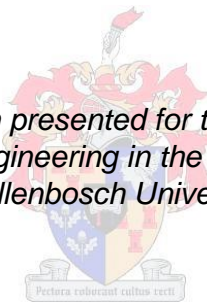# A Subcritical and Transcritical Carbon Dioxide Refrigeration System Utilizing Multiple Expansion Devices

By
Marna Fourie

Dissertation presented for the degree of
Master of Science in Engineering in the Faculty of Engineering at Stellenbosch University

Promoter: Robert T Dobson

March 2014

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I

Abstract

Carbon dioxide is a natural gas that has been used as a refrigerant as far back as the 1850s. It was then primarily used because it was easily obtainable and non-toxic. Due to the high operating pressure and relatively low critical temperature (and high critical pressure) of CO_2, it was gradually phased out and replaced with hydrochlorofluorocarbons and chlorofluorocarbons. With the discovery of the greenhouse effect, greenhouse gasses and ozone depleting gasses, the Montreal Protocol and the Kyoto Protocol were placed into effect. These two protocols call for the reduction in use of certain greenhouse gasses and the complete exclusion of others. The focus turned to natural gasses that are more environmentally friendly and easier to come by. Carbon dioxide is one such gas. However, CO_2 has a low critical temperature and high pressure, 33.98 °C and 73.77 bar respectively. CO_2 refrigeration systems are more effective, have a greater coefficient of performance and have a greater operating temperature difference (over the gas-cooler) when used under transcritical conditions. It is preferred to have transcritical CO_2 refrigeration systems. The drawback of such a system is the extreme operating conditions. A special system must be designed, built and commissioned at the University of Stellenbosch. The system will demand the use or manufacture of unique, specialised components. Most of the components utilised are extremely expensive and/or difficult to come by, or designed and manufactured specifically for this system.

The CO_2 system at the University of Stellenbosch is exceptional and stands out from conventional systems in that:

- it can operate under both the subcritical and transcritical conditions;
- it has multiple expansion units, which can be alternated, giving a total of four different operating configurations;
- the system can run with fully automated controllers or as a static system;
- the internal heat exchanger can be included or excluded from the system;
- all the heat exchangers can run in counter flow or parallel flow; and
- The system has multiple testing points, for both temperature and pressure, to give the operator accurate measurements to be used in comparison with design software, etc.

Three simulation programs are given that describe the physical system. The first simulation program is a steady-state simulator used to aid in the design of the heat exchangers and the capillary tube. The second simulator is a steady-state program that determines the mass flow rate in the capillary tube. The third simulation is a transient program, programmed to determine the steady-state conditions of a system, given set initial conditions and a transient start-up.

Opsomming

Koolstofdioksied is 'n natuurlike gas wat gebruik word as 'n verkoelingsmiddel so ver terug as die 1850's. Dit is toe hoofsaaklik gebruik omdat dit maklik verkrygbaar en nie giftig is nie. As gevolg van die hoë werksdruk en relatief lae kritiese temperatuur (en hoë kritiese druk) van CO_2, is dit geleidelik uitgefaseer en vervang met hidrochloorfluoorkoolstof en chloorfluoorkoolstowwe. Met die ontdekking van die kweekhuiseffek, kweekhuisgasse en osoon-afbrekende gasse, is die Montreal-protokol en die Kyoto-protokol in werking gestel. Hierdie twee protokolle vereis die afname in die gebruik van sekere kweekhuisgasse en die algehele uitsluiting van ander. Die fokus het verskuif na natuurlike gasse wat omgewingsvriendeliker en makliker is om te bekom. Koolstofdioksied is so 'n gas. Maar CO_2 het 'n lae kritiese temperatuur en hoë kritiese druk, 33.98 °C en 73.77 bar onderskeidelik. CO_2 verkoelingstelsels is meer effektief, het 'n groter koëffisiënt van werksverrigting en het 'n groter bedryfstemperatuur-verskil (oor die gas-verkoeler) wanneer dit gebruik word onder transkritiese toestande. Dit is dus verkieslik om transkritiese CO_2 verkoelingstelsels te hê. Die nadeel van so 'n stelsel is die relatief uiterste bedryfstoestande. 'n Spesiale stelsel moet ontwerp word, dikwels met die gebruik of vervaardiging van unieke, hoogs gespesialiseerde komponente. So 'n stelsel is ontwerp vir die Universiteit van Stellenbosch. Die meeste van die komponente wat gebruik is, is baie duur en/of moeilik om te bekom, of is spesifiek ontwerp en vervaardig vir hierdie stelsel.

Die CO_2-stelsel by die Universiteit van Stellenbosch is uitsonderlik en staan uit bo konvensionele stelsels deurdat:

- dit kan funksioneer onder beide subkritiese en transkritiese toestande;
- dit verskeie gasuitsettings-eenhede het, wat afgewissel kan word, wat 'n totaal van vier verskillende bedryfskonfigurasies gee;
- die stelsel bedryf kan word met volle outomatiese beheerders of as 'n statiese stelsel;
- die interne hitte-uitruiler óf in-, óf uitgesluit kan word van die stelsel;
- al die hitte-uitruilers bedryf kan word in kontra-vloei of parallelle vloei; en
- die stelsel verskeie toetspunte het, vir beide temperatuur en druk, wat die operateur akkurate metings gee om te vergelyk met die ontwerp sagteware, ens., wat gebruik word.

Drie gegewe simulasieprogramme beskryf die fisiese stelsel. In een simulasie word 'n bestendige toestand-simulator gebruik om te help met die ontwerp van die hitteruilers en die kapillêre buis. Die tweede simulasie program is 'n bestendige toestands-program wat die algehele vloeitempo in die kapillêre buis bepaal. Die laaste simulasie is 'n vlugtige program, geprogrammeer om die tydelike toestande van 'n stelsel te bepaal, gegewe vasgestelde aanvanklike toestande en 'n kortstondige aanskakeling.

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NOMENCLATURE

A	Area $[\mathrm{m}^2]$
A/C	Air-Conditioning
ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineers
ATEL	Acute-Toxicity Exposure Limit
CAD	Computer Assisted Drawing
CCS	Carbon Capture Sequestration
C_f	Friction Coefficient (Fanning)
CFC	Chlorofluorocarbon
COP	Coefficient of Performance
c_p	Specific Isobaric Heat Capacity $\left[\frac{\mathrm{kJ}}{\mathrm{kg\,K}}\right]$
d	Diameter $[\mathrm{m}]$
$\dot{E}$	Rate of Energy Flow $[\mathrm{W}]$
EEV	Electronic Expansion Valve
EES	Engineering Equation Solver
ETFL	Elevated Temperature Flame Limit
FC	Fluorocarbon
g	Gravitational Force $\left[\frac{\mathrm{kg\,m}}{\mathrm{s}^2}\right]$
GWP	Global Warming Potential
GUI	Graphical User Interface
h	Specific Enthalpy $\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
ht	Convective Heat Transfer Coefficient $\left[\frac{\mathrm{W}}{\mathrm{m}^2\,\mathrm{K}}\right]$
ID	Internal Diameter
IHX	Internal Heat Exchanger
k	Thermo Conductivity $\left[\frac{\mathrm{W}}{\mathrm{K\,m}}\right]$
L	Length $[\mathrm{m}]$

LFL	Lower Flammability Limit
LMTD	Log Mean Temperature Difference
m	Mass [kg]
$\dot{m}$	Mass Flow Rate $\left[\frac{\text{kg}}{\text{s}}\right]$
MS	Microsoft
N	Number of Control Volumes
NASA	National Aeronautic and Space Administration
Nu	Nusselt Number; $\text{Nu} = 0.023\text{Re}^{0.8}\,\text{Pr}^{\,0.4}$
OD	Outside Diameter
ODL	Oxygen Deprivation Limit
ODP	Ozone Depletion Potential
OEL	Occupational Exposure Limit
P	Pressure [bar]
$\wp$	Perimeter [m]
PAG	Polyalkaline Glycol
ppm	Parts Per Minute
Pr	Prandtl Number
PWM	Pulse Width Modulation
q	Heat Transfer Rate [W]
Q	Heat Transfer [kJ]
$\dot{Q}$	Heat Transfer Rate [W]
r	Radius [m]
R	Thermal Resistance $\left[\frac{\text{K}}{\text{W}}\right]$
RCL	Refrigeration Concentration Limit
Re	Reynolds Number; $\text{Re} = (\dot{m}\,d)/(A\,\mu)$
s	Specific Entropy $\left[\frac{\text{kJ}}{\text{kg K}}\right]$
SM	System Monitoring
SMS	Short Message Service
t	Time [s]
T	Temperature [°C] or [K]
TEV	Thermostatic Expansion Valve
TWA	Time-Weighted Average

U	Internal Energy $[\mathrm{kJ}]$
u	Specific Internal Energy $\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right]$
v	Velocity $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
VBA	Visual Basic for Applications
WCF	Worst Case of Formulation of Flammability
WCFF	Worst Case of Fractional for Flammability
x	Mass Fraction $[\%]$
z	Control Volume Length $[\mathrm{m}]$

CHEMICAL SYMBOLS

$C_2F_4C\ell_2$	1,2-Dichlorotetrafluoroethane (R114)
C_3H_8	Propane (R290)
H_2O	Water, Steam (R718)
H_2O	Water (R718)
O_2	Oxygen (R732)
$CH_3CH_2CH_2CH_3$	Butane (R600)
CH_4	Methane (R50)
CO_2	Carbon Dioxide (R744)
$CC\ell_2F_2$	Dichlorodifluoromethane, Freon-12 (R12)
$CC\ell_3F$	Trichlorofluoromethane, Freon-11 (R11)
$CH(CH_3)_2CH_3$	Isobutane (R600a)
$CHC\ell F_2$	Chlorodifluoromethane, Freon-22 (R22)
NH_3	Ammonia (R717)
NH_3	Ammonia (R717)

SUPERSCRIPTS

t	Time

XIII

SUBSCRIPTS

a	Actual
Acc	Accumulator
Amb	Ambient
c	Cold
comp	Compressor
CT	Capillary Tube
disc	Discharge
ev	Evaporator
F	Friction
g	Gas
G	Gravity
GC	Gas-Cooler
go	Gas Only
h	Homogeneous
h	Hot
i	Inner
i	i[th] Control Volume
IHX	Internal Heat Exchanger
ℓ	Liquid
ℓo	Liquid Only
M	Momentum
o	Outer
P	Pressure
r	Refrigerant
rec	Receiver
ss	Stainless Steel
suc	Suction
w	Water
x	Cross-Sectional

GREEK LETTERS

$\sum F_z$	Sum of the Forces
α	Void Fraction; $\alpha = v_g \, / \, (v_g + v_\ell)$
ΔE	Change in Energy $[\mathrm{kJ}]$
Δm	Change in Mass $[\mathrm{kg}]$
Δt	Change in Time / Time Step $[\mathrm{s}]$
ΔT	Change in Temperature $[\mathrm{K \, or \, °C}]$
ΔP	Change in Pressure $[\mathrm{MPa}]$
ΔU	Change in Internal Energy $[\mathrm{kJ}]$
Δz	Change in Length $[\mathrm{m}]$
η	Efficiency [%]
μ	Dynamic Viscosity $[\mathrm{\mu Pa \, s}]$
ρ	Density $[\mathrm{kg/m^3}]$
τ	Shear Stress $[\mathrm{N/m^2}]$
ϕ	Angle of Control Volume to the Horizontal, [degrees]
Φ	Two-Phase Friction Multiplier

CHAPTER 1 INTRODUCTION

A CO_2 refrigeration system is extremely versatile in the sense that it can be used in the smallest, residential systems to extremely large industrial projects. CO_2 can be used on its own, or in composite, multiplex systems. It can safely be used in the food industry, as CO_2 is non-toxic. As the focus shifts from synthetic refrigerants, to inherently safe, natural refrigerants, CO_2 is becoming more and more popular.

The project includes the design, building and commissioning of a multi-use test platform. The unit runs on CO_2 refrigerant and uses water to heat and cool the heat exchangers. Multiple types of expansion units are included in the system. Also included in the project is a steady-state simulation program, a steady-state simulation for a capillary tube and a transient simulation program for the system as a whole, excluding the capillary tube. The steady-state simulation program for the system is used to aid in the design of the experimental system. The steady-state capillary tube simulation program determines the mass flow rate through the capillary tube for specified operating conditions. The transient simulation program determines the transient change in the system from start up to steady-state conditions.

1.1 MOTIVATION

Common refrigerants used in commercial, industrial and residential systems pose a risk either to the environment or the people working around them. Carbon dioxide is both environmentally friendly, and is intrinsically safe. It is non-toxic, non-explosive, non-flammable and non-corrosive. In addition, a natural gas is easily available in abundance.

Using CO_2 as a refrigerant does pose several problems. It has a very high operating pressure and a relatively low critical temperature. This calls for supercritical CO_2 in the refrigeration system, as ambient air is most often used to cool the refrigerant in the condenser. Special components need be designed and manufactured for the high operating temperatures. Normal refrigeration systems only use one type of expansion unit. The system at Stellenbosch University is specifically designed and built to withstand the high operating temperatures and pressure. It also has multiple expansion units, allowing for complex system layouts and extensive testing of the system and the ability to draw a comparison between the different types of expansion units.

The equations of state for CO_2 properties are extremely complex. As such, it does not fall in the scope of the project to attempt to recreate the equations of state and fluid properties of CO_2. Instead, an *add-in* from MegaWatSoft is used. The add-in is used in Microsoft Excel, along with Visual Basic for Applications (VBA).

1.2 OBJECTIVES

Objectives include firstly the review of existing work regarding CO_2 refrigeration systems. The review gives a good indication of progress made on CO_2 refrigeration systems and the research based off them.

Secondly, the development of three simulation programs for the project. The first is a steady-state simulation of a basic refrigeration system that must aid in the design of the final, experimental refrigeration system by determining lengths of the heat exchangers and the capillary tube. The second simulation is a steady-state program for the capillary tube that must determine the mass flow rate through the capillary tube. The third simulation is a transient program of the full system – minus the capillary tube. This simulation determines the steady-state conditions in the system, through a transient process from initial conditions.

Thirdly, the design, building and commissioning of an operational carbon dioxide refrigeration system. The experimental system needs to run on carbon dioxide, with water cooled/heated tube-in-tube heat exchangers. The system needs to be safe, fully controllable and must include several temperature and pressure measuring points. The multiple expansion devices must be easily interchangeable.

The three objectives are amplified as follows.

1.2.1 REVIEW OF LITERATURE

The literature review considers available information on existing research projects across the world. It covers the theory behind a basic refrigeration system, the properties of different refrigerants, carbon dioxide as a refrigerant, the different types of expansion units used in refrigeration systems, and finally the review covers basics of the complex carbon dioxide refrigeration system built at Stellenbosch University.

1.2.2 SIMULATIONS

The first is a steady-state simulation program that can aid in the design of the heat exchangers and the capillary tubes for the refrigeration system. The steady-state simulator uses ideal system assumptions. It is not meant to be an ideal, complete system design simulator, but only to aid the user and give a "starting point" for the final design.

The second is a steady-state simulation for the capillary tube. It determines the ideal mass flow rate for a specific tube, under specific operating conditions.

The third is a transient simulator that simulates the refrigeration system from start-up conditions to steady-state conditions. The capillary tube is not included in this simulation.

All the simulations are written in Microsoft Excel, with VBA. The steady-state simulation for the system and the transient simulation makes use of a specialised *add-in* to determine the fluid properties and the equations of state for CO_2. The capillary tube simulation uses vast data tables, based off the *add-in* for the fluid properties and the equations of state.

1.2.3 DESIGNING, BUILDING AND COMMISSIONING THE CARBON DIOXIDE REFRIGERATION SYSTEM

Design and build of a "state of the art" system that captures the operating conditions and safety considerations. The system must be usable for future testing and practical sessions for students or as a training platform for companies. The system includes multiple expansion units, namely electronic expansion valves and capillary tubes. It gives 14 configurations, not including the use of more than one capillary tube at the same time. The system also has the capability to run under multiple operating conditions. It can run subcritically or transcritically, the heat exchangers can have parallel or counter flow configuration, the internal heat exchanger can be included or excluded, the system can have one source for heating and cooling water or two separate sources, etc.

CHAPTER 2 LITERATURE REVIEW

The literature review briefly covers the basics of refrigeration, its effects on global warming and ozone depletion and the ideal qualities of a refrigerant. Carbon dioxide (CO_2) is considered as a refrigerant and the possible hazards and safety issues are highlighted. Differences between subcritical and transcritical operation and transient and steady-state operation are briefly discussed. The review is concluded with a look at the components typically used in a CO_2 refrigeration system.

2.1 GENERAL INFORMATION

Carbon Dioxide, or CO_2, is often seen as a main contributor to global warming. CO_2 is naturally present as part of Earth's carbon cycle and has always been kept in a balance by nature. It is through human intervention and development that CO_2 levels are increasing beyond the point where nature can balance it out. This primarily happens through industrial processes and de-forestation. The main contributor to the release of CO_2 is the burning of fossil fuels for electricity, transportation and other industrial processes – such as the making of cement.

CO_2 emission can be reduced by:

- Energy efficiency;
- Energy conservation;
- Switching to renewable fuel sources; and
- Carbon capture and sequestration.

Carbon capture and sequestration (CCS) is a highly appealing method to reduce total CO_2 emission into the atmosphere. CCS usually involves capturing CO_2 emission from a power plant or industrial plant, transporting the compressed CO_2 – usually through pipes – and pumping the CO_2 into deep underground cavities. These cavities are usually a few kilometres deep and are located in an air pocket beneath solid, non-porous rock formations. This method, while it greatly reduces CO_2 emission, merely places the CO_2 out-of-sight. If the CO_2 should leak from the pocket and make it into the atmosphere, it will have detrimental effects on the environment. (EPA - United States Environmental Protection Agency, 2013)

To give an example of the devastation CO_2 can cause, consider a 500 MW coal-fired power plant produces 3 million tons of CO_2 per year. This is equal to planting over 62 million trees and then waiting ten years for them to grow. (EPA - United States Environmental Protection Agency, 2013)

2.1.1 REFRIGERANTS, THE PAST AND PRESENT

The mid-1800s saw the birth of mechanical type refrigeration systems. These machines used refrigerants such as sulphur dioxide, methyl chloride, ether, carbon dioxide, as well as wine, brandy, vinegar, etc. These systems were primarily used to manufacture ice throughout the year, competing with natural ice that was harvested and shipped seasonally. In the late 1800s, most refrigeration systems used either ammonia or carbon dioxide (CO_2). CO_2 was cheaper than ammonia and non-toxic, which made it the working fluid of choice for storing food and drink. Carbon dioxide refrigeration systems were often found on-board ships that were carrying meat on long voyages.

The downside of using CO_2 was its need for relatively cold water in the condenser to ensure that the CO_2 temperature (and pressure) did not increase too much and cause the system to run into the supercritical region. In the 1930s, the cascade system was invented, using an ammonia cycle to cool the CO_2 condenser. This system allowed for the simultaneous freezing of foods and the cooling of cold storage rooms. Other benefits include using less of the toxic ammonia and the ability to use smaller – and thus cheaper – compressors.

The invention of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) also followed in the 1930s. These refrigerants – e.g. R12 that was invented by Midgley (Sneader, 2005) – had all the desirable properties for a refrigerant of that time. They are non-toxic, non-flammable, have good thermodynamic properties and do not have extreme temperature or pressure operating conditions – such as the extreme high operating pressure of CO_2. Thus, CFCs, such as R12, R11, R114 and R502, quickly became the refrigerant of choice. While ammonia was still used in many industrial applications, CO_2 was quickly abandoned.

The following table lists a group of common refrigerants, their refrigerant type, their designated ASHRAE number, their molecular formula and their chemical or commercial name. CO_2 has the ASHRAE number of R744.

Table 2.1 ASHRAE Clarification of Refrigerants

TYPE	ASHRAE NUMBER	MOLECULAR FORMULA	CHEMICAL/COMMERCIAL NAME
CFC	R11	$CC\ell_3F$	Trichlorofluoromethane (Freon-11)
CFC	R12	$CC\ell_2F_2$	Dichlorodifluoromethane (Freon-12)
HCFC	R22	$CHC\ell F_2$	Chlorodifluoromethane (Freon-22)

Table 2.2 ASHRAE Clarifications of Refrigerants (Contd.)

TYPE	ASHRAE NUMBER	MOLECULAR FORMULA	CHEMICAL/COMMERCIAL NAME
HC	R50	CH_4	Methane
CFC	R114	$C_2F_4C\ell_2$	1,2-Dichlorotetrafluoroethane
HCFC	R502	48.8% $CH_2C\ell_2F_2$ 51.2% $C_2F_5C\ell$	R22/115 (48.8/51.2)
HC	R290	C_3H_8 or $CH_3CH_2CH_3$	Propane
HFC	R404A	$44 \pm 2\%\ C_2HF_5$ $52 \pm 1\%\ C_2H_3F_3$ $4 \pm 2\%\ C_2H_2F_4$	R125/143a/134a (44±2/52±1/4±2)
HC	R600	C_4H_{10} or $CH_3CH_2CH_2CH_3$	Butane
HC	R600a	C_4H_{10} or $CH(CH_3)_2CH_3$	Isobutane
Natural	R717	NH_3	Ammonia
Natural	R718	H_2O	Water/Steam
Natural	R729	78.082% N_2 20.945% O_2 0.934% Ar	Air
Natural	R732	O_2	Oxygen
Natural	R744	CO_2	Carbon Dioxide

(American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 2010)

2.1.2 GLOBAL WARMING AND OZONE DEPLETION

In the mid- to late 1900s, scientists discovered that the protective ozone layer in the upper atmosphere was thinning. Chemists Frank Sherwood Rowland and Mario Moline began their studies on the impact of CFCs on the Earth's atmosphere in 1973. They discovered that CFCs would remain stable in the atmosphere until they are eventually broken down by ultraviolet radiation. Then the chlorine proceeds to act as a catalyst to the destruction of the ozone layer. In 1985 British Antarctic Survey scientists Farman, Gardiner and Shanklin, published results of abnormally low ozone concentrations above Halley Bay, Antarctica. NASA later revealed matching data collected from their satellite recordings, suggestion CFCs are to be blamed. (Chemistry, Assembly of Mathematical and Physical Sciences (U.S.). Panel on Atmospheric; Chemistry, National Research Council (U.S.). Panel on Atmospheric, 1976)

In 1989, the Montreal Protocol for Substances that Deplete the Ozone Layer was placed into effect. The Protocol included the management plan to phase out CFCs by 1996. It also included a plan to phase out HCFCs by

2030. Hydrofluorocarbons (HFCs) and fluorocarbons (FCs) quickly re-placed the HCFCs and CFCs. The thermodynamic properties of fluorocar-bons are, however, inferior to the properties of CFCs.

While the Montreal Protocol addressed all ozone depleting gasses, the Kyoto Protocol addresses all greenhouse gasses. The intention of the Kyoto Protocol is to reduce the global emissions of greenhouse gasses. These gasses absorb and emit thermal infrared radiation, creating a rise in temperature, as would occur in a typical greenhouse. All participating coun-tries to the protocol were given specific requirements: either to reduce their greenhouse gas emissions or to cap their emissions level. The first period for implementing the Kyoto Protocol is from 2008 to 2012. To date, most countries who agreed to the Kyoto Protocol have ratified the goals set upon them. Only the United States of America and Canada have not ratified the Protocol – with Canada withdrawing from it. (Chemistry, Assembly of Mathematical and Physical Sciences (U.S.). Panel on Atmospheric; Chemistry, National Research Council (U.S.). Panel on Atmospheric, 1976) (Daniel & Velders, 2007)

Table 2.3 Global Impact of Refrigerants

TYPE	ASHRAE NUMBER	ATMOSPHERIC LIFETIME (YEARS)	ODP	GWP
CFC	R11	45	1	4 750
CFC	R12	100	1	10 900
HCFC	R22	12	0.05	1 810
HC	R50	12 ± 3	<0(smog)	25
CFC	R114	300	1	10 040
HCFC	R502	876.256	0.24968	4 657
HC	R290	12 ± 3	0	3
HFC	R404A	40.36	0	3 922
HC	R600	12 ± 3	0	<25
HC	R600a	-	0	<25
Natural	R717	<0.019105	0	0
Natural	R718	-	0	0.2 ± 0.2
Natural	R729	-	-	0
Natural	R732	-	-	<0
Natural	R744	29 300 - 36 100	0	1

(Forster, et al., 2007) (Daniel & Velders, 2007) (Hundy, et al., 2008)

All gasses have both an ozone depletion potential (ODP) and a global warming potential (GWP). Carbon dioxide – while it has a long lifetime – has no ODP and is used as the baseline when comparing GWPs. Table 2.3 lists and compares the ASHRAE number, Atmospheric Lifetime, ODP and GWP of common refrigerants.

An example on how the GWP works: CO_2 has a GWP of 1, while R11 has a GWP of 4 750. This means that every 1 kg of R11 released into the atmosphere is equivalent to 4 750 kg of CO_2 released into the atmosphere. The focus on ODP and GWP in the recent years have brought on renewed interest in gasses that are both natural and environmentally friendly.

2.1.3 REFRIGERANT PROPERTIES

The challenge today is to find a gas that proves to be an "ideal" refrigerant. Such a gas should have good thermophysical properties, higher than ambient condensing temperature, low freezing temperature, high critical temperature, low critical pressure, good heat transfer properties, no ODP or GWP, should be non-toxic, non-flammable, non-corrosive, non-explosive, should have low system cost and should be easily obtainable. Table 2.4 compares the critical temperature and pressure for the various refrigerants listed in Table 2.1 above.

Table 2.4 Critical Properties of Refrigerants

ASHRAE NUMBER	CRITICAL TEMPERATURE [°C]	CRITICAL PRESSURE, ABSOLUTE [bar]
R11	197.96	44.08
R12	111.97	41.36
R22	96.14	49.90
R50	-82.3	46.40
R114	145.68	32.57
R502	80.73	40.19
R290	44.10	0.42
R404A	72.14	37.35
R600	0 ± 1	1.52
R600a	-11.7	1.35
R717	132.4	112.8
R718	373.95	220.6
R729	28.97	-1.93
R732	-	0.32
R744	31.04	73.8

As can be seen, older, synthetic CFCs and HCFCs refrigerants have high critical temperatures and low critical pressures – an ideal combination. HFCs, FCs and natural refrigerants do not all share this combination. An example is R717 (ammonia), which has a high critical temperature but also a high critical pressure and R744 (CO_2), which has a low critical temperature and a high critical pressure. However, due to scientific advancements, high-pressure ranges within a refrigeration cycle are no longer a problem

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as components can be made and tested to withstand these extreme pressures.

2.1.4 REFRIGERANT QUALITY CARBON DIOXIDE

Natural gasses must be of a high purity for them to be optimally used as a refrigerant. Linde Gas – a major supplier of refrigerants – supplies both natural and synthetic refrigerants. R717 (ammonia) is supplied 99.98% pure, R744 (carbon dioxide) is supplied >99.9% pure and hydrocarbon refrigerants are supplied with at least 97.5% purity. (The Linde Group, 2012)

After the discovery of the harmful effects of synthetic halocarbons, the world renewed its interest in natural gasses as refrigerants. R744 (carbon dioxide) does not display many of the negative characteristics that are common with other refrigerants. CO_2 is non-toxic, non-flammable, has no ODP, a GWP of 1 and is non-corrosive. Unlike R717 (ammonia) – which is corrosive and can only be used with steel and aluminium tubing – CO_2 can be used with copper tubing. Copper tubing is the industry standard and provides a more comfortable working material for manufacturers of refrigeration and air-conditioning systems. Furthermore, CO_2 is relatively cheap, easily available and has excellent thermal properties. The only downside of CO_2 is the high operating pressure.

2.1.5 BASICS OF REFRIGERATION SYSTEMS

The goal of a refrigeration system is to cool a product/room to a lower temperature than the ambient conditions. Such a system usually has one heat exchanger used to cool the room or product and one heat exchanger that gets rid of the excess heat, usually by venting said heat to the environment. In some cases, however, the excess heat can be fully utilised to e.g. heat water for a geyser or a boiler.

Any substance can, theoretically, be used as a refrigerant. However, only a handful contains the desirable characteristics of a good refrigerant, as listed above in section 2.1.4 Refrigerant Quality Carbon Dioxide.

Refrigeration systems are used today in a vast variety of applications: From large industrial or commercial units, to smaller domestic units down to the small A/Cs in a car. Figure 2.1 shows the basic layout of a refrigeration system. The system consists of a compressor, a condenser, an expansion unit – an electronic valve in this instance – and an evaporator. Some units also make use of an internal heat exchanger. In a supercritical system, a gas-cooler replaces the condenser.
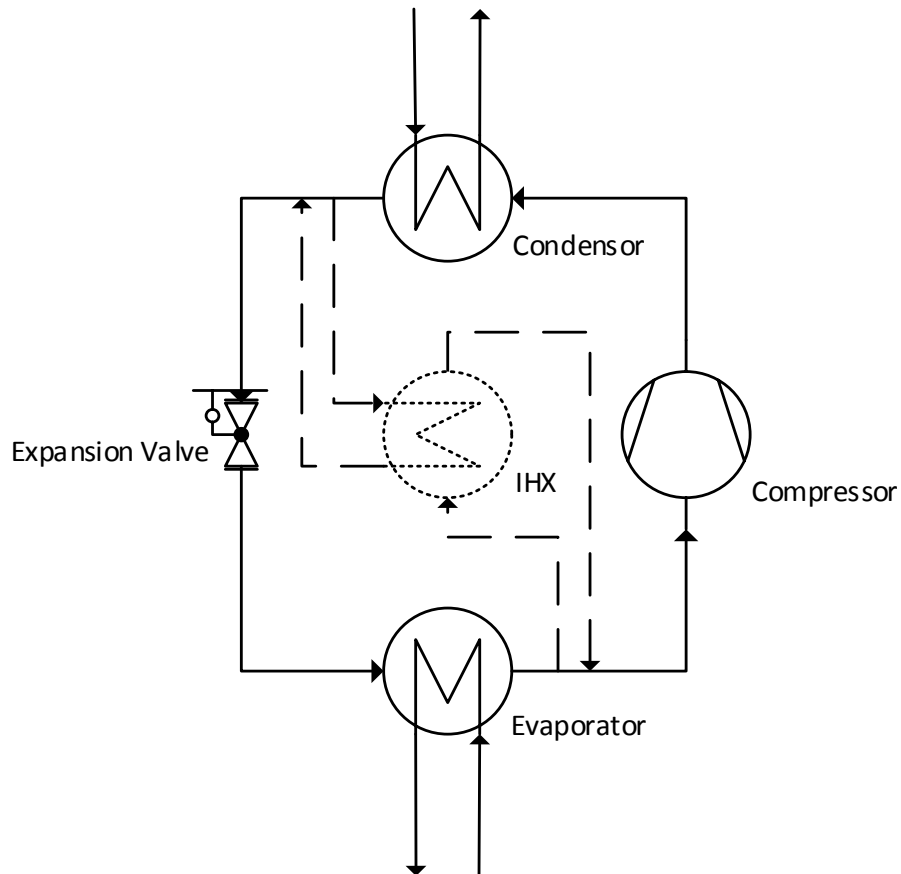
Figure 2.1 Basic Layout of a Typical Refrigeration System

In the basic system shown above, the two heat exchangers are heated/cooled with water. The water flows in a counter flow direction compared to the gas flow. Extra components that can be added to the refrigeration system, depending on the system requirements, include suction line accumulator, receiver, oil separator, oil cooler, oil reservoir, filters and driers, high-to-mid-pressure expansion valves amongst others.

A carbon dioxide system is unique in the sense that the operating pressures in the high-pressure line are extremely high, while the critical temperature of the gas remains relatively low.

The refrigerant is not toxic, but can prove deadly in high enough concentrations through asphyxiation. In addition, during a safety discharge on the low-pressure side, the refrigerant exiting the system is below 0 °C and exiting at a high velocity. The discharge can cause severe injuries to skin or eyes.

Typical components used in a refrigeration system are designed with a maximum operating pressure of 46 bar. In terms of CO_2, this is merely

12 °C. Ambient conditions in South Africa are around 20 °C and thus ambient air cannot be used to cool the CO_2 in the condenser. Typical gas-cooler or condenser outlet temperature (for carbon dioxide) is 35 °C and this is above the critical temperature of the gas. Specialised components – that can operate under higher pressures – need be specifically designed for CO_2 systems. Danfoss Group, known for their various refrigeration components and complete systems, has designed certain components for CO_2 refrigeration cycles. These components include expansion valves, controllers, and small compressors (although the Danfoss line of CO_2 compressors has been discontinued).

ESK Schultze also has a couple of components specifically for CO_2: suction line accumulators, filter/driers, oil receivers, etc. However, most of their CO_2 components still focus on subcritical CO_2 refrigeration systems. Recently, in 2013, ESK has started to focus more on transcritical CO_2 refrigeration systems.

Temprite has specialised components for the oil line of CO_2 systems that can operate safely under transcritical conditions. Their units will be used within this project's experimental setup. They are the only company, known to the author, which manufactures and supplies transcritical oil units.

The compressor is a special unit supplied by Dorin, which is no longer in production. The compressor is designed to operate transcritically, but it can be configured to run subcritically.

Even the oil used for a CO_2 system is exceptional. Polyalkaline Glycol (PAG) oil is used with the CO_2 compressor in this project. It is an expensive oil that is not always readily available.

Most of the CO_2 refrigeration systems are theoretical in design only. The majority of them are never realised, built and tested. As such, it is very difficult to find information on test systems. A handful of systems have recently been installed in South Africa. However, most of these systems operate on a cascade method, or as a basic subcritical system where the air must be pre-cooled before entering the condenser. In South Africa, Pick 'n Pay, Macro and Coca Cola have a couple of CO_2 units installed. Woolworths seems to be the leader in the use of CO_2 refrigeration systems.

2.1.6 COMMON SAFETY HAZARDS

Several different properties must be taken into account when one considers the safety of a refrigerant. Is the refrigerant toxic? Is it flammable? Does it pose a risk to the environment? Is it an explosive hazard? Moreover, many more questions can be asked. The environmental risks have been addressed in 2.1.2 Global Warming and Ozone Depletion. This chapter focuses on the toxicity and flammability of refrigerants.

All refrigerants are classified within the ASHRAE 34 standard, (American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 2010) Table 2.5 lists a selection of refrigerants and their occupational exposure limit (OEL) in parts per million (ppm) and group classification as per their ASHRAE 34 classifications. The OEL is – according to an excerpt from ASHRAE 34 – "the time-weighted average (TWA) concentration for a normal eight-hour workday and a 40-hour workweek to which nearly all workers can be repeatedly exposed without adverse effect…"

Table 2.5 Hazards Rating of Refrigerants

ASHRAE NUMBER	TOXIC	FLAMMABLE	OEL PPM V/V	ASHRAE 34 GROUP
R11	No	No	C1 000	A1
R12	No	No	1 000	A1
R22	No	No	1 000	A1
R50	No	Yes	1 000	A3
R114	No	No	1 000	A1
R502	No	No	1 000	A1
R290	No	Yes	1 000	A3
R404A	No	No	1000	A1
R600	No	Yes	1 000	A3
R600a	No	Yes	1 000	A3
R717	Yes	Yes	25	B2
R718	No	No	-	A1
R729	No	No	-	-
R732	No	No	-	-
R744	No	No	5000	A1

(Calm, 1996) (American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 2010)

The ASHRAE 34 classification exists of an "A" or "B" – depending on toxicity – and number "1", "2" or "3" – depending on the flammability of the refrigerant.

2.1.6.1 TOXICITY

Refrigerants designated as Class A have a lower level of toxicity with a recommended OEL of 400 ppm or greater. Refrigerants designated as Class B have a higher degree of toxicity, with a recommended OEL of less than 400 ppm.

Some refrigerants classified as non-toxic can still be harmful in very high concentrations. An example is carbon dioxide. It is non-toxic in all regards,

but can cause asphyxiation and have an anaesthetic effect in high concentrations.

2.1.6.2 FLAMMABILITY

Flammability is classified into one of three classes – with the optional sub-class "2L" based on lower flammability limit testing, heat of combustion and the optional burning velocity measurement. Class 1 shows no flame propagation, class 2 shows some flame propagation and class 3 shows some flame propagation, mare than that of class 2. Further, information that is more detailed is available in Appendix H - Flammability

Table 2.6 Safety Group Classification

Safety Group

Increase Flammability →	Higher Flammability	A3	B3
	Lower Flammability	A2	B2
		A2L	B2L
	No Flame Propagation	A1	B1
		Lower Toxicity	Higher Toxicity

Table 2.6 shows the relationship between flammability and toxicity and their classification into the eight safety groups.

2.1.6.3 OTHER HAZARDS

Special components are required that can handle both the temperature and the pressure requirements of a CO_2 refrigeration system. Each component and its connection must be pressure tested to at least 1.3 times the operating pressure of that component. Failure of the component to handle the operating pressure can lead to catastrophic failure. In the case of a pressure vessel or valve, the unit might explode. Shrapnel from the unit can cause severe damage to any personnel in the vicinity. Tubing and fittings

undergo the same conditions and must also be certified and/or tested under the extreme conditions before the system is made operational.

Extreme temperatures in typical refrigeration systems also pose a threat to personnel. If tubing or a component is below 0 °C, the operator may obtain a "cold burn" upon touching the unit. The same is true for units and tubing that has a very high temperature. In severe cases, all units and tubing must be thoroughly insulated and barriers be placed to prevent the operator or any other personnel from touching the components.

Some refrigerants pose an explosive risk. When the refrigerant is exposed to a specific catalyst, it may react to the materials around it and explode.

In a typical refrigeration system, certain safety devices are installed to prevent the excessive increase in pressure. A safety pressure-relief valve is usually included. This unit can either be manually or electronically triggered, depending on the design of the unit. Once triggered, the valve discharges refrigerant to the environment to prevent a pressure build-up in the system. The discharge of the refrigerant can be harmful to personnel or the environment and special precautions must be made in such a case. Refrigerant discharge must always be directed away from personnel and sensitive equipment. In most cases, the discharge must take place outside the building. In the case of CO_2, the discharge must be to the outside environment or to a low point, such as a trench or drain. It is, however, extremely important that the CO_2 is not allowed to accumulate, as it will pose the risk of asphyxiation.

Under Chapter 5.5 is a list of safety precautions utilised in this project, along with a description of what each process includes.

2.2 CARBON DIOXIDE (R744) AS A REFRIGERANT FOR THIS PROJECT

Carbon dioxide is the refrigerant of choice for this project. Part of the project focus is on environmentally friendliness. CO_2 is one of the few natural refrigerants that are non-flammable and non-toxic. It has excellent thermophysical properties, has good heat transfer properties, is readily available at a low cost, is environmentally friendly and is non-corrosive.

CO_2 is, however, not an ideal gas to use for refrigeration purposes. The use of CO_2 in modern times is in its infancy. Therefore, initial system costs are still high. This will change as the industry focusses more on CO_2 refrigeration systems, but remains problematic for the time being.

Carbon dioxide has a low critical temperature, meaning that it cannot be used in all applications – for example using ambient air to cool the condenser where the ambient air temperature is above CO_2 critical temperature – while operating subcritically. The solution is to run a transcritical CO_2 system.

Most "high pressure" refrigeration components are rated with a maximum operating pressure of 46 bar. The critical pressure of CO_2 is 73.8 bar. The respective temperatures for 46 bar and 73.8 bar is 10.87 °C and 31.04 °C. Thus, special components need be designed, manufactured and utilized for a typical CO_2 system – especially for the high-pressure side of the system.

The following is a summary of the properties of CO_2, as listed in Section 2.1 above:

1. General Information
    1.1. Type of Refrigerant: Natural Gas
    1.2. ASHRAE Number: R744
    1.3. Molecular Formula: CO_2
    1.4. Chemical/Commercial Name: Carbon Dioxide
    1.5. Critical Temperature: 31.04 °C
    1.6. Critical Pressure: 73.8 bar

2. Environmental Properties
    2.1. Atmospheric Lifetime: 29 300 - 36 100 years
    2.2. ODP: 0
    2.3. GWP: 1

3. ASHRAE Classification
    3.1. Classification: A1
    3.2. ATEL: 40 000 ppm
    3.3. ODL: 140 000 ppm
    3.4. OEL: 5 000 ppm
    3.5. RCL: 40 000 ppm

ATEL – or acute-toxicity exposure limit – is the limit at which the refrigerant will pose a severe risk to any employees working in normally occupied, enclosed spaces.

ODL – oxygen deprivation limit – is the concentration of the refrigerant that will result in insufficient oxygen for normal breathing.

RCL – refrigeration concentration limit – the limit intended to reduce the risk of acute toxicity, asphyxiation and flammability hazards in normally occupied, enclosed spaces.

All limits and their classifications as in ASHRAE 34 (American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 2010).

2.3 THEORY OF OPERATION OF EXPERIMENTAL REFRIGERATION SYSTEM

This section deals with the design basics for the University of Stellenbosch's CO_2 refrigeration system. First is the difference between subcritical and transcritical operation, which is illustrated with pressure-enthalpy and temperature-entropy diagrams. This is followed by a short discussion of simulations and software used in the industry. Lastly, the theory covers some of the components used in the system.

2.3.1 SUBCRITICAL AND TRANSCRITICAL OPERATION

All traditional refrigeration systems operate subcritically. Transcritical operation is usually not needed. Carbon dioxide can be used successfully transcritically, with a high COP and a large temperature drop across the gas-cooler in the high-pressure region.

"Transcritical" and "subcritical" refers to the high-pressure side of the systems, as it operates above or below the critical point, respectively. Super-critical refers to the state of the refrigerant above the critical point. Super-heated and subcooled refers to the refrigerant state that is below the critical point and outside the two-phase region. Superheated refrigerant is vapour that is heated beyond the vapour saturation point, while subcooled refrigerant is liquid cooled below the liquid saturation point that is still in liquid form.

The four graphs given as Figure 2.2 to 2.5 show the difference between a typical transcritical CO_2 system and a typical subcritical CO_2 system. The graphs reflect ideal conditions throughout the system and using the steady-state simulation developed for this project. Refer to SC Simulation in Chapter 4.2. First is the pressure versus enthalpy diagrams of transcritical (Figure 2.2) and subcritical (Figure 2.3) systems, respectively. The two graphs shows the following: constant specific volume (blue); constant entropy (green); constant temperature (thin black); saturated liquid (thick black, left side); saturated vapour (thick black, right side) and the system operating conditions (red).

The pressure (P, [bar]) is indicated on the vertical axis, on a logarithmic scale. The horizontal axis is specific enthalpy (h, [kJ/kg K]). The pressure axis is given in bar and not MPa, as this is what the add-in uses and thus also what is used in all the simulation programs

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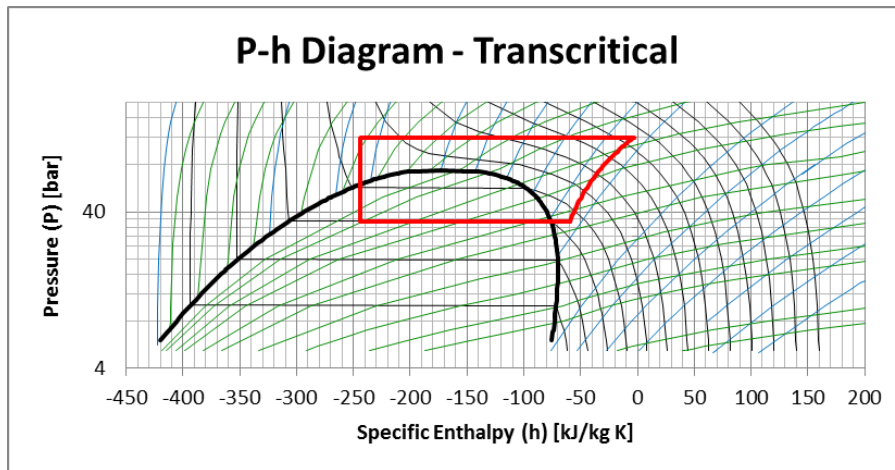
Figure 2.2 P-h Diagram – Transcritical System
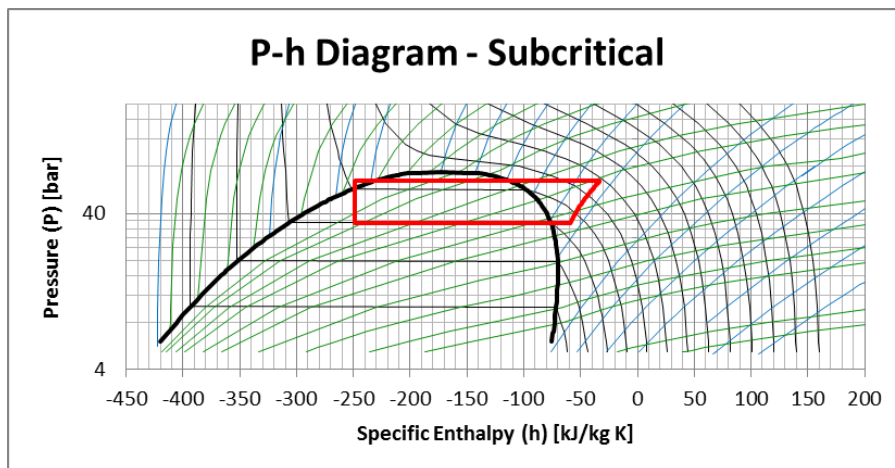
Figure 2.3 P-h Diagram – Subcritical System

In Figure 2.2, transcritical operation shows the high-pressure line above the critical point, and in Figure 2.3 the subcritical operation shows the high-pressure line below the critical point.

Figures 2.4 and 2.5 show: constant specific volume (blue); constant enthalpy (green); constant pressure (thin black); saturated liquid (thick black, left side); saturated vapour (thick black, right side) and the system operating conditions (red). The vertical axis is temperature (T, [°C]) and the horizontal axis is specific entropy (s, $[\text{kJ/kg K}]$).

17
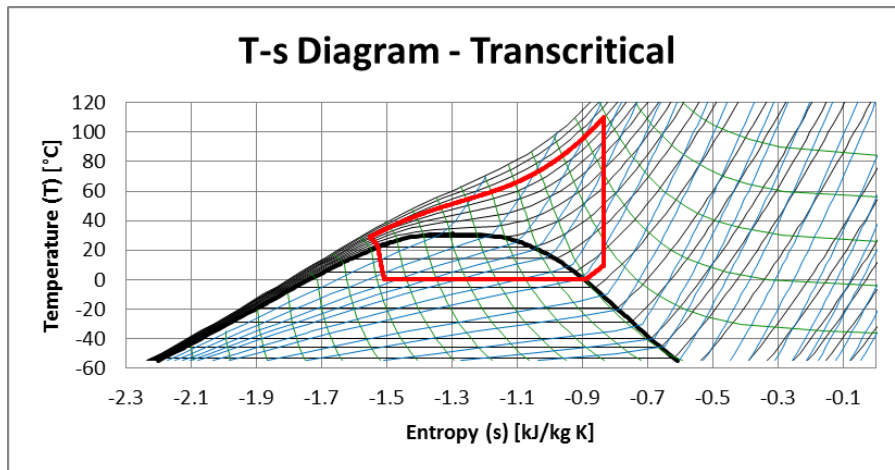
Figure 2.4 T-s Diagram – Transcritical System

Again, the difference between transcritical operation in Figure 2.4 and sub-critical operation in Figure 2.5 is obvious.
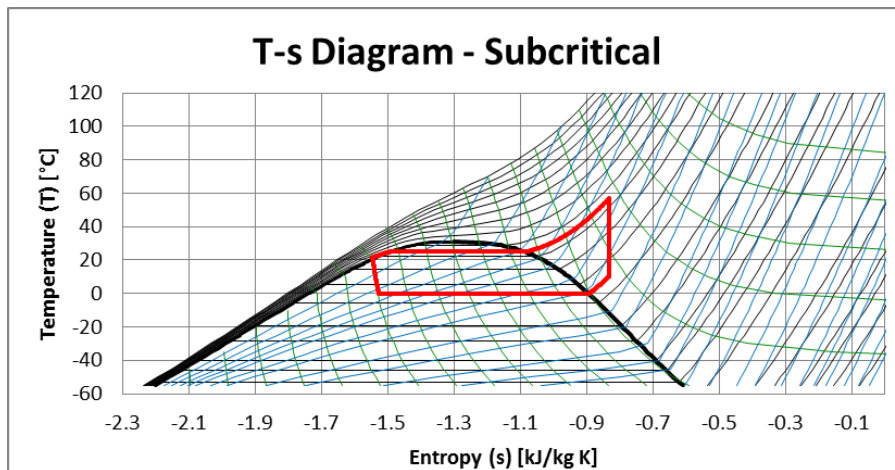
Figure 2.5 T-s Diagram – Subcritical System

In subcritical conditions, specifically in the two-phase region, there is only one free variable. Pressure and temperature are directly related to each other. In any other region, pressure and temperature are no longer directly related. Two free variables exist, i.e. at least two variables are required to determine either the pressure or the temperature at the given operating condition.

In the supercritical region, both vapour and liquid exists, but in such a state that they are indistinguishable from each other.

Danfoss conducted an experiment in 2003, to show the different phases and their appearances. A small vessel made in the Danfoss R&R Laboratory in Nordborg, Denmark has been tested and the results are well documented. A short video is available of the test and below is a selection of screenshots from the video. All images and the video itself are protected by copyright to Danfoss Industrial Refrigeration A/S.

Figure 2.6 (a) shows a photo of the test unit. The chamber has a large glass view port – through-which the operator can view the state of the CO_2. High pressure lines lead to the top of the unit and a discharge line leads from the left of the unit. Pressure is measured with both a high pressure gauge and a pressure transducer. The unit is charged with vapour CO_2 – keeping the discharge line shut-off – until the unit cell is sufficiently, visually filled with both liquid and vapour CO_2. Figure 2.6 (b) shows a graph with the four different phases of CO_2. At the start of the test, the unit is filled with CO_2 at room temperature and it's corresponding pressure on the diagram – such that it lies on the liquid-vapour line.
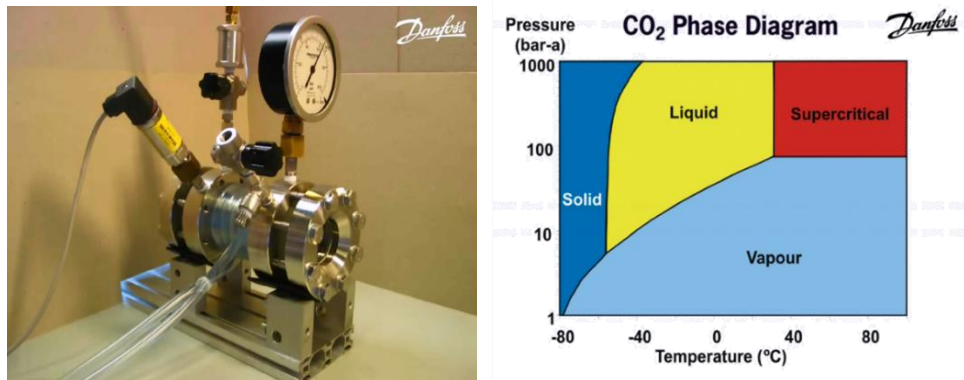
Figure 2.6(a) View of the CO_2 Test Vessel   (b) Phase Diagram of CO_2

Figure 2.7 (a) shows the state of the CO_2 inside the unit at the starting position. Both liquid and vapour are present in the unit and they are in equilibrium.

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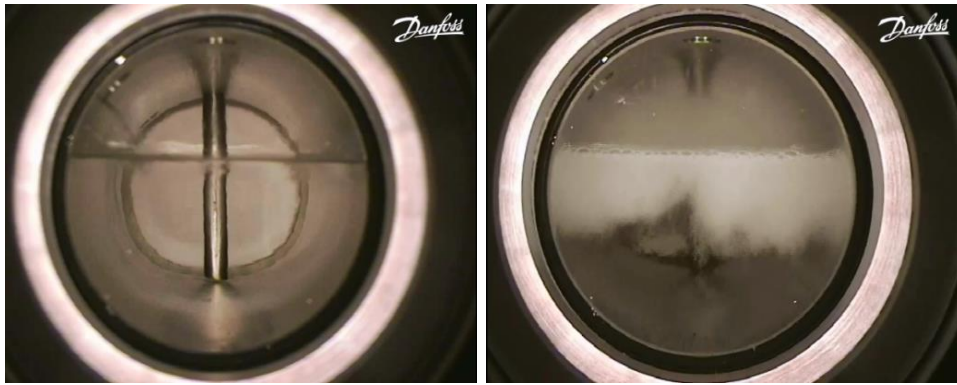
Figure 2.7(a) Start Position   (b) Reducing Pressure – Danfoss

The pressure is first reduced in the unit, by opening the discharge vent valve. Opening the valve allows vapour CO_2 to escape. Temperature is reduced via forced evaporation, as liquid boils to produce more vapour. This is visible in Figure 2.7(b).
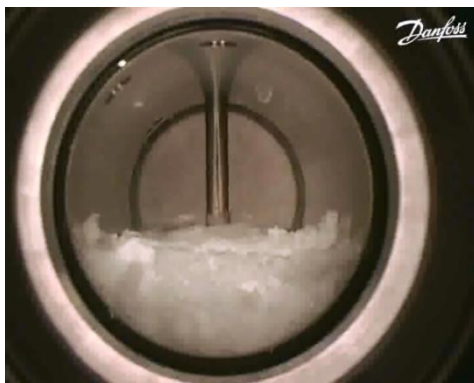
Figure 2.8 Triple Point – Danfoss

The triple point is reached at 5.2 bar-absolute and -56.6 °C. All three phases are present in the unit. As the pressure is reduced even further, only solid CO_2 will be present. Figure 2.8 shows the three phases in the test unit.

For the next test, the unit is returned to its starting conditions. Pressure is then increased by adding more CO_2 vapour to the unit. Figure 2.9 (a) shows the unit just below the critical point. The different phases are highlighted in Figure 2.9 (b) – red showing vapour and blue showing liquid.
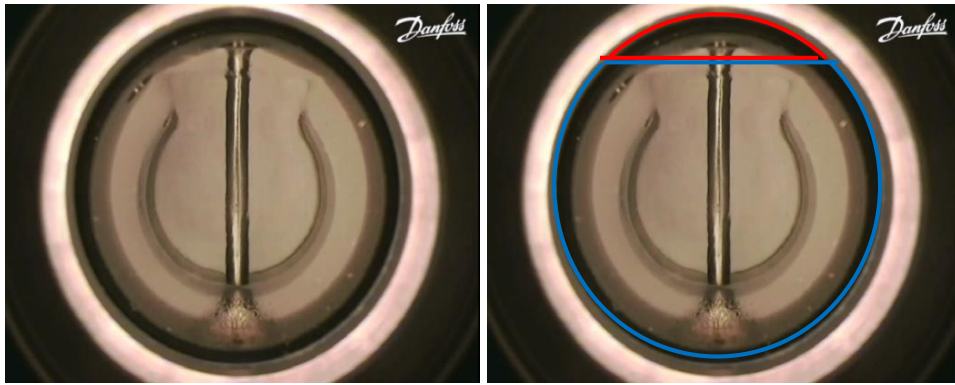
Figure 2.9(a) Below Critical Point   (b) Below Critical Point (2) - Danfoss

As the pressure is increased further, the CO_2 passes through the critical point. The CO_2 is now in the supercritical phase. In Figure 2.10, CO_2 is present in such a state that liquid and vapour are not discernible from each other.
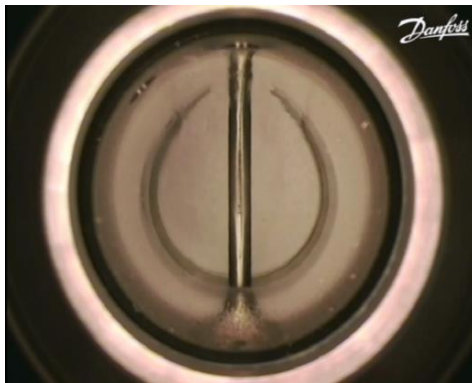
Figure 2.10 Above Critical Point

The discharge valve is then opened, allowing the pressure in the unit to reduce. Just below critical point, vapour and liquid separate and vapour bubbles form and escape to top. The boiling is visible in Figure 2.11 (a).
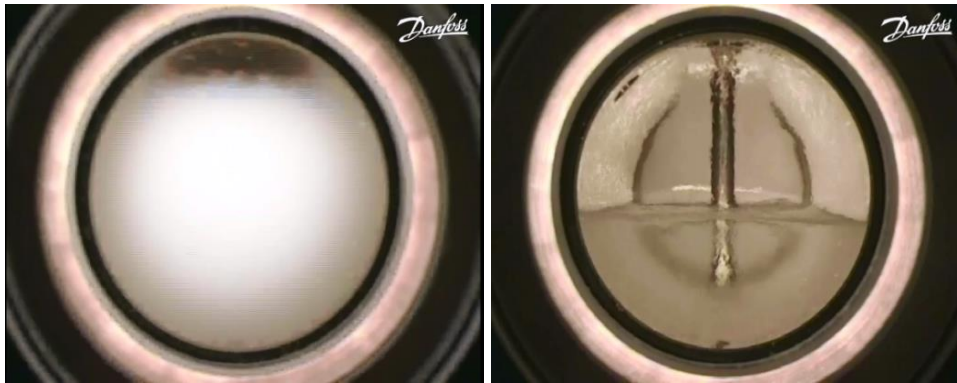
Figure 2.11(a) Lowering Pressure   (b) End Condition - Danfoss

Moments later, the two phases are clearly visible again. The unit now has conditions similar to the start-up conditions, as can be seen in Figure 2.11 (b).

(Danfoss - Refrigeration and Air Conditioning, 2011)

2.3.1 SIMULATIONS AND SOFTWARE

Software often used for system simulation includes FORTRAN, Engineering Equation Solver (EES) and Matlab. Complete software packages are available, at incredible cost, from a handful of researchers and companies. These program are often designed to simulate very specific conditions, e.g. to determine the required length of a capillary tube. (Madsen, et al., 2005) Danfoss has several software packages available for free download from their website. These programs, however, aim to aid the user in choosing the correct Danfoss products for their system. Most of the programs do not have the option of choosing CO_2 as a refrigerant.

For CO_2 specifically, there are a couple of *add-in*s and software packages available for purchase from a variety of resources. These packages are designed to aid programmers who wish to simulate a CO_2 system. The most affordable package is an *add-in* for Microsoft Excel. With the *add-in*, the user can determine all the fluid and transport properties of CO_2. Mega-WatSoft sells this *add-in*. The author developed all the simulation programs for this project in Microsoft Excel, using VBA code and the MegaWatSoft *add-in*.

2.3.2 EXPANSION DEVICES

The purpose of an expansion device is to regulate the flow from the high-pressure side to the low-pressure side. There are several types of expansion devices. A short list on the types of expansion devices available are: capillary tubes, thermostatic expansion valves, electronic expansion valves, low- and high-pressure float valves.

One of the most common types of expansion device in use is a capillary tube, which is a tube with a very small inner diameter, when compared to the tubing in the rest of the system. Tubing with a higher inner surface resistance will also restrict the flow more. The thinner the tube, the shorter the capillary tube needs to be. Capillary tubes are commonly found in smaller systems – such as in domestic or small commercial fridges. While a capillary tube has a fixed inner diameter and a fixed length it can still be successfully utilised with a high coefficient of performance (COP) and under a variety of operating conditions.

Thermostatic expansion valves (TEVs) have a built-in mechanism that detects the amount of superheat in the suction line of the compressor. Superheat is generally used to ensure that no liquid refrigerant enters the compressor. Should the amount of superheat reduce, the TEV will start to close and allow less refrigerant to pass through, until the desired amount of superheat is once again achieved. TEVs have robust, physical connections to the system. This calls for careful planning on system layout.

Electronic expansion valves (EEVs) offer a finer degree of control over a wider range of capacities, compared to TEVs. Furthermore, EEVs have electrical connections that allow for more flexible system layouts. The biggest drawback of an EEV is that it must be used in conjunction with a controller and various sensors. The controller receives information of the refrigerant state - via pressure and/or temperature sensors placed on, before or after the evaporator – and sends an output signal to the EEV. This signal controls the opening degree of the valve. Two basic types of EEVs exist: a continuous flow type and a pulse width modulating (PWM) type. With the continuous flow type, the opening degree of the valve – or rather, the size of the orifice in the valve – is controlled with a stepper motor. The PWM type valve receives an electric signal that either completely opens or completely closes the valve. Other forms of EEVs also exist, combining characteristics of the PWM and continuous flow types.

Some evaporators – specifically flooded evaporators – require the tubing within the evaporator to be wetted at all times. A float valve can be installed in the evaporator to ensure that the evaporator is constantly filled with a sufficient amount of fluid. This type of valve is called a low-pressure float valve and cannot close completely. The valve will continuously allow refrigerant to flow into the evaporator, increasing the amount of fluid before the compressor inlet. It can prove problematic if the fluid reaches the compressor inlet. A high-pressure float valve is basically the exact opposite of a low-pressure float valve. The high-pressure variant ensures that there is no fluid

within the condenser by controlling the pressure within the condenser. (Hundy, et al., 2008)

CO_2 transcritical systems can have a wide variety of expansion devices. Li and Groll (2005) studied a cycle with an ejector-expansion device. They concluded that a CO_2 cycle could be improved by more than 16% just by utilizing the ejector-expansion device. Sarkar (2009) made a theoretical study into cycle optimization using a vortex tube expansion device. While these studies are not directly connected to the project at hand, it does give insight into theoretical study procedures. These kinds of studies, where other types of expansion devices are used, also offer an understanding of the different kinds of expansion devices and their individual influence on CO_2 cycles.

Agrawal and Bhattacharyya (2008) made a study on the performance of capillary tubes against expansion valves in a transcritical CO_2 system. Their study included using several capillary tubes with varying diameters, lengths and internal surface roughness'. Their system consisted of a basic refrigeration system, only including a liquid separator before the compressor - to protect the compressor. They set up a mathematical model wherein they calculate the necessary parameters in order to obtain the optimum capillary tube length. The mathematical model was then generated into a computer simulation of the system. In their conclusion, they state that it is indeed possible to theoretically determine the optimum length of capillary tube necessary for optimum performance of a CO_2 system.

2.3.3 HEAT EXCHANGERS

A heat exchanger is a unit designed to transfer heat from one medium to another as efficiently as possible. There are large varieties of heat exchangers currently on the market. E.g. plate, plate and shell, plate fin, pillow plate, shell and tube, tube-in-tube, fluid, dynamic scraped surface, phase-change heat exchangers and waste heat recovery units. These are all examples of heat exchangers with a physical wall between the two mediums. Some heat exchangers, where the same medium is used – but usually in different phases – are called a direct contact heat exchanger. The optimum type of heat exchanger to use depends on the heating and cooling media and the goal of the heat exchanger. An example: Cold air is blown through a refrigeration unit that stores food. A normal method to cool the air would be with a forced air a fin heat exchanger.

The heat exchangers used in this project are all tube-in-tube heat exchangers. These are effective units when water is used as the cooling medium in a condenser/gas-cooler and as a heating medium in the evaporator. While a multi-pass tube-in-tube, or otherwise known as a tube in shell heat exchanger, would be more effective and efficient, a basic tube-in-tube unit is easier to simulate. It, therefore, provides an easier learning curve for students studying the refrigeration system.

The heat exchanger in the system can run with either parallel or counter flow configurations. The flow direction of the water is easily changed by moving the water lines to the respective in- and outlets. The internal heat exchanger has fixed lines, but the direction of flow can be selected with special three-way valves. Figure 2.12 shows the difference between parallel flow (left) and counter flow (right) tube-in-tube heat exchangers.
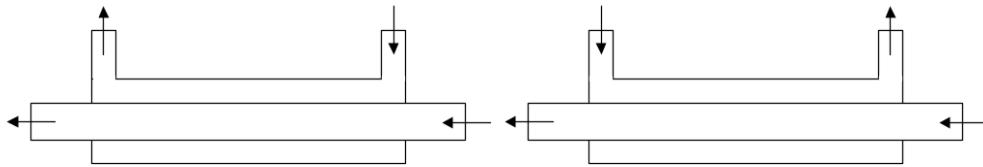
Figure 2.12 Tube-in-tube Heat Exchangers: Parallel Flow (Left) and Counter Flow (Right)

The most basic equations for a tube-in-tube heat exchanger are as follows:

The heat transferred Q is a function of the temperature difference ΔT and the heat transfer coefficient UA, or the temperature difference and the total thermal resistance R_{total}.

$$Q = \Delta T \, UA = \frac{\Delta T}{R_{\text{total}}} \quad [\text{W}] \tag{2.1}$$

Where the temperature difference ΔT is

$$\Delta T = (T_{\text{hot}} - T_{\text{cold}}) \quad [\text{K}] \tag{2.2}$$

The thermal resistance R_{total} is

$$R_{\text{total}} = R_{\text{conv},1} + R_{\text{wall}} + R_{\text{conv},2} \tag{2.3}$$

$$= \frac{1}{h_{\text{in}} \, A_{\text{z,in}}} + \frac{\ln\frac{r_{\text{o}}}{r_{\text{i}}}}{2\pi \, k \, L} + \frac{1}{h_{\text{out}} \, A_{\text{z,out}}} \tag{2.4}$$

Tao, et al. (2010) did an experimental study of the performance of CO_2 in a residential air-conditioning system. An internal heat exchanger (IHX), a liquid receiver and an oil separator into their system and an electronic expansion valve were included into the system. The system makes use of air-cooled fin-and-tube heat exchangers. The objective was to study the effects of working conditions on the system COP. This included changing the inlet temperatures of the air at both the evaporator and the gas cooler and changing the velocities of the air at both units. The conclusion was that the COP of the system is minimally affected by changes in both air velocity and air temperature at the evaporator side. It was found that the COP is greatly affected with change in air temperature and change in air velocity at the gas-cooler side. The COP is also affected by evaporator temperature, with

up to a 27% increase and a 20% decrease in COP. Finally, the design, using a fin-and-tube heat exchanger for the gas cooler, must be optimized to enhance heat transfer and thus improve the COP. From this research, the author noted that it would be far more effective and less problematic to use water-cooled heat exchangers.

2.3.4 ELECTRONIC CONTROLLERS

The electronic controllers are designed to monitor and control certain conditions of the refrigeration cycle. Each controller can be programmed to monitor their inputs and to optionally send out control signals to e.g. electronic valves. The inputs can be from temperature sensors, from pressure transducers, or from degrees that valves are open, (that is sent in the form of a voltage or amperage signal).

The controllers used with the system:

- AK-SM 350
  A system-monitoring unit with optional communication modules. The optional units include connection to a printer, a SMS module and a modem.

- AK-CC 550
  Primarily used to control the pressure in the low-pressure line. The unit includes compressor ON/OFF control, defrost control, product temperature measurement and control, a "door is open" function, and a light ON/OFF control. The controller has built-in algorithms that allows it to accurately control the amount of superheat in the system. This is called Adaptive Superheat Control. Further, the unit directly controls an electronic expansion valve, called the AKVH, and is capable of monitoring several temperature and pressure points.

- EKC 326A
  The EKC controller is used to control the pressure in the gas-cooler/high-pressure line. It does so by sending control signals to a high- and mid-pressure valve (the ICMTS and the CCM, respectively). It also has the capability to monitor several temperature and pressure points.

- GD Gas Detector
  The GD unit is a specialised carbon dioxide detector, accurately measuring the level of CO_2 at its location. It has low and high level alarms that is factory set to 5000 and 9000 ppm. These values can, however, be set to any values.

The AK-SM 350 unit is designed primarily to be a monitoring system. It can, however be used to program other controllers. The AK-SM 350 itself will not directly control any devices and will only generate alarm outputs.

The other controllers can be set to control certain aspects of the system, such as the opening degree of a valve. Each controller also has built-in alarm sub-routines. For the EKC 326A and the AK-CC 550, the alarm signals are sent to the AK-SM 350, where they are interpreted and the relative alarms signals are sent out.

The GD gas detector constantly monitors carbon dioxide levels. It has its own alarm relay, which is linked to alarm lights and a buzzer. The level of CO_2 is also sent from the GD unit to the AK-SM 350 controller.

The compressor is controlled by a VLT Micro Drive unit, which is in turn controlled by the AK-CC 550. The VLT drive controls the frequency of the motor. This is necessary to ensure a smooth, oscillation free start-up of the system. Frequency control can also be used to limit the refrigerant flow through the compressor, restricting the maximum and minimum pressure levels.

CHAPTER 3 THEORETICAL MODELLING

The equations used for the theoretical modelling are the equations of change, rules of heat transfer, the equations of state and the equations for the transport properties. These equations are applied to each 1-dimensional control volume of the discretised system.

3.1 GENERAL CONSERVATION EQUATIONS

The system simulated consists of a compressor, gas-cooler, internal heat exchanger, capillary tube, evaporator and an accumulator. Figure 3.1 Complete Discretised System is the system with each component discretised into a number of control volumes. The compressor and accumulator are seen as one control volume each. The equations (conservation of mass, energy and momentum) are applied to each individual control volume as shown by Figure 3.2, Figure 3.3 and Figure 3.4; for example the conservation of mass for the gas-cooler, the conservation of momentum for the capillary tube and the conservation of energy for the gas-cooler and the capillary tube.
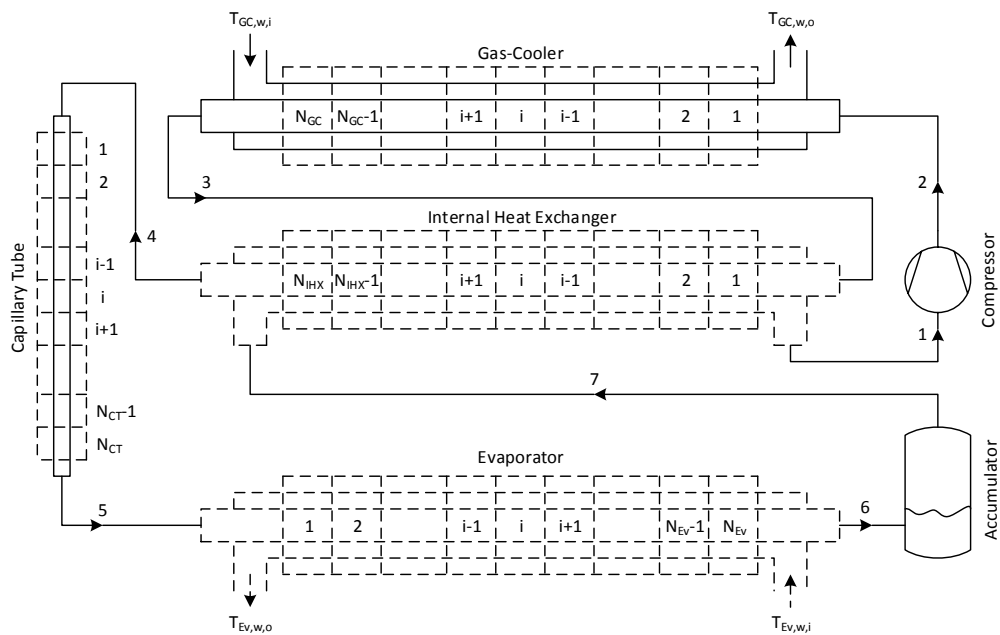
Figure 3.1 Complete Discretised System

The conservation of mass, for the gas-cooler, is illustrated in Figure 3.2 Gas-Cooler Conservation of Mass. The mass flow rate entering the gas-cooler does not necessarily equal the mass flow rate exiting the gas-cooler, for both the water and the refrigerant, under transient conditions.

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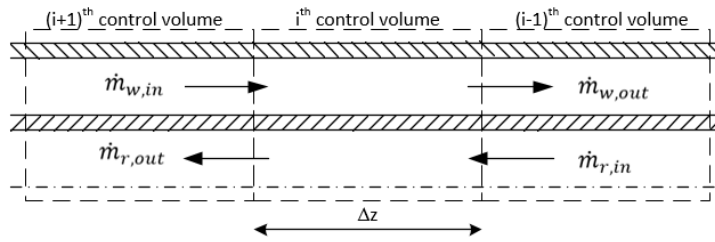
Figure 3.2 Gas-Cooler Conservation of Mass

For steady-state situations, the term on the left-hand side of each equation is set equal to zero.

With reference to Figure 3.2 the conservation of mass is:

$$\frac{\Delta m}{\Delta t} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \quad \left[\frac{\text{kg}}{\text{s}}\right] \qquad\qquad (\,3.1\,)$$

Figure 3.3 Gas-Cooler Conservation of Energy shows the conservation of energy for the i$^{\text{th}}$ control volume of the gas-cooler.
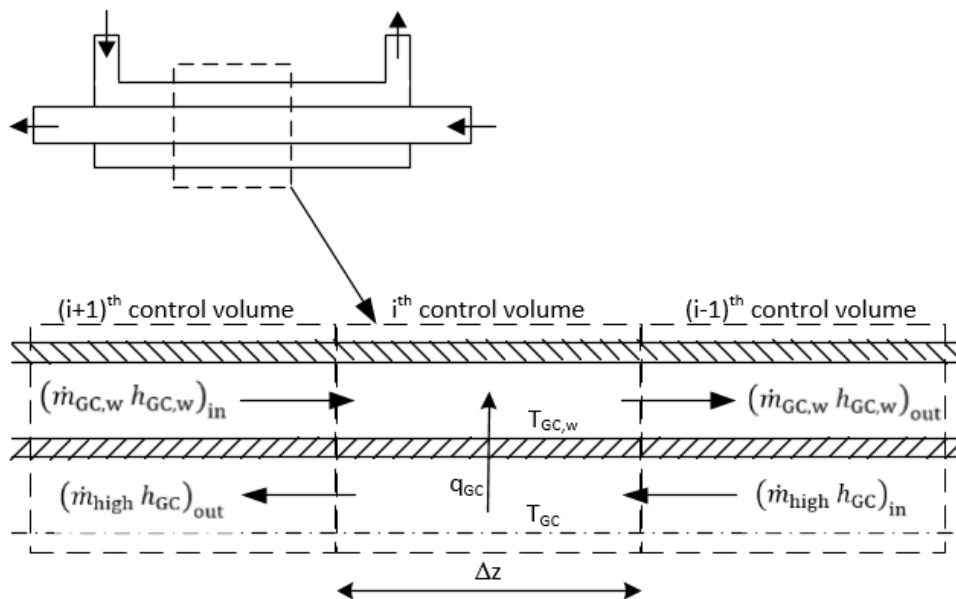
Figure 3.3 Gas-Cooler Conservation of Energy

The control volume is the same for all three heat exchangers, with the exception of the internal heat exchanger where there is refrigerant in both tubes.

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The capillary tube has a similar control volume, Figure 3.4 Capillary Tube Conservation of Energy. There is, however, no heat transfer from the refrigerant to the environment, as the capillary tube is assumed to be perfectly insulated.
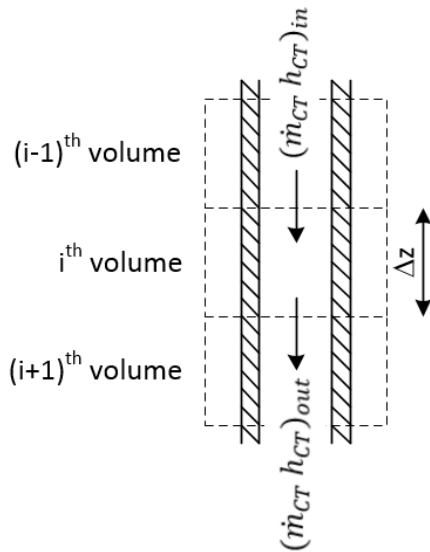
Figure 3.4 Capillary Tube Conservation of Energy

Conservation of energy is: the rate of change in energy ΔE is equal to the rate of energy entering the control volume $\dot{E}_{\text{in}}$ minus the rate of energy exiting the control volume $\dot{E}_{\text{out}}$.

$$\frac{\Delta E}{\Delta t} = \dot{E}_{\text{in}} - \dot{E}_{out} \ \ [\text{W}] \tag{3.2}$$

Ignoring kinetic and potential energy and work of expansion and work against friction,

$$\frac{\Delta E}{\Delta t} = (\dot{m} \, h)_{\text{in}} - (\dot{m} \, h)_{\text{out}} \pm \dot{Q} \ \ [\text{W}] \tag{3.3}$$

Figure 3.5 Capillary Tube Conservation of Momentum shows the conservation of momentum for the capillary tube, neglecting gravity. The pressure drop in heat exchangers is assumed zero, which is why the conservation of momentum is not applicable to the heat exchangers.

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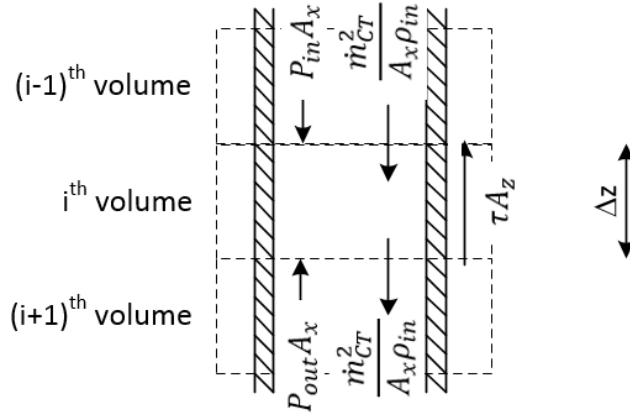
Figure 3.5 Capillary Tube Conservation of Momentum

The conservation of momentum is:

$$\frac{\Delta m\, v}{\Delta t} = (\dot{m}\, v)_{\text{in}} - (\dot{m}\, v)_{\text{out}} + \sum F_z \quad [\text{N}]$$ (3.4)

The sum of the forces is the pressure difference acting on the control volume $\left(P_{i-\frac{1}{2}} - P_{i+\frac{1}{2}}\right) A_{\text{x}}$, the shear force on the wall of the tube $\tau_{\text{h}}\, \wp\, \Delta z$ and the effect of gravity $\rho_{\text{h}}\, A_{\text{x}}\, \Delta z\, \text{g}\, \sin\phi$.

$$\sum F_z = \left(P_{i-\frac{1}{2}} - P_{i+\frac{1}{2}}\right) A_{\text{x}} - \overline{\tau}\, \wp\, \Delta z - \overline{\rho}\, A_{\text{x}}\, \Delta z\, \text{g}\, \sin\phi \quad [\text{N}]$$ (3.5)

The cross-sectional area is $A_{\text{x}} = 2\pi\, r_i^2$ and the perimeter is $\wp = 2\pi\, r_i$, where r_{i} is the inner radius of the tubing, $\overline{\tau}$ is the shear stress, and Δz is the length of the control volume. The density is $\overline{\rho}$. The gravitational force is given as $\text{g}\, \sin\phi$, where $\sin\phi$ is used to correct to the angle of the tube.

Carbon dioxide is not ideal over the range of pressure and temperature of the system and hence the Span and Wagner (2003) equation of state has been used. It may be given as

$$\begin{aligned}
\alpha(\tau,\delta) &= \alpha^{\text{o}}(\tau,\delta) + \alpha^{\text{r}}(\tau,\delta) \\
&= \alpha^{\text{o}}(\tau,\delta) + n_1\, \delta\, \tau^{0.250} + n_2\, \delta\, \tau^{1.250} + n_3\, \delta\, \tau^{1.500} + n_4\, \delta^3\, \tau^{0.250} \\
&+ n_5\, \delta^7\, \tau^{0.875} + n_6\, \delta\, \tau^{2.375}\, e^{-\delta} + n_7\, \delta^2\, \tau^{2.000}\, e^{-\delta} + n_8 \delta^5\, \tau^{2.125}\, e^{-\delta} \\
&+ n_9\, \delta\, \tau^{3.500}\, e^{-\delta^2} + n_{10}\, \delta\, \tau^{6.500}\, e^{-\delta^2} \\
&+ n_{11}\, \delta^4\, \tau^{4.750}\, e^{-\delta^2} + n_{12}\, \delta^2\, \tau^{12.50}\, e^{-\delta^3}
\end{aligned}$$ (3.6)

MegaWatSoft – a company that specialises in engineering and scientific software for calculations in computational thermodynamics – has a Microsoft Excel *add-in* that can be used to determine the thermodynamic and transport properties of CO_2. The *add-in* uses the equations of state, as presented by Span and Wagner (2003). For transport properties, the *add-in* makes use of

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equations presented in "The Viscosity of Carbon Dioxide" (Fenghour, et al., 1998) and "The Transport Properties of Carbon Dioxide" (Vesovic, et al., 1988).

The MS Excel *add-in* has a validity range of 0 to 100 MPa and -56.8 to 750.6 °C. The *add-in* can calculate values for 29 different variables, while supplying 11 different combinations of input variables. Furthermore, the *add-in* provides these functions and variables in either SI units or English units. A list of property IDs and constant values used within the *add-in* is available in Appendix A – MegaWatSoft Inc. Microsoft Excel *Add-in* Information.

3.2 WATER/STEAM PROPERTIES

The properties for water and steam are taken from the steam tables in the text-book Heat and Mass Transfer – A Practical Approach. (Cengel, 2006) In Excel, a lookup-table is used to determine the corresponding H_2O properties.

3.3 TRANSIENT EQUATIONS OF CHANGE

Transient equations are basically equations that include the change a property or group of properties undergoes with time. Within a transient simulation, the equations of change thus retain their Δt component. The goal of the transient equations is to determine the properties for the current control volume, for the next time step.

For the explicit method of the finite difference approach, it is important to keep the time steps small. Larger time steps will cause instability in the program.

Looking first at the equation for the conservation of mass:

$$\frac{\Delta m}{\Delta t} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \quad \left[\frac{\text{kg}}{\text{s}}\right] \tag{3.7}$$

In terms of a transient solution, the explicit numerical solution,

$$m_i^{t+\Delta t} = m_i^t + \Delta t \left(\dot{m}_{i-\frac{1}{2}}^t - \dot{m}_{i+\frac{1}{2}}^t \right) \quad [\text{kg}] \tag{3.8}$$

Likewise, for the conservation of momentum:

$$\frac{\Delta mv}{\Delta t} = (\dot{m}v)_{\text{in}} - (\dot{m}v)_{\text{out}} + \sum F_{\text{z}} \quad [\text{N}] \tag{3.9}$$

$$m_i^{t+\Delta t} \, v_i^{t+\Delta t} = m_i^t \, v_i^t + \Delta t \left(\dot{m}_{i-\frac{1}{2}}^t \, v_{i-\frac{1}{2}}^t - \dot{m}_{i+\frac{1}{2}}^t \, v_{i+\frac{1}{2}}^t + \left(P_{i-\frac{1}{2}}^t - P_{i+\frac{1}{2}}^t \right) A_x - \right.$$
$$\left. \bar{\tau}_i^t \, \wp \, \Delta z - \bar{\rho}_i^t \, A_x \, \Delta z \, \text{g} \sin \phi_i \right) \quad [\text{N s}] \tag{3.10}$$

For the conservation of energy,

$$\frac{\Delta U}{\Delta t} = (\dot{m}\,h)_{\text{in}} - (\dot{m}\,h)_{\text{out}} \pm \dot{Q} \quad \left[\frac{\text{kJ}}{\text{s}}\right] \tag{3.11}$$

$$U_i^{t+\Delta t} = U_i^t + \Delta t \left(\dot{m}_{i-\frac{1}{2}}^t h_{i-\frac{1}{2}}^t - \dot{m}_{i+\frac{1}{2}}^t h_{i+\frac{1}{2}}^t \pm \dot{Q}_i^t \right) \ [\text{kJ}] \tag{3.12}$$

Converting to specific internal energy u,

$$u_i^{t+\Delta t} m_i^{t+\Delta t} = u_i^t m_i^t + \Delta t \left(\dot{m}_{i-\frac{1}{2}}^t h_{i-\frac{1}{2}}^t - \dot{m}_{i+\frac{1}{2}}^t h_{i+\frac{1}{2}}^t \pm \dot{Q}_i^t \right) \ \left[\frac{\text{kJ}}{\text{kg}}\right] \tag{3.13}$$

Furthermore, with $m_i^{t+\Delta t}$ know from the conservation of continuity above,

$$u_i^{t+\Delta t} = \frac{u_i^t m_i^t}{m_i^{t+\Delta t}} + \frac{\Delta t}{m_i^{t+\Delta t}} \left(\dot{m}_{i-\frac{1}{2}}^t h_{i-\frac{1}{2}}^t - \dot{m}_{i+\frac{1}{2}}^t h_{i+\frac{1}{2}}^t \pm \dot{Q}_i^t \right) \ \left[\frac{\text{kJ}}{\text{kg}}\right] \tag{3.14}$$

The thermal heat transfer rate $\dot{Q}$ is assumed to have an always-positive value. Heat transferred to the control volume adds to the equation, while heat transferred out of the control volume is detracted from the equation, hence the "$\pm$" symbol.

3.4 STEADY-STATE CONSERVATION EQUATIONS

Referring back to the conservation equations, the term on the left-hand side of each equation is set equal to zero – as $\Delta t \to \infty$. For mass

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \quad \left[\frac{\text{kg}}{\text{s}}\right] \tag{3.15}$$

Alternatively, the mass flow rate $\dot{m}$ is constant.

For momentum,

$$0 = (\dot{m}\,v)_{\text{in}} - (\dot{m}v)_{\text{out}} + \sum F_z \ [\text{N}] \tag{3.16}$$

Substituting $v_{\text{in}} = \frac{\dot{m}_{\text{in}}}{\bar{\rho}_{\text{in}}\,A_x}, \ v_{\text{out}} = \frac{\dot{m}_{\text{out}}}{\bar{\rho}_{\text{out}}\,A_x}$ and equation (3.5)

$$0 = \left(\frac{\dot{m}^2}{\bar{\rho}\,A_x}\right)_{\text{in}} - \left(\frac{\dot{m}^2}{\bar{\rho}\,A_x}\right)_{\text{out}} + (P_{\text{in}} - P_{\text{out}})\,A_x - \bar{\tau}\,\wp\,\Delta z - \bar{\rho}\,A_x\,\Delta z\,\text{g}\,\sin\phi \ [\text{N}] \tag{3.17}$$

Dividing by $(A_x\,\Delta z)$

$$0 = -\frac{1}{\Delta z\,A_x^2}\left(\frac{\dot{m}_{\text{out}}^2}{\bar{\rho}_{\text{out}}} - \frac{\dot{m}_{\text{in}}^2}{\bar{\rho}_{\text{in}}}\right) - \frac{P_{\text{out}} - P_{\text{in}}}{\Delta z} - \frac{\bar{\tau}\,\wp}{A_x} - \bar{\rho}\,\text{g}\,\sin\phi \ \left[\frac{\text{Pa}}{\text{m}}\right] \tag{3.18}$$

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This gives the equation in terms of pressure gradients: momentum, pressure, friction and gravity, respectively,

$$0 = -\left(\frac{\Delta P}{\Delta z}\right)_{\text{M}} - \left(\frac{\Delta P}{\Delta z}\right)_{\text{P}} - \left(\frac{\Delta P}{\Delta z}\right)_{\text{F}} - \left(\frac{\Delta P}{\Delta z}\right)_{\text{G}} \left[\frac{\text{Pa}}{\text{m}}\right]$$
(3.19)

For a heat exchanger, the Log Mean Temperature Difference (LMTD) method is used. Here the LMTD temperature in a control volume of the gas-cooler $T_{\text{LMTD,GC},i}$ is given as

$$\Delta T_{\text{LMTD,GC},i} = \frac{\Delta T_{\text{GC,in},i} - \Delta T_{\text{GC,out},i}}{\ln\left(\frac{\Delta T_{\text{GC,in},i}}{\Delta T_{\text{GC,out},i}}\right)}$$
(3.20)

Where the temperature on the "in" side of the control volume is: for parallel flow

$$\Delta T_{\text{GC,in},i} = T_{\text{GC,hot},i-\frac{1}{2}} - T_{\text{GC,cold},i-\frac{1}{2}}$$
(3.21)

And for counter flow

$$\Delta T_{\text{GC,in},i} = T_{\text{GC,hot},i-\frac{1}{2}} - T_{\text{GC,cold},i}$$
(3.22)

In addition, the temperature on the "out" side of the control volume is for parallel flow

$$\Delta T_{\text{GC,out},i} = T_{\text{GC,hot},i} - T_{\text{GC,cold},i}$$
(3.23)

For counter flow

$$\Delta T_{\text{GC,out},i} = T_{\text{GC,hot},i} - T_{\text{GC,cold},i-\frac{1}{2}}$$
(3.24)

3.5 TWO-PHASE FLOW EQUATIONS

Various methods to model two-phase flow exist. It is important to choose the correct method – with respect to the application.

Agrawal and Bhattacharyya (2008) studied the flow model present within capillary tubes in a refrigeration system. They compared homogeneous two-phase flow with separated two-phase flow and concluded that the separated two-phase flow models give reasonably accurate predictions. The discrepancy between the separated flow model and the homogeneous model is at most only 8% higher. They recommend in their conclusion that a homogeneous model can used and is preferred for its simplicity. Therefore, a homogeneous flow model is adopted for this project, wherever two-phase flow is present.

With a homogeneous model, the two-phase mixture within a control volume is replaced with a single-phase, incompressible fluid. (Dobson, 2013) It is further assumed that both phases are well mixed and have equal velocities. Thus,

$$v_\mathrm{g} = v_\ell = v \ \left[\frac{\mathrm{m}}{\mathrm{s}}\right] \tag{3.25}$$

Furthermore, there exists a homogeneous density ρ_h and void fraction α_h given as

$$\rho_\mathrm{h} = \rho_\mathrm{g}\,\alpha_\mathrm{h} + \rho_\ell\,(1 - \alpha_\mathrm{h}) \ \left[\frac{\mathrm{kg}}{\mathrm{s}}\right] \tag{3.26}$$

And

$$\alpha_\mathrm{h} = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_\mathrm{g}}{\rho_\ell}\right)\left(\frac{v_\mathrm{g}}{v_\ell}\right)} \tag{3.27}$$

Void fraction is also given as

$$\alpha = \frac{v_\mathrm{g}}{v_\mathrm{g} + v_\ell} \tag{3.28}$$

It can easily be seen that – when substituting α_h into ρ_h – homogeneous density is

$$\rho_\mathrm{h} = \frac{1}{\frac{x}{\rho_\mathrm{g}} + \frac{1-x}{\rho_\ell}} \ \left[\frac{\mathrm{kg}}{\mathrm{s}}\right] \tag{3.29}$$

Where x is the mass fraction,

$$x = \frac{\dot{m}_\mathrm{g}}{\dot{m}_\mathrm{g} + \dot{m}_\ell} \tag{3.30}$$

A homogeneous dynamic viscosity μ_h is be defined as

$$\mu_\mathrm{h} = \frac{1}{\frac{x}{\mu_\mathrm{g}} + \frac{1-x}{\mu_\ell}} \ [\mu\mathrm{Pa\ s}] \tag{3.31}$$

Similarly, the homogeneous thermal conductivity k_h is expressed as,

$$k_\mathrm{h} = \alpha_\mathrm{h}\,k_\mathrm{g} + (1 - \alpha_\mathrm{h})\,k_\ell \ \left[\frac{\mathrm{W}}{\mathrm{K\,m}}\right] \tag{3.32}$$

In addition, the specific isobaric heat capacity c_p is

$$c_{p,\mathrm{h}} = x\,c_{p,\mathrm{g}} + (1 - x)\,c_{p,\ell} \ \left[\frac{\mathrm{kJ}}{\mathrm{kg\,K}}\right] \tag{3.33}$$

Lastly, the homogeneous enthalpy h_h is

$$h_{\mathrm{h}} = \left(h_{\mathrm{g}} - h_{\ell}\right) x + h_{\ell} \text{ Alternatively } h_{\mathrm{h}} = h_{\mathrm{g}} x + (1 - x) h_{\ell} \quad \left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right] \tag{3.34}$$

Evaluating the conservation equations (3.18) to (3.19):

$$0 = -\frac{1}{\Delta z\, A_x^2} \left(\frac{\dot{m}_{\mathrm{out}}^2}{\bar{\rho}_{\mathrm{out}}} - \frac{\dot{m}_{\mathrm{in}}^2}{\bar{\rho}_{\mathrm{in}}}\right) - \frac{P_{\mathrm{out}} - P_{\mathrm{in}}}{\Delta z} - \frac{\bar{\tau}\, \wp}{A_x} - \bar{\rho}\, \mathrm{g}\, \sin\phi \quad \left[\frac{\mathrm{N}}{\mathrm{m}^3} \text{ or } \frac{\mathrm{Pa}}{\mathrm{m}}\right] \tag{3.35}$$

$$0 = -\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{M}} - \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{P}} - \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F}} - \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{G}} \quad \left[\frac{\mathrm{Pa}}{\mathrm{m}}\right] \tag{3.36}$$

The friction pressure gradient $\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F}}$ in the modified conservation of motion equation is often expressed along with a two-phase friction multiplier ϕ_i^2. (Dobson, 2013)

$$\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F}} = \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F},i} \phi_i^2 \quad \left[\frac{\mathrm{Pa}}{\mathrm{m}}\right] \tag{3.37}$$

Where $i = \ell$ (liquid), ℓo (liquid only), g (gas) or go (gas only)

For homogeneous flow, the friction multiplier is derived as:

$$\frac{\tau_{\mathrm{h}}\, \wp}{A_{\mathrm{x}}} = \left(C_{f\mathrm{h}}\, \rho_{\mathrm{h}}\, \frac{v_{\mathrm{h}}^2}{2}\right) \frac{\wp}{A_{\mathrm{x}}} \quad \left[\frac{\mathrm{N}}{\mathrm{m}^3}\right] \tag{3.38}$$

$$= \left(C_{f\ell\mathrm{o}}\, \rho_{\ell}\, \frac{v_{\ell\mathrm{o}}^2}{2}\, \frac{\wp}{A_{\mathrm{x}}}\right) \left(\frac{C_{f\mathrm{h}}}{C_{f\ell\mathrm{o}}}\, \frac{\rho_{\mathrm{h}}}{\rho_{\ell}}\, \frac{v_{\mathrm{h}}^2}{v_{\ell\mathrm{o}}^2}\right) \quad \left[\frac{N}{m^3}\right] \tag{3.39}$$

Substituting $v = \frac{\dot{m}}{\rho\, A_{\mathrm{x}}}$ and simplifying,

$$= \left(C_{f\ell\mathrm{o}}\, \rho_{\ell}\, \frac{v_{\ell\mathrm{o}}^2}{2}\, \frac{\wp}{A_{\mathrm{x}}}\right) \left(\frac{C_{f\mathrm{h}}}{C_{f\ell\mathrm{o}}}\right) \left(\frac{\rho_{\ell}}{\rho_{\mathrm{h}}}\right) \quad \left[\frac{N}{m^3}\right] \tag{3.40}$$

$$= \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F},\ell\mathrm{o}} \phi_{\ell\mathrm{o}}^2 \quad \left[\frac{N}{m^3}\right] \tag{3.41}$$

Where the two-phase friction multiplier is now,

$$\phi_{\ell\mathrm{o}}^2 = \left(\frac{C_{f\mathrm{h}}}{C_{f\ell\mathrm{o}}}\right) \left(\frac{\rho_{\ell}}{\rho_{\mathrm{h}}}\right) \tag{3.42}$$

The friction pressure gradient for liquid only is,

$$\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{F},\ell\mathrm{o}} = C_{f\ell\mathrm{o}}\, \rho_{\ell}\, \frac{v_{\ell\mathrm{o}}^2}{2}\, \frac{\wp}{A_{\mathrm{x}}} \quad \left[\frac{\mathrm{Pa}}{\mathrm{m}}\right] \tag{3.43}$$

Using a Blassius type friction factor,

$$C_{f\ell\text{o}} = 0.078\,\text{Re}_{\ell\text{o}}^{-0.25}$$

(3.44)

In addition, the Reynolds Re number is a function of the mass flow rate $\dot{m}$, the inner diameter of the tube d_i, the dynamic viscosity μ and the cross-sectional area of the tube A_x:

$$\text{Re}_{\ell\text{o}} = \frac{\dot{m}d_i}{\mu_\ell\,A_x}$$

(3.45)

Likewise, for homogeneous values,

$$C_{f\text{h}} = 0.078\,\text{Re}_\text{h}^{-0.25}$$

(3.46)

$$\text{Re}_\text{h} = \frac{\dot{m}\,d_\text{i}}{\mu_\text{h}\,A_x}\ \ [\#]$$

(3.47)

Often, the homogeneous two-phase friction multiplier is, by assuming $C_{f\text{h}} \approx C_{f\ell o}$,

$$\phi_{\ell\text{o}}^2 \approx \frac{\rho_\ell}{\rho_\text{h}}$$

(3.48)

All two-phase theory derived from "Two-Phase Flow and Heat Transfer" (Dobson, 2013).

3.6 COMPRESSOR EFFICIENCY

The compressor used in the physical system is a semi-hermetic, single stage compressor. The compressor, while designed to run transcritically, can also be used with certain, restrictive, subcritical conditions. The compressor has an electrical and mechanical efficiency. If the compressor were assumed to run 100% efficiently, then the CO_2 would undergo isentropic compression. However, with the efficiencies not equal to 100%, the following equation is to be used: (Stoecker & Jones, 1982)

The actual volumetric efficiency of the compressor $\eta_{v\text{a}}$ is given as

$$\eta_{v\text{a}} = \frac{\dot{V}_{\text{suc,comp}}}{\dot{V}_{\text{disp,comp}}}\,100\,[\%],$$

(3.49)

Where $\dot{V}_{\text{suc,comp}}$ is the volume flow rate entering the compressor, and $\dot{V}_{\text{disp,comp}}$ is the displacement rate of the compressor. The value for the volumetric flow rate entering the compressor can only be determined after extensive testing of the compressor.

The compressor efficiency used in the transient simulation is,

$$\eta = \frac{\left(\frac{P_{in}}{P_{out}}\right)^{\frac{\kappa-1}{\kappa}} - 1}{\left(\frac{T_{out}}{T_{in}}\right) - 1} \ [\%] \tag{3.50}$$

Where the isentropic exponent is, (Fahlgren, 2003)

$$\kappa = f(P_{in}, h_{in}) \tag{3.51}$$

CHAPTER 4 COMPUTER SIMULATION PROGRAMS

Certain properties of CO_2 change dramatically near the critical point. Furthermore, upon testing CO_2 it must be noted that the state of CO_2 changes dramatically once it passes through the critical point. Where the CO_2 would be a mixture of fluid and vapour below the critical point, it changes into a mixture where neither fluid nor vapour is present. Instead, it is a mixture similar to superheated vapour, but with certain properties closer to that of its fluid form. Section 2.3.1 Subcritical and Transcritical Operation covers the test performed by Danfoss that further describes the phenomenon.

Three programs where developed to simulate various CO_2 systems and subsystems: The first is a steady-state simulation that determines the operating conditions, given certain physical and start-up conditions. The second simulation program is a simulator program, given the initial conditions, determines the operating conditions from start to steady-state conditions. The third simulation program simulates the operating conditions within a capillary tube.

Microsoft Excel is extensively used for all the simulation programs. The state and transport properties of CO_2 are determined from the CO2Tables *add-in* from MegaWatSoft. While MS Excel is a powerful program that can easily be utilised for complex simulations, it does have its limitations. MS Excel was not created with complex, large simulation programs in mind. Therefore, while it can solve the simulations, it takes a long time. As an example, for the transient simulation, one time step of 0.01 seconds can take up to a minute real time to calculate. Furthermore, it is advisable not to use the computer running the simulator for any other intensive programs. It is also advisable not to use any other Microsoft Office program and specifically not to use the copy-paste function. It is impossible to use MS Excel – a separate workbook – while the simulator is running. MS Excel can also be used to store an expansive database. The information in the database can then easily be accessed by VBA macros in MS Excel.

4.1 STEADY-STATE SIMULATION – SC SIMULATION V.5

The first simulation is a small steady-state simulator, using final operating conditions to determine the size of the heat exchangers and capillary tubes required in the system.
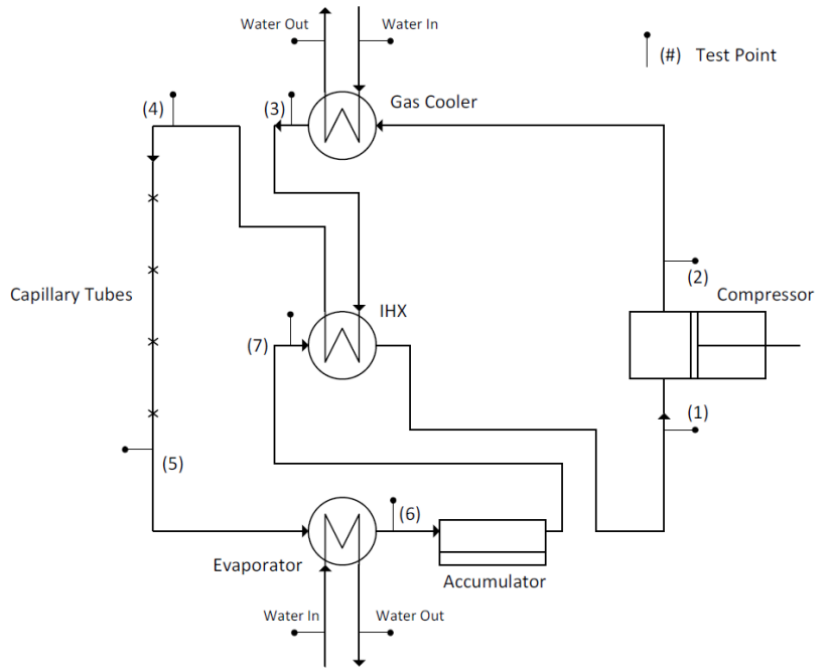
Figure 4.1 SC Simulation v.5 System Layout with Sensors

Figure 4.1 SC Simulation v.5 System Layout with Sensors is a line drawing layout of the system and its components to be simulated in SC Simulation v.5.

4.1.1 STEADY-STATE SIMULATION FLOW DIAGRAM

The steady-state simulator starts with a graphical user interface (GUI) that allows the user to choose desired operating conditions for the system. The simulation then calculates steady-state conditions for those operating conditions. The simulator ends with another GUI that shows the results to the user.

The following figure shows the flow diagram of the simulator.
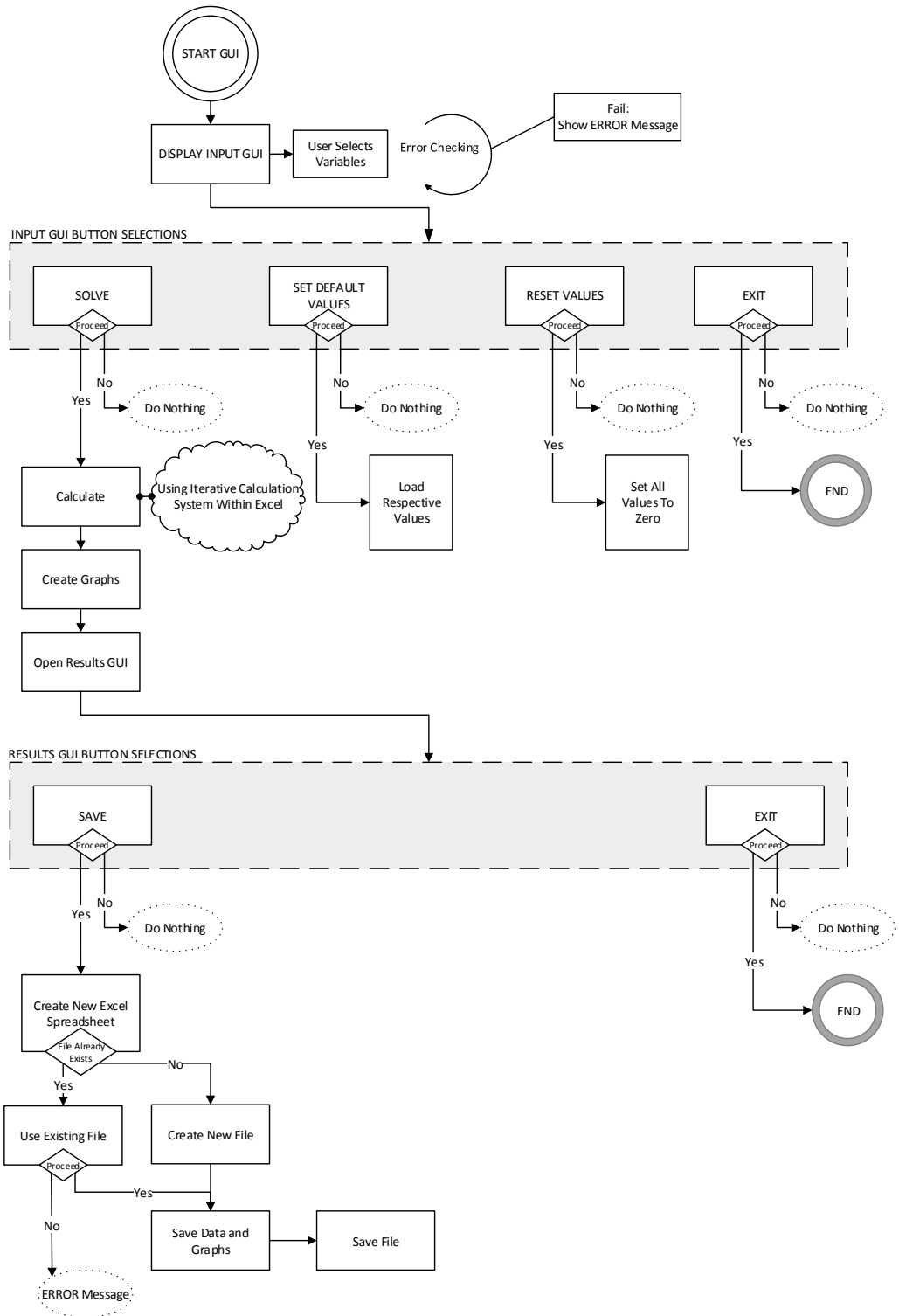
Figure 4.2 Steady-State Simulation Flow Chart

4.1.2 SC SIMULATION V.5 SCREENSHOTS AND STEP-THROUGH GUIDE

The simulation runs in MS Excel. As the user opens the simulator file (SC Simulation v.5.xlsm), he/she is greeted with a welcome screen, shown in Figure 4.3 Steady-State Simulation Welcome Screen.
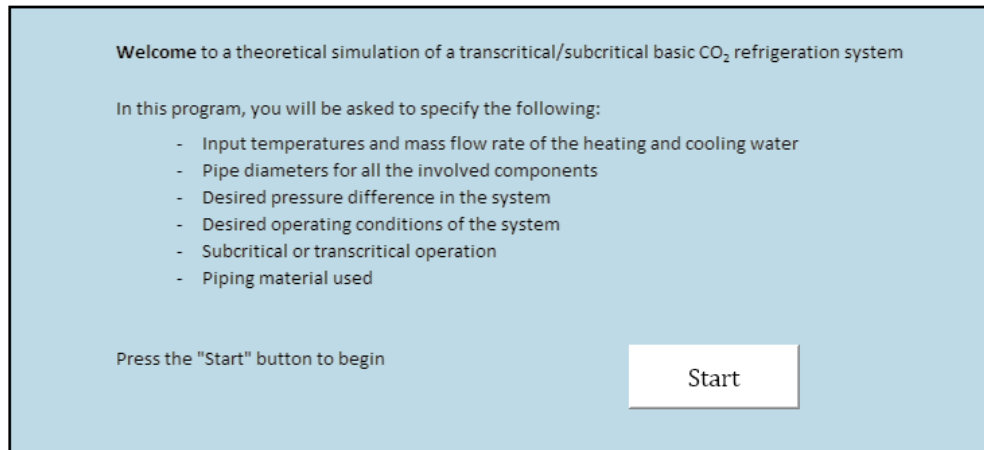
Figure 4.3 Steady-State Simulation Welcome Screen

Once pressing the "Start" button, the input screen, Figure 4.4, appears. Here the user chooses the desired operating conditions of the system, tubing diameters and tubing material, transcritical or subcritical operation, counter of parallel flow heat exchangers and to include or exclude the internal heat exchanger.
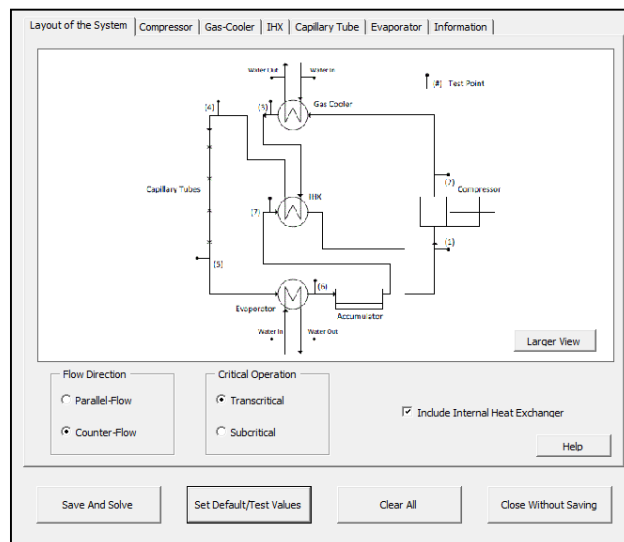
Figure 4.4 Steady-State Simulation Input Screen

The program will always check to ensure that the operator makes physically viable choices. I.e. The inner tube of the gas-cooler must not be equal to or greater than the outer tube of the gas-cooler.

This small, steady-state simulator uses the assumption that the system is running under all ideal conditions. It assumes the following:

- No pressure losses in the system due to friction, apart from in the capillary tube
- No heat transfer losses to the environment
- The compressor is running at 100% efficiency

Once the simulator has finished its calculations, a "Results" pop-up window appears for the user. This window contains numerical and graphical results of the simulation.
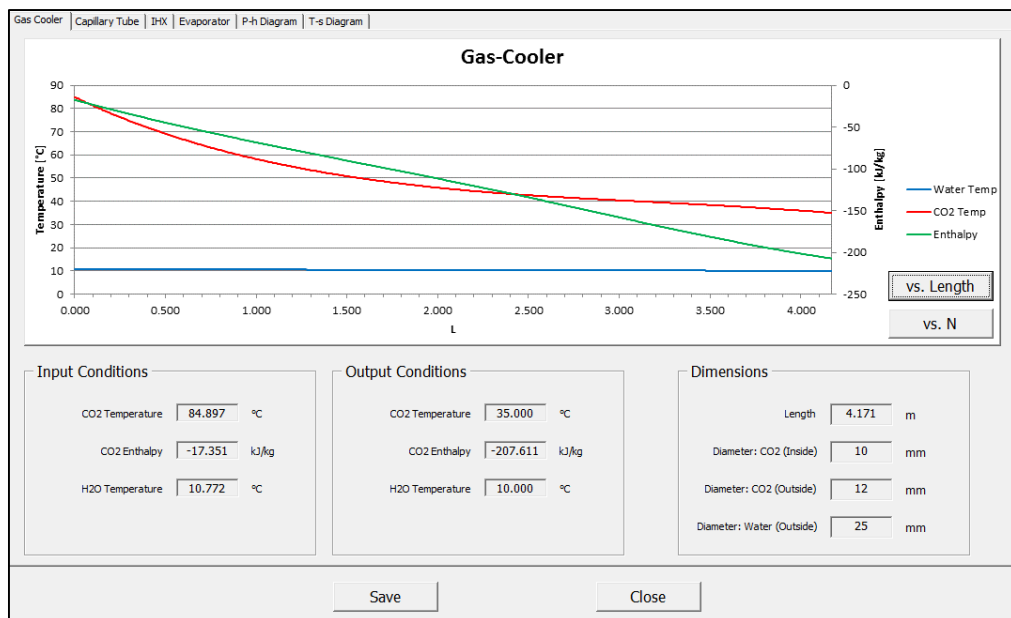
Figure 4.5 SC Simulation v.5 Gas-Cooler Results

Figure 4.5 is the results tab for the simulator, showing the gas-cooler results.

4.2 TRANSIENT SIMULATION – T SIMULATION V.1.5

The second simulation is a transient program. The simulation is written in Microsoft Excel. While the simulation seems to give accurate answers in the MS Excel environment, it takes a long time to calculate only a few time steps.

Given initial start-up conditions, this program simulates the thermal-hydraulic response of the system, until steady-state conditions have been attained. The system simulated is given in Figure 4.6 T Simulation v.1.5 System Layout with Sensors and uses the applicable theory as outlined in Chapter 3.3 Transient Equations of Change.
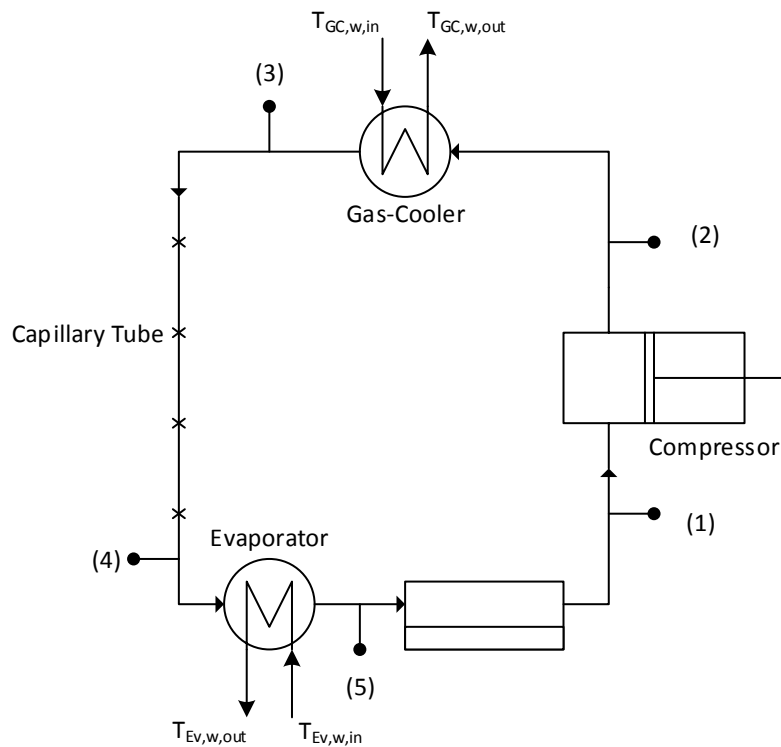
Figure 4.6 T Simulation v.1.5 System Layout with Sensors
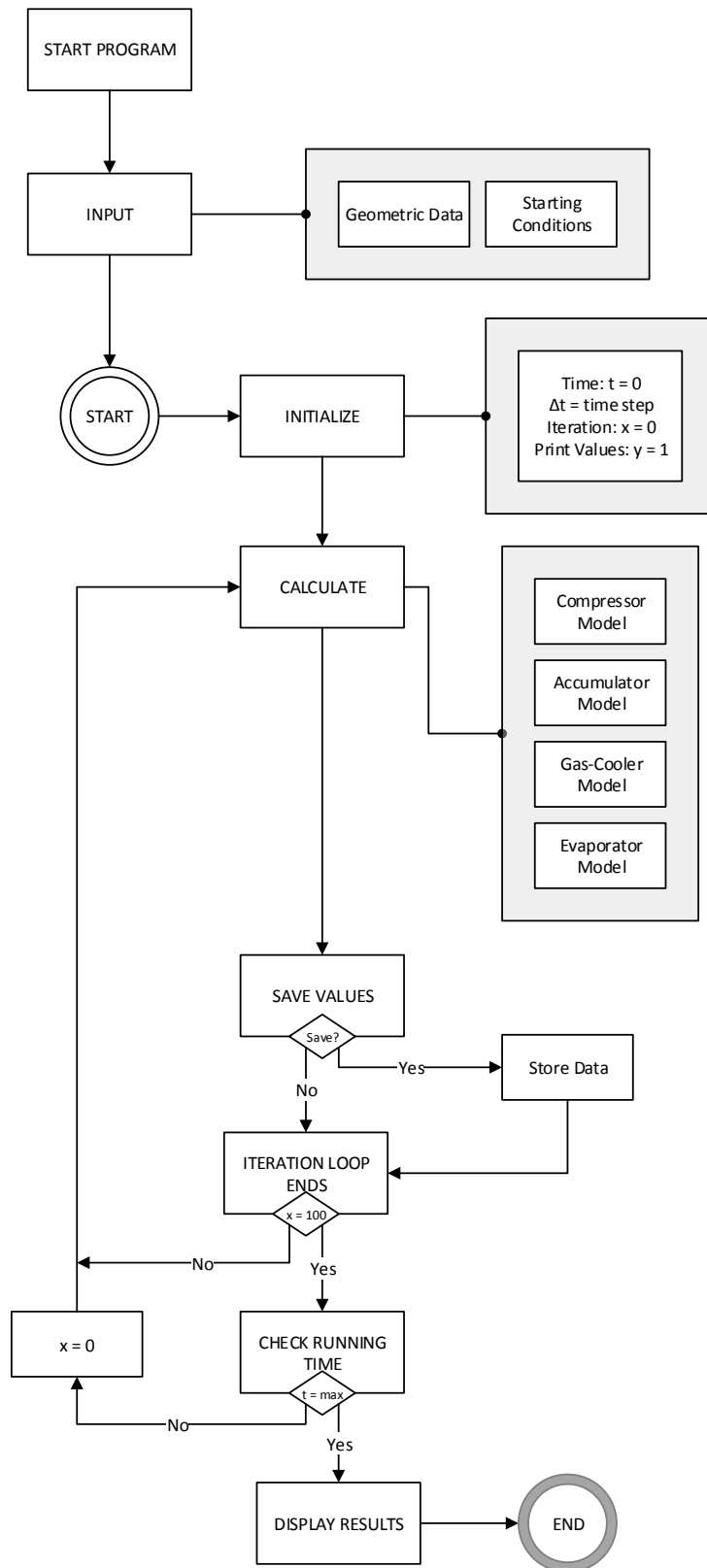
Figure 4.7 T Simulation v.1.5 Flow Diagram

4.3 CAPILLARY TUBE – CT SIMULATION V.1

The capillary tube simulation program is fundamentally the same as the capillary tube model used in SC Simulation. The program then determines the mass flow rate through the capillary tube, as required to meet the operating conditions. A difference between the CT Simulation and the SC Simulation program is that the former uses a large database to determine the fluid and physical properties of CO_2 where the latter uses the *add-in*. The database is based on the CO2Tables *add-in*. An excerpt of the database is shown in Appendix F - CO_2 Fluids Properties Database Excerpt.

Figure 4.8 is the discretised capillary tube model used in the simulator.
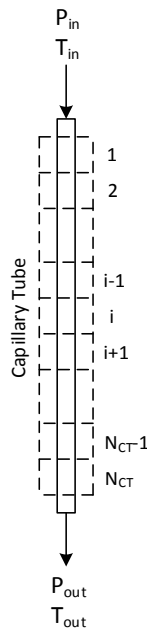
Figure 4.8 CT Simulation v.1 Discretised Capillary Tube

4.3.1 CT SIMULATION FLOW DIAGRAM

The capillary tube simulator starts by opening a graphical user interface (GUI) asking the user to choose between the transcritical and subcritical operating region. The user then chooses initial conditions for the simulation. The simulator ends with another GUI displaying the results of the simulation. Figure 4.9 CT Simulation v.1 Flow Diagram is the flow diagram of the capillary tube simulation.
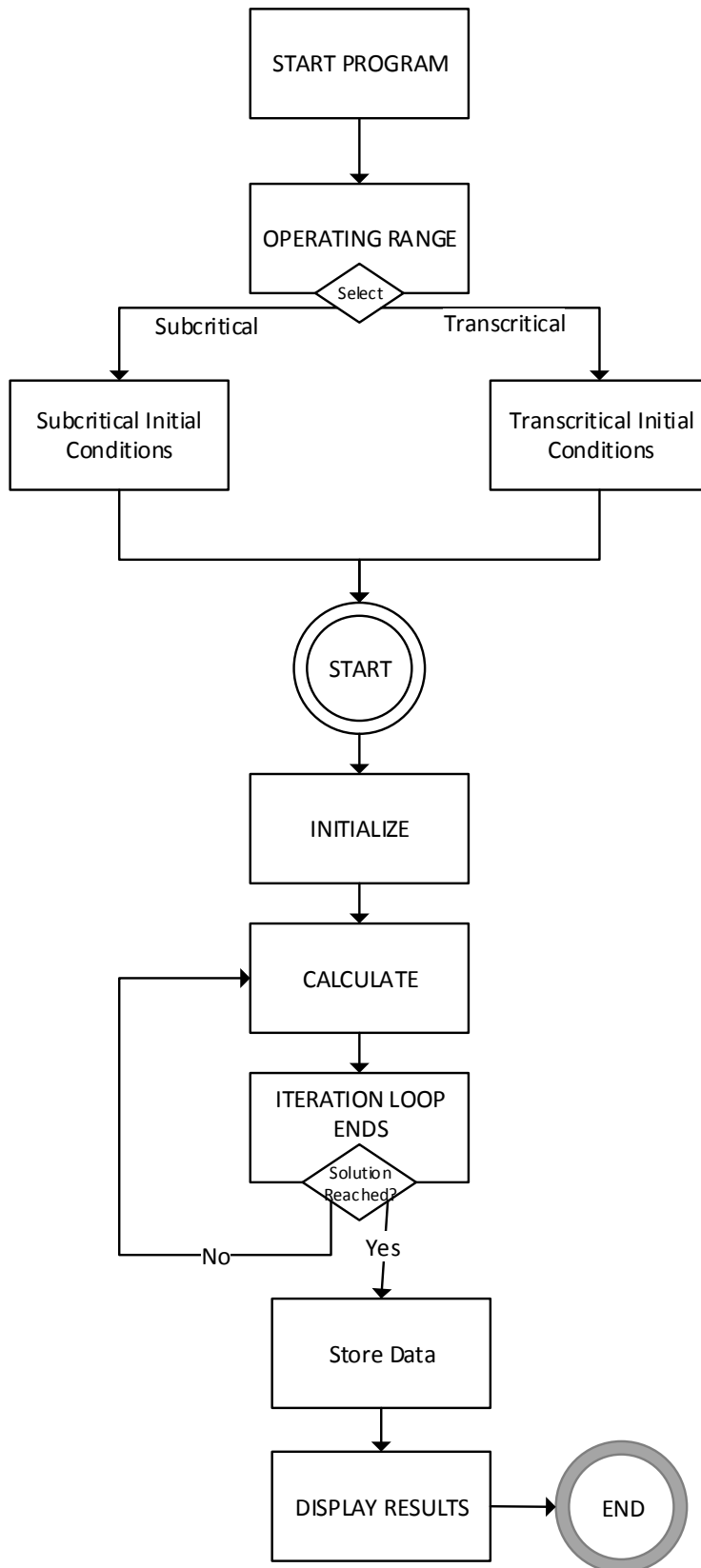
Figure 4.9 CT Simulation v.1 Flow Diagram

4.3.2 CT SIMULATION SCREENSHOTS AND STEP-THROUGH GUIDE

The following screenshots are from the CT Simulation program, from start-up to shut-down. A short description and instructions on how to use the program follows each figure.

As the file "Capillary Tine.xlsm" is opened, it immediately displays a *pop-up* with the choice between a subcritical and a transcritical operation region. Figure 4.10 displays the choose between subcritical or transcritical operation. The choice can be reversed on the next screen:

Figure 4.10 CT Simulation v.1 Subcritical and Transcritical Choice *Pop-Up*

Figure 4.11 (a) is the subcritical inputs screen and (b) is the transcritical input screen.

Figure 4.11 CT Simulation v.1 (a) Subcritical Inputs (b) Transcritical Inputs

The inputs for the subcritical operation include gas-cooler temperature, evaporator temperature, mass fraction of the CO_2 exiting the gas-cooler, the inner diameter of the capillary tube and the length of the capillary tube. The inputs for the transcritical system are similar, but instead of mass fraction, the pressure is chosen, and gas-cooler temperature is specifically the temperature of the CO_2 exiting the gas-cooler.

Because of the complex equations, the large database and the number of iterations, the simulation program takes a few hours to run. It is advisable to let the program run and to not have any other intensive program running on the computer.
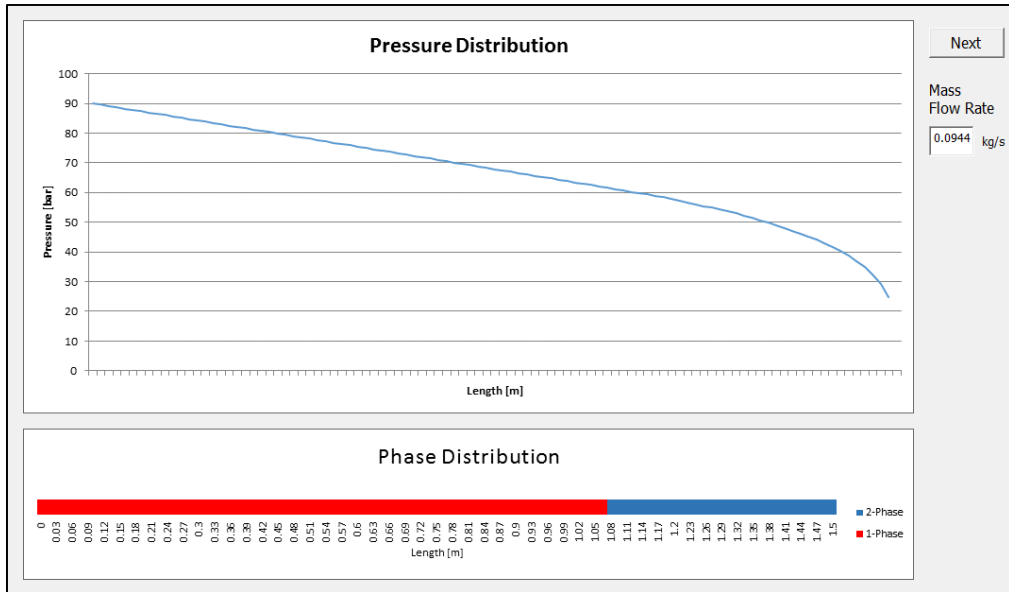
Figure 4.12 CT Simulation v.1 Results Screenshot

On this screen, the results are displayed. The graph contains the temperature and pressure distribution throughout the tube. It also includes a bar that indicate where single-phase and where two-phase flow is present in the tube. Numerical values include the mass flow rate, as calculated.

CHAPTER 5 EXPERIMENTAL SET-UP

The design, building and commissioning of a carbon dioxide refrigeration system; the use of CO_2 in commercial refrigeration systems, in modern times, is still a new and most companies guard their secrets closely. There are also very limited parts and components available on the market that can be used with the specialised field that is CO_2 refrigeration.

The design of the system includes: an exchangeable capillary tube expansion unit; an evaporator; a gas-cooler/condenser; an internal heat exchanger; safe, flexible operating conditions; an electronic expansion valve – as used commercially; full monitoring and control systems.

5.1 REFRIGERATION SYSTEM LAYOUT AND ITS COMPONENTS

The system consists of primary components (the heat exchangers, the compressor, the expansion units, etc.) and secondary components (the stand-by cooling unit, the evaporator defrost unit, etc.)

The heat exchangers and the capillary tubes were designed in SC Simulation v.5. All the heat exchangers are tube-in-tube units, with the gas-cooler/condenser and evaporator units cooled and heated by water, respectively. Figure 5.1 is the final layout of the system, including safety valves and the positions of the pressure transducers. Each component is numbered and is described in Table 5.1 Components Used in the System and Table 5.2 Components Used in the System (Contd.)

Twenty-six components are numbered and described. The fittings used throughout the system, as well as the tubing, have the same pressure and temperature ratings throughout. The same is true for the 3-way and 2-way shut off valves.
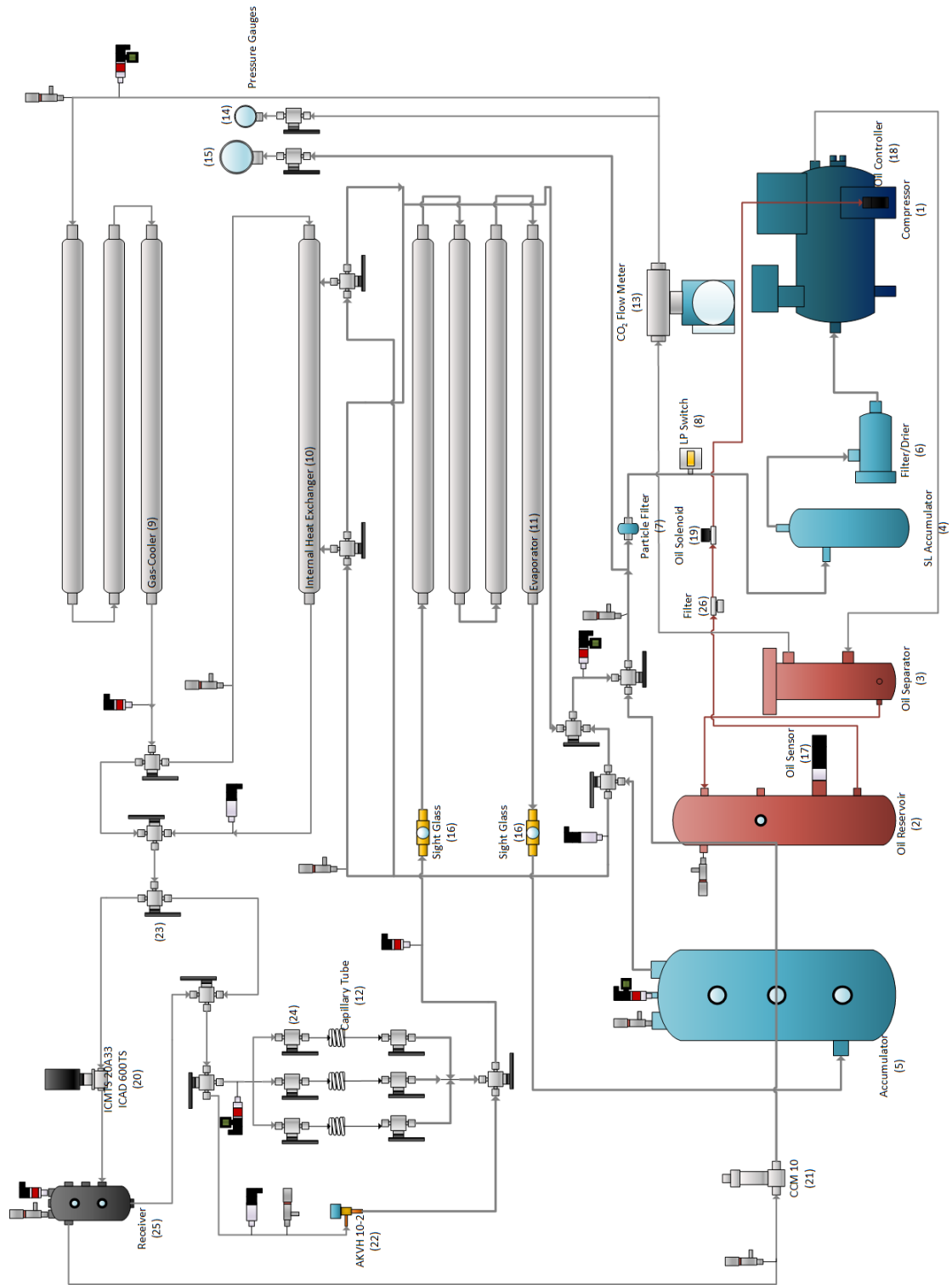
Figure 5.1 Full Layout of the Experimental System

Table 5.1 lists the mechanical components, their supplier, their function in the system and their maximum pressure rating.

Table 5.1 Components Used in the System

#	SUPPLIER	UNIT	MAXIMUM PRESSURE	USE IN SYSTEM
1	Dorin	TCS-340/D	163 bar (discharge) 100 bar (suction)	Provides compression
2	Temprite USA	Oil Reservoir RES 7	130 bar	Store oil for the compressor
3		Oil Separator 133A	130 bar	Separates oil from refrigerant
4	ESK Schultze	Suction Line Accumulator FA-12U-CDH	100 bar	Removes any liquid from the refrigerant
5		Accumulator SGS 32	60 bar	Stores refrigerant Separates liquid from vapour
6		Filter/Drier FT1-22-CD	60 bar	Dries refrigerant and removes particles
7		Particle Filter F-10L	60 bar	Removes particles and dirt
8	WIKA	LP Switch DCM 63	63 bar	Low-pressure protective switch
9	Custom	Gas-cooler	200 bar	Cooling of gas in high-pressure line
10	Custom	Internal Heat Exchanger	200 bar	Provide extra cooling and superheat
11	Custom	Evaporator	200 bar	Evaporation of liquid in the low-pressure line
12	Custom	Capillary Tubes	340 bar	Expansion of gas from the high-pressure line to the low-pressure line
13	Endress & Hauser	CO_2 Flow Meter	130 bar	Measure flow rate of CO_2
14	WIKA	Pressure Gauge (High)	160 bar	Measure pressure in high-pressure line
15		Pressure Gauge (Low)	60 bar	Measure pressure in low-pressure line
16	ESK Schultze	Sight Glasses	46 bar	Give visual reference of refrigerant
17	HB Products	Oil Sensor	130 bar	Detect oil in oil reservoir and separator

Table 5.2 Components Used in the System (Contd.)

#	SUPPLIER	UNIT	MAXIMUM PRESSURE	USE IN SYSTEM
18	HB Prod-ucts	Oil Controller	130 bar	Control oil flow to the compressor
19	HB Prod-ucts	Oil Solenoid	130 bar	Regulate oil flow to the compressor
20	Danfoss	Electronic Ex-pansion Valve ICMTS 20A33	140 bar	Expand high-pressure gas to mid-pressure
21	Danfoss	Electronic Ex-pansion Valve CCM 10	90 bar	Expand mid-pressure gas to low pressure
22	Danfoss	Electronic Ex-pansion Valve AKVH 10-2	90 bar	Expand high- or mid-pres-sure gas to low-pressure
23	Swagelok	3-Way Valve	206 bar	3-Way directional valve
24	Swagelok	2-Way Valve	206 bar	Shut off valve
25	Custom	Receiver	180 bar	Separate vapour and liq-uid in the mid-pressure line
26	Swagelok	Filter	350 bar	Filters out dirt particles in the oil line

The following component configurations are possible

1. Full expansion valve configuration
    a. ICMTS valve (High-pressure to mid-pressure)
    b. CCM valve (Mid-pressure to low-pressure)
    c. AKVH valve (High- or Mid-pressure to low-pressure)

2. Expansion valves with capillary tube
    a. ICMTS (High-pressure to mid-pressure)
    b. CCM (Mid-pressure to low-pressure)
    c. Capillary tube (High- or Mid-pressure to low-pressure)

3. AKVH only (High- or Mid-pressure to low-pressure)

4. Capillary tube only (High- or Mid-pressure to low-pressure)

5.1.1 TUBE-IN-TUBE HEAT EXCHANGERS

The three heat exchangers where designed with the steady-state simulator SC Simulation v.5. Tubing diameters were chosen based on readily available tubing and convenience. SC Simulation v.5 was run with different operating conditions and the sizes for the heat exchangers were chosen around these results. Table 5.3 lists the final dimensions for the internal heat exchangers:

Table 5.3 Heat Exchangers - Dimensions

HEAT EXCHANGER	INNER TUBE ID	INNER TUBE OD	OUTER TUBE ID	OUTER TUBE OD	LENGTH
Gas-Cooler	10 mm	12 mm	21 mm	25 mm	3
Evaporator	15 mm	18 mm	21 mm	25 mm	4
Internal Heat Exchange	15 mm	18 mm	21 mm	25 mm	1

5.1.2 CAPILLARY TUBES

The performance of a capillary tube in a subcritical system is very different from the performance of one in a transcritical system. In a subcritical system, the phases within the capillary tube can be either subcooled or two-phase flow. In some subcritical systems – where there is no subcooling – the only type of flow will be two-phase. A certain degree of subcooling is usually desired in a subcritical system, since it increases the COP of the system. For a transcritical system, there will always be supercritical, subcooled and two-phase flow.

The pressure gradient is much higher in 2-phase regions, due to the high density of the mixture compared to the density of single-phase transcritical gas. Thus, the length of a capillary tube for a transcritical system will be a considerably longer than for a subcritical system – assuming equal inner diameter and capillary tube inlet and outlet conditions.

Three capillary tubes have been installed on the system. However, due to the use of Swagelok fittings, the existing capillary lines can easily be switched with others. The three tubes installed have the following properties:

1. 1 m Length, 3 mm OD, 2 mm ID

2. 1.5 m Length, 3 mm OD, 2 mm ID

3. 1 m Length, 6 mm OD, 3 mm ID

5.1.3 RECEIVER – CUSTOM DESIGN UNIT

The high-pressure receiver is a custom unit designed by the author, along with Mainstream Refrigeration. Most receivers cannot withstand the extremely high pressures, while this unit is specifically designed for it. Figure 5.2 shows the final design drawings for the receiver. A larger format of the drawing is available in the addendum.
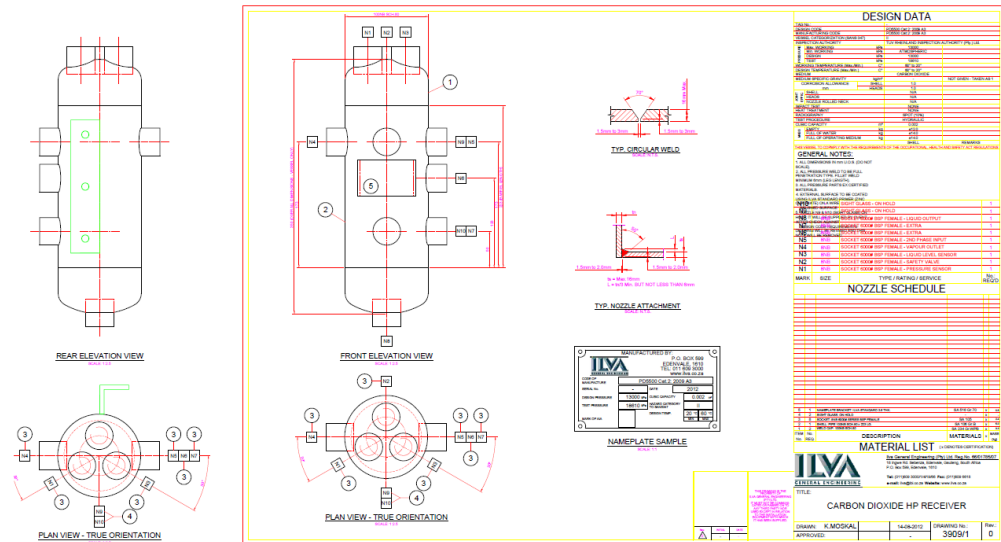
Figure 5.2 Receiver Design Drawing

The receiver has been designed to conform to the standard and code for pressure vessels.

5.1.4 ELECTRONIC EXPANSION VALVES

There are three electronic expansion valves present in the system. Their function is to create a pressure difference over the valve body. They can either they can maintain a constant high-range pressure, or a constant mid-range pressure, depending on the settings programmed into their respective controllers.

The first valve is the ICMTS 20A33 valve. This valve has a large – directly programmable – actuator called an ICAD 600 TS. Its duty is to maintain a constant pressure in the high-pressure line, throughout the gas-cooler and internal heat exchanger.

The second valve that will always be used in conjunction with the ICMTS is the CCM 10 valve. This valve has a built-in actuator. The duty of the valve is to reduce the pressure in the mid-pressure line to equal that of the low-

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pressure line. The valve can also be used to maintain a constant pressure in the mid-pressure line.

The third valve, that can be used with or without the ICMTS, is the AKVH 10-2 valve. This valve is the main pressure-reducing valve in the system. The capillary tubes will perform the same function and as such, the system will operate with the one or the other.

5.1.5 EVAPORATOR DEFROST UNIT

Typical temperatures inside the evaporator during operation are often below 0 °C. As water freezes at 0 °C, frost will accumulate on any exposed tubing with refrigerant at or below 0 °C. The water used in the evaporator will also freeze if the input temperature of the water is not sufficiently high enough to prevent frost. It is typical for the water used in the evaporator to be recycled back to the evaporator. Thus, eventually, the overall water temperature will drop enough for frost to form. Most conventional refrigeration systems have some form of a defrost cycle included in the operation of the system.

The controllers used in the experimental system laid out in this project can be programmed with three types of defrost. The first type of defrost is "Natural". Here, the heating fluid or vapour running through the evaporator is kept running while the system itself switches off, allowing for natural defrost of the evaporator. The second type of defrost is "Hotgas". With this type of defrost, extra solenoid valves are installed in the system and is used to divert hot gas from the discharge side of the compressor through the evaporator. In simple terms, the temperature of the refrigerant through the evaporator is increased such that it melts the frost collecting on the tubing.

The third type of defrost – and the type used with the system in this project – is "Electric" defrost. When the defrost cycle starts, a heating element on the evaporator is activated. There are various settings the controller can be programmed to regarding defrost – including start-up time, activation type, when to end the defrost cycle, etc.

A heat tracing cable from Unitemp is utilized and fixed to the evaporator with aluminium tape. The heat tracing cable can heat up to 65 °C. The aluminium tape provides excellent heat resistance, thus ensuring the least amount of heat to be lost to the environment during the defrost cycle. The controller – specifically, the AK-CC 550 controller – is programmed to start a defrost cycle once every 6 hours. The defrost cycle is programmed to stop after 30 minutes. The defrost cycle can also be programmed to activate at a certain temperature and to remain on until a secondary set temperature is reached.

5.1.6 ACCUMULATOR STAND-BY COOLER

The low-pressure line is restricted to 46 bar. If the pressure should rise above this range, the expansion valves – if used – will close and the safety pressure-relief valves will blow off CO_2 and the system will shut down. This is undesirable, as it will require the system to be charged with CO_2 after every blow-off.

A back-up system will be ready to cool the refrigerant in the accumulator, should the pressure in the low-pressure line increase above 40 bar. The stand-by system consists of a refrigeration system running on R404A. The cooling systems evaporator coils are turned around the main systems accumulator. The R404A refrigeration system itself will be located outside of the testing lab, as it will generate a significant heat while it is running. Danfoss sponsored the unit, while Refrigeration Solutions built housing and connected the controllers for the unit.

5.2 ELECTRONIC SENSORS AND CONTROL SYSTEM

The system is controlled and monitored with three Danfoss controller units; the AK-CC 550; EKC 326A; and the AK-SM 350. A VLT micro drive unit is utilised to control the frequency of the compressor. A gas detector unit, the GDC IR Gas Detector, monitors the levels of CO_2 in the test environment. A full report on the programming and utilization of the controllers is available, compiled by the author of this report. (Fourie, 2012) The report is included in the Addendum.

The main controller is primarily a monitoring unit, called the AK-SM 350. (SM – System Monitor) All the measured variables throughout the system are stored in the AK-SM 350. The AK-SM 350 can be used to program the other controllers, or each controller can be programmed individually, either directly on the controller itself or remotely via a computer. The AK-CC 550 controller monitors the temperature and pressure conditions within and around the refrigeration appliance, including the temperature of the product refrigerated. It also sends signals to an AKVH expansion valve to control the expansion of the gas and the amount of superheat reaching the compressor. The EKC 326A controller monitors the temperature and pressure in the high-pressure and mid-pressure line and sends signals to the ICMTS type valve and the CCM valve to accurately control the pressure in the gas-cooler and in the high-pressure receiver.

Nearly all the components in system require shielded, insulated cables. Two, three and four core cables are used throughout the system, apart from the 3-phase power cable for the compressor.

Sensors and other components used in the system:

- Pressure Transducers
    - AKS 2050
    - MBS 33
    - PTI-S Standard Industrial

- Temperature Sensors
    - Pt-1000, Length: 3 m
    - Pt-1000, Length: 5 m

- Electronic Valves
    - ICMTS 20A33
    - ICAD 600TS – Motor for ICMTS
    - AKVH 10-2 (With Coil)
    - CCM 10

- Alarm Units
    - Red Strobe Light
    - Red and Orange Static Lights
    - Buzzers

- Power Supplies
    - 230 V a.c.
    - 24 V a.c.
    - 15 V d.c.

- Other
    - Green Static Light (Power ON)
    - Electric Heating Unit (Defrost System)

The major electronic components are the Danfoss controllers. Details of the controllers are given in Chapter 5.3.1, 5.3.2 and 5.3.3.

Figure 5.3 Electronic Schematic shows how the controllers are connected to each other.
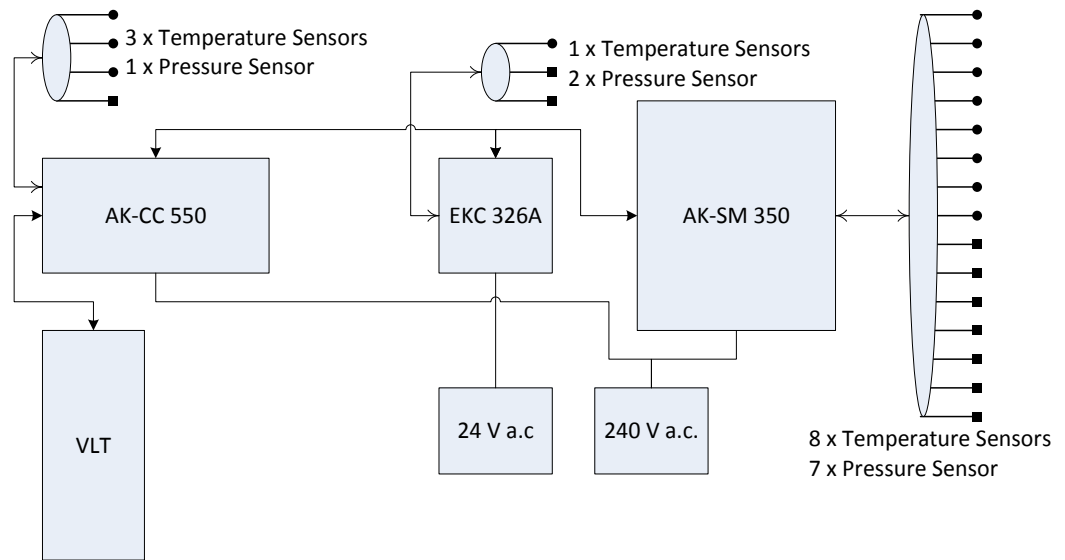
Figure 5.3 Electronic Schematic

A larger format of the schematic, with full connections in detail, is available at the back of this report, in Appendix Figure B.1Diagram of the Experimental System.

5.3 ELECTRONIC SCHEMATIC DIAGRAM

A controller box, made up of Plexiglas, encases the controllers and other electronic components. This enables the user/operator to easily see the read-outs on the controller screens, while keeping him/her safe from live electric connections. Figure 5.4 is a photograph of the controller box.

Hinges allow the lid to remain open while the user operates the controllers. Two locks on the lid keep unauthorised personnel from tampering with the electronic components.

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Figure 5.4 Photograph of Controller Box

5.3.1 AK-SM 350 – SYSTEM MONITORING UNIT

The AK-SM 350 is a system-monitoring unit. It has the capability to communicate with several controllers and monitor up to 65 measuring points. The controller can also be connected directly to a computer and a printer, or can connect indirectly via a modem. The controller also has an Ethernet TCP/IP connection that must be installed when other software is used to monitor or program the controller. It is important to note that any printer used must be HP PCL-3 compatible. The power supply for the AK-SM 350 is 230 V a.c. and 10 VA.

5.3.1.1 COMPONENTS ON AK-SM 350

Table 5.4 lists the components connected to the controller, their basic information and their use in the system.

The PTI-S transducers are powered by an external 15 V D.C. supply, while the MBS 33 transducers are power by the AK-SM 350's internal 12 V D.C. supply. PT-1000 temperature sensors require no extra power supply. The alarm lights and buzzer are connected to a 240 V A.C. supply.

Table 5.4 AK-SM 350 Components

COMPONENT TYPE	PART CODE	MIN/MAX RATINGS	USE IN SYSTEM
Pressure Transducer	PTI-S	0 – 160 bar	AKVH inlet pressure
	PTI-S	0 – 160 bar	IHX low-pressure line
	PTI-S	0 – 160 bar	IHX high-pressure line
	MBS 33	0 – 250 bar	Capillary Tube inlet pressure
	MBS 33	0 – 250 bar	Compressor discharge pressure
	MBS 33	0 – 60 bar	Compressor suction pressure
	MBS 33	0 – 60 bar	Accumulator pressure
Temperature Sensor	Pt-1000	~ 3.9 Ohm/°C	Gas-Cooler inlet temperature
	Pt-1000	~ 3.9 Ohm/°C	IHX high-temperature line
	Pt-1000	~ 3.9 Ohm/°C	Capillary Tube inlet temperature
	Pt-1000	~ 3.9 Ohm/°C	AKHV inlet temperature
	Pt-1000	~ 3.9 Ohm/°C	Evaporator inlet temperature
	Pt-1000	~ 3.9 Ohm/°C	IHX low-temperature line
	Pt-1000	~ 3.9 Ohm/°C	Compressor suction temperature
	Pt-1000	~ 3.9 Ohm/°C	Gas-cooler water out temperature
Alarm: Orange Light with Buzzer	————	————	A buzzer and light to indicate that a high level alarm has been triggered anywhere in the system
Alarm: Red Strobe	————	————	Secondary alarm light – higher visibility
CO_2 Flow Meter	CNGmass DCI	————	CO_2 mass flow rate meter
Controller	AK-CC 550	————	Appliance control
Controller	EKC 326A	————	Pressure regulator

5.3.1.2 AK-SM 350 ALARM SETTINGS

Table 5.5 lists the various high and low level alarms:

Table 5.5 AK-SM 350 Alarms

PARAMETER	LOW	HIGH
Temperature Sensors ($T_{GC,i}$)	40 °C	160 °C
Temperature Sensors ($T_{IHX,h,o}$, $T_{CT,i}$)	0 °C	70 °C
Temperature Sensors ($T_{EV,I}$, $T_{IHX,c,o}$)	-20 °C	10 °C
Temperature Sensors (T_{suc})	-20 °C	13 °C
Temperature Sensors ($T_{CCM,o}$)	-20 °C	60 °C
Temperature Sensors ($T_{GC,w,o}$)	-10 °C	75 °C
Pressure Transducers ($P_{IHX,h,i}$, $P_{CT,i}$, P_{disc})	35 bar	120 bar
Pressure Transducers ($P_{AKVH,i}$)	35 bar	95 bar
Pressure Transducers (P_{Acc}, P_{Suc})	15 bar	45 bar
Pressure Transducers ($P_{IHX,c,i}$)	20 bar	50 bar
"I'm Alive" Alarm	5 hours	None
Gas Detector	5000	9000

A secondary, red, strobe light is also connected to "Relay 1" on the AK-SM 350. This light signals an alarm along with the static, orange light and buzzer at the back of the frame – the red light is positioned high up on the frame of the system to ensure clear visibility from any position.

The "I'm Alive" alarm is an alarm that signals briefly every 5 hours to indicate that the system is still on and running.

5.3.2 AK-CC 550 – APPLIANCE CONTROL

The AK-CC 550 is a controller used to control an AKVH expansion valve and monitor and maintain set conditions on the low-pressure line (Evaporator line). The controller used in this instance utilizes "Application 1". (For more information, one may refer to the instruction manual for the controller. (Danfoss, 2013)) With this application additional, optional, controls include:

compressor on/off; alarm on/off; defrost on/off; fan on/off; and light (door light) on/off. The AK-CC 550 requires a 230 V a.c. supply.

Components of the AK-CC 550 include pressure transducers, temperature sensors, the AKVH expansion valve, the VLT Micro Drive, a defrost unit for the evaporator and alarm units. Table 5.6 highlights the basic information of each component.

Table 5.6 AK-CC 550 Components

COMPONENT TYPE	DESIGNATED CODE	MIN/MAX RATINGS	USE IN SYSTEM
Pressure Transducer (AKS 2050)	P_e	-1 – 59 bar	Evaporator pressure
Temperature Sensor (Pt-1000)	T_{S2}	~ 3.9 Ohm/°C	S2 – Evaporator outlet temperature
Temperature Sensor (Pt-1000)	T_{S3}	~ 3.9 Ohm/°C	S3 – water temperature outlet/inlet
Temperature Sensor (Pt-1000)	T_{S4}	~ 3.9 Ohm/°C	S4 – water temperature outlet/inlet
Electronic Expansion Valve	AKVH	————	Valve for pressure expansion
Alarm Light with Buzzer	————	————	A buzzer and light to indicate that a high level alarm has been triggered anywhere in the system
Compressor Drive	VLT Micro Drive	————	Switches the compressor On/Off
Electric Defrost Unit	————	Heats to 65 °C	Switches the electric defrost unit On/Off

The set temperature of the water (desired output temperature of the water in the gas-cooler) is set directly on the AK-CC 550.

5.3.3 EKC 326A – HIGH-PRESSURE REGULATION

The EKC 326A is used to control the ICMTS 20A33 valve with ICAD 600TS motor and the CCM 10 valve. It also monitors and maintains set operating conditions of the high-pressure line and the conditions in the high-pressure receiver. It requires a 24 V supply.

Components connected to the EKC 326A unit includes pressure transducers, temperature sensors, the ICAD motor, the CCM valve and alarm units. Table 5.7 highlights these components.

Table 5.7 EKC 326A Components

COMPONENT TYPE	DESIGNATED CODE	MIN/MAX RATINGS	USE IN SYSTEM
Pressure Transducer (AKS 2050)	P_{Rec}	-1 – 99 bar	Receiver pressure
Pressure Transducer (AKS 2050)	P_{GC}	-1 – 159 bar	Gas-cooler pressure
Temperature Sensor (Pt-1000)	T_{S2}	~ 3.9 Ohm/°C	$S_{GC}2$ – gas-cooler outlet temperature
Valve Motor	ICAD 600TS	_____	Power the ICMTS valve
Electronic Expansion Valve	CCM 10	90 bar MWP 50 bar MOPD -40 to 40 °C	Valve for pressure expansion
Alarm Light with Buzzer	_____	_____	A buzzer and light to indicate that a high level alarm has been triggered anywhere in the system

5.3.4 VLT MICRO DRIVE – COMPRESSOR FREQUENCY CONTROL

The VLT Micro Drive is used to control the compressor speed, specifically the frequency of the compressor. It is a basic unit that can communicate directly to a computer or indirectly with the AK-SM 350. The VLT drive requires a 3-phase power supply, with a maximum current of 16.8 A, intermittent. It uses the 3-phase supply to control and power the compressor.

5.3.5 GDC IR – CO_2 GAS DETECTOR

The gas detector used for the system is specialized for the detection of CO_2. It has an infrared sensor. The unit has an LCD display, showing the current CO_2 levels and the status of the alarms. The gas detector has two lights and a buzzer connected to it, a yellow light for a low-level alarm and a red light with a buzzer for the high-level alarm. The GDC IR is connected to the AK-SM 350 controller via a TP connection. The gas detector requires a supply of 12 – 24 V a.c. It is important to note that the gas detector has a stabilisation period – if it has been off for a lengthy period. The stabilisation period for this gas detector is 10 seconds. However, if the gas detector has been in long-term storage, the stabilisation period can be 1-2 hours.

5.3.5.1 GAS DETECTOR ALARM SETTINGS

The gas detector has two alarm settings, a low level and a high-level alarm. These settings can be changed on the gas-cooler mother PCB, but they are kept on factory default settings. The current settings and their respective alarms are shown in Table 5.8:

Table 5.8 Gas Detector Alarms

	CO₂ PPM	YELLOW LIGHT	RED LIGHT AND BUZZER	ALARM ON THE AK-SM 350
Low Level Alarm	5000	Yes	No	Yes
High Level Alarm	9000	No	Yes	Yes

The gas detector is connected to the AK-SM 350 via a Third Party (TP) connection. Thus, the AK-SM will be able to monitor and record measurements from the gas detector, as well as the alarms set off by the gas detector.

5.3.5.2 SETTINGS ON GAS DETECTOR

The following settings are set on the gas detector.

Table 5.9 Gas Detector Settings

PARAMETER	SETTING VALUE
Low level alarm delay	0 minutes
High level alarm delay	0 minutes
Low level alarm	5000 ppm
High level alarm	9000 ppm
Address	01:019

65

5.4 CALLIBRATIONS AND PRESSURE TESTING

Several components throughout the system have been specifically tested to withstand operating pressure conditions. The components are all the units that required welding and the safety pressure-relief valves. The following two tables, Table 5.10 and Table 5.11, list the components and the pressure they were tested to.

Table 5.10 Test Pressures of Components

COMPONENT	MAXIMUM OPERATING PRESSURE	MAXIMUM ALLOWABLE OPERATING PRESSURE	TEST PRESSURE
Accumulator Valve #1	46 bar	60 bar	47.65 bar
Accumulator Valve #2	46 bar	60 bar	48.57 bar
Particle Filter (F10L-CD)	46 bar	45 bar @ 50 °C	46.81 bar
CCM 10	90 bar	90 bar	91.56 bar
ICMTS 20A33	120 bar	140 bar	122.66 bar
Suction Line Accumulator	46 bar	100 bar @ 100 °C 75 bar @ (-10 to -50 °C)	49 bar
Filter/Drier	46 bar	60 bar @ 70 °C 45 bar @ (-10 to -40 °C)	47.41 bar

The safety pressure-relief valves have been calibrated using PAG oil.

Table 5.11 Test Pressures of Safety Relief Valves

VALVE NUMBER	DESIRED RELIEF PRESSURE	SET RELIEF PRESSURE
#1 (Yellow Spring)	46 bar	45.96 bar
#2 (Yellow Spring)	46 bar	46.25 bar
#3 (Yellow Spring)	46 bar	46.32 bar
#4 (Purple Spring)	90 bar	90.62 bar
#5 (Purple Spring)	90 bar	90.62 bar
#6 (Purple Spring)	90 bar	90.38 bar
#7 (Orange Spring)	130 bar	129.85 bar
#8 (Orange Spring)	130 bar	130.07 bar
#9 (Orange Spring)	130 bar	130.48 bar

The full report on the testing of these components is available in the Addendum. The pressure test was performed with a calibrated, certified system.

Pressure and leak testing was performed extensively before the system as charged with CO_2. Argon gas was used up to a maximum pressure of 120 bar in the high-pressure line, 90 bar in the mid-pressure line and 46 bar in the low-pressure line. Each component and fitting was tested with liquid-leak-test fluid.

5.5 SAFETY PRECAUTIONS

CO_2 sensors used in industry typically use infrared sensors. Complete gas detection units for CO_2 – such as the one produced by Danfoss – also has a built-in alarm with both low and high limits. Factory settings for the low and high limit is 5 000 ppm and 9 000 ppm, respectively. The value of 5 000 ppm corresponding with the Occupational Exposure Limit (OEL) value as set by the ASHRAE 34 standard. The 9 000 ppm complies with EN 378/prEN 378 standard. As CO_2 is heavier than air, any room with a CO_2 system must be well ventilated and special care must be taken to avoid areas such as crawl-ways or below-ground work areas.

Special precautions must be taken with the placement of the safety pressure-relief valves, because of the low critical temperature and typical evaporator temperatures of CO_2 systems. Any relief valves placed within an area where fluid CO_2 is present poses the possible hazard of freezing up during emergency discharge. Failure to vent excess CO_2 will cause a build-up in pressure and can lead to catastrophic failure of the system.

Whenever one of the controllers or the GD unit signals for an alarm, a strobe light and a buzzer will signal until the alarm is either dismissed or resolved. The buzzers have a decibel range of 55 to 85 dB. Each buzzer can have one of ten tones.

During start-up of the system, the main alarm buzzer will signal for 1 minute. There is also an "I'm Alive" alarm. This alarm signals every 5 hours to indicate that the system is still running.

The compressor has a built-in safety chip that monitors oil temperature and refrigerant temperature in the compressor. A "fail" signal from either of those units will keep the compressor from switching on. Additionally, a low-pressure switch is situated on the low-pressure line before the compressor. If the pressure in the suction line falls too low, the switch will send a "fail" signal to the AK-CC 550 controller and the compressor will switch off.

5.6 TESTING THE SYSTEM

Before switching the system on, the operator must first ensure that all the valves are switched to the correct positions and that the pressure readings on the pressure gauges are still in the correct range. The operator must ensure

that the controllers are programmed correctly, by checking each setting in the three controllers manually. The water temperatures must be set and allowed to reach set temperatures. The next step is to switch on the water pumps and to adjust the water flow rates by adjusting the shut-off valves for each line. The final safety check is to ensure that all personnel have hearing protection.

As the system is switched on by supplying power to the controllers and the VLT drive, the AK-SM controller will send an alarm signal for 1 minute. Once the controllers have completed their start-up sequence, both the AK-CC and the EKC controllers must be switched on by changing setting "r12" to "1" on both. The system will then switch the compressor on and start regulation. If any changes must be made (e.g. changing desired operating conditions of the system), then the relative parameters in the controllers must be programmed before "r12" is set to "1".

The refrigeration system can operate under various different configurations. To change to a different configuration, the system must be completely shut off. Valve positions are changed and the system is allowed to equalize.

5.7 DATA COLLECTION

The AK-SM 350 controller records all the information sent to it. It takes temperature and pressure readings every 15 minutes. This information can be stored for one year. The controller has the capability to generate graphs to display the readings of each input vs. time. The raw data, as well as the graphs the controller generates, can be sent to a printer or a PC.

The measurements from the water flow meters are not recorded via any controller and must be manually recorded during testing. Only one recording is necessary once the water flow is initiated.

CHAPTER 6 RESULTS

The results from testing the three simulations, the steady-state simulation, the transient simulation and the capillary tube simulation, and the experimental system are presented in the sections to follow.

6.1 STEADY-STATE SIMULATION RESULTS

The steady-state simulation program SC Simulation v.5 runs in just under 1 minute. It presents its results in numerical and graphical form. Below is an example of results for the gas-cooler. Here the system runs subcritically, with the IHX excluded, stainless steel tubing and counter-flow heat exchangers.

Input and output conditions were chosen as:

Table 6.1 Initial Conditions of Steady-State Simulation

CHOSEN CONDITIONS			CALCULATED VALUES		
Parameter	Value	Unit	Parameter	Value	Unit
$T_{GC,r,o}$	30	°C	$T_{GC,r,i}$	64.608	°C
$T_{GC,w,i}$	15	°C	$T_{GC,w,o}$	15.589	°C
$T_{Ev,i}$	-10	°C	$T_{Ev,o}$	-10	°C
$T_{Ev,w,i}$	15	°C	$T_{Ev,w,o}$	14.571	°C
$d_{GC,r,i}$	10	mm	L_{GC}	5.470	m
$d_{GC,r,o}$	12	mm	L_{Ev}	5.712	m
$d_{GC,w,i}$	21	mm	L_{CT}	0.266	m
$d_{Ev,i}$	15	mm			
$d_{Ev,o}$	18	mm			
$d_{Ev,w,i}$	21	mm			
$d_{CT,i}$	2	mm	$\dot{m}_{CO2}$	0.0693	kg/s
$\dot{m}_{GC,w}$	15	kg/s	P_{high}	72.133	bar
$\dot{m}_{Ev,w}$	5	kg/s	P_{low}	26.485	bar

Figure 6.1 Simulated Temperature and Enthalpy for the Gas-Cooler – SC Simulation is the result of the gas-cooler in the steady-state simulation. Displayed on the graph is the temperature distribution for the water and the refrigerant across the length of the gas-cooler. Additionally, the specific enthalpy for the refrigerant is also shown. Numerical results include all the chosen input and output operating conditions, the tubing diameters, the calculated water outlet temperature and the calculated length of the gas-cooler.
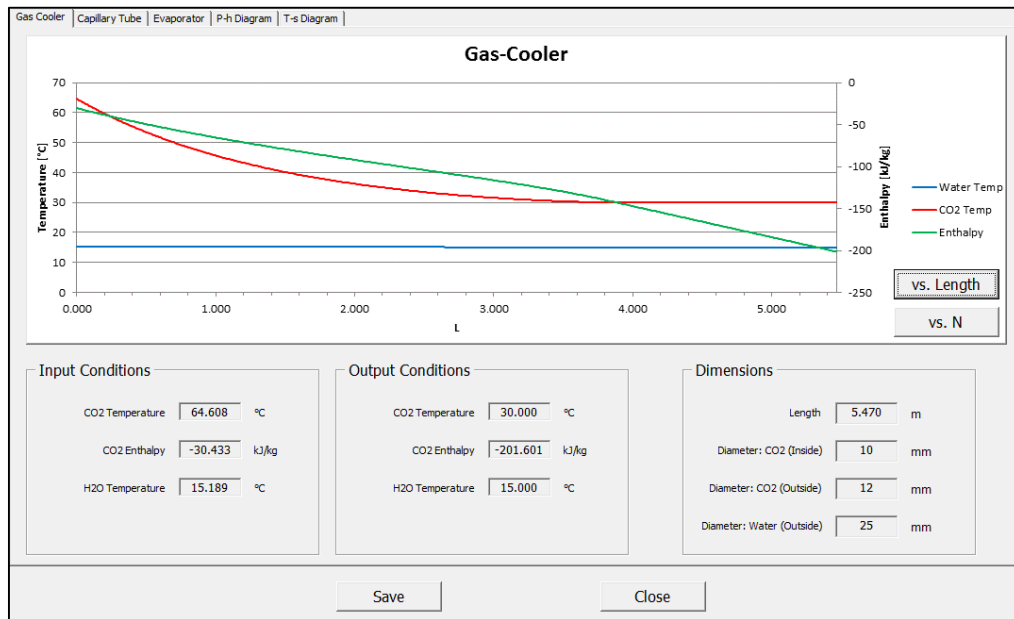
Figure 6.1 Simulated Temperature and Enthalpy for the Gas-Cooler – SC Simulation v.5

The Figure 6.2 shows the system operating conditions for the steady-state simulation, on a P-h-diagram. The red line shows where the system operates. The operating pressure in the high-pressure line is just below the critical pressure for CO_2.
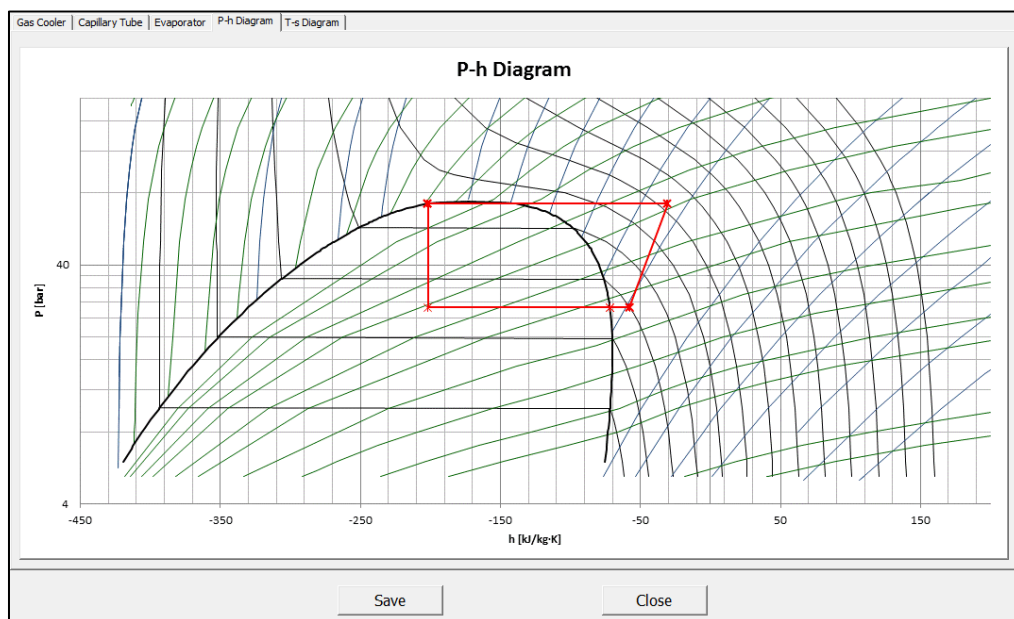
Figure 6.2 P-h Diagram for the SC Simulation v.5

6.2 TRANSIENT SIMULATION RESULTS

The transient simulation starts with the following initial conditions, in Table 6.2.

Table 6.2 Transient Simulation Initial Conditions

INITIAL CONDITIONS						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,o}$	22	°C		$T_{GC,r,i}$	22	°C
$T_{GC,w,i}$	35	°C		$T_{GC,w,o}$	24	°C
$T_{Ev,i}$	22	°C		$T_{Ev,o}$	22	°C
$T_{Ev,w,i}$	15	°C		$T_{Ev,w,o}$	8	°C
$d_{GC,r,i}$	10	mm		L_{GC}	4.6	m
$d_{GC,r,o}$	12	mm		L_{Ev}	6.674	m
$d_{GC,w,i}$	20	mm		$\dot{m}_{GC,w}$	15	kg/s
$d_{Ev,i}$	15	mm		$\dot{m}_{Ev,w}$	5	kg/s
$d_{Ev,o}$	18	mm		t	0	s
$d_{Ev,w,i}$	20	mm		Δt	0.01	s
P_{high}	60	bar		P_{low}	60	bar

The program starts with the initial conditions, and determines the transient solution with 0.01 s time steps. After reaching steady-state, the results are:

Table 6.3 Transient Simulation Steady-State Conditions

STEADY-STATE CONDITIONS						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,o}$	54.61625	°C		$T_{GC,r,i}$	100.4007	°C
$T_{GC,w,i}$	35.0091	°C		$T_{GC,w,o}$	35.23622	°C
$T_{Ev,i}$	-10.9783	°C		$T_{Ev,o}$	-7.90296	°C
$T_{Ev,w,i}$	13.09015	°C		$T_{Ev,w,o}$	1.083515	°C
				t	60	s
P_{high}	103.2247	bar		P_{low}	25.75439	bar

The P-h diagram in Figure 6.3 for the T Simulation v.1.5 shows that the gas-cooler does not have sufficient cooling. This can be drastically improved by lowering the water temperature of the cooling water.
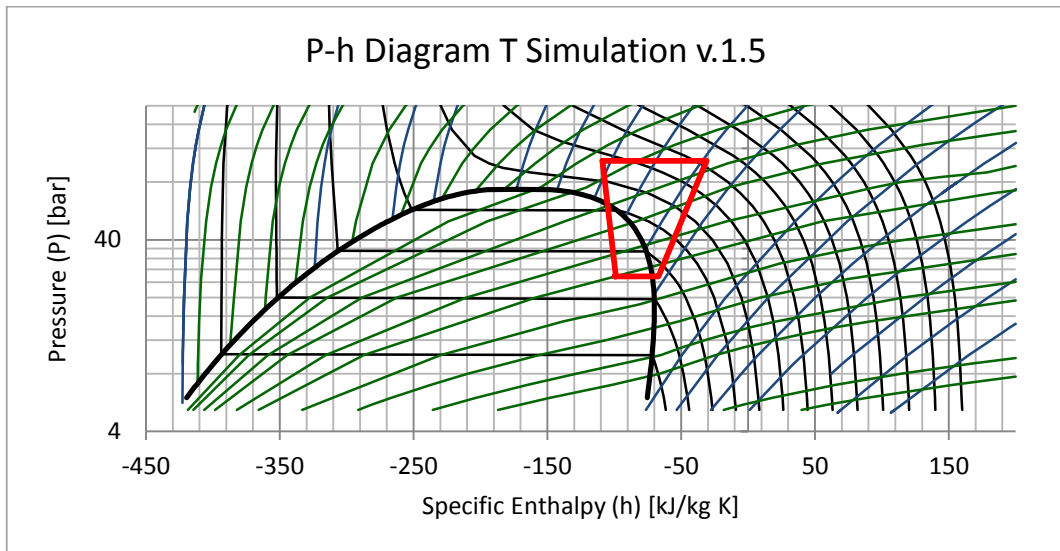
Figure 6.3 T Simulation P-h Diagram

Figure 6.4 is the temperature distribution in the gas-cooler. Lowering the flow rate of the cooling water will allow a greater temperature difference over the water.
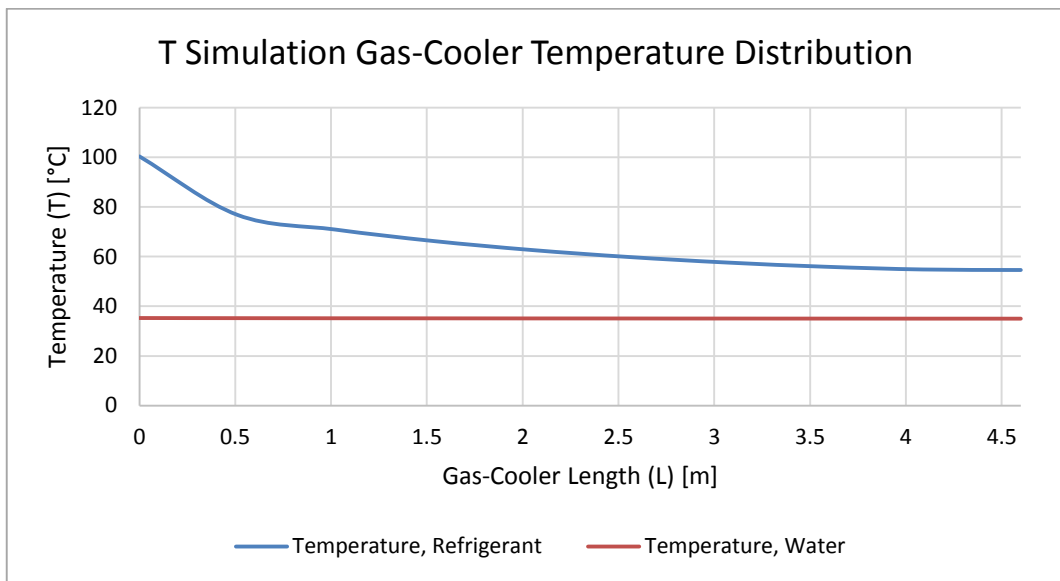
Figure 6.4 T Simulation Gas-Cooler – Temperature Distribution

6.3 CAPILLARY TUBE SIMULATION RESULTS

The capillary tube simulation program uses MS Excel and an extensive data-base to determine operating conditions, fluid properties and equations of state. The goal of the program is to determine the mass flow required in a capillary tube for specific operating conditions and capillary tube dimensions. The chosen initial conditions for the capillary tube is given in Table 6.4.

Table 6.4 Capillary Tube Simulation Initial Conditions

PARAMETER	SYMBOL	VALUE	UNITS
Transcritical/Subcritical	-	Transcritical	-
Refrigerant Temperature, In	$T_{CT,i=1}$	25	°C
Refrigerant Temperature, Out	$T_{CT,i=N}$	−10	°C
Pressure, High (Transcritical Only)	P_{high}	90	bar
Capillary Tube Inside Diameter	$d_{CT,i}$	2	mm
Capillary Tube Length	L_{CT}	1	m

The capillary tube program presents the results in the form of a GUI, as in Figure 6.5 Capillary Tube Simulation Graphical User Interface. The figure is a repetition of Figure 4.12. The bar at the bottom of the display shows the length of the capillary tube. The red indicates single phase, while the blue region indicates 2-phase flow. The mass flow rate for the chosen conditions is calculated to be 0.0944 kg/s .
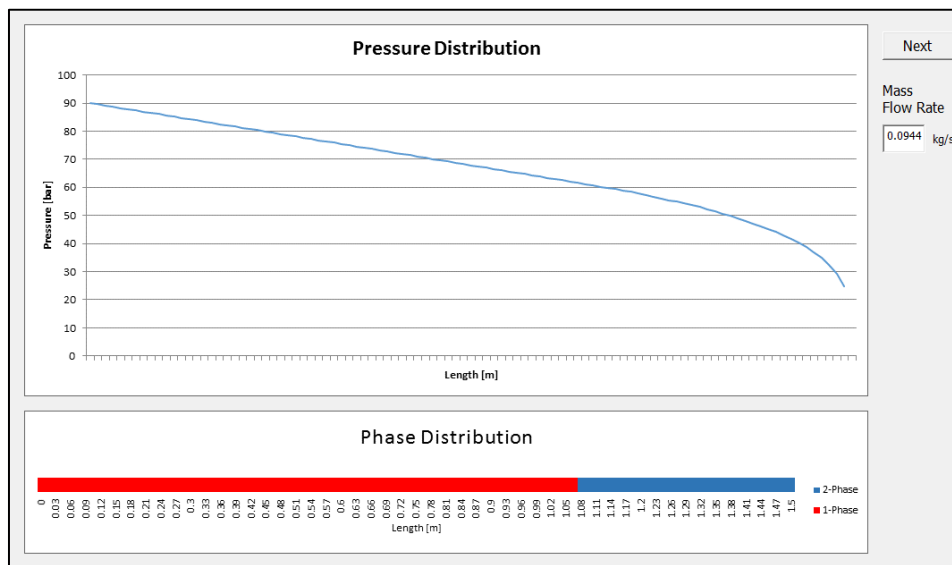
Figure 6.5 Capillary Tube Simulation Graphical User Interface

6.4 EXPERIMENTAL REFRIGERATION SYSTEM

The experimental refrigeration system was tested under various operating conditions and configurations. Six tests were performed for the following conditions:

1. Test 1
    1.1. ICMTS, CCM, AKVH;
    1.2. Transcritical region;
    1.3. Counter flow heat exchangers;
    1.4. Internal heat exchanger included; and
    1.5. Heating and cooling water supplied by two sources.

2. Test 2
    2.1. Capillary tube: 1 m length, 2 mm ID;
    2.2. Subcritical region;
    2.3. Counter flow heat exchangers;
    2.4. Internal heat exchanger included; and
    2.5. Heating and cooling water supplied by two sources.

3. Test 3
    3.1. Capillary tube: 1 m length, 2 mm ID;
    3.2. Subcritical flow;
    3.3. Counter flow heat exchangers;
    3.4. Internal heat exchanger excluded; and
    3.5. Heating and cooling water supplied by the same source.

4. Test 4
    4.1. Capillary tube: 1.5 m length, 2 mm ID;
    4.2. Subcritical flow;
    4.3. Counter flow heat exchangers;
    4.4. Internal heat exchanger excluded; and
    4.5. Heating and cooling water supplied by the same source.

5. Test 5
    5.1. ICMTS, CCM, capillary tube: 1 m length, 2 mm ID;
    5.2. Transcritical region;
    5.3. Counter flow heat exchangers;
    5.4. Internal heat exchanger excluded; and
    5.5. Heating and cooling water supplied by the same source.

6. Test 6
    6.1. ICMTS, CCM, AKVH;
    6.2. Transcritical region;
    6.3. Counter flow heat exchangers;
    6.4. Internal heat exchanger excluded; and
    6.5. Heating and cooling water supplied by the same source.

Test 3 and 4 were repeated to confirm repeatability of the tests. Table 6.5 lists the operating condition for the various tests. Included is the discharge and suction pressures, the receiver pressure, the gas-cooler outlet temperature, the evaporator temperature and the hot and cold water temperatures.

Table 6.5 Experimental System Test 5 Operating Conditions

TEST #	DISCHARGE PRESSURE	RECEIVER PRESSURE	SUCTION PRESSURE	GAS-COOLER OUTLET TEMPERATURE	EVAPORATOR TEMPERATURE	HOT WATER TEMPERATURE	COLD WATER TEMPERATURE
	bar	bar	bar	°C	°C	°C	°C
Test 1	103	53	19.5	49.9	-13.2	42.5	8.9
Test 2	64.4	-	22.5	25.8	-11.3	15	14.9
Test 3	63.6	-	21.8	25.8	-13.2	16.5	16.5
Test 4	64.7	-	20.3	26.4	-15.7	17.5	17.5
Test 5	77.7	57.4	17.4	30.4	-18.1	19.1	20.6
Test 6	79.1	60.5	17.6	31.1	-18.3	19.2	19.2
Test 3 Repeated	62.7	-	21.3	25.6	-13.7	18.6	18.6
Test 4 Repeated	64.2	-	20	26.3	-15.3	19	19

The slight variations in inlet water temperatures are enough to cause the minor discrepancies between tests 3 and 4 and their repeated tests. Results of test 5 are in Table 6.6. The results for the other tests are in Appendix H - Experimental System Test Results.

Table 6.6 Experimental System Test 5 Measured Values

MEASURED VALUES						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,i}$	95.6	°C		$T_{GC,r,o}$	30.4	°C
$T_{GC,w,i}$	19.1	°C		$T_{GC,w,o}$	20.6	°C
$T_{Ev,i}$	-18.1	°C		$T_{Ev,o}$	15.1	°C
$T_{Ev,w,i}$	20.6	°C		$T_{Ev,w,o}$	14.2	°C
$T_{IHX,h,i}$	-	°C		$T_{IHX,h,o}$	-	°C
$T_{IHX,c,i}$	-	°C		$T_{IHX,c,o}$	-	°C
$T_{CCM,o}$	-20.3	°C		$T_{CT,i}$	21	°C
T_{suc}	-1.9	°C		$T_{ambient}$	20.5	°C
$\dot{m}_{ref}$	0.0225	kg/s		P_{high}	77.7	bar
$\dot{m}_{GC,w}$	0.7337505	kg/s		P_{med}	57.4	bar
$\dot{m}_{Ev,w}$	0.169711	kg/s		P_{low}	17.4	bar

Representing these operating conditions on a Pressure-Enthalpy diagram is Figure 6.6 Pressure-Enthalpy Diagram of Test Results – Test 5.
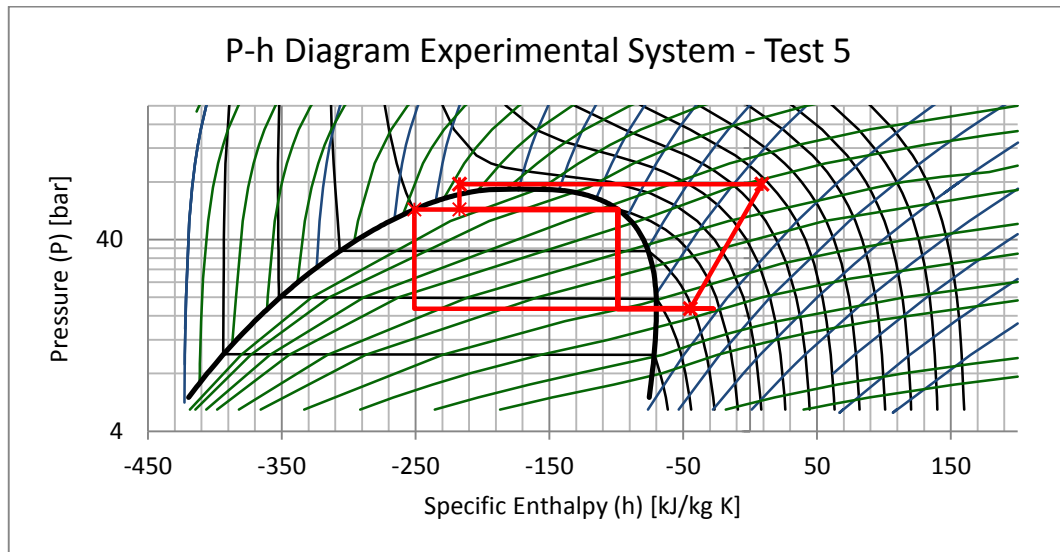
Figure 6.6 Pressure-Enthalpy Diagram of Test Results – Test 5

Further cooling in the system can be achieved by lowering the temperature of the water entering the gas-cooler.

The high-pressure line has a pressure of 77.7 bar. This value is extremely close to the critical pressure of CO_2, making these operating conditions difficult, but not impossible, to simulate.

Running the steady-state simulation, SC Simulation v.5, with the same input and output conditions given and using a capillary tube only.

Table 6.7 Steady-State and Experimental System - Test 5 Comparison

PARAMETER	VALUE	UNITS
L_{GC}	4.8433777	m
L_{Ev}	6.549870897	m
L_{CT}	1.741758598	m

It is important to realise the effect of the ICMTS and the CCM valves in the system. These two valves are not included in the SC Simulation program, which is why the results for the steady-state simulation does not conform to the actual system in this specific test. Figure 6.7 Steady-State P-h Diagram of Experimental System – Test 5 is the SC Simulations' representation of the operating conditions of test 5. The effect of the CCM valve is not included, but the effect of the ICMTS valve and the high-pressure receiver is included as best possible.
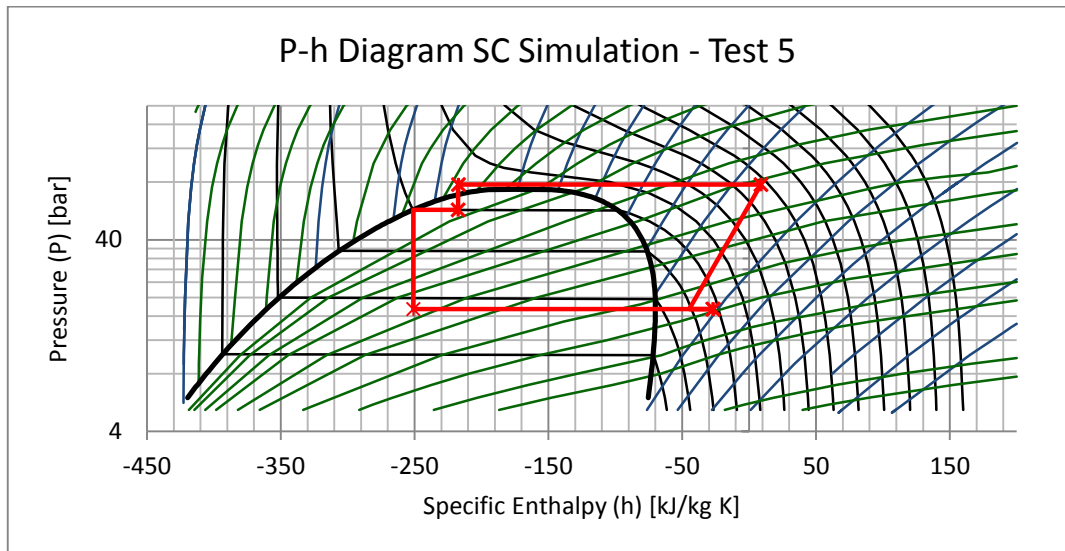
Figure 6.7 Steady-State P-h Diagram of Experimental System – Test 5

The actual efficiency of the compressor is determined by,

$$\eta = \frac{h_{in} - h_{out,ideal}}{h_{in} - h_{out,actual}} \ 100 \ \%$$

(6.1)

Table 6.8 Actual Compressor Efficiency

TEST #	h_{in}	$h_{out,actual}$	$h_{out,ideal}$	η
1	-44.6597	34.7975	44.2157	89.40%
2	-32.4173	23.3943	23.2089	99.67%
3	-33.6657	26.7533	22.7776	93.42%
4	-30.3519	31.2711	32.5357	97.99%
5	-31.5167	27.8704	26.1242	97.06%
6	-29.1396	32.5686	34.5957	96.82%
7	-44.7209	8.83569	33.8238	68.19%

These experimentally determined values appear to be far too high. It is, therefore, recommended that further tests be performed on the system in an attempt to determine the actual efficiencies.

CHAPTER 7 DISCUSSION, CONCLUSION AND RECOMMENDATIONS

It is important to draw a comparison between the physical system and the theory applied to the simulation programs. One effective method is to determine the energy balance of each and compare the results.

The energy transfer rates for the transient simulation program, under various operating conditions are in Table 7.1. The average energy balance for the transient simulation program is relatively large. This is due to the assumptions made within the simulation: no losses to the environment; no pressure drop over the fittings, tubing and heat exchangers; and in inaccurate compressor efficiency model.

Table 7.1 Transient Simulation Energy Transfer Rates

TEST #	$\dot{Q}_{Ev}$ [kW]	$\dot{Q}_{IHX,c}$ [kW]	$\dot{W}_{comp}$ [kW]	$\dot{Q}_{GC}$ [kW]	$\dot{Q}_{IHX,h}$ [kW]	$\Delta\dot{Q}$ [kW]
1	2.518193	-	1.904782	4.917142	-	0.494167
2	2.518186	-	1.904782	4.917143	-	0.494176
3	2.023009	-	2.183215	4.906572	-	0.700348
4	0.711659	-	1.661939	3.399026	-	1.025428
5	2.071446	-	2.342224	5.061333	-	0.647663
6	4.265078	-	2.53085	6.567739	-	-0.22819
7	4.721458	-	2.03276	6.383633	-	-0.37058
					Average Energy Balance	0.394716

The steady-state simulation's heat transfer rates are in Table 7.2. The average energy balance for the steady-state simulation program is zero, because the system is assumed to have no losses, no pressure drops, a compressor efficiency of 100% and all round ideal conditions.

Table 7.2 Steady-State Simulation Energy Transfer Rates

TEST #	$\dot{Q}_{Ev}$ [kW]	$\dot{Q}_{IHX,c}$ [kW]	$\dot{W}_{comp}$ [kW]	$\dot{Q}_{GC}$ [kW]	$\dot{Q}_{IHX,h}$ [kW]	$\Delta\dot{Q}$ [kW]
1	10.76384	0.900136	2.587354	13.3512	0.900136	0
2	10.80435	-	3.820072	14.62443	-	0
3	9.913341	-	2.997982	12.91132	-	0
4	12.38132	1.1254	-7.55775	4.82357	1.1254	0
5	12.78034	1.1254	1.044573	13.82491	1.1254	0
6	11.93242	-	2.666145	14.59857	-	0
7	14.33418	-	4.835992	19.17019	-	0
					Average Energy Balance	0

The energy transfer rates from the experimental system tests are in Table 7.3. The experimental system does not follow any ideal conditions. Measured compressor

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work and heat transfer across the components add up to produce a non-zero av-erage energy balance. This critically accounts for the losses to the environment and the losses within the system. Considering test 6 and test 7 as outliers, the average energy balance is -0.01888 kW.

Table 7.3 Experimental System Energy Transfer Rates

TEST #	$\dot{Q}_{Ev}$ [kW]	$\dot{Q}_{IHX,c}$ [kW]	$\dot{W}_{comp}$ [kW]	$\dot{Q}_{GC}$ [kW]	$\dot{Q}_{IHX,h}$ [kW]	$\Delta\dot{Q}$ [kW]
1	2.089552	0.155535	1.720858	4.010702	0.018008	0.062765
2	2.089311	-	1.802499	3.852017	-	-0.03979
3	2.067347	-	1.64328	3.632438	-	-0.07819
4	1.973797	-	1.712327	3.658892	-	-0.02723
5	2.137496	-	1.779255	3.904788	-	-0.01196
6	5.033315	-	1.205024	5.08789	-	-1.15045
7	5.420061	-	1.601982	5.511982	-	-1.51006
Average Energy Balance						-0.39356

The energy transfer rates are shown in Figure 7.1. The energy transfer rate "in" includes the heat applied to the evaporator plus the work done by the compressor and the heat applied to the internal heat exchanger (if included). The heat transfer rate "out" includes the heat removed by the gas-cooler water and the internal heat exchanger (if included), but excludes the heat lost through the insulation to the environment.
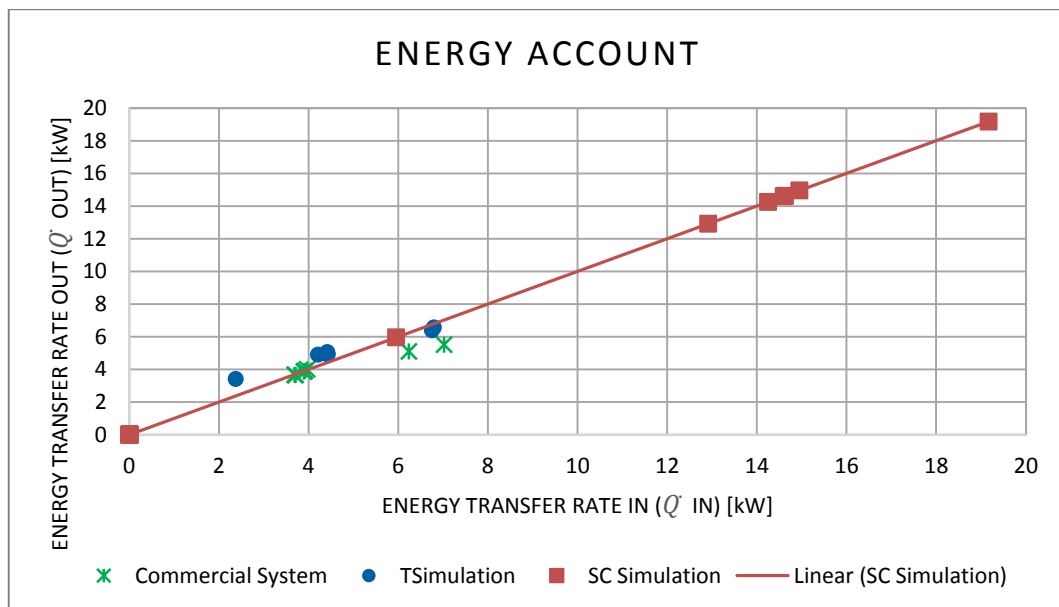
Figure 7.1 Energy Transfer Rate In and Out of the System

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The experimental refrigeration system, as in Figure 7.3, is operational and tests so far have produced excellent results. The energy balance of the system is very close to ideal, and the energy difference is accounted for in the minor losses within the system and to the environment.

The goal for the project was to study the literature surrounding carbon dioxide refrigeration systems and from this gain knowledge on the workings of both a basic system and a more complex system, such as the one built for this project. A steady-state simulation program was developed and this program was used to aid in the design of the tube-in-tube heat exchangers and the capillary tubes for the experimental system. A transient program simulates the experimental system's start-up and run until it reaches steady state, when using only a capillary tube. The third simulation program is a steady-state program that determines the mass flow rate through a capillary tube, given specified operating conditions. The simulation provide similar values to the other simulation programs and the experimental system.

Carbon dioxide was chosen for its environmentally friendly characteristics and its apparently high COP when running under both subcritical and transcritical conditions. The experimental system at the University of Stellenbosch offers a vast range of operating conditions and makes use of specialised components to ensure the operator is both in control of the system and safe at all times.

A basic refrigeration system consist of a compressor, a condenser (or gas-cooler) an evaporator, an expansion unit (usually an electronic expansion valve) and optionally include an internal heat exchanger. The experimental system consist of all these components, along with multiple expansion units, a high-pressure receiver, an accumulator with added stand-by cooling unit, an oil management system, filters and driers, flow meters, pressure gauges, sight glasses, a CO_2 sensor and several temperature and pressure sensors that are connected to three controllers. The controllers monitor the system (through the pressure and temperature sensors) and control the expansion valves and the compressor.

The experimental system has been tested, to ensure that it functions correctly. Some interesting findings include:

- Very high compressor efficiency has been measured;
- Lower COP values than anticipated (as in Table 7.4 below);
- Heat transfer rates and energy transfer rates are lower than determined for an ideal system;
- Heat exchangers have a high efficiency; and
- Large amount of frost build-up on the components and tubing forming quickly after start-up (as Figure 7.2 below).

Table 7.4 COP of the Experimental System

TEST #	$\dot{Q}_{Ev}$ [kW]	$\dot{W}_{comp}$ [kW]	$\dot{Q}_{GC}$ [kW]	COP
1	2.089552	1.720858	4.010702	1.214
2	2.089311	1.802499	3.852017	1.159
3	2.067347	1.64328	3.632438	1.258
4	1.973797	1.712327	3.658892	1.153
5	2.137496	1.779255	3.904788	1.201
6	5.033315	1.205024	5.08789	4.177
7	5.420061	1.601982	5.511982	3.383

The COP of conventional systems are usually around 2.5. For the experimental system' tests, the COP was on average 1.94. (1.2 if the last two tests are seen as outliers) As with any system, however, the maximum COP is not that easy to determine. Hundreds of tests will be required if the maximum COP is to be determined experimentally. The transient simulation program tests have an average COP of 1.2, while the steady-state simulation program tests have an average COP of 4.05. Neither of these values will be the maximum COP for the system, instead the actual value will most likely between the two values. It is possible to theoretically determine the maximum COP of a system. The development of a program that can calculate the COP of a CO2 system can make for an interesting project for students who will continue with studies on this experimental system.
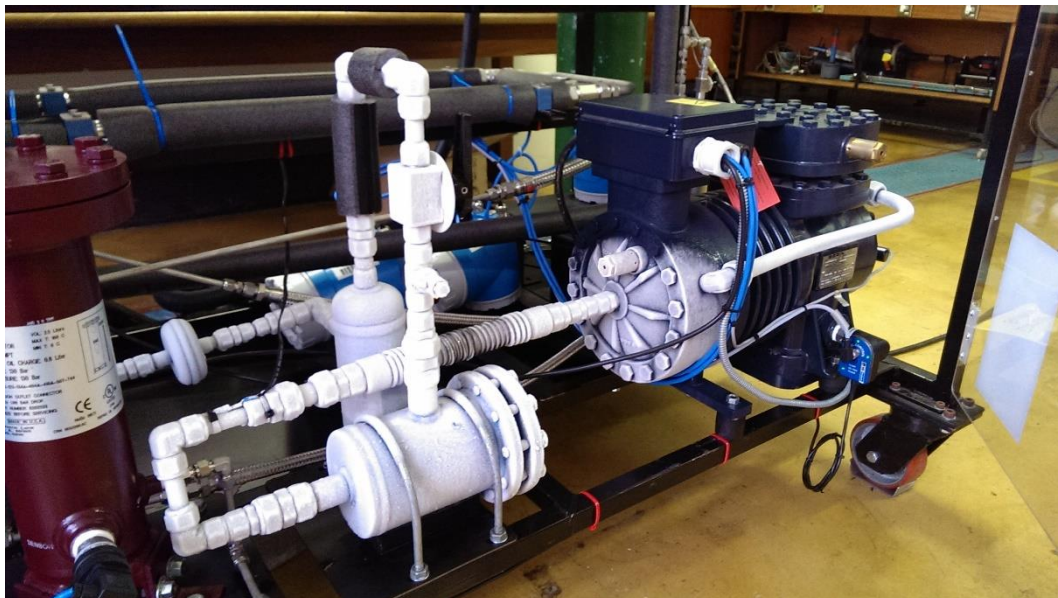
Figure 7.2 Experimental System - Frost Build-up on Equipment and Tubing

The efficiency of the compressor in the experimental system (as determined in Chapter 6.4) is very high, with the highest 99.67%. Further tests of the system will be provide more evidence to either support the high efficiency or provide a lower actual efficiency and a reason why previous calculations could be erroneous.

The experimental system can now be used for a vast variety of tests. The following is small portion of the tests that can be performed on the system:

- Compare different expansion units
- Calculate COP for each configuration
- Determine the effect of water flow rate and temperatures on the system
- Determine the effect of using multiple capillary tubes simultaneously
- Compare counter flow with parallel flow heat exchangers
- Manual operation of the system vs. automated control
- Test different insulation materials and methods
- Testing the efficiency and power consumption of the compressor

Figure 7.3 Photo of Experimental Refrigeration System

Each component can be tested individually for its performance, e.g. the oil separator. Removing a capillary tube and placing another kind of expansion unit in its place will offer an even larger variety of tests to be performed on the system. While further tests can be performed on the system as-is, there are several optional extras that can be added to the system. These include:

- An SMS module
- A printer
- A modem
- Additional monitoring units
- Thermocouples within the water lines
- A CO_2 liquid-level detector in the mid-pressure receiver
- Advanced compressor safety and control
- Differential pressure sensors over oil units and heat exchangers

Having built a complex experimental system that includes a number of operating configurations, such as automatic, or capillary tube expansion device options and operating conditions under either/both subcritical and transcritical; it is recommended that the system be maintained and made available as a test facility for industry. This will assist in speeding-up, so to say, the conversion of the South African refrigeration situation to a more environmentally friendly lower global warming country.

Appendix A -   MEGAWATSOFT INC. MS EXCEL *ADD-IN* INFORMATION

MegaWatSoft Inc. provides the add-in for the price of $400 – price correct at date of publish of this report. Version 2.0 of the *add-in* is used. Version 3.0 only includes 64-bit functionality and version 4.0 was not available at the time.

A.1 CALCULABLE PROPERTIES

The Excel *add-in* has the capability to calculate 29 different properties. Appendix Table A.1 contains a list of the properties, their property ID, the SI unit and their English unit.

Appendix Table A.1 Calculable Properties of MS Excel *Add-in*

PROPERTY NAME	PROPERTY ID	SI UNIT	ENGLISH UNIT
Pressure	p	bar	psi
Temperature	t	°C	°F
Density	d	$\frac{kg}{m^3}$	$\frac{lb}{ft^3}$
Specific Volume	v	$\frac{m^3}{kg}$	$\frac{ft^3}{lb}$
Specific Enthalpy	h	$\frac{kJ}{kg}$	$\frac{Btu}{lb}$
Specific Entropy	s	$\frac{kJ}{kg\,K}$	$\frac{Btu}{lb\,°F}$
Specific Internal Energy	u	$\frac{kJ}{kg}$	$\frac{Btu}{lb}$
Specific Isobaric Heat Capacity	cp	$\frac{kJ}{kg\,K}$	$\frac{Btu}{lb\,°F}$
Specific Isochoric Heat Capacity	cv	$\frac{kJ}{kg\,K}$	$\frac{Btu}{lb\,°F}$
Speed of Sound	w	$\frac{m}{s}$	$\frac{ft}{s}$
Isentropic Exponent	kappa	#	#

Appendix Table A.2 Calculable Properties of MS Excel *Add-in* (Contd.)

PROPERTY NAME	PROPERTY ID	SI UNIT	ENGLISH UNIT
Specific Helmholtz Free Energy	H	$\frac{kJ}{kg}$	$\frac{Btu}{lb}$
Specific Gibbs Free Energy	G	$\frac{kJ}{kg}$	$\frac{Btu}{lb}$
Compressibility Factor	Z	#	#
Quality (vapour mass fraction)	x	%	%
Isobaric Volume Expansion Coefficient	alpha	$\frac{1}{K}$	$\frac{1}{°F}$
Isothermal Compressibility	Kt	$\frac{1}{MPa}$	$\frac{1}{psi}$
Partial Derivative $\left(\frac{dP}{dT}\right)_v$	dpdt	$\frac{MPa}{K}$	$\frac{psi}{°F}$
Partial Derivative $\left(\frac{dP}{dV}\right)_T$	dpdv	$\frac{MPa\,kg}{m^3}$	$\frac{psi\,lb}{ft^3}$
Partial Derivative $\left(\frac{dV}{dT}\right)_p$	dvdt	$\frac{m^3}{kg\,K}$	$\frac{ft^3}{lb\,°F}$
Partial Derivative $\left(\frac{dV}{dP}\right)_T$	dvdp	$\frac{m^3}{kg\,MPa}$	$\frac{ft^3}{lb\,psi}$
Isothermal Joule-Thomson Coefficient	iJTC	$\frac{kJ}{kg\,MPa}$	$\frac{Btu}{lb\,psi}$
Joule-Thomson Coefficient	JTC	$\frac{K}{MPa}$	$\frac{°F}{psi}$
Melting Pressure	pm	bar	psi
Dynamic Viscosity	dv	μPa s	$\frac{lb}{ft\,h}$
Kinematic Viscosity	kv	$\frac{μm^2}{s}$	$\frac{ft^2}{h}$
Thermal Conductivity	tc	$\frac{W}{K\,m}$	$\frac{Btu}{h\,ft\,°F}$
Thermal Diffusivity	td	$\frac{μm^2}{s}$	$\frac{ft^2}{h}$
Prandtl Number	Pr	#	#

A.2 AVAILABLE FUNCTIONS

The following two lists detail the available functions within the *add-in*. These functions are used within MS Excel in the exact same manner as any of the built-in functions in MS Excel. The functions for SI units are listed, followed by the functions for English units and finally a few examples are given using the SI unit functions.

1. **SatCO2siPX** (pressure, quality, propertyID)
2. **SatCO2siTX** (temperature , quality, propertyID)
3. **CO2siPT** (pressure, temperature, propertyID)
4. **CO2siPH** (pressure, enthalpy, propertyID)
5. **CO2siPS** (pressure, entropy, propertyID)
6. **CO2siPU** (pressure , internal energy, propertyID)
7. **CO2siPV** (pressure , volume, propertyID)
8. **CO2siTH** (temperature, enthalpy, propertyID)
9. **CO2siTS** (temperature, entropy, propertyID)
10. **CO2siTU** (temperature , internal energy, propertyID)
11. **CO2siTV** (temperature , volume, propertyID)

1. **SatCO2ePX** (pressure, quality, propertyID)
2. **SatCO2eTX** (temperature , quality, propertyID)
3. **CO2ePT** (pressure, temperature, propertyID)
4. **CO2ePH** (pressure, enthalpy, propertyID)
5. **CO2ePS** (pressure, entropy, propertyID)
6. **CO2ePU** (pressure , internal energy, propertyID)
7. **CO2ePV** (pressure , volume, propertyID)
8. **CO2eTH** (temperature, enthalpy, propertyID)
9. **CO2eTS** (temperature, entropy, propertyID)
10. **CO2eTU** (temperature , internal energy, propertyID)
11. **CO2eTV** (temperature , volume, propertyID)

A.3 EXAMPLES

Selecting the subcritical, superheated point where pressure is equal to 50 bar and temperature is equal to 50 °C:

$$\text{CO2siPT}(50,50, "h") = -25.5511229880 \, \frac{\text{kJ}}{\text{kg}} \qquad\qquad (\text{A.1})$$

$$\text{CO2siPT}(50,50, "s") = -0.7745949655 \, \frac{\text{kJ}}{\text{kg K}} \qquad\qquad (\text{A.2})$$

$$\text{CO2siPT}(50,50, "d") = 104.8571045546 \, \frac{\text{kg}}{\text{m}^3} \qquad\qquad (\text{A.3})$$

$$\text{CO2siPT}(50,50, "tc") = 0.022087 \, \frac{\text{W}}{\text{K m}} \qquad\qquad (\text{A.4})$$

$$\text{CO2siPT}(50,50, "Pr") = 1.05497 \qquad\qquad (\text{A.5})$$

A.4 USED CONSTANTS

The following constants are used in the MS Excel *add-in*.

Specific gas constant:

$$R = 0.188923 \, \frac{\text{kJ}}{\text{kg K}} = 0.04512348 \, \frac{\text{Btu}}{\text{lb °F}} \qquad\qquad (\text{A.6})$$

Molar gas constant:

$$R_m = 8.31451 \, \frac{\text{J}}{\text{mol K}} = 1.985609881262 \, \frac{\text{Btu}}{\text{lbmol°F}} \qquad\qquad (\text{A.7})$$

Molar mass:

$$M = 44.010 \, \frac{\text{g}}{\text{mol}} = 44.010 \, \frac{\text{lbm}}{\text{lbmol}} \qquad\qquad (\text{A.8})$$

Critical Temperature:

$$T_c = 304.1282 \, \text{K} = 87.76076 \, \text{°F} \qquad\qquad (\text{A.9})$$

Critical Pressure:

$$P_c = 7.377 \, \text{MPa} = 1069.936763938 \, \text{psi} \qquad\qquad (\text{A.10})$$

Critical Density:

$$\rho_c = 467.6 \ \frac{\text{kg}}{\text{m}^3} = 29.192158821326 \ \frac{\text{lb}}{\text{ft}^3} \qquad\qquad (\text{ A.11 })$$

Triple-point Temperature:

$$T_t = 216.6 \ \text{K} = -69.79 \ \text{°F} \qquad\qquad (\text{ A.12 })$$

Triple-point Pressure:

$$P_t = 0.51796 \ \text{MPa} = 75.123281313454 \ \text{psi} \qquad\qquad (\text{ A.13 })$$

Appendix B -   SCHEMATICAL DIAGRAM OF THE EXPERIMENTAL SYSTEM
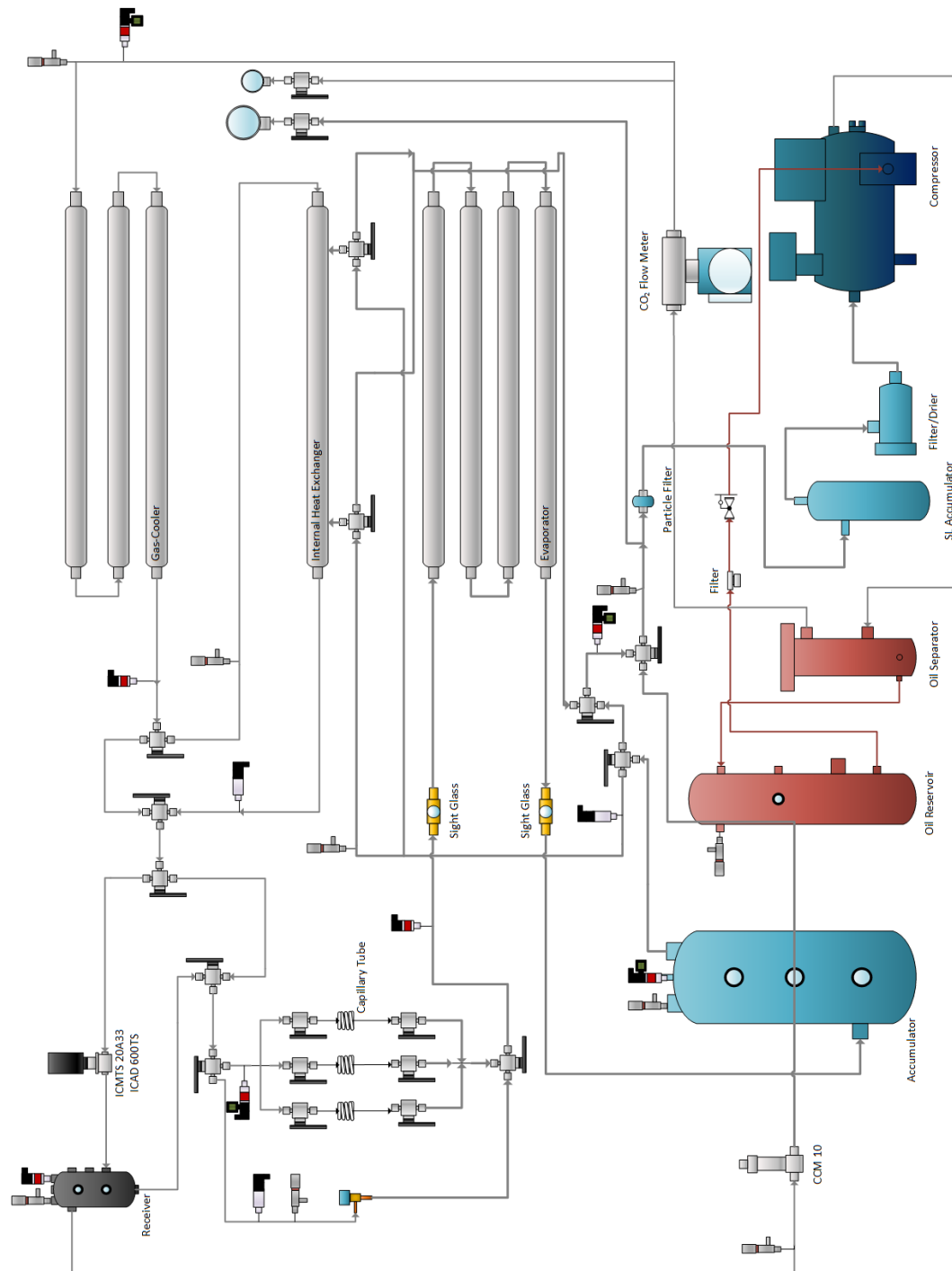
Appendix Figure B.1 Diagram of the Experimental System

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Appendix C -   ELECTRONIC SCHEMATIC DIAGRAM
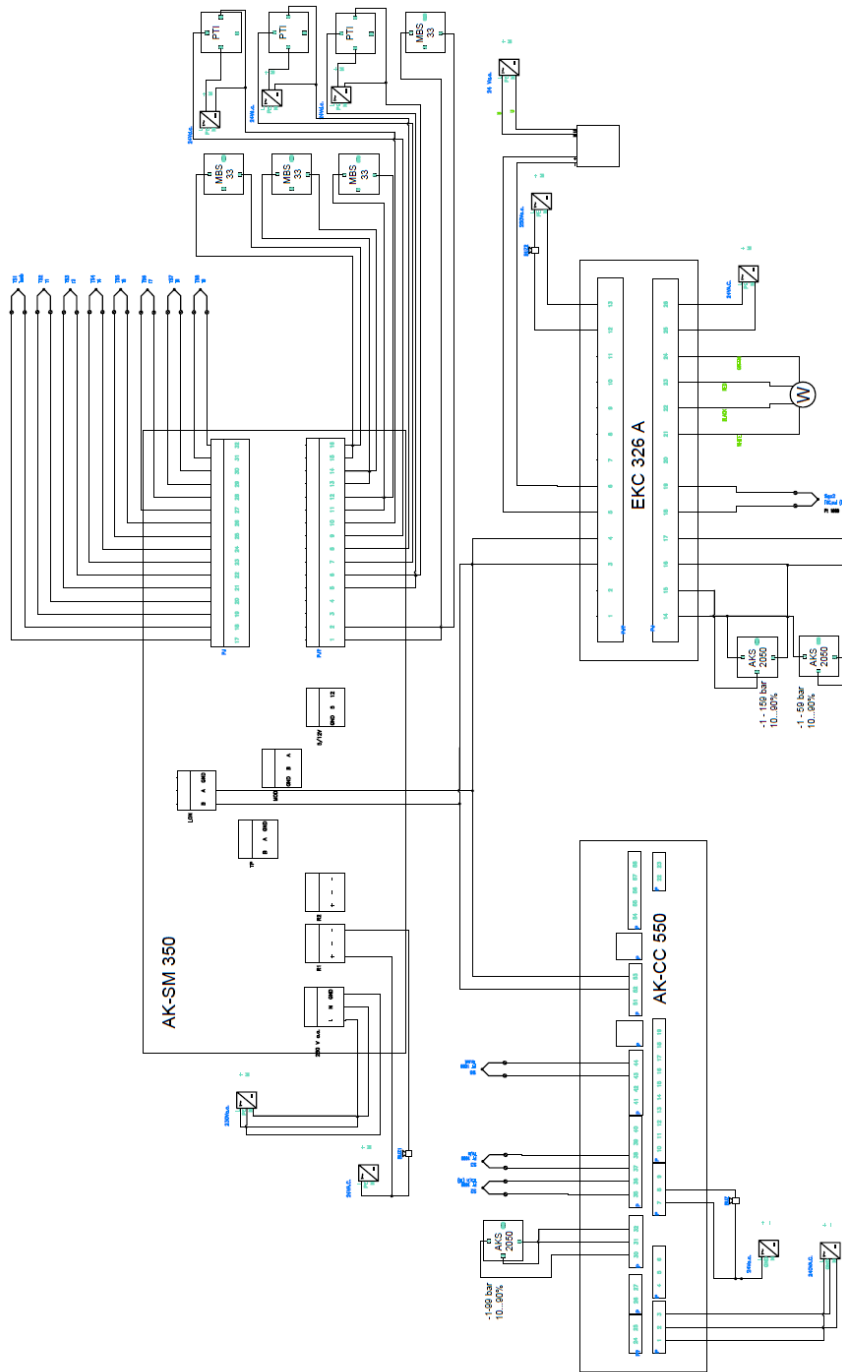
Appendix Figure C.1 Electronic Layout Diagram

Appendix D -   SC SIMULATION DESCRIPTION

The following is a brief introduction to the Microsoft Excel program called SC Simulation v.5.

When the program is opened, the user is greeted with the following screen:

Welcome to a theoretical simulation of a transcritical/subcritical basic CO_2 refrigeration system

In this program, you will be asked to specify the following:

- Input temperatures and mass flow rate of the heating and cooling water
- Pipe diameters for all the involved components
- Desired pressure difference in the system
- Desired operating conditions of the system
- Subcritical or transcritical operation
- Piping material used

Press the "Start" button to begin

Start

Appendix Figure D.1 SC Simulation Welcome Page

Once the user presses "Start", a pop-up screen is displayed. At the top of the screen, there are seven tabs. The first tab, "Layout of the System", is the first to open by default. It shows a simplified layout of the system, as well as the first choices the user must make regarding the system operation. The user must choose between Parallel Flow and Counter Flow. Parallel Flow is selected by default. Next, the user must choose between Transcritical and Subcritical Operation. Transcritical Operation is selected by default. Lastly, the user must choose whether he/she would like to include an Internal Heat Exchanger in their system. Should the user decide not to include the IHX, then the IHX tab will disappear from the selection.

After these selections have been made, the user can either switch between each of the respective tabs and select custom operating parameters and pipe diameters or choose to press the "Set Default/Test Values". This will change all the variable parameters to a set of default values, taking in account the choices the user has made on the "Layout of the System" tab.

Once all the variable parameters have been set, the user may proceed to press the "Save and Solve" button. This will save the chosen parameters to the simulation file and continue to solve the simulation problem.

The user may also choose to press "Clear All". This will empty all the chosen parameters, apart from those on the "Layout of the System" tab. To close the window and return to the Microsoft Excel environment, press "Close Without Saving".

The user will be prompted with each of the buttons whether he/she truly wishes to continue. If the user tries to press, "Save and Solve" without all the variable parameters set, then the user will receive an error message. Lastly, once the user presses "Save and Solve", and all the parameters have been correctly selected, then the user will be asked to kindly be patient whilst the simulation runs in the background.
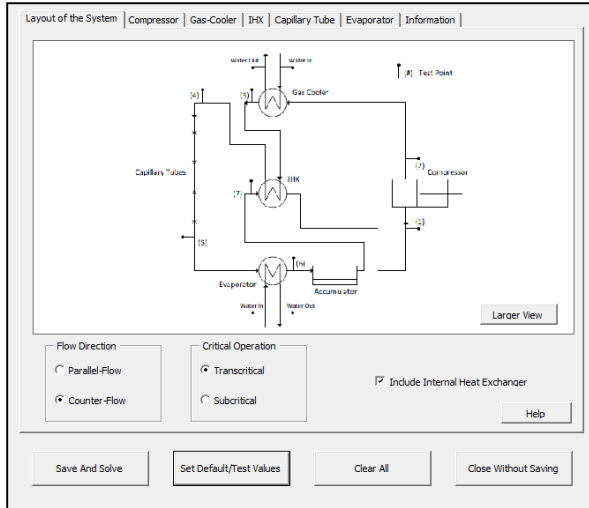
Appendix Figure D.2 Layout of the System (tab)

If the user chooses to set all the variable parameters manually, then he/she may proceed to page through the tabs. Appendix Figure D.3 to Appendix Figure D.7 shows a screenshot of each tab, with default/test values already chosen.

Showing the Compressor tab. The tab has basic information regarding the compressor and a photo of the compressor the author will be using in the experimental setup.

Appendix Figure D.3 Compressor (tab)

Showing the gas-cooler tab. On the tab: variable parameters, a simple control volume of the gas-cooler, P-h-diagram showing where the gas-cooler will operate.
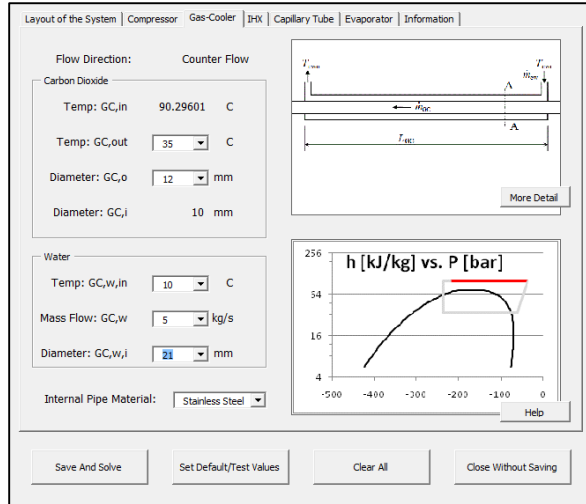
Appendix Figure D.4 Gas-Cooler (tab)

Showing the IHX tab. The layout of the tab is similar to that of the gas-cooler.
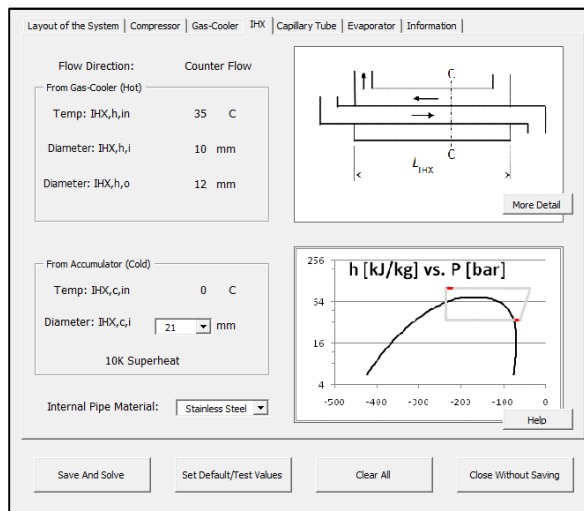
Appendix Figure D.5 IHX (tab)

Showing the Capillary Tube tab. The layout of the tab is similar to that of the gas-cooler.
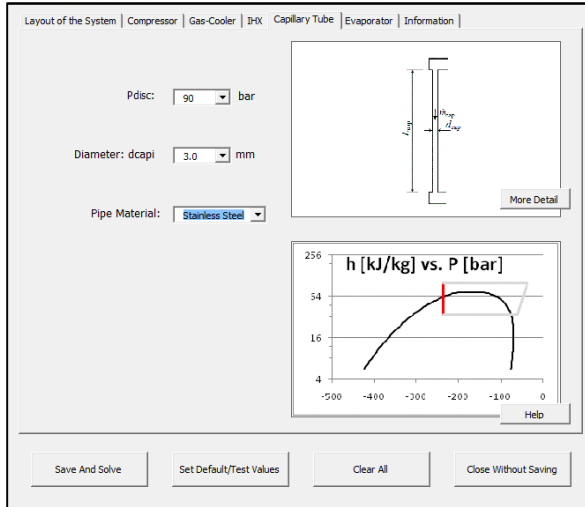
Appendix Figure D.6 Capillary Tube (tab)

Showing the Evaporator tab. The layout of the tab is similar to that of the gas-cooler.
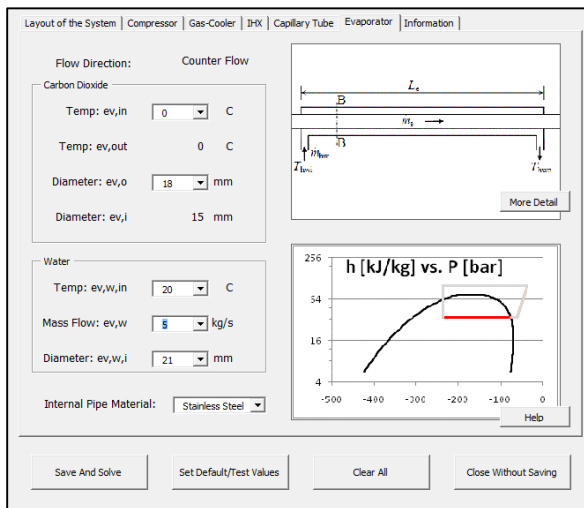
Appendix Figure D.7 Evaporator (tab)

The final tab shows basic information regarding the program. It also has the option to show a more detailed system layout schematic, included in Appendix B – Schematic Diagram of the Re-frigeration System.
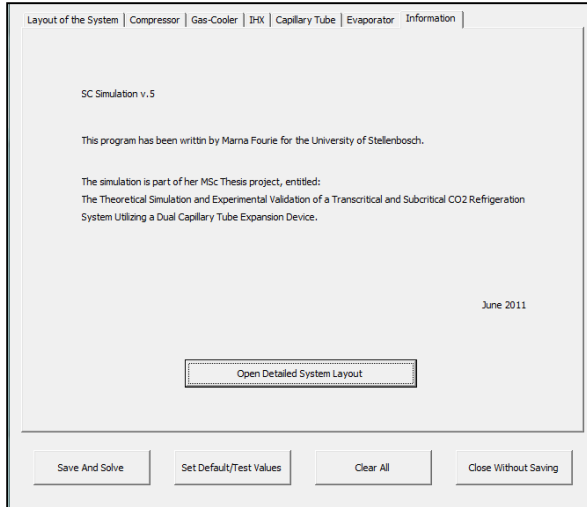
Appendix Figure D.8 Information (tab)

Each of the tabs had both a "Help" button and a "More Detail" button. The help button will open a window that holds an explanation of the respective tab and all the error messages you might receive on that tab.

The following, Appendix Figure D.9 Gas-Cooler Details, shows an example of what the user will see when he/she presses the "More Detail" button on the gas-cooler tab.
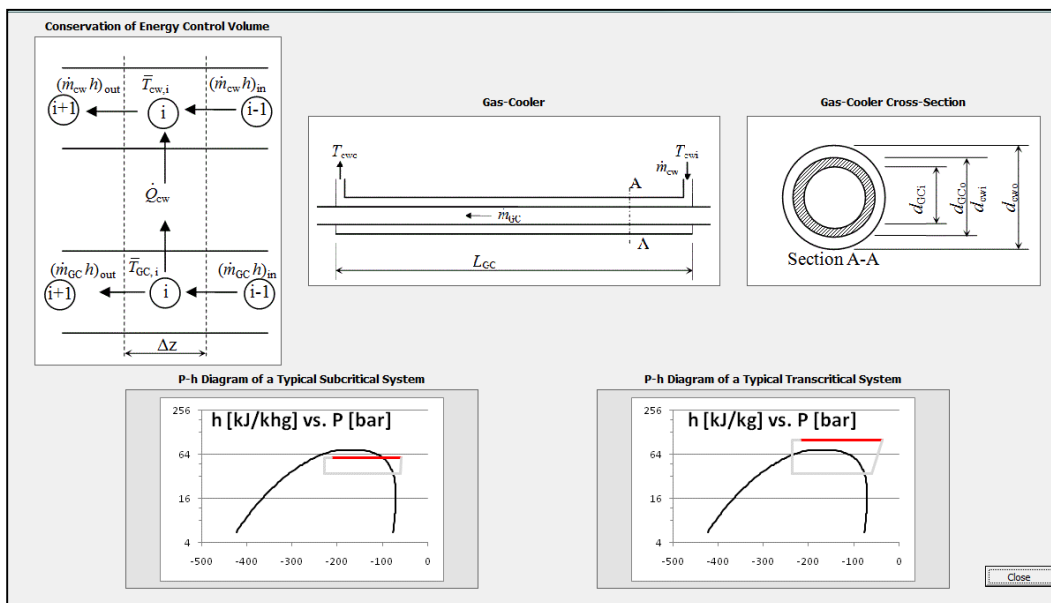
Appendix Figure D.9 Gas-Cooler Detail

Once the program has finished running, it will bring up a new pop-up screen. This screen will show the user the results of the simulation.

Screen, as with the previous, is divided up into different tabs. "Gas-Cooler" is the default tab. Appendix Figure D.10 to Figure D.15 shows each of the respective tabs and the results the simulation obtained.

In the results below, the program had to solve a default counter flow, transient system, with the IHX included.

The first tab, "Gas-Cooler", showing a graph with the refrigerant temperature, refrigerant enthalpy and the water temperature and input and output values of the gas-cooler as well as the diameter and length of the piping used.
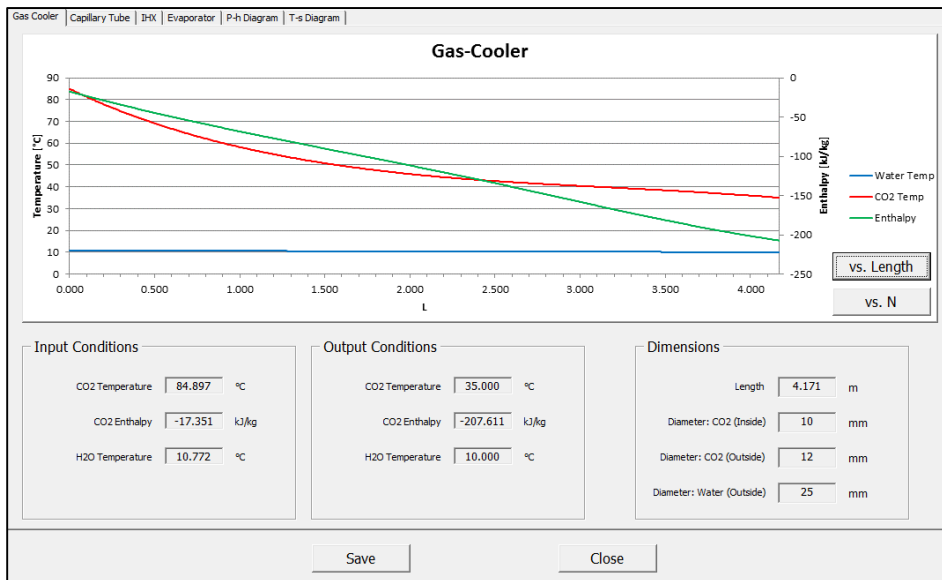
Appendix Figure D.10 Gas-Cooler Results (tab)

The second tab is "Capillary Tube", showing a graph with the refrigerant temperature and refrigerant pressure. The input and output values of the capillary tube as well as the diameter and length of the tube used is also included.
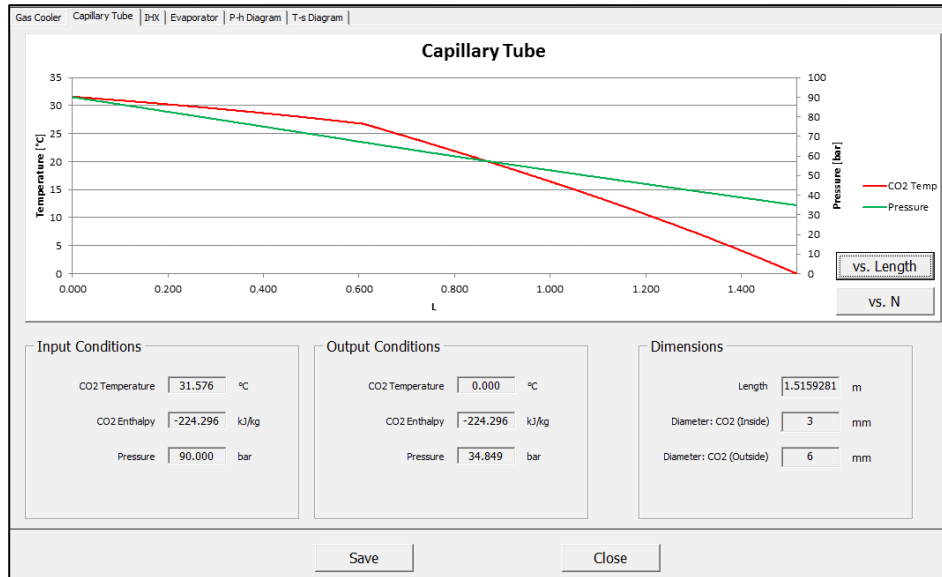
Appendix Figure D.11 Capillary Tube Results (tab)

The third tab is the "IHX". It shows a graph with the hot refrigerant temperature, hot refrigerant enthalpy, cold refrigerant temperature, cold refrigerant enthalpy and input and output values of the IHX as well as the diameter and length of the piping used. Hot refers to the refrigerant coming from the gas-cooler and cold refers to the refrigerant coming from the evaporator.
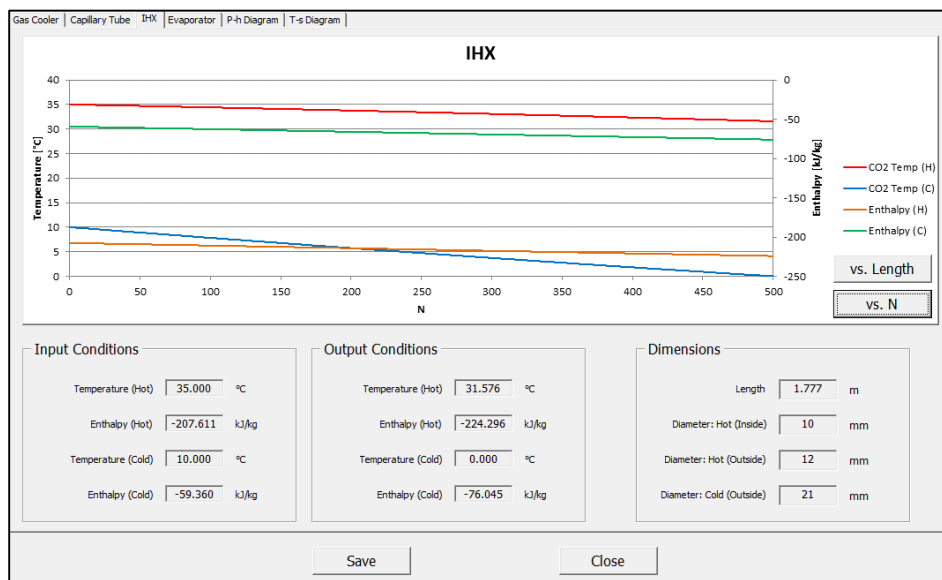
Appendix Figure D.12 IHX Results (tab)

The fourth tab showing the "Evaporator". It has a graph and values similar to that of the gas-cooler

Appendix Figure D.13 Evaporator Results (tab)

The second to last tab, "P-h Diagram", shows the pressure-enthalpy diagram of the solved system.
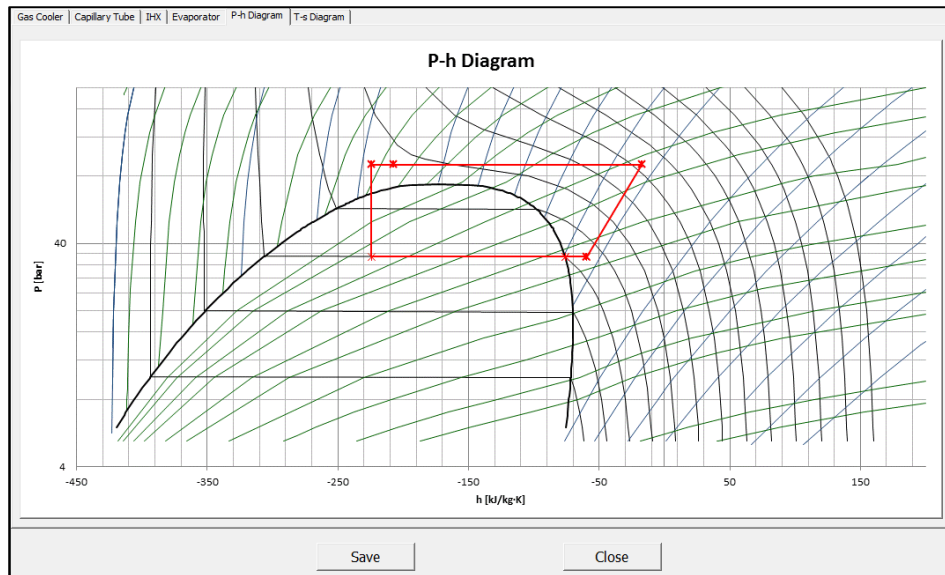
Appendix Figure D.14 P-h Diagram Results (tab)

The last tab, "T-s Diagram", shows the temperature-entropy diagram of the solved system.
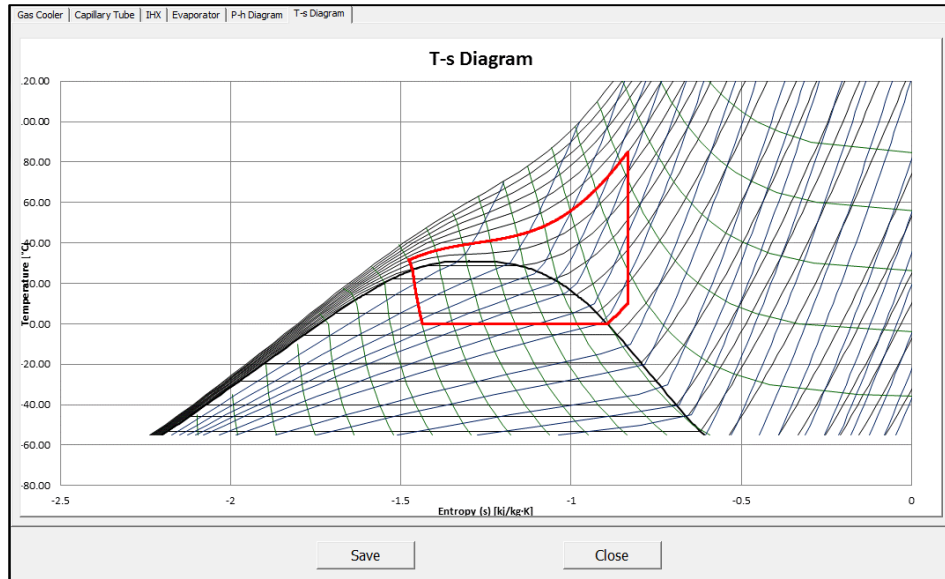
Appendix Figure D.15 Diagram Results (tab)

Appendix E - SAMPLE CALCULATIONS

The following sections detail the equations as used in the various simulation programs, for the gas-cooler and capillary tube only.

E.1 STEADY-STATE – SC SIMULATION V.5

The steady-state simulation – as outlined in Chapter 4.1 Steady-State Simulation – SC Simulation v.5 – utilizes the conservation equations, heat transfer equations and equations of state for CO_2 and water. This section outlines the equations used within the steady-state simulation.

Parameters chosen by the user for the gas-cooler – in SC Simulation v.5 – are in Appendix Table E.1:

Appendix Table E.1 Steady-State Simulation - Input Values, Gas-Cooler

PARAMETER	SYMBOL	VALUE	UNITS
Refrigerant Temperature, Out	$T_{GC,r,i=N}$	35	°C
Water Temperature, In	$T_{GC,w,i=1}$	10	°C
Inner Tube Diameter, Inside	$d_{GC,r,i}$	10	mm
Inner Tube Diameter, Outside	$d_{GC,r,o}$	12	mm
Outer Tube Diameter, Inside	$d_{GC,w,i}$	20	mm
Pressure, High	P_{high}	120	bar
Mass Flow Rate, Water	$\dot{m}_{GC,w}$	5	$\frac{kg}{s}$
Flow Direction	-	Counter Flow	-
Pipe Material	-	Stainless Steel	-
Critical State	-	Transcritical	-
Internal Heat Exchanger		Not Included	

Constant values,

Appendix Table E.2 Steady-State Simulation - Constant Values, Gas-Cooler

PARAMETER	SYMBOL	VALUE	UNITS
Volumetric Flow Rate	$\dot{V}_{comp}$	3.5	$\frac{m^3}{h}$
Number of Cells	N	500	-
Thermal conductivity, Stainless Steel	k_{ss}	16	$\frac{W}{m\,K}$

The basic calculations for physical parameters,

Inner pipe area on the refrigerant side $A_{GC,r,i}$ the inner pipe area on the water side $A_{GC,r,o}$,

$$A_{GC,r,i} = 2\pi d_{GC,r,i} = 62.8318.. \cdot 10^{-3} \ m^2 \tag{E.1}$$

$$A_{GC,r,o} = 2\pi d_{GC,r,o} = 75.3982.. \cdot 10^{-3} \ m^2 \tag{E.2}$$

The cross-sectional area of the refrigerant $A_{GC,r,x}$ the cross-sectional area of the water,

$$A_{GC,r,x} = \frac{\pi}{4} \ d_{GC,r,i}^2 = 78.5398.. \cdot 10^{-6} \ m^2 \tag{E.3}$$

$$A_{GC,w,x} = \frac{\pi}{4} \left(d_{GC,w,i}^2 - d_{GC,r,o}^2 \right) = 201.0619.. \cdot 10^{-6} \ m^2 \tag{E.4}$$

The hydraulic diameter of the water $d_{GC,w,h}$

$$d_{GC,w,h} = d_{GC,w,i} - d_{GC,r,o} = 8 \ mm \tag{E.5}$$

The mass flow rate in the heat exchanger is constant,

$$\dot{m}_{GC} = \frac{\dot{V}_{comp}}{3600} \ \rho_{suc} = 85.0780.. \cdot 10^{-3} \ \frac{kg}{s} \tag{E.6}$$

Where ρ_{suc} is the density of the refrigerant entering the compressor and

$$\rho_{suc} = f\left(P_{low}, T_{ev,r,i=N} + 10 \right) = 83.4562.. \ \frac{kg}{m^3} \tag{E.7}$$

The tube-in-tube heat exchanger is divided into N number of cells. For parallel flow, both the water and the CO_2 enters the heat exchanger at cell $i = 1$. With counter flow, CO_2 enters at $i = 1$, while the water enters at $i = N$. The conditions of the CO_2 and water entering the tube, assuming counter flow,

Enthalpy for the CO_2 entering and exiting the heat exchanger,

$$h_{GC,i=1} = f\left(P_{high}, T_{GC,r,i=1} \right) = -15.0984.. \ \frac{kJ}{kg} \tag{E.8}$$

$$h_{GC,i=N} = f\left(P_{high}, T_{(GC,r,i=N)} \right) = -226.873.. \frac{kJ}{kg} \tag{E.9}$$

The change in enthalpy,

$$\Delta h_{GC} = \frac{h_{GC,i=N} - h_{GC,i=1}}{N} = -423.55.. \cdot 10^{-3} \ \frac{kJ}{kg} \tag{E.10}$$

Thus, the distribution of enthalpy throughout the gas-cooler is given by the following two equations,

$$h_{GC,out,i=N} = h_{GC,in,i=N} + \Delta h_{GC} = -15.5219..\frac{kJ}{kg} \qquad \text{(E.11)}$$

And

$$h_{GC,in,i=N} = h_{GC,out,i-(N-1)} = -15.0984..\frac{kJ}{kg} \qquad \text{(E.12)}$$

The temperature of the refrigerant entering the heat exchanger,

$$T_{GC,r,in,i} = f\left(P_{high}, h_{GC,in,i=1}\right) = 102.17\,°C \qquad \text{(E.13)}$$

$$T_{GC,r,out,i} = f\left(P_{high}, h_{GC,out,i=1}\right) = 35°C \qquad \text{(E.14)}$$

The temperature distribution for the water in the gas-cooler is not known from the initial, chosen conditions. An output temperature value is assumed for start-up and then the correct value is determined via an iterative method. This is especially so when counter flow is selected. Assuming the change in temperature for the water, $\Delta T_{GC,w,i}$, is known (as a value for it was assumed), we get the following equations for the water temperature:

$$T_{GC,w,in,i=N} = T_{GC,w,out,i-(N-1)} = 10.818\,°C \qquad \text{(E.15)}$$

$$T_{GC,w,out,i=N} = T_{GC,w,in,i=N} + \Delta T_{GC,w,i=N}^{guess} = 10.820\,°C \qquad \text{(E.16)}$$

Thus, the difference in temperature, between the water and the refrigerant is:

$$\Delta T_{GC,in,i=N} = \text{abs}\left(T_{GC,w,out,i=N} - T_{GC,r,in,i=N}\right) = 91.7375..\,°C \qquad \text{(E.17)}$$

$$\Delta T_{GC,out,i=N} = \text{abs}\left(T_{GC,w,in,i=N} - T_{GC,r,out,i=N}\right) = 91.4896..\,°C \qquad \text{(E.18)}$$

Using the Log Mean Temperature Difference

$$\Delta T_{\mathrm{lm,GC},i=N} = \frac{\Delta T_{\mathrm{GC,in},i=N} - \Delta T_{\mathrm{GC,out},i=N}}{\ln\left(\frac{\Delta T_{\mathrm{GC,in},i=N}}{\Delta T_{\mathrm{GC,out},i=N}}\right)} = 91.6135..\,°C \qquad (\text{ E.19 })$$

The saturation properties for the CO_2 in the control volume is as follows, for saturated vapour and saturated liquid, respectively.

For enthalpy

$$h_{\mathrm{GC,g},i=N} = f\big(P_{\mathrm{high}}, x = 100\big) = \text{"Out of Range"} \qquad (\text{ E.20 })$$

$$h_{\mathrm{GC},\ell,i=N} = f\big(P_{\mathrm{high}}, x = 0\big) = \text{"Out of Range"} \qquad (\text{ E.21 })$$

For dynamic viscosity

$$\mu_{\mathrm{GC,r,g},i=N} = f\big(P_{\mathrm{high}}, x = 100\big) = \text{"Out of Range"} \qquad (\text{ E.22 })$$

$$\mu_{\mathrm{GC,r},\ell,i=N} = f\big(P_{\mathrm{high}}, x = 0\big) = \text{"Out of Range"} \qquad (\text{ E.23 })$$

SAT is a fictitious property used to simplify the logical conditions necessary for other equations.

$$h_{\mathrm{GC,in},i=N} \geq h_{\mathrm{GC,g},i=N} \qquad (\text{ E.24 })$$

Thus

$$SAT_{i=N} = 2 \qquad (\text{ E.25 })$$

If $SAT = 2$, then the refrigerant is superheated or in the supercritical region. If $SAT = 0$, then the refrigerant is supercooled. If $SAT = 1$, then the refrigerant is within the two-phase region.

The mass ratio of the CO_2 $x_{\mathrm{GC},i}$, is defined as follows:

$$x_{\mathrm{GC},i=N} = 200 \quad (SAT_{i=N} = 2) \qquad (\text{ E.26 })$$

It is worth a note that the mass ratio only has a value between 0 and 1 – or 0 and 100 % as in this simulation - and that the "200" and "-200" value are for logical declarations only.

Assuming a homogeneous two-phase flow model, the following homogeneous values for dynamic viscosity can be determined:

$$\mu_{\mathrm{GC,r,h},i=N} = f\big(P_{\mathrm{high}}, h_{\mathrm{GC,in},i=N}\big) = 23.6212..\,\mu Pa\,s\,(SAT_{i=N} = 2) \qquad (\text{ E.27 })$$

For water, we use a lookup table.

$$\mu_{\text{GC,w},i=N} = f_{\text{lookup}}(T_{\text{GC,w},i=N}) = 1.307.. \, \text{mPa s} \qquad (\text{ E.28 })$$

The Prandtl number for the refrigerant and the water is defined as,

$$\text{Pr}_{\text{GC,r},i=N} = f(P_{\text{high}}, T_{\text{GC,r},i=N}) = 1.2134.. \qquad (\text{ E.29 })$$

$$\text{Pr}_{\text{GC,w},i=N} = f_{\text{lookup}}(T_{\text{GC,w},i=N}) = 9.47.. \qquad (\text{ E.30 })$$

The thermal conductivity

$$k_{\text{GC,r},i=N} = f(P_{\text{high}}, T_{\text{GC,r},i=N}) = 0.0330.. \frac{\text{W}}{\text{K m}} \qquad (\text{ E.31 })$$

$$k_{\text{GC,w},i=N} = f_{\text{lookup}}(T_{\text{GC,w},i=N}) = 0.58.. \frac{\text{W}}{\text{K m}} \qquad (\text{ E.32 })$$

The specific isobaric heat capacity for the water: $\qquad (\text{ E.33 })$

$$c_{\text{p,w},i=N} = f_{\text{lookup}}(T_{\text{GC,w},i=N}) = 4.193.. \frac{\text{kJ}}{\text{kg K}} \qquad (\text{ E.34 })$$

Using the information available, we can determine the heat transfer rate of the refrigerant:

$$q_{\text{GC},i=N} = \dot{m}_{GC,r} \cdot \Delta h_{\text{GC},i} \cdot 1000 = 34.3659.. \, \text{W} \qquad (\text{ E.35 })$$

Using this value, we can finally determine the temperature difference in the water line:

$$\Delta T^{\text{calculated}}_{\text{GC,w},i=N} = \frac{q_{\text{GC},i=N}}{\dot{m}_{\text{GC,w}} \cdot c_{\text{p,w},i=N}} = 0.001639 \, °\text{C} \qquad (\text{ E.36 })$$

We can now compare the initial guessed value for $\Delta T_{\text{GC,w},i=N}$. If the values are equal, then the original assumption was correct. If not, then one can change the guessed value and iterate until,

$$\Delta T^{\text{guess}}_{\text{GC,w},i=N} = \Delta T^{\text{calculated}}_{\text{GC,w},i=N} = 0.001639 \, °\text{C} \qquad (\text{ E.37 })$$

Continuing to determine the length of each control volume:

The thermal resistance is used:

$$R_{\mathrm{GC},i=N} = \frac{\Delta T_{\mathrm{lm,GC},i=N}}{q_{\mathrm{GC},i=N}} = 2.6658..\frac{\mathrm{K}}{\mathrm{W}}$$ (E.38)

The Reynolds number and Nusselt number:

$$Re_{\mathrm{GC,r},i=N} = \frac{\dot{m}_{\mathrm{GC,r}}\, d_{\mathrm{GC,i}}}{A_{\mathrm{GC,x,r}}\, \mu_{\mathrm{GC,r},i=N}} = 437.3523..\cdot 10^3$$ (E.39)

$$Nu_{\mathrm{GC,r},i=N} = 0.023\, Re_{\mathrm{GC,r},i=N}^{0.8}\ Pr_{\mathrm{GC,r},i=N}^{\,0.4} = 809.1023$$ (E.40)

This gives a heat transfer coefficient of

$$ht_{\mathrm{GC,r},i=N} = Nu_{\mathrm{GC,r},i=N}\, \frac{k_{\mathrm{GC,r},i=N}}{d_{\mathrm{GC,i}}} = 2677.664..\frac{\mathrm{W}}{\mathrm{K\,m^2}}\ (Re_{GC,r,i} \geq 1183)$$ (E.41)

The heat transfer coefficient is usually donated as h in literature, but it is donated as ht here to easily distinguish it from the h of specific enthalpy.

The same equations can be used for the water:

The Reynolds number and Nusselt number:

$$Re_{\mathrm{GC,w},i=N} = \frac{\dot{m}_{\mathrm{GC,w}}\, d_{\mathrm{GC,w,h}}}{A_{\mathrm{GC,x,w}}\, \mu_{\mathrm{GC,w},i=N}} = 152.213..10^3$$ (E.42)

$$Nu_{\mathrm{GC,w},i=N} = 0.023\, Re_{\mathrm{GC,w},i=N}^{0.8}\ Pr_{\mathrm{GC,w},i=N}^{\,0.4} = 791.1002..$$ (E.43)

This gives a heat transfer coefficient of

$$ht_{\mathrm{GC,w},i=N} = Nu_{\mathrm{GC,w},i=N}\, \frac{k_{\mathrm{GC,w},i=N}}{d_{\mathrm{GC,w,h}}} = 57.354..10^3\, \frac{\mathrm{W}}{\mathrm{K\,m^2}}\big(Re_{\mathrm{GC,w},i=N} \geq 1183\big)$$

(E.44)

With all this information, we can finally determine the length of the control volumes:

$$\Delta z_{\mathrm{GC},i=N} = \frac{1}{R_{\mathrm{GC},i=N}}\left[\frac{1}{h_{\mathrm{GC,r},i=N}\, A_{\mathrm{GC,i}}} + \frac{1}{h_{\mathrm{GC,w},i=N}\, A_{\mathrm{GC,o}}} + \frac{\ln\frac{d_{\mathrm{GC,o}}}{d_{\mathrm{GC,i}}}}{2\pi\, k_{\mathrm{ss}}}\right] = 0.005313\ \mathrm{m}$$

(E.45)

Finally, the summation of all the Δz

$$\sum \Delta z_{\mathrm{GC},i} = 3.8727..\,\mathrm{m}$$ (E.46)

E.2 TRANSIENT – T SIMULATION V.1.5

The transient simulation program determines the final, steady-state conditions from an initial start-up condition. The equations to follow are for the gas-cooler in the system, as it has reached steady-state. Values, as determined by other components is in Appendix Table E.3.

Appendix Table E.3 T Simulation Gas-Cooler Conditions

PARAMETER	VALUE	UNIT
P_{high}	103.224	bar
$\rho_{\text{GC},i}$	222.625	kg/m^3
$\rho_{\text{GC,w},i}$	994.1	kg/m^3
Vol_{GC}	$3.6128..10^{-5}$	m^3
$Vol_{\text{GC,w}}$	$9.2488..10^{-5}$	m^3

The first property to determine is the refrigerant mass m_{GC} and the water mass $m_{\text{GC,w}}$ in each control volume of the gas-cooler.

$$m_{\text{GC},i=1} = \rho_{\text{GC},i=1}\, Vol_{\text{GC}} = 8.043..10^{-3}\ \text{[kg]} \qquad (\text{ E.47 })$$

$$m_{\text{GC,w},i=1} = \rho_{\text{GC,w},i=1}\, Vol_{\text{GC,w}} = 9.194..10^{-3}\text{[kg]} \qquad (\text{ E.48 })$$

The pseudo value SAT_{GC}, $SAT_{\text{GC},i=1} = 0$ (subcooled/transcritical)

The internal energy $U_{\text{GC},i=1}$ is determined using the specific internal energy $u_{\text{GC},i=1}$ of the previous time step and the newly calculated mass.

$$U_{\text{GC},i=1} = u_{\text{GC},i=1}\, m_{\text{GC},i=1} = -0.6257..\ \text{[kJ]} \qquad (\text{ E.49 })$$

The saturated liquid and saturated vapour values for enthalpy $h_{\text{GC,f},i}$ & $h_{\text{GC,g},i}$ and density $\rho_{\text{GC,f},i}$ & $\rho_{\text{GC,g},i}$, respectively, are:

$$h_{\text{GC,f},i=1} = f_{\text{SatCO2siPX}}\big(P_{\text{high},i=1}, x = 0\big) = \text{"Not in range"} \qquad (\text{ E.50 })$$

$$h_{\text{GC,g},i=1} = f_{\text{SatCO2siPX}}\big(P_{\text{high},i=1}, x = 100\big) = \text{"Not in range"} \qquad (\text{ E.51 })$$

$$\rho_{\text{GC,f},i=1} = f_{\text{SatCO2siPX}}\big(P_{\text{high},i=1}, x = 0\big) = \text{"Not in range"} \qquad (\text{ E.52 })$$

$$\rho_{\text{GC,g},i=1} = f_{\text{SatCO2siPX}}\big(P_{\text{high},i=1}, x = 100\big) = \text{"Not in range"} \qquad (\text{ E.53 })$$

The homogeneous enthalpy $h_{\text{GC,h},i}$

$$h_{\text{GC,h},i=1} = f_{\text{CO2siPU}}\left(P_{\text{high},i=1}, u_{\text{GC},i=1}\right) = -31.1455.. \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.54 })$$

The specific internal energy entering and exiting each control volume,

$$u_{\text{GC,in},i=1} = u_{\text{comp,out}} = -57.4742.. \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.55 })$$

$$u_{\text{GC,out},i=1} = u_{\text{GC},i=1} = -77.5123.. \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.56 })$$

The specific enthalpy entering and exiting each control volume,

$$h_{\text{GC,in},i=1} = h_{\text{comp,out},i=1} = -4.8623.. \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.57 })$$

$$h_{\text{GC,in},i=2} = h_{\text{GC},i=1} = -31.1455 \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.58 })$$

$$h_{\text{GC,out},i=1} = h_{\text{GC},i=1} = -31.1454.. \left[\frac{\text{kJ}}{\text{kg}}\right] \qquad (\text{ E.59 })$$

If the refrigerant is in the two-phase region, the dynamic viscosity $\mu_{\text{GC},i}$ and thermal conductivity $k_{\text{GC},i}$ are determined as follows,

$$\mu_{\text{GC,r,f},i=1} = f_{\text{CO2siPH}}\left(P_{\text{high},i=1}, h_{\text{GC,f},i=1} - 0.0001\right) = \text{"Not in range"} \qquad (\text{ E.60 })$$

$$\mu_{\text{GC,r,g},i=1} = f_{\text{CO2siPH}}\left(P_{\text{high},i=1}, h_{\text{GC,g},i=1} + 0.0001\right) = \text{"Not in range"} \qquad (\text{ E.61 })$$

$$\mu_{\text{GC,r},i=1} = f_{\text{CO2siPU}}\left(P_{\text{high},i=1}, SU_{\text{GC},i=1}\right) = 22.2488.. \ [\mu\text{Pa s}] \qquad (\text{ E.62 })$$

$$k_{\text{GC,r,f},i=1} = f_{\text{CO2siPH}}\left(P_{\text{high},i=1}, h_{\text{GC,f},i=1} - 0.001\right) = \text{"Not in range"} \qquad (\text{ E.63 })$$

$$k_{\text{GC,r,g},i=1} = f_{\text{CO2siPH}}\left(P_{\text{high},i=1}, h_{\text{GC,g},i=1} + 0.001\right) \ \text{"Not in range"} \qquad (\text{ E.64 })$$

$$k_{\text{GC,r},i=1} = f_{\text{CO2siPT}}\left(P_{\text{high},i=1}, T_{\text{GC},i=1}\right) = 0.3063..\,10^{-3} \left[\frac{\text{W}}{\text{K m}}\right] \qquad (\text{ E.65 })$$

The Prandtl number $\text{Pr}_{GC,i}$ for the refrigerant and the water respectively,

$$\text{Pr}_{\text{GC,r},i=1} = f_{\text{CO2siPT}}\left(P_{\text{high},i=1}, T_{\text{GC},i=1}\right) = 1.1125.. \qquad (\text{ E.66 })$$

$$\text{Pr}_{\text{GC},\text{w},i=1} = f_{\text{lookup table}}(T_{\text{GC},\text{w},i=1}) = 4.885 \tag{E.67}$$

The dynamic viscosity of the water,

$$\mu_{\text{GC},\text{w},i=1} = f_{\text{lookup table}}(T_{\text{GC},\text{w},i=1}) = 0.721 \,[\text{mPa s}] \tag{E.68}$$

Thus, the Reynolds numbers for the refrigerant and water are,

$$Re_{\text{GC},\text{r},i=1} = \frac{\dot{m}_{\text{high},i=1}\, d_{\text{i},\text{GC}}}{A_{\text{x},\text{GC}}\, \mu_{\text{GC},\text{r},i=1}\, 10^{-6}} \quad 342.439..\,10^3 \tag{E.69}$$

$$Re_{\text{GC},\text{w},i=1} = \frac{\dot{m}_{\text{GC},\text{w},i=1}\, d_{\text{o},\text{GC}}}{A_{\text{xw},\text{GC}}\, \mu_{\text{GC},\text{w},i=1}\, 10^{-3}} \quad 579.087..\,10^3 \tag{E.70}$$

The Nusselt number is,

$$Nu_{\text{GC},\text{r},i=1} = 0.023\, Re_{\text{GC},\text{r},i=1}^{0.8} \quad \text{Pr}_{\text{GC},\text{r},i=1}^{0.4} = 654.5698.. \tag{E.71}$$

$$Nu_{\text{GC},\text{w},i=1} = 0.023\, Re_{\text{GC},\text{w},i=1}^{0.8} \quad \text{Pr}_{\text{GC},\text{w},i=1}^{0.4} = 1767.9457.. \tag{E.72}$$

$$Re_{\text{GC},\text{r},i=1} \geq 1183$$

$$h_{\text{GC},\text{r},i=1} = \frac{Nu_{\text{GC},\text{r},i=1}\, k_{\text{GC},\text{r},i=1}}{d_{\text{i},\text{GC}}} = 1968.3540..\left[\frac{\text{W}}{\text{K m}^2}\right] \tag{E.73}$$

$$Re_{\text{GC},\text{w},i=1} \geq 1183$$

$$h_{\text{GC},\text{w},i=1} = \frac{Nu_{\text{GC},\text{w},i=1}\, k_{\text{GC},\text{w},i=1}}{d_{\text{hw},\text{GC}}} = 137.6787..\,10^3 \left[\frac{\text{W}}{\text{K m}^2}\right] \tag{E.74}$$

$$R_{\text{GC},i=1} = \frac{1}{h_{\text{GC},\text{r},i=1}\, A_{\text{i},\text{GC}}} + \frac{1}{h_{\text{GC},\text{w},i=1}\, A_{\text{o},\text{GC}}} + \frac{\ln\frac{d_{\text{o},\text{GC}}}{d_{\text{i},\text{GC}}}}{2\pi\, k_{\text{ss}}\, \Delta z_{\text{GC}}} = 0.3951..\,10^{-3}\left[\frac{\text{K}}{\text{W}}\right] \tag{E.75}$$

$$q_{\text{GC},i=1} = \frac{T_{\text{GC},i=1} - T_{\text{GC},\text{w},i=1}}{R_{\text{GC},i=1}} = 1648.3594\,[\text{W}] \tag{E.76}$$

$$\sum \dot{m}_{\text{GC},i=1} = \dot{m}_{\text{GC},\text{in},i=1} - \dot{m}_{\text{GC},\text{out},i=1} = 2.9788..\,10^{-3}\left[\frac{\text{kg}}{\text{s}}\right] \tag{E.77}$$

$$m_{\text{GC},i=1}^{\text{new}} = m_{\text{GC},i=1} + \Delta t \sum \dot{m}_{\text{GC},i=1} = 8.0728..\,10^{-3}\,[\text{kg}] \tag{E.78}$$

$$\rho_{\text{GC},i=1}^{\text{new}} = \frac{m_{\text{GC},i=1}^{\text{new}}}{Vol_{\text{GC}}} = 223.4500.. \left[\frac{\text{kg}}{\text{m}^3}\right] \tag{E.79}$$

$$(\dot{m}h)_{\text{GC,in},i=1} = \dot{m}_{\text{high,in},i=1} \, h_{\text{GC,in},i=1} = -0.2793.. \left[\frac{\text{kJ}}{\text{s}}\right] \tag{E.80}$$

$$(\dot{m}h)_{\text{GC,out},i=1} = \dot{m}_{\text{high,out},i=1} \, h_{\text{GC,out},i=1} = -1.6967.. \left[\frac{\text{kJ}}{\text{s}}\right] \tag{E.81}$$

$$\sum \dot{E}_{\text{GC,r},i=1} = (\dot{m}h)_{\text{GC,in},i=1} - (\dot{m}h)_{\text{GC,out},i=1} - \frac{q_{\text{GC},i=1}}{1000}$$

$$= -0.2309.. [\text{kW}] \tag{E.82}$$

$$U_{\text{GC},i=1}^{\text{new}} = u_{\text{GC},i=1} \, m_{\text{GC},i=1} + \Delta t \, \sum \dot{E}_{\text{GC,r},i=1} = -0.6257.. [\text{kJ}] \tag{E.83}$$

$$u_{\text{GC},i=1}^{\text{new}} = \frac{U_{\text{GC},i=1}^{\text{new}}}{m_{\text{GC},i=1}^{\text{new}}} = -77.5124.. \left[\frac{\text{kJ}}{\text{kg}}\right] \tag{E.87}$$

E.3 CAPILLARY TUBE – CT SIMULATION V.1

The capillary tube simulation model is designed to determine the mass flow rate of the refrigerant passing through a capillary tube. This section gives a sample solution to the simulation.

The following sample is based on a transcritical input condition and values required to start the capillary tube simulation is in Appendix Table E.4.

Appendix Table E.4 Capillary Tube Simulation - Initial Conditions

PARAMETER	SYMBOL	VALUE	UNITS
Transcritical/Subcritical	-	Transcritical	-
Refrigerant Temperature, In	$T_{CT,i=1}$	35	°C
Refrigerant Temperature, Out	$T_{CT,i=N}$	0	°C
Pressure, High (Transcritical Only)	P_{high}	120	bar
Capillary Tube Inside Diameter	$d_{CT,i}$	2	mm
Capillary Tube Length	L_{CT}	1	m

Constant values hard-coded into the simulation is in Appendix Table E.5.

Appendix Table E.5 Capillary Tube Simulation - Constant Values

PARAMETER	SYMBOL	VALUE	UNITS
Number of Cells	N	100	-
Pipe Material	-	Stainless Steel	-
Thermo conductivity, Stainless Steel	k_{ss}	16	$\frac{W}{mK}$

The first values calculated is the cross sectional area $A_{CT,x}$ specific enthalpy h_{CT}, the low pressure P_{low}, the density initial conditions $\rho_{CT,i=N}$ and $\rho_{CT,i=1}$.

$$A_{CT,x} = \frac{\pi}{4} d_{CT,i}^2 = 3.1416.. \cdot 10^{-6} \text{ m}^2 \qquad (\text{E.88})$$

$$h_{CT} = f_{table}\left(P_{high}, T_{CT,i=1}\right) = -254.4914.. \frac{kJ}{kg} \qquad (\text{E.89})$$

$$P_{low} = f_{table}\left(T_{CT,i=N}\right) = 34.8488.. \text{ bar} \qquad (\text{E.90})$$

The pressure distribution is assumed to be constant from one control volume to the next, while the length of each control volume is not necessarily constant.

Thus,

$$\Delta P_{\mathrm{CT}} = \frac{P_{\mathrm{CT,in}} - P_{\mathrm{CT,out}}}{N-1} = 0.8601..\,\mathrm{bar} \qquad (\text{E.91})$$

The pressure in each control volume is,

$$P_{\mathrm{CT},i=1} = 120\,\mathrm{bar} \qquad (\text{E.92})$$

$$P_{\mathrm{CT},i=2} = P_{\mathrm{CT},i=1} - \Delta P_{\mathrm{CT}} = 119.1398..\,\mathrm{bar} \qquad (\text{E.93})$$

The corresponding temperatures,

$$T_{\mathrm{CT},i=1} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i=1}, h_{\mathrm{CT}}\big) = 24.9966..\,°\mathrm{C} \qquad (\text{E.94})$$

The pseudo parameter SAT,

$$SAT_{\mathrm{CT},i=1} = 0 \quad (h_{\mathrm{CT}} < h_{\mathrm{CT},\ell,i=1}) \qquad (\text{E.95})$$

Thus, the mass ratio becomes,

$$x_{\mathrm{CT},i} = 200 \quad (SAT_{\mathrm{CT},i} = 0) \qquad (\text{E.96})$$

The specific enthalpy for saturated vapour and saturated liquid,

$$h_{\mathrm{CT,g},i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 100\big) = \text{"Not in range"} \qquad (\text{E.97})$$

$$h_{\mathrm{CT},\ell,i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 0\big) = \text{"Not in range"} \qquad (\text{E.98})$$

The density and dynamic viscosity of saturated vapour and saturated liquid,

$$\rho_{\mathrm{CT,g},i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 100\big) = \text{"Not in range"} \qquad (\text{E.99})$$

$$\rho_{\mathrm{CT},\ell,i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 0\big) = \text{"Not in range"} \qquad (\text{E.100})$$

$$\mu_{\mathrm{CT,g},i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 100\big) = \text{"Not in range"} \qquad (\text{E.101})$$

$$\mu_{\mathrm{CT},\ell,i} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i}, x = 0\big) = \text{"Not in range"} \qquad (\text{E.102})$$

The homogeneous density and dynamic viscosity,

$$\rho_{\mathrm{CT,h},i=1} = f_{\mathrm{table}}\big(P_{\mathrm{CT},i=1}, h_{\mathrm{CT}}\big) = 845.3483..\,\frac{\mathrm{kg}}{\mathrm{m}^3} \qquad (\text{E.103})$$

$$\mu_{\text{CT,h},i=1} = f_{\text{table}}\big(P_{\text{CT},i=1}, h_{\text{CT}}\big) = 79.3214.. \,\mu\text{Pa s} \qquad (\text{ E.104 })$$

The Reynolds number,

$$Re_{\text{CT},i=1} = \frac{\dot{m}_{\text{CT}}\, d_{\text{CT},i}}{A_{\text{CT,x}}\, \mu_{\text{CT,h},i=1}} = 980.2581.. \,10^3 \qquad (\text{ E.105 })$$

With the Reynolds number $Re_{\text{CT},i=1} > 1183$,

$$C_{\text{f}\ell\text{o,CT},i=1} = 0.078\, Re_{\text{CT},i=1}^{-0.25} = 2.4789.. \,10^{-3} \qquad (\text{ E.106 })$$

$$u_{\ell\text{o,CT},i=1}^2 = \left(\frac{\dot{m}_{\text{CT}}}{\rho_{\text{CT,h},i=1}\, A_{\text{CT,x}}}\right)^2 = 2.1146.. \,10^3 \left(\frac{\text{m}}{\text{s}}\right)^2 \qquad (\text{ E.107 })$$

The friction pressure gradient,

$$\left(\frac{\Delta P}{\Delta z}\right)_{\text{F},\ell\text{o,CT},i=1} = -C_{\text{f}\ell\text{o,CT},i=1}\, \rho_{\text{CT,h},i=1}\, u_{\ell\text{o,CT},i=1}^2 \frac{p_{\text{CT}}\, \Delta z_{\text{CT}}}{2\, A_{(\text{CT,x})}}$$

$$= 44.3126.. \,10^3 \frac{\text{Pa}}{\text{m}} \qquad (\text{ E.108 })$$

The two-phase friction multiplier,

$$\phi_{\ell\text{0,CT},i=1}^2 = 1 \qquad (\text{ E.109 })$$

$$\left(\frac{\Delta P}{\Delta z}\right)_{\text{F,CT},i=1} = \left(\frac{\Delta F}{\Delta z}\right)_{\text{F},\ell\text{o,CT},i=1} \phi_{\ell\text{0,CT},i=1}^2 = 44.3126.. \,10^3 \frac{\text{Pa}}{\text{m}} \qquad (\text{ E.110 })$$

To determine the momentum pressure gradient, one needs to determine the saturated vapour and liquid densities of the refrigerant exiting the control volume.

$$\rho_{\text{CT,out,g},i=1} = f\big(P_{\text{CT},i=2}, x = 100\big) = \text{"Not in range"} \qquad (\text{ E.111 })$$

$$\rho_{\text{CT,out,}\ell,i=1} = f\big(P_{\text{CT},i=2}, x = 0\big) = \text{"Not in range"} \qquad (\text{ E.112 })$$

The exiting homogeneous density is as follows,

$$\rho_{\text{CT,out,h},i=1} = f\big(P_{\text{CT},i=1}, h_{\text{CT}}\big) = 844.7429.. \,\text{kg/m}^3 \qquad (\text{ E.113 })$$

Thus, the momentum pressure gradient,

$$\left(\frac{\Delta P}{\Delta z}\right)_{\text{M,CT},i=1} = \frac{\dot{m}_{\text{CT}}^2}{A_{\text{CT,x}}^2}\left(\frac{1}{\rho_{\text{CT,out},i=1}} - \frac{1}{\rho_{\text{CT},i=1}}\right) = 1.2811.. \,10^3 \frac{\text{Pa}}{\text{m}} \qquad (\text{ E.114 })$$

The new pressure for the control volume is

$$P_{\text{CT},i=2}^{\text{new}} = \frac{\left(-\left(\frac{\Delta P}{\Delta z}\right)_{\text{F,CT},i=1} - \left(\frac{\Delta P}{\Delta z}\right)_{\text{M,CT},i=1}\right)}{100\,000} + P_{\text{CT},i=1} = 119.544\ \text{bar} \qquad (\text{E.115})$$

The pressure in the next control volume is updated,

$$P_{\text{CT},i=2} = P_{\text{CT},i=1}^{\text{new}} \qquad (\text{E.116})$$

The simulation runs through all the control volumes until it determines $P_{\text{CT},i=N}$

If the calculated value for the outlet pressure is equal to the actual value, then the simulation ends. Otherwise, the mass flow rate is adjusted and the program iterates through all the control volumes again.

Appendix F -   CO₂ FLUIDS PROPERTIES DATABASE EXCERPT

Appendix Table F.1 is an excerpt of the database used to determine fluid properties and state of carbon dioxide.

Appendix Table F.1 CO₂ Fluid Properties Database Excerpt

P	h	T	u	s	rho	k	μ	Pr	x
bar	$\frac{kJ}{kg}$	°C	$\frac{kJ}{kg}$	$\frac{kJ}{kg\,K}$	$\frac{kg}{m^3}$	$\frac{W}{K\,m}$	µPa s		%
20	-350	-19.5007	-352.09	-1.89702	956.9343	0.124358	132.9892	2316.471	0.405595
20	-349.5	-19.5007	-351.654	-1.89505	928.4158	0.120899	130.8597	2343.012	0.58358
20	-349	-19.5007	-351.218	-1.89308	901.5479	0.11764	128.7973	2368.369	0.761566
20	-348.5	-19.5007	-350.783	-1.89111	876.1913	0.114564	126.7989	2392.598	0.939551
68.4	-61.5	44.2836	-100.037	-0.92956	177.4924	0.024948	18.86089	1.591916	200
68.4	-61	44.52184	-99.6597	-0.92799	176.9285	0.024939	18.85557	1.581677	200
68.4	-60.5	44.76163	-99.2825	-0.92641	176.3681	0.02493	18.85048	1.571604	200
68.4	-60	45.00295	-98.9054	-0.92484	175.811	0.024922	18.84559	1.561695	200
117	-273	17.336	-286.128	-1.65107	891.2267	0.102179	89.46934	2.088514	200
117	-272.5	17.54542	-285.647	-1.64935	889.9672	0.101931	89.16401	2.090462	200
117	-272	17.75444	-285.165	-1.64763	888.7055	0.101684	88.85976	2.092451	200
117	-271.5	17.96306	-284.684	-1.64591	887.4416	0.101436	88.55661	2.094481	200
170	148	237.0244	59.55481	-0.5493	192.2095	0.041893	27.89181	0.842273	200
170	148.5	237.4197	59.94972	-0.54832	191.9813	0.041914	27.90081	0.841831	200
170	149	237.8152	60.34466	-0.54734	191.7538	0.041934	27.90984	0.841391	200

Appendix G - REFRIGERANT FLAMMABILITY

For classification, a single-compound or a refrigerant blend is tested in air at 60 °C and 101.3 kPa. In the case of a refrigerant blend, the worst case of formulation for flammability (WCF) and the worst case of fractionation for flammability (WCFF) is assumed and tested. The classifications, as a direct excerpt from the ASHRAE 34-2010 standard, are as follows:

1. Class 1

   1.1. A single-compound refrigerant has no flame propagation when tested;
   1.2. The WCF of a refrigerant blend has no flame propagation when tested;
   1.3. The WCFF of a refrigerant blend shows no flame propagation when tested.

2. Class 2

   2.1. A single-compound refrigerant must meet all three the following conditions:
       2.1.1. It exhibits flame propagation;
       2.1.2. It has a lower flammability limit (LFL) greater than $0.10\ \mathrm{kg/m^3}$;
       2.1.3. It has a heat of combustion less than 19 000 $\mathrm{kJ/kg}$.

   2.2. The WCF of a refrigerant blend must meet all three the following conditions:
       2.2.1. It exhibits flame propagation;
       2.2.2. It has a LFL greater than 0.10 $\mathrm{kg/m^3}$;
       2.2.3. It has a heat of combustion less than 19 000 $\mathrm{kJ/kg}$.

   2.3. The WCFF of a refrigerant blend must meet all three the following conditions:
       2.3.1. It exhibits flame propagation;
       2.3.2. It has a LFL greater than 0.10 $\mathrm{kg/m^3}$;
       2.3.3. It has a heat of combustion less than 19 000 $\mathrm{kJ/kg}$.

3. Subclass 2L
   The refrigerant must meet the additional condition of having a maximum burning velocity of less than or equal to 10 $\mathrm{cm/s}$ when tested at 23.0 °C and 101.3 kPa.

4. Class 3

   4.1. A single-compound refrigerant must meet both of the following conditions:
       4.1.1. It exhibits flame propagation;
       4.1.2. It has a LFL less than or equal to 0.10 $\mathrm{kg/m^3}$ or it has a heat of combustion greater than or equal to 19 000 $\mathrm{kJ/kg}$.

   4.2. The WCF of a refrigerant blend must meet both of the following conditions:
       4.2.1. It exhibits flame propagation;
       4.2.2. It has a LFL less than or equal to 0.10 $\mathrm{kg/m^3}$ or it has a heat of combustion greater than or equal to 19 000 $\mathrm{kJ/kg}$.

4.3. The WCFF of a refrigerant blend must meet both of the following conditions:

4.3.1. It exhibits flame propagation;

4.3.2. It has a LFL less than or equal to 0.10 $\mathrm{kg/m^3}$ or it has a heat of combustion greater than or equal to 19 000 $\mathrm{kJ/kg}$.

For single-compound refrigerants and refrigerant blends with no LFL at 23.0 °C and 101.3 kPa, an elevated temperature flame limit (ETFL) at 60 °C shall be used in lieu of the LFL to determine the flammability classification of the refrigerant.

Appendix Table G.1 Safety Group Classification

Safety Group

Higher Flammability	A3	B3
Lower Flammability	A2	B2
	A2L	B2L
No Flame Propagation	A1	B1

Increase Flammability ↑

Lower Toxicity   Higher Toxicity

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Appendix H - EXPERIMENTAL SYSTEM TEST RESULTS

The six test results for the experimental system is listed in Appendix Table H.1 to Appendix Table H.6.

Six tests, as mentioned in Chapter 6.4 Experimental Refrigeration System, were performed with the following configurations:

1. Test 1
    1.1. ICMTS, CCM, AKVH;
    1.2. Transcritical region;
    1.3. Counter flow heat exchangers;
    1.4. Internal heat exchanger included; and
    1.5. Heating and cooling water supplied by two sources.

2. Test 2
    2.1. Capillary tube #1;
    2.2. Subcritical region;
    2.3. Counter flow heat exchangers;
    2.4. Internal heat exchanger included; and
    2.5. Heating and cooling water supplied by two sources.

3. Test 3
    3.1. Capillary tube #1;
    3.2. Subcritical flow;
    3.3. Counter flow heat exchangers;
    3.4. Internal heat exchanger excluded; and
    3.5. Heating and cooling water supplied by the same source.

4. Test 4
    4.1. Capillary tube #2;
    4.2. Subcritical flow;
    4.3. Counter flow heat exchangers;
    4.4. Internal heat exchanger excluded; and
    4.5. Heating and cooling water supplied by the same source.

5. Test 5
    5.1. ICMTS, CCM, capillary tube #1;
    5.2. Transcritical region;
    5.3. Counter flow heat exchangers;
    5.4. Internal heat exchanger excluded; and
    5.5. Heating and cooling water supplied by the same source.

6. Test 6
    6.1. ICMTS, CCM, AKVH;
    6.2. Transcritical region;
    6.3. Counter flow heat exchangers;
    6.4. Internal heat exchanger excluded; and
    6.5. Heating and cooling water supplied by the same source.

Test 3 and 4 were repeated to confirm repeatability of the tests.

Appendix Table H.1 Test 1 Measured Results

MEASURED VALUES – TEST 1					
Parameter	Value	Unit	Parameter	Value	Unit
$T_{GC,r,i}$	99.1	°C	$T_{GC,r,o}$	25.8	°C
$T_{GC,w,i}$	15	°C	$T_{GC,w,o}$	16.3	°C
$T_{Ev,i}$	-11.3	°C	$T_{Ev,o}$	10.8	°C
$T_{Ev,w,i}$	14.9	°C	$T_{Ev,w,o}$	15	°C
$T_{IHX,h,i}$	25.8	°C	$T_{IHX,h,o}$	25.9	°C
$T_{IHX,c,i}$	10.8	°C	$T_{IHX,c,o}$	15.4	°C
$T_{CCM,o}$	-	°C	$T_{CT,i}$	25.6	°C
T_{suc}	16.2	°C	$T_{ambient}$	22	°C
$\dot{m}_{ref}$	1.85	kg/min	P_{high}	64.4	bar
$\dot{m}_{GC,w}$	43.9	ℓ/min	P_{med}	-	bar
$\dot{m}_{Ev,w}$	10.2	ℓ/min	P_{low}	22.5	bar

Appendix Table H.2 Test 2 Measured Results

MEASURED VALUES – TEST 2					
Parameter	Value	Unit	Parameter	Value	Unit
$T_{GC,r,i}$	101.4	°C	$T_{GC,r,o}$	25.8	°C
$T_{GC,w,i}$	16.5	°C	$T_{GC,w,o}$	18	°C
$T_{Ev,i}$	-13.2	°C	$T_{Ev,o}$	13	°C
$T_{Ev,w,i}$	16.5	°C	$T_{Ev,w,o}$	14.3	°C
$T_{IHX,h,i}$	-	°C	$T_{IHX,h,o}$	-	°C
$T_{IHX,c,i}$	-	°C	$T_{IHX,c,o}$	-	°C
$T_{CCM,o}$	-	°C	$T_{CT,i}$	25.3	°C
T_{suc}	14.2	°C	$T_{ambient}$	22.5	°C
$\dot{m}_{ref}$	1.79	kg/min	P_{high}	63.6	bar
$\dot{m}_{GC,w}$	43.9	ℓ/min	P_{med}	-	bar
$\dot{m}_{Ev,w}$	10.2	ℓ/min	P_{low}	21.8	bar

Appendix Table H.3 Test 3 Measured Results

MEASURED VALUES – TEST 3						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,i}$	105.8	°C		$T_{GC,r,o}$	26.4	°C
$T_{GC,w,i}$	17.5	°C		$T_{GC,w,o}$	18.6	°C
$T_{Ev,i}$	-15.7	°C		$T_{Ev,o}$	14.3	°C
$T_{Ev,w,i}$	17.5	°C		$T_{Ev,w,o}$	14.8	°C
$T_{IHX,h,i}$	-	°C		$T_{IHX,h,o}$	-	°C
$T_{IHX,c,i}$	-	°C		$T_{IHX,c,o}$	-	°C
$T_{CCM,o}$	-	°C		$T_{CT,i}$	25.7	°C
T_{suc}	15.5	°C		$T_{ambient}$	22.3	°C
$\dot{m}_{ref}$	1.6	kg/min		P_{high}	64.7	bar
$\dot{m}_{GC,w}$	43.9	ℓ/min		P_{med}	-	bar
$\dot{m}_{Ev,w}$	10.2	ℓ/min		P_{low}	20.3	bar

Appendix Table H.4 Test 4 Measured Results

MEASURED VALUES – TEST 4						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,i}$	101.8	°C		$T_{GC,r,o}$	25.6	°C
$T_{GC,w,i}$	18.6	°C		$T_{GC,w,o}$	19.6	°C
$T_{Ev,i}$	-13.7	°C		$T_{Ev,o}$	15.2	°C
$T_{Ev,w,i}$	18.6	°C		$T_{Ev,w,o}$	16.6	°C
$T_{IHX,h,i}$	-	°C		$T_{IHX,h,o}$	-	°C
$T_{IHX,c,i}$	-	°C		$T_{IHX,c,o}$	-	°C
$T_{CCM,o}$	-	°C		$T_{CT,i}$	25.3	°C
T_{suc}	15.6	°C		$T_{ambient}$	20.2	°C
$\dot{m}_{ref}$	1.72	kg/min		P_{high}	62.7	bar
$\dot{m}_{GC,w}$	44.1	ℓ/min		P_{med}	-	bar
$\dot{m}_{Ev,w}$	10.4	ℓ/min		P_{low}	21.3	bar

Appendix Table H.5 Test 5 Measured Results

MEASURED VALUES – TEST 5						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,i}$	106.6	°C		$T_{GC,r,o}$	26.3	°C
$T_{GC,w,i}$	19	°C		$T_{GC,w,o}$	20.3	°C
$T_{Ev,i}$	-15.3	°C		$T_{Ev,o}$	15.7	°C
$T_{Ev,w,i}$	19	°C		$T_{Ev,w,o}$	16.9	°C
$T_{IHX,h,i}$	-	°C		$T_{IHX,h,o}$	-	°C
$T_{IHX,c,i}$	-	°C		$T_{IHX,c,o}$	-	°C
$T_{CCM,o}$	-	°C		$T_{CT,i}$	26.1	°C
T_{suc}	16.3	°C		$T_{ambient}$	20.5	°C
$\dot{m}_{ref}$	1.5	kg/min		P_{high}	64.2	bar
$\dot{m}_{GC,w}$	44.1	ℓ/min		P_{med}	-	bar
$\dot{m}_{Ev,w}$	10.2	ℓ/min		P_{low}	20	bar

Appendix Table H.6 Test 6 Measured Results

MEASURED VALUES – TEST 6						
Parameter	Value	Unit		Parameter	Value	Unit
$T_{GC,r,i}$	95.6	°C		$T_{GC,r,o}$	49.9	°C
$T_{GC,w,i}$	42.5	°C		$T_{GC,w,o}$	44.4	°C
$T_{Ev,i}$	-13.2	°C		$T_{Ev,o}$	11	°C
$T_{Ev,w,i}$	9.8	°C		$T_{Ev,w,o}$	8.9	°C
$T_{IHX,h,i}$	49.9	°C		$T_{IHX,h,o}$	48.3	°C
$T_{IHX,c,i}$	11	°C		$T_{IHX,c,o}$	28	°C
$T_{CCM,o}$	-17.5	°C		$T_{AKVH,i}$	17.2	°C
T_{suc}	1.1	°C		$T_{ambient}$	23	°C
$\dot{m}_{ref}$	1.5	kg/min		P_{high}	103	bar
$\dot{m}_{GC,w}$	23.5	ℓ/min		P_{med}	53	bar
$\dot{m}_{Ev,w}$	18.6	ℓ/min		P_{low}	19.5	bar

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CHAPTER 9 ADDENDUM

Documents included in the addendum are:

1. Controller programming

2. Welding pressure testing report

3. Large format of the receiver design drawing

All items are available on request and are not for examination purposes.