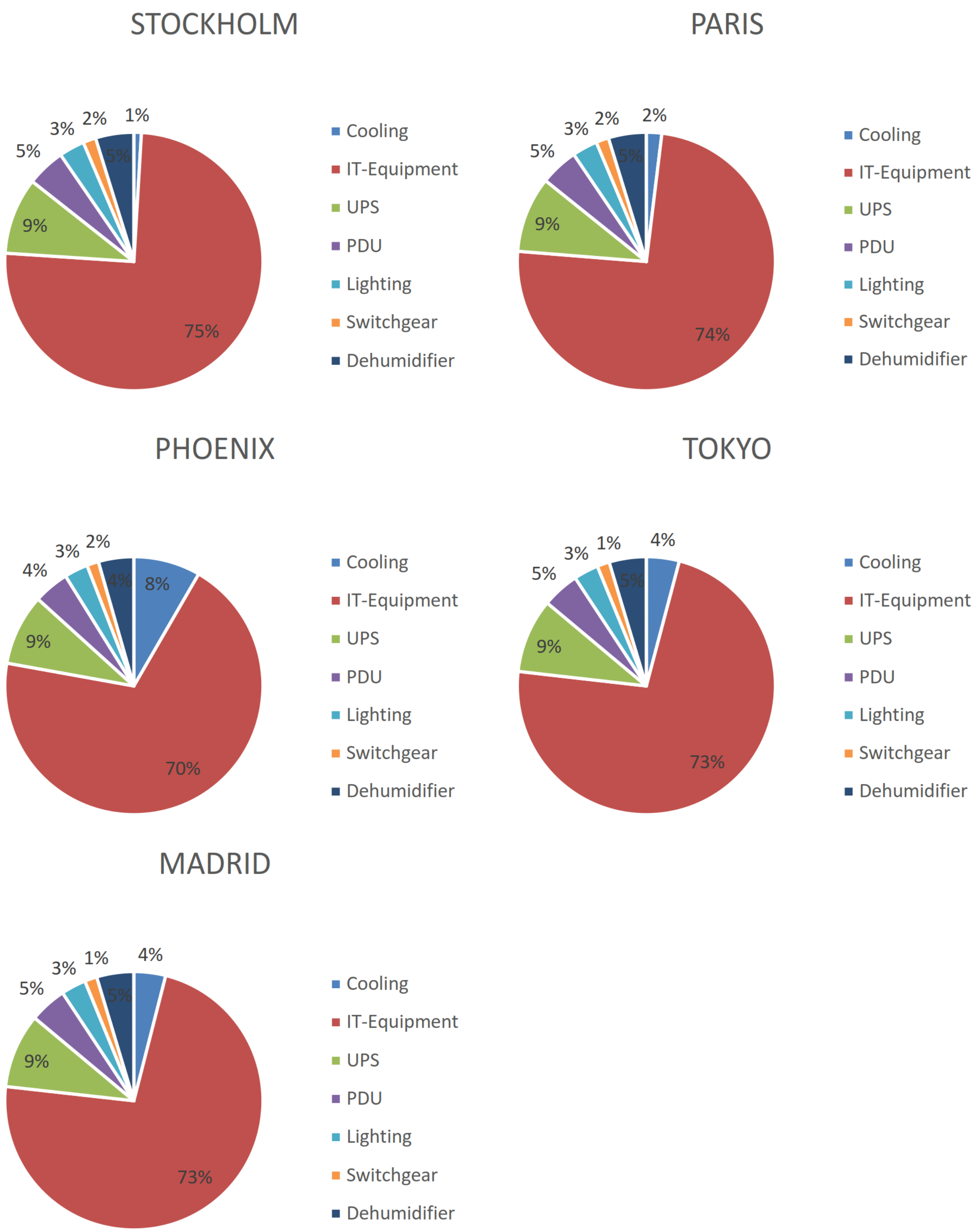

Figure 57 *- Energy distribution in the indirect R290-system*

The energy distribution for the R22-system is shown in Figure 58.

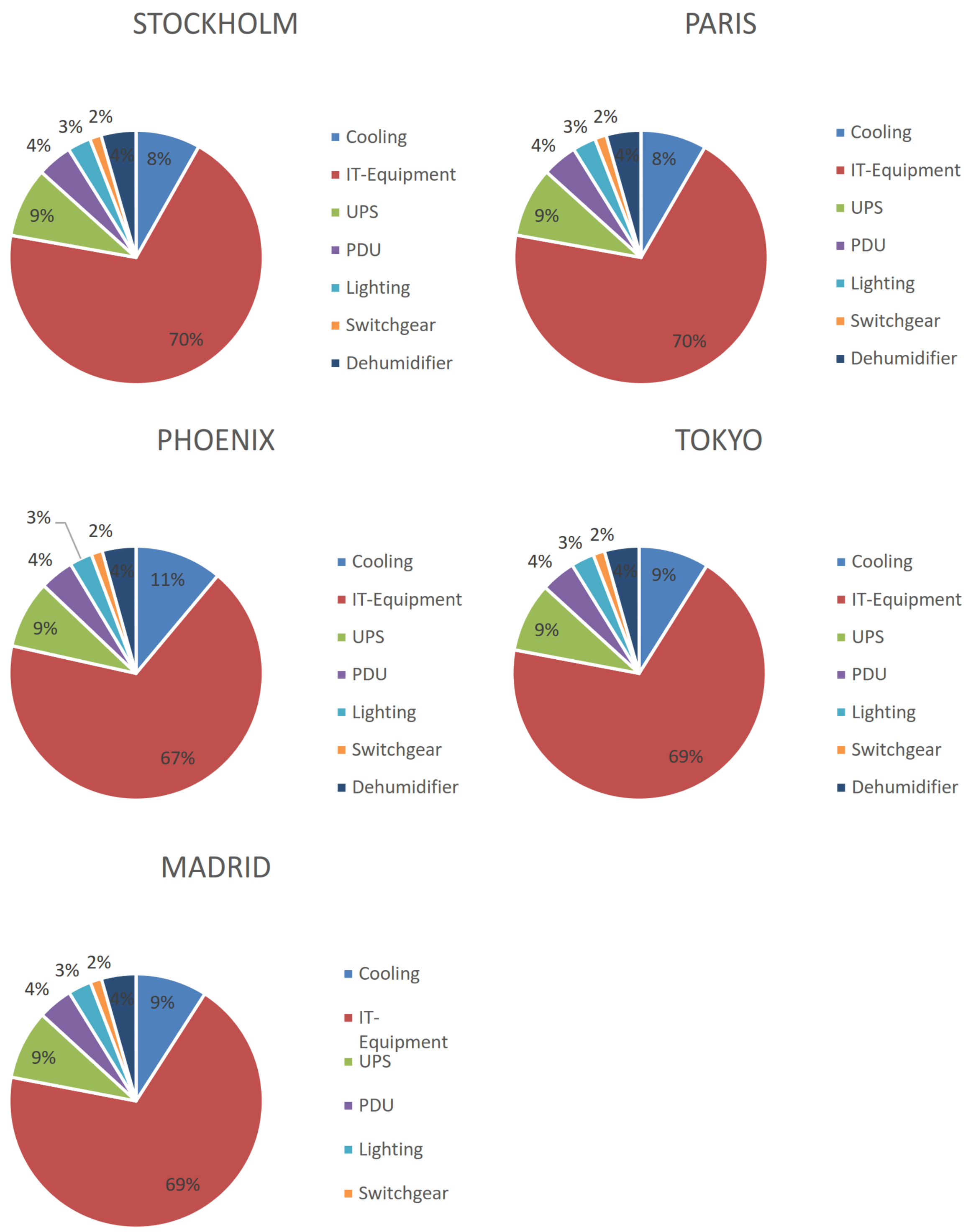

Figure 58 - *Energy distribution in the R22-system*

The energy distribution for the R22-system with economizer is shown in Figure 59.

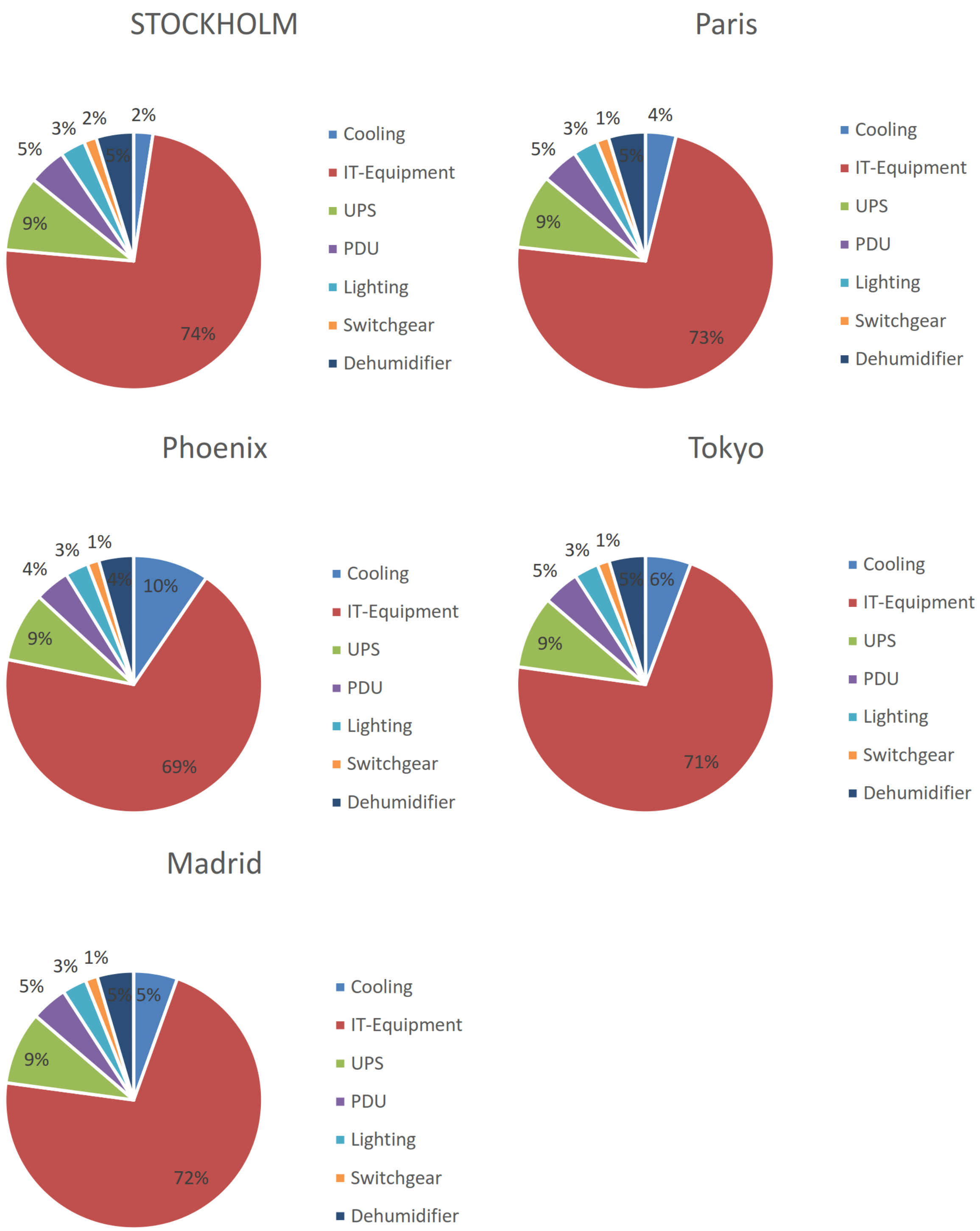

Figure 59 *- Energy distribution in the R22-system with economizer*