



Thermal-fluid modelling of the transient behaviour of a capacity-controlled underground refrigeration unit

DJ Botha

 **[orcid.org/ 0000-0002-9851-774X](https://orcid.org/0000-0002-9851-774X)**

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Supervisor: Prof M van Eldik
Co-supervisor: Dr PvZ Venter

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ABSTRACT

South African mining operations are continuously expanding deeper underground to reach the available ore bodies, resulting in a higher demand for cooling. The traditional cooling approach by using a vast chilled water network with a surface fridge plant, becomes less effective for deep mining operations, especially in remote underground locations. Spot cooling in remote areas has become a necessity to comply with mining regulations. The modular and mobile Air Cooling Unit (ACU) (patent 2007/04679) was specifically developed to achieve more effective spot cooling. The ACU extracts energy from the air and reject it into service water via a vapour compression cycle. The ACU is dependent on a constant water flowrate to stay operational. Unfortunately the water supply fluctuates and results in the ACU tripping on over pressure as a safety mechanism. To improve operations and reduce down time, infinite capacity control of the compressor has been implemented to cope with the fluctuating water flow.

The effect of the infinite capacity control strategy on the thermal fluid dynamics of the refrigeration cycle under varying environmental conditions is not well known. This study investigated the refrigeration cycle's reaction when subjected to a fluctuating feedwater or air supply.

A detailed thermal fluid model of the ACU was developed using Flownex[®]. Each of the main cycle components were first individually modelled, and their accuracy verified using third-party software. These components were then integrated in a cycle model, the accuracy of which was validated against both surface and underground data. The model was then used to simulate transient scenarios of feedwater and air availability. The effect on the refrigeration cycle and its response was observed.

The study identified the operational limits for feedwater and air supply, accurately captured the response of compressor infinite capacity control to fluctuating feedwater supply and determined the obtainable cooling duty for limited feedwater availability. The operational limits that led to ACU downtime were restricted feedwater supply, less than 1 kg/s at 21°C, and total fan failure. In response to fluctuating feedwater supply, the compressor actively decreased capacity for reduced feedwater supply and increased capacity when the feedwater supply was restored. Obtainable cooling duty did not dramatically improve when feedwater was oversupplied. Undersupplying feedwater below 4 kg/s at 21°C resulted in a dramatic decrease in obtainable cooling due to compressor capacity decrease. The study concluded that the compressor infinite capacity control strategy is advantageous to maintain continuous ACU operation in the mining environment.

Keywords: Refrigeration; Air Cooling Unit; Thermal-fluid modelling

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NOMENCLATURE

This nomenclature describes abbreviations, symbols, subscripts, units, metric prefixes and frequently used terms used in this dissertation.

Abbreviations used in this dissertation are listed in Table 1 below:

Table 1: Abbreviations Used in this Dissertation

Abbreviation	Description
ACU	Air Cooling Unit
Avg	Average
BEP	Best Efficiency Point
B.I.C.	Bushveld Igneous Complex
COP	Coefficient of Performance
CFD	Computational Fluid Dynamics
TC	Condensing Temperature
EES	Engineering Equation Solver
Eq.	Equation
TO	Evaporating Temperature
FAT	Factory Acceptance Test
E.g.	For Instance
Max	Maximum
Min	Minimum
SC	Subcool
SH	Superheat
Surf	Surface
TEV	Thermostatic Expansion Valve
TP	Two-phase
Und	Underground
VCS	Vapour Compression System
VRT	Virgin Rock Temperature

Table 2 lists the variables used in the equations of this dissertation:

Table 2: Variables Used in this Dissertation

Variable	Unit	Description
A	[m ²]	Area
j	[-]	Colburn J-factor
D or d	[m]	Diameter
s	[m]	Distance
h	[J/kg]	Enthalpy
x	[-]	Fluid quality
f	[-]	Friction factor
$\dot{Q}$	[W]	Heat transfer rate
K	[-]	K-value for secondary losses
Pt	[m]	Lateral tube pitch
I	[A]	Line current
V	[V]	Line-to-line voltage
h_{liq}	[W-m ² -K]	Liquid convective heat transfer coefficient
Pl	[m]	Longitudinal tube pitch
$\dot{m}$	[kg/s]	Mass flowrate

Variable	Unit	Description
P	[W]	Power
Pr	[-]	Prandtl number
p	[Pa] or [bar]	Pressure
$\dot{W}$	[W]	Rate of work performed
Re	[-]	Reynolds number
N	[rev/s]	Rotational speed of the compressor
C_p	[J/kg-K]	Specific heat capacity at constant pressure
T	[K]	Temperature
t	[s]	Time
h_{tp}	[W-m ² -K]	Two-phase convective heat transfer coefficient
v	[m/s]	Velocity
$\forall$	[m ³]	Volume

The Greek symbols used as variables in the equations of this dissertation are listed separately in Table 3 below:

Table 3: Greek Symbols Used in this Dissertation

Variable	Unit	Symbol Name	Description
ρ	[kg/m ³]	Rho	Density
θ	[°]	Theta	Phase angle

Table 4 lists the subscripts used in the equations of this dissertation:

Table 4: Subscripts Used in this Dissertation

Subscript	Description
e	Exit
n, m	Final increment
i, j	Increment number
i	Inlet
in	Inlet flow to increment
o	Outside
0	Stagnation

Units used in this dissertation are listed in Table 5 below:

Table 5: Units Used in this Dissertation

Unit	Description
[A]	Ampere
[bar]	Bar
[°]	Degree
[°C]	Degree Celsius
[g]	Gram
[K]	Kelvin
[L]	Litre
[m]	Meter
[Pa]	Pascal
[%]	Percentage as a unit of proportion
[rev]	Revolutions
[s]	Second
[t]	Tonne

<i>Unit</i>	<i>Description</i>
[V]	Volt
[W]	Watt

Metric prefixes used in this dissertation are listed in Table 6 below:

Table 6: Metric Prefixes Used in this Dissertation

<i>Unit</i>	<i>Magnitude</i>	<i>Description</i>
[k]	$\times 10^3$	kilo
[m]	$\times 10^{-3}$	milli

The terms frequently used in this dissertation are listed Table 7 below:

Table 7: Frequently Used Terms in this Dissertation

<i>Term</i>	<i>Description</i>
Dry bulb temperature	The temperature of air measured by a thermometer freely exposed to the air but shielded from radiation and moisture. [Noun Phrase]
Flow network	A network constructed using Flownex elements to create a flow circuit with thermodynamic and fluid mechanic properties. [Noun Phrase]
Flownex element	Building blocks in the Flownex domain that are used to create flow networks. [Noun Phrase]
Spot cooling	Confining cooling to a particular spot in an underground mine as opposed to cooling the entire network. [Verb Phrase]
Thermal gradient	The ratio of the temperature difference and the distance between two points. [Noun Phrase]
Wet bulb temperature	The temperature measured by a thermometer covered in a water-soaked cloth over which air is passed. [Noun Phrase]

CHAPTER 1: INTRODUCTION

1.1 Background

The writer Tim Winton once wrote: "You never really think much about breathing. Until it's all you ever think about" (May, 2009). Air conditioning and ventilation are as essential to mining as breathing is to life.

The abundance of mineral resources below the surface of its vast plains, has offered South Africa the opportunity to extract many of its resources from the earth's surface using open pit mining techniques. The majority of its deposits, however, are located deeper within the earth's crust, necessitating the use of underground mining techniques such as room and pillar, narrow vein stoping and large-scale mechanised mining. Mineral resource extraction has progressed to such an extent that the country currently holds six of the ten deepest mines in the world. AngloGold Ashanti's Mponeng gold mine located South-West of Johannesburg is the deepest gold mine in the world descending to depths exceeding 4 km (Njini, 2021).

Mining at greater depths presents air conditioning and ventilation challenges. As one descends deeper into the earth's crust, the geothermal gradient increases due to Virgin Rock Temperature (VRT) rising, auto compression of air at greater depths, mining operations and poor heat conductivity of the ore mined (Le Roux, 1970).

The 1995 Leon Report released by the Commission of Inquiry into Safety and Health in the Mining Industry highlighted the lack of emphasis in the South African mining industry on the importance of occupational health and safety (Stanton, 1995). Following this report, the Department of Mineral Resources and Energy enforced regulations under the mine health and safety act that prohibits mining operations exceeding wet bulb temperatures of 32.5°C and dry bulb temperatures of 37°C (LexisNexis, 2017). Thermal stress causes inefficient and unproductive workers and may even result in death (Le Roux, 1970).

Air cooling is essential in South African mines at depths exceeding 1500 m. Typically, underground mining operations rely on surface fridge plant infrastructure to maintain cool temperatures. These cooling systems have proven ineffective for deep mining operations in remote underground locations. Over great distances a significant amount of heat is added to descending cooling pipes, diminishing the cooling effect. This presented a need for a more effective cooling system for deep mining operations (Van Eldik, 2006).

This need inspired the development of the 100 kW Mk1 ACU. This tried and tested innovation to mine air-conditioning has proven effective in addressing heat challenges in remote underground

locations. The success of the Mk1 ACU inspired the development of the larger 250 kW Mk2 ACU, the focus subject of this study. The distinguishing feature of this cooling system is the spot cooling technique. The process involves extracting energy from the air and transferring it into water via the vapour compression cycle. Subsequently this water, now carrying the heat load, is efficiently pumped out of the mine alongside the rest of the underground water. It is an effective way to move the heat load from where the underground air temperature is too high, to an area where this heat can be managed, and cooled (Van Eldik, 2006).

The unit's refrigeration cycle contains a semi-hermetic compact screw compressor, a tube-in-tube, refrigerant-to-water, counterflow condenser with a stainless-steel inner pipe and a carbon steel annulus, straight charged, externally equalized, Thermostatic Expansion Valves (TEVs) and dual, epoxy-coated, wavy aluminium-finned, rifled bore copper-coiled, air-to-refrigerant crossflow evaporators. A standard 22 kW mine fan draws air in through the fins of the evaporators, expelling cooled air towards the workplace. The unit has a nominal cooling capacity of 290 kW (at compressor design conditions), achievable at a wet bulb air temperature of 31.5°C, a refrigerant condensing temperature of 50°C, and an evaporating temperature of 10°C. Under these conditions, the water outlet temperature is maintained at 40°C, with a flowrate of 3.6 l/s at an inlet temperature of 15°C, and 6 l/s at an inlet temperature of 25°C. The unit's nominal electrical power input is 75.5 kW, resulting in a cooling Coefficient of Performance (COP) of 3.8 (M-Tech Industrial, 2022).

In the underground environment the unit needs to endure harsh operating conditions while delivering the cooling required. When operating conditions move outside tolerable limits, the ACU is equipped with safeguards to protect it from component damage. These safeguards include sensory switches that break the control circuit, shutting down the compressor and the fan. The compressor has built-in overheating, over current, locked rotor and single phasing protection. In addition to this, a low - and high pressure trip switch monitors the inlet and outlet refrigerant pressure of the compressor.

There are several factors that trigger ACU trip switches, of which the most prominent include fan failure, evaporator airside blockage, refrigerant leakage, electrical supply that moves above or below threshold percentages and inadequate feedwater supply. The only factor mentioned with no means of simple prevention is fluctuating feedwater supply, since water is a scarce commodity underground and a constant sufficient supply cannot be guaranteed. When there is an inadequate supply of feedwater, the feedwater's capacity to effectively remove heat from the refrigerant in the condenser is reduced. As a result, the refrigerant pressure and temperature increase, aiming to establish a temperature difference between the water and the refrigerant gas. This temperature difference is necessary to reject the energy stored in the refrigerant. The high temperature and pressure refrigerant continues through the cycle, entering the compressor. This gas is difficult to

compress, resulting in a higher outlet pressure in the compressor. If the compressor discharge pressure rises above a critical setpoint, the high-pressure switch breaks the control circuit (Van Eldik, 2006).

Mining regulations dictate that heavy machinery like the ACU may only be reset by qualified personnel. Since the unit typically operates at great depths, reset may take significant time. Mining operations cannot afford operation disruptions like ACU downtime. This necessitates continuous, effective operation of the ACU regardless of the state of the water supply (Van Eldik, 2006).

A step capacity approach of 25/50/75/100% was implemented to achieve better continuous operation of the ACU under fluctuating feedwater supply. Limiting compressor capacity when less feedwater was available, reduced the refrigerant mass flowrate in the cycle. Essentially, the obtainable cooling duty is scaled down based on the cycle's ability to reject heat into the water. Given that the compressor discharge pressure decreases for less refrigerant circulated, the trip switch is never triggered, and continuous operation is maintained. Implementing infinite capacity control provided more precise control by allowing the compressor capacity to settle within the entire 25 to 100% range rather than being limited to four points under step capacity control. This refined control improved obtainable cooling duty since compressor capacity settled higher than it could under step control when feedwater supply reductions occurred.

A better understanding of the unit response under infinite capacity control would aid in identifying operational limits for feedwater and air supply, as well as determining the obtainable cooling duty within these constraints. Knowing at what conditions trips occur would help prevent ACU downtime in the future.

1.2 Problem Statement

The impact of the infinite capacity control strategy on the thermal fluid dynamics of the ACU refrigeration cycle under varying environmental conditions was not well known.

1.3 Purpose of the Study

This study investigated the refrigeration cycle's reaction when subjected to fluctuating feedwater or air supply. Studying the thermal fluid effects of these environmental conditions helped identify operational limits and obtainable cooling duty within these constraints. The investigation is conducted through thermal fluid modelling of the complete cycle.

1.4 Goals

The goals of this study include:

- Develop a thermal fluid model of each of the relevant ACU components using Flownex simulation environment one-dimensional thermal modelling software. Flownex offers advanced transient simulation capabilities, crucial to the completion of this study.
- Verifying the accuracy of component models consisting of a compressor, a condenser, evaporators and a fan.
- Validating the accuracy of the integrated model consisting of the developed component models.
- Investigating the transient behaviour of the integrated model under the influence of fluctuating feedwater and air supply. Specific points of interest include:
 - Operational limits for feedwater and air supply.
 - The compressor's response to fluctuating feedwater supply under infinite capacity control.
 - The obtainable cooling duty for limited feedwater availability.

1.5 Methodology

The method of completing this study include:

- Literature Study
 - Investigate the sources of heat in mines.
 - Investigate various deep level mine refrigeration systems.
 - Determine the current state of research on modelling the vapour compression cycle, doing a systems level as well as component-specific investigation.
- Model development
 - Creating component models of the main ACU components using Flownex software. These components include the compressor, condenser, expansion valves, evaporators, fan and interconnecting piping.
- Verification
 - The compressor model's accuracy is verified using Bitzer® software. Bitzer generates compressor performance information using performance chart data.
 - The condenser model's accuracy is verified for a single tube using EES software.
 - The evaporator model's accuracy is verified using coil designer software developed by the ACU evaporator manufacturer, HC Heat-Exchangers®.
 - The fan model is verified using fan manufacturer data.
- Validation
 - The integrated model's accuracy is validated for surface and underground conditions comparing simulation results to real-world performance data.

- Transient investigation
 - Modelling various scenarios to investigate the effect of fluctuating feedwater and air mass flowrate on the unit.

NOTE: The ACU is a patented design (patent number 2007/04679).
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CHAPTER 2: LITERATURE STUDY

This chapter reviews literature pertinent to the ACU being investigated in this study. The source of heat in mines are identified with emphasis on the heat situation in South-African mines since these conditions inspired the development of the ACU. Like various other mine cooling systems, the ACU removes heat using a vapour compression cycle. The workings of the vapour compression cycle are broken down into functional groups and rationalised. Since supply side water is indispensable to the ACU cooling system, the use of water as a cooling medium in other mine cooling systems is reviewed. Furthermore, this chapter reviews various alternatives to model the ACU and its components mathematically. Points of focus include:

- Sources of heat in mines investigating the geological background and geothermal gradients of Witwatersrand Basin and the Bushveld Complex.
- The workings of the vapour compression cycle.
- The use of water as a cooling medium.
- Mathematical modelling of ACU components.

2.1 Sources of Heat in Mines

South Africa is endowed with significant mineral riches, therefore, mining is one of the country's primary economic foundations (Jones, 2018). The nation was the world's leading gold producer until 2007, but it dropped to the seventh position by 2017 (Ober, 2018). Rising production costs, slow economic development and labour concerns have hindered South African mines' productivity and profitability (Neingo & Tholana, 2016). The average grade of South African gold mining ore has plummeted from 12 g/t in 1970 to 5 g/t in 2014. Since then, most high-grade gold deposits have been depleted and mines are now focusing on lower-grade, deep-lying orebodies (Musingwini, 2014).

The Witwatersrand Basin and the Bushveld Complex are two of the most mined regions in South Africa. Figure 1 below displays the Witwatersrand Basin and the Bushveld Complex in South-Africa along with the location of the town in each with the most prominent mining activity highlighted in red.

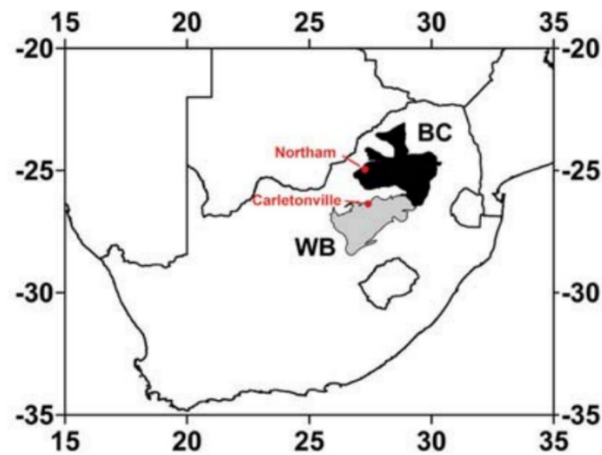


Figure 1: Depiction of the Witwatersrand Basin and the Bushveld Complex with prominent mining towns in each (Jones, 2018)

The Witwatersrand Basin is a geological formation which surfaces in the Witwatersrand region of South-Africa. It contains the world's largest known gold deposits and has produced about 40,000 t, which accounts for around 22% of all surface gold, globally (Norman & Whitfield, 2013). The Bushveld Complex hosts the world's largest reserves of platinum (and associated platinum group elements), chromium and vanadium (Cawthorn, 2007).

Mining can be described as violent destructive work, excavating rock faces through blasting, hammering and drilling, generating a great deal of friction (Le Roux, 1970). The law of conservation of energy stipulates that energy cannot be created or destroyed but can only change from one form to another (Reisel, 2021). The energy that is not absorbed by the rock, is released primarily as heat and sound. There are various other sources heating the air of which the most prominent include rock walls, water, active machinery, lighting, people, explosives, electric cables, oxidation of minerals and rock movement (Le Roux, 1970).

VRT is a major source of heat in deep mining systems, while auto-compression adds an extra 10°C/km rise in dry-bulb temperature relative to the surface (Torres & Singh, 2011). The earth is made up of several layers of which the lithosphere is the surface layer and the outermost of the upper mantle (Johnson *et al.*, 2006). The lithosphere is on average 100 km thick and the layer in which mining takes place. Beneath the lithosphere, the barysphere is the hot innermost layer of the earth consisting of semi-molten rock. The lithosphere's bounding entities have extreme temperature differences that causes dramatic fluctuation in the geothermal gradient of the lithosphere (Jones, 1988).

Over the last five decades, technological advances have enabled mining at ever-deeper levels. Mine refrigeration and ventilation are key concerns when considering deep level mining since it is required to manage and regulate air temperature in the working environment. As mining depth

increases, so does VRT, resulting in higher heat loads on underground operations (Jones, 2003). This is especially prominent in the Witwatersrand gold fields, which contain some of the world's deepest mines, with mining reaching 4 km of depth, and the Bushveld Complex, where platinum mines exceed 2 km of depth. Chromium and vanadium mines are still opencast, but underground mining can occur in the future. Since the geothermal gradient of a mine is dependent on the heat conduction capabilities of the rock (Le Roux, 1970), proper planning demands knowledge of both the VRT and the thermal characteristics of the rocks surrounding the underground operation (Jones, 2018).

2.1.1 Geological Background and Rock Characteristics for the Witwatersrand Basin and the Bushveld Complex

Figure 2 and Figure 3 below contain geological maps of the Witwatersrand Basin and the Bushveld Complex provided to aid in the discussion of the geological background of these areas.

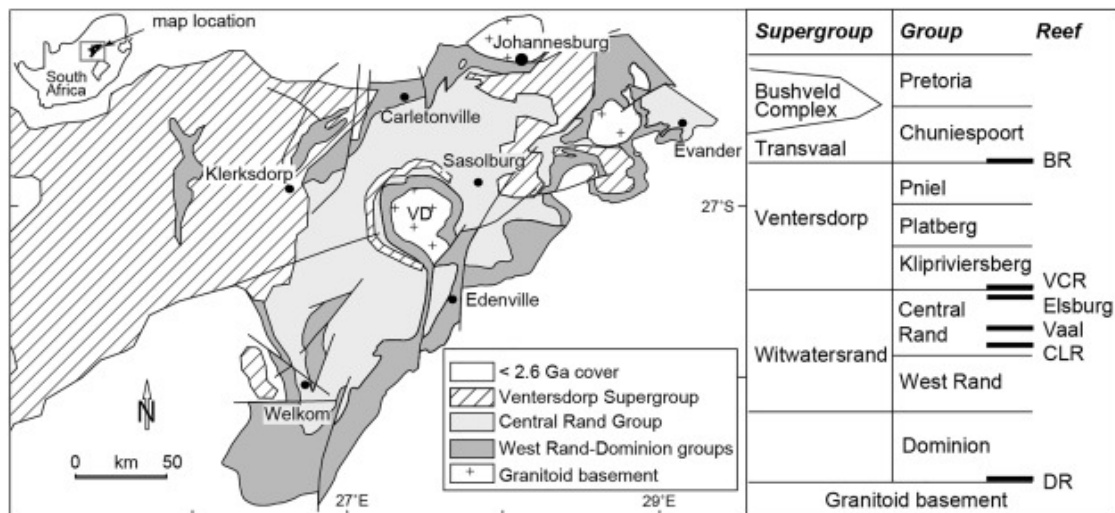


Figure 2: Schematic geological map of the Witwatersrand Basin (Da Costa *et al.*, 2020)

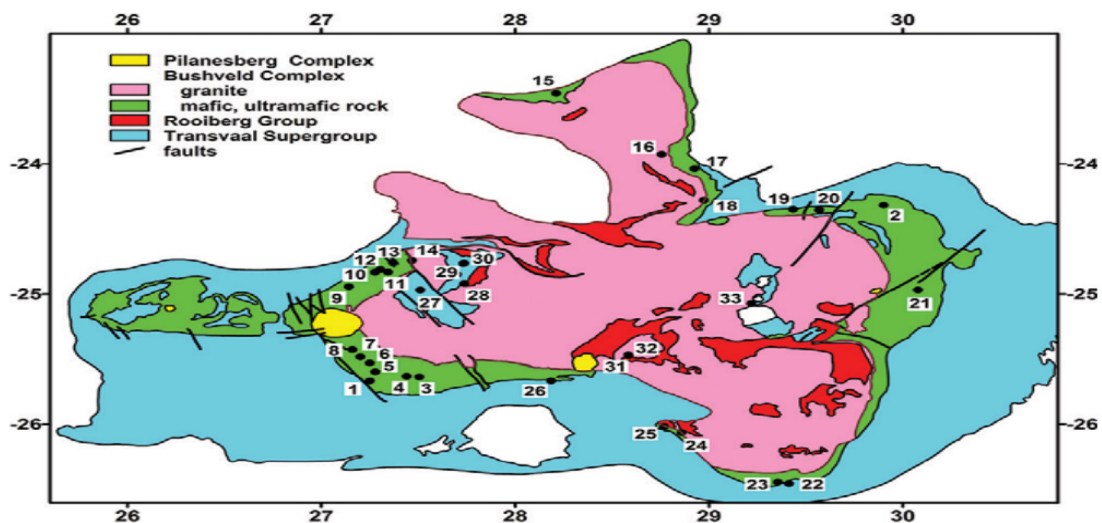


Figure 3: Schematic geological map of the Bushveld Complex (Jones, 2018)

The Bushveld Complex (Figure 3) is located in the Kaapvaal Craton geological landscape. Its Archaean cratonic core, represented by basement granitic rocks and greenstone belts, formed around 3700 to 2650 Mega-annum and had mostly stabilised by that time (Eglington & Armstrong, 2004). During this period, relatively undeformed stratified basins subsequently formed on the craton. The oldest, the Witwatersrand Basin (Figure 2), hosts the largely volcanic dominion group (indicated in dark grey on Figure 2) that formed around 3100 Mega-annum, overlain by the largely sedimentary Witwatersrand Supergroup (indicated in dark and light grey on Figure 2) around 3000 to 2700 Mega-annum, and followed by the largely volcanic Ventersdorp Supergroup that formed around 2700 Mega-annum (indicated by diagonally striped lines on Figure 2). Subsequent deposition of platform strata of the Transvaal Supergroup that formed around 2650 to 2200 Mega-annum occurred in the Transvaal Basin, which overlies the Witwatersrand Basin and an extensive area of Archaean basement rocks further north (Johnson *et al.*, 2006). Extrusion of the Rooiberg Group (indicated in red on Figure 3) that formed around 2070 Mega-annum and emplacement of plutonic rocks of the Bushveld Complex (Figure 3) into the northern section of the Transvaal Basin (indicated in blue on Figure 3) were the next major geological events to influence this area of the Kaapvaal Craton (Zeh *et al.*, 2015). Sedimentary rocks from the 300-200 Mega-annum old Karoo Supergroup cover parts of the Witwatersrand Basin and the southern Bushveld Complex. Figure 4 below contains a table of essentially the whole 'thermal conductivity stratigraphy' of these geological formations.

Thermal conductivity of major rock types occurring above the platinum mining areas of the Bushveld Complex and gold mining areas of the Witwatersrand Basin

Rock unit/sub-unit	Rock type	$K \pm s.d.$ (W/mK)	Range (W/mK)	N_K
KAROO SUPERGROUP				
Ecce Group	sandstone	3.43 ± 0.68	1.80–4.78	69
	shale	2.02 ± 0.46	0.81–2.96	118
	coal	0.42 ± 0.10	0.24–0.71	49
BUSHVELD COMPLEX				
Bushveld granite	granite	3.43 ± 0.41	2.68–4.00	103
Upper Zone	diorite	2.17 ± 0.07	2.04–2.27	8
	anorthosite	2.09 ± 0.08	2.00–2.22	7
	gabbro, ferrogabbro	2.39 ± 0.22	1.94–3.07	122
	pyroxenite	3.38 ± 0.33	2.40–4.39	29
Main-Critical Zone	anorthosite	1.92 ± 0.12	1.72–2.25	144
	norite	2.28 ± 0.15	1.86–2.94	360
	pyroxenite	3.45 ± 0.35	2.41–4.21	89
	chromitite	2.45 ± 0.18	2.17–2.90	16
Lower Zone	pyroxenite, dunite, harzburgite	4.13 ± 0.21	2.56–4.98	39
TRANSVAAL SUPERGROUP				
Pretoria Group	lava	3.53 ± 0.37	2.90–4.25	21
	quartzite, chert	6.96 ± 0.54	5.77–7.71	29
	siltstone	4.09 ± 0.69	3.07–5.50	19
	shale	2.57 ± 0.34	1.90–3.14	28
Chuniespoort Group	dolomite	5.09 ± 0.63	3.19–7.53	57
VENTERSDORP SUPERGROUP				
	lava	3.41 ± 0.47	2.59–5.06	153
	quartzite, conglomerate	5.20 ± 0.68	3.71–6.59	34
	shale	3.58 ± 0.32	3.01–3.92	5
WITWATERSRAND SUPERGROUP				
Central Rand Group	quartzite, conglomerate	6.38 ± 0.79	3.72–8.21	336
	shale	4.77 ± 1.20	2.76–6.38	10
DOMINION GROUP				
	lava	3.34 ± 0.45	2.28–5.09	251

Figure 4: Thermal conductivity of major rock types in the Bushveld Complex and Witwatersrand Basin (Jones, 2003, 2018)

*In the table: K represents mean thermal conductivity; s.d. represents standard deviation; and N_K is for number of conductivity measurements.

The findings show that several rocks in the Bushveld Complex have conductivities that are one-third to one-half of those in the Witwatersrand Basin and the overlying Transvaal Supergroup. This has significant implications for comprehending the two locations' geothermal gradients.

2.1.2 Geothermal Gradients for the Witwatersrand Basin and the Bushveld Complex

The first rock temperature investigations revealed that the geothermal gradients in the Witwatersrand are rather low (Bullard, 1939). The quartzite of the Witwatersrand Basin is a good conductor of heat and thus the rock temperature does not increase drastically with increased depth in the Rand mines (Le Roux, 1970). Routine VRT measurements taken in mines since 1950 down to maximum rock-breaking depth, where the VRT surpasses 65°C, have corroborated this (Jones, 2003). Early measurements into Bushveld Complex revealed that the geothermal gradients were around twice those observed in the Witwatersrand Basin (Carte & Van Rooyen, 1971). Figure 5 below illustrates the VRT in various mining areas of the Witwatersrand Basin and Bushveld Complex for a depth of up to 3 km.

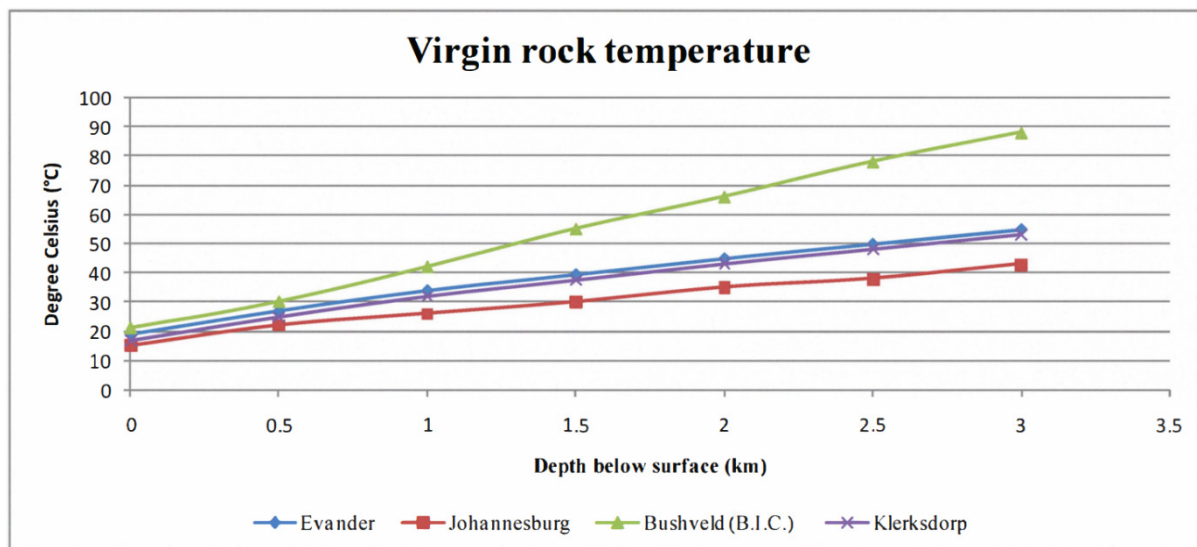


Figure 5: Virgin rock temperature rise for increased mining depth in the Witwatersrand Basin and Bushveld Complex (Van Rensburg et al., 2011)

The graph indicates that the geothermal gradient remains fairly constant across the Witwatersrand Basin indicating a geothermal gradient of about 12°C/km vertical depth. The temperature increase gradient of the Bushveld Igneous Complex (B.I.C.) mines is significantly higher as indicated by the green plot. Figure 6 below illustrates more comprehensive data of the VRT in various mining areas of the Bushveld Complex for a depth of up to 800 m.

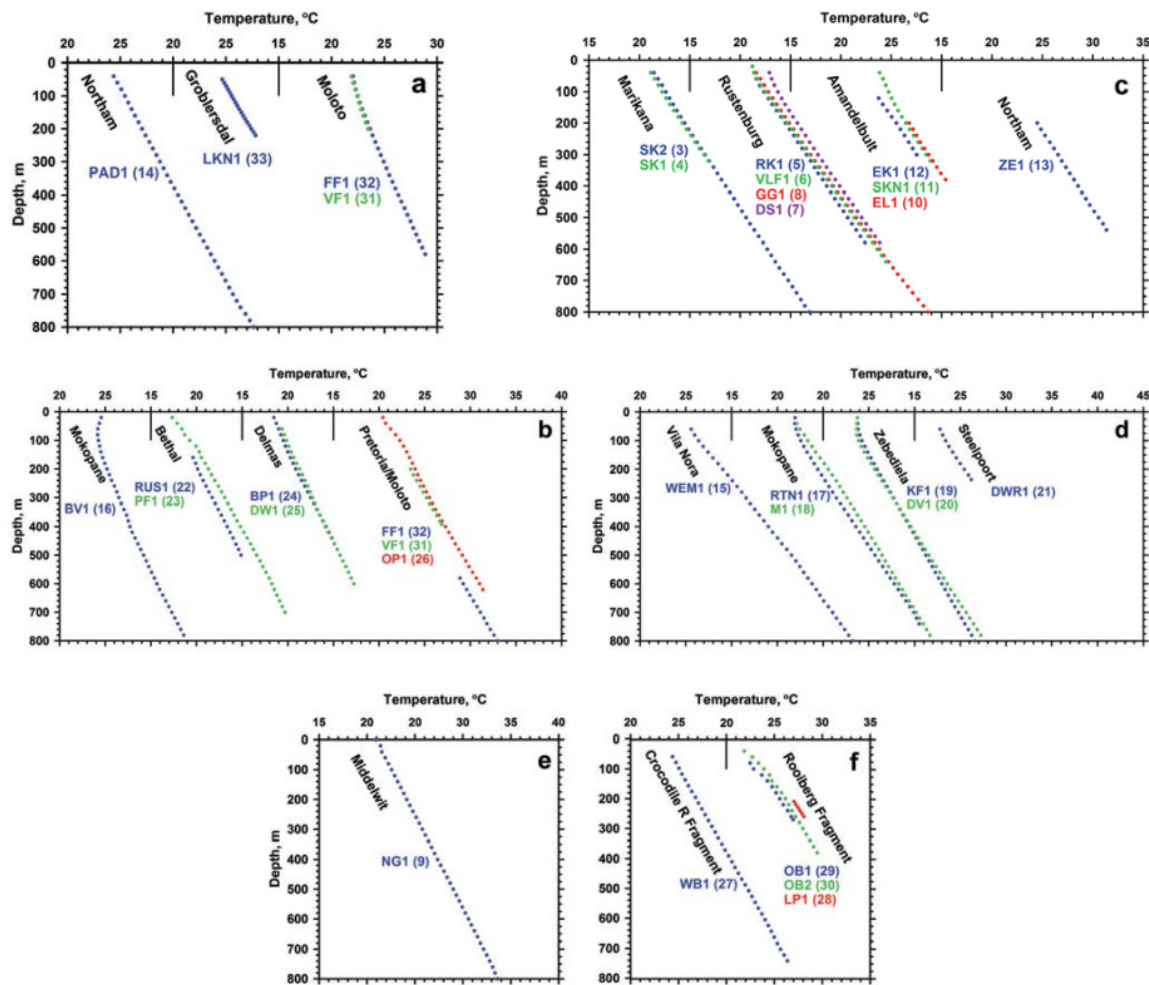


Figure 6: Virgin rock temperature rise for increased mining depth in the Bushveld complex (Jones, 2018)

In Figure 6, charts a and b show temperature profiles in Bushveld granite and the Complex's upper zone, respectively. The data from the Main Zone (Figure 6, charts c and d) are arranged clockwise from Marikana via the Complex's western and northern branches to the Steelpoort area. Figure 6, chart e depicts data from a single borehole in the Bottom Zone. Figure 6, chart f depicts the findings of Transvaal Supergroup and Rooiberg Group lithological units found in geological inliers, or pieces, in the Complex's interior (Figure 3). The geothermal gradients across various parts of the Bushveld Complex are fairly constant, although it is significantly higher than that of the Witwatersrand Basin with a geothermal gradient of about 20.71°C/km.

Geothermal gradients in the Bushveld Complex's platinum mining sites (17.7-24.5°C/km) are almost double that of the Witwatersrand Basin's rocks covering gold reefs (9.6-12.9°C/km). The geothermal heat flow in the Bushveld Complex platinum mining sites (45 mW/m²) is lower than in the Witwatersrand gold fields (51 mW/m²). The fundamental cause for the Complex's large thermal gradients is the comparatively low thermal conductivity of the component rocks in comparison to the Witwatersrand Basin. The VRT in the Northam region, the deepest location explored in the Bushveld Complex, is up to 30°C higher than the VRT in the Carletonville area, where the deepest

gold mining occurs. This is owing to both greater surface temperatures and thermal gradients in the Bushveld (Jones, 2018).

Mining is prohibited by the Mine Health and Safety Act for dry bulb temperatures exceeding 37°C (LexisNexis, 2017). Reviewing Figure 5 as an emissary to the Witwatersrand Basin, refrigeration is required for deep mines in the region since temperatures exceed allowable limits within 1.5 km of depth. The Bushveld Complex, represented by Figure 6, indicates most mines deeper than 800 m are on the brink of, or already have refrigeration needs. The literature suggests the future of South-African mining will depend on its ability to maintain a cool working environment in deep level mining.

2.2 Deep Level Mine Refrigeration Systems

In mining operations, refrigeration systems play a crucial role in both cooling and heat rejection processes. These systems employ different types of heat exchangers to achieve their goals. Indirect contact heat exchanger banks, typically operating in closed-circuit systems, are primarily utilized for cooling purposes. On the other hand, direct contact spray chambers, functioning in open-circuit setups, are versatile, serving both air cooling and heat rejection needs.

Deep level mines rely on three key cooling mechanisms: indirect contact heat exchangers, spray towers, and spray chambers, all contributing to maintaining optimal air temperatures. The deployment of banks of indirect contact heat exchangers is mainly reserved for closed-circuit water systems operating in clean air environments. Cooling within these mining systems is achieved through surface, underground, or mobile refrigeration plants, all of which incorporate the vapour compression cycle in their operations to efficiently manage temperature control (Hooman *et al.*, 2015).

Notably, efficient heat rejection is facilitated through the use of spray towers and chambers, both of which fall under the category of direct contact heat exchangers. Spray towers employ a process in which water is sprayed into the air, establishing direct contact with the warm air and promoting heat transfer via evaporation. Similarly, spray chambers involve the direct interaction between the cooling medium, often water and the warm air stream, achieving effective heat rejection through this immediate exchange. These mechanisms are instrumental in achieving efficient temperature regulation in deep-level mines, thus contributing to a conducive working environment and ensuring the safety and productivity of mining operations (Hooman *et al.*, 2015).

2.2.1 Vapour Compression Cycle

The ACU uses the vapour compression cycle, also known as the direct expansion cycle, to achieve cooling. This is the most widely used refrigeration solution in the world due to its ability to remove significant heat load with little refrigerant mass flowrate and its high energy efficiency, which implies

excellent COP abilities (McPherson, 1993). Figure 7 below illustrates a simplified vapour compression cycle to aid in visualisation of the discussion.

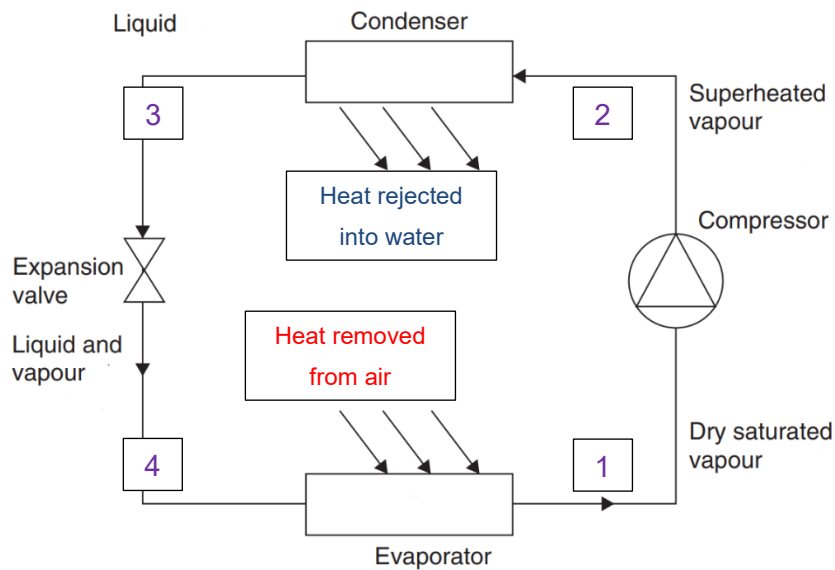


Figure 7: Simple vapour compression cycle (Hundy *et al.*, 2008)

This is a closed loop system that exploits the temperature variations observed when a working fluid (refrigerant) changes phase as it continually circulates through an evaporator, compressor, condenser and expansion valve to simultaneously achieve cooling and heating (Le Roux, 1970).

A good refrigerant should have a unique set of properties to ensure phase change occurs with minimal fluctuation in applied pressure and energy. These properties include high latent heat, good conductivity, low specific volume as a saturated gas and low freezing point (Gabrielii & Vamling, 1997).

Referring to Figure 7, warm refrigerant vapour leaves the evaporator and enters the compressor (Figure 7, region 1) where the gas is compressed from a low to high temperature and pressure (Hundy *et al.*, 2008). The degree of superheat achieved depends on the efficiency of the compressor (Hundy *et al.*, 2016). The warm refrigerant vapour leaves the compressor and flows in at the top of a condenser, (Figure 7, region 2) allowing condensed liquid to collect at the bottom (Trott & Welch, 2000). During phase change, latent heat is transferred to the cooling water. Refrigerant is cooled with water to the point where condensing temperature is met, condensing the refrigerant to a saturated liquid. The removal of more heat sub-cools the refrigerant liquid to a lower temperature (Hundy *et al.*, 2008). The degree of sub-cool is defined as the temperature difference between the outlet of the condenser and the saturated liquid point. The effectiveness of the entire cycle is dependent on the effectiveness of heat transfer taking place between the refrigerant and the cooling water (Hundy *et al.*, 2016).

The saturated refrigerant liquid collects at the bottom of the condenser (Figure 7, region 3) and passes through the expansion valve, where it is throttled to a low pressure and corresponding low temperature in a constant enthalpy process resembling a two-phase mixture when it enters the evaporator (Figure 7, region 4) (Hundy *et al.*, 2008).

Heat is extracted from the air and transferred to the refrigerant in the evaporator. The added heat elevates refrigerant temperature, evaporating the refrigerant to a saturated vapour (Trott & Welch, 2000). The addition of more heat superheats the refrigerant to a higher temperature. The degree of superheat is defined as the temperature difference between the outlet of the evaporator and the saturated vapour point (Hundy *et al.*, 2016).

The cycle repeats in an infinite closed loop for as long as the compressor is running. Heat is exchanged between the refrigerant and air to achieve cooling and between the feedwater and refrigerant to obtain heating. The feedwater is pumped away, removing heat from the control volume.

2.2.2 Incorporating Water to Achieve Cooling

Mine refrigeration is a means of removing the heat load from confined areas in deep-level-mining operations. The ACU achieves this, moving the heat load into the supply side water where it is moved to an area where it can be managed and cooled. Water is an excellent heat transfer medium compared to other readily available alternatives like air due to its unique high specific heat capacity.

2.2.2.1 Chilled Water Reticulation

A typical deep mine water reticulation layout is depicted in Figure 8 below. Observing the figure, chilled water reticulation is the process of supplying cold water to mining levels and returning hot water to the surface for cooling in an open loop cycle.

2.2.2.3 Ice Plant Technology

Surface refrigeration machines produce ice, which is transferred to underground ice dams, from which cold water is pumped throughout the mine. The sensible heat of the fluid remains the same as the fluid changes phase allowing the fluid to have a better cooling capacity when it reaches underground. The fundamental advantage of employing ice slurries is that the amount of return cooling water is decreased by a factor of four, resulting in much lower pumping expenses to the surface (Sheer, Butterworth & Ramsden, 2001).

According to (Greyling, 2008), AngloGold Ashanti's Mponeng mine employs ice technology, providing 70% dry ice slurry to underground workings. Whilst it is feasible to produce dry ice for usage underground, increasing the amount of energy that can be extracted from the air, there are practical issues for mine use. The major difficulty is transporting dry ice. Whilst a system may potentially be developed to move solid ice, it will need extensive re-design of existing mining infrastructure, including the water piping system. When ice slurry is utilised, however, the mining pipe system may stay basically unmodified.

2.2.2.4 Localised Cooling

Mining starts with the recovery of easily accessible resources and gradually progresses to more difficult resources, which indefinitely leads to increasingly large distances from the mine shaft to the working areas (Du Plessis *et al.*, 2013). The result is a decrease in the effectivity of primary and secondary ventilation and cooling, which usually consists of the main ventilation fans and bulk air coolers (Sasmito *et al.*, 2015). Subsequently, mobile cooling units are often used as tertiary or in-stope cooling in warm underground working areas (Du Plessis *et al.*, 2013). A fan forces warm air over the heat exchanger. The air exits the mobile cooling unit at a lower temperature and moves to the work area. Warm water is dumped onto the footwell upon exiting the mobile cooling unit, ultimately accumulating with used mining water at the settlers.

Mobile cooling units are subject to corrosion, inadequate supply water temperature and flow, and extensive ducting lengths. Because of the extreme environment, mobile cooling units require extensive maintenance to avoid becoming inefficient. Inadequate maintenance procedures contribute to increased inefficiency over time. The inevitable conclusion is that inefficient mobile cooling units negatively impact the very ambient conditions they intend to improve. Inefficient mobile cooling devices waste water and power due to the unnecessary usage of fans and high water consumption (Van Staden *et al.*, 2020).

2.3 Mathematical Modelling of ACU Components

This section investigates methods to mathematically model component specific and integrated vapour compression cycles similar to those used in the ACU. Furthermore, this section provides a background on R407C, the ACU refrigerant.

These main components of the ideal ACU include:

1. A capacity controlled, semi-hermetic, oil injected screw compressor performing isentropic compression.
2. A tube-in-tube, refrigerant-to-water, counterflow condenser with a stainless-steel inner pipe and a carbon steel annulus performing heat rejection and condensation.
3. Straight charged, externally equalized, TEVs performing isenthalpic expansion.
4. Epoxy-coated, wavy aluminium-finned, rifled bore copper-coiled, air-to-refrigerant crossflow evaporators performing isobaric heat absorption and evaporation.

The system is driven by refrigerant mass flowrate through the expansion valve and compressor as well as the fluctuating pressure of two-phase fluid through the heat exchangers.

Up to date, dynamic modelling of Vapour Compression Systems (VCS) is done using physics- or data-based approaches. Physics-based modelling paradigms are based on first principles. These models include lumped parameter approaches that attempt to model the dynamic variations in phase transition points, moving boundary approaches that attempt to model the dynamic variations in phase transition points, and finite-control volume approaches that use discretized models that include temperature and parameter gradients in an effort to achieve greater accuracy. These models require tedious calibration and validation with experimental data. Data-driven techniques provide faster model generation but are limited to the system and conditions that generated the data (Rasmussen, 2012).

The domain of sensible dynamic modelling alternatives exists between these two poles. At one pole, models are purely governed by the conservation equations. At the other pole are empirical models that rely on transient experimental data to create dynamic models. This is known as time-series analysis or system identification and allows model creation abiding to a time frequency domain. The majority of modelling efforts described in literature fall somewhere in the middle, as the bulk is governed by physics, but include semi-empirical features, such as heat transfer correlations and efficiency maps, created using data from experimental means. Pure data dependent empirical models are favoured for their expedience, however, these models are rigid and limited to the tested system and conditions that supplied the data for the model. Pure conservation equations models are more difficult and time consuming to create, requiring extensive

tuning and validation; however, they are robust accommodating changing operating conditions or changes in system parameters (Rasmussen, 2012).

2.3.1 Modelling the VCS

This section presents modelling approaches for a typical VCS, discussing and classifying published approaches. Literature suggests that modelling two-phase heat exchanger dynamics pose the greatest challenge in VCS model development. The heat exchangers determine how the system interprets pressure using the mass flowrates as model inputs. The compressor and expansion valve devices use boundary pressures to determine mass flowrate. Proper integration of the main VCS components is important since each component determines outlet fluid enthalpy to be used by subsequent component models.

A hybrid dynamic model allowing the transient simulation of an air/water heat pump is described by Rasmussen and Shenoy (2012) and Wu *et al* (2015). The refrigeration system described, consists of an evaporator, a condenser, an expansion valve, a variable speed scroll compressor and a receiver, circulating R410A. The model allows for variation in evaporator and condenser geometry and boundary condition inputs. This is due to the employment of a switching mechanism between different configurations based on a phase stability test, which is intended to verify the system simulation's continuation. The model's inputs include the temperatures and mass flowrates of the inflow secondary fluids, the compressor rotation speed, and the position of the expansion valve. The overall refrigerant mass is determined by the initial conditions maintained in the heat exchangers and receiver and remains constant. Time variations in the refrigerant thermodynamic state are calculated as outputs at the inlet and outlet of each sub-system, based on the position within the heat exchangers. The time variations of the heat exchanger's walls and secondary fluid temperature profiles are also calculated. The single-phase domain was simulated using the Lemmon equation of state. The two-phase domain is calculated using empirical polynomials from available data as functions of temperature and quality. The condenser and evaporators are based on the same heat exchanger model of which the main assumptions include:

1. The momentum balance equation is not included because it is assumed that the refrigerant pressure is uniform.
2. The liquid and vapour in the refrigerant's two-phase zone are in thermodynamic equilibrium.
3. Axial conduction effects are ignored.
4. The secondary fluid is taken as a single-phase fluid. It is assumed that its density and heat capacity are constant.

The model indicated good agreement with experimental results. The evaporator component under-predicted the pressure loss. To maintain flexibility of the heat exchanger model, it is proposed to

model the pressure loss over the evaporator using two lumped pressure loss models either side of the receiver and evaporator, allowing the pressure in the receiver and evaporator to remain uniform. A numerical model of a variable displacement vapour compression cycle driven by a linear compressor running on R134a refrigerant is presented by Li *et al.*, 2022. The model includes detailed Matlab®/Simulink® models of a variable displacement compressor, a tube-in-tube refrigerant-to-water condenser, an air-to-refrigerant microchannel evaporator, a TEV as well as the refrigerant distribution. The overall results were within 10% of experimental data. The model operates based on the following assumptions:

1. There is no variation in refrigerant enthalpy across the expansion valve, thus the condenser outlet enthalpy is assumed to be the same as the evaporator inlet enthalpy.
2. The compressor wall temperature is assumed to have a constant heat flux relating to the ambient temperature.
3. The body pressure of the compressor is constant.
4. There is no pressure-drop in the heat exchanger.
5. Reed valves are modelled as rigid bodies.

The strategies investigated are diverse in nature, covering the spectrum between the physics- and data-based poles in the domain of dynamic modelling alternatives. The remainder of this sub-chapter reviews component specific modelling strategies.

2.3.2 Compressor Modelling

The compressor compresses the superheated refrigerant in the vapour compression cycle to high-pressure vapour (Hundy *et al.*, 2016). The ACU is equipped with a compact screw compressor, which is a popular configuration among refrigeration systems due to its limited components, robustness and efficiency (Rowinski *et al.*, 2018). Unlike scroll compressors, the orientation of the screw compressors does not affect their operation, allowing the ACU to be turned vertical when descending down mine shafts without risking damage to the compressor (Majgaonkar, 2008).

The operational capacity of the ACU compressor is controlled by a slide valve system and not the commonly used variable speed control. Variable speed control uses an inverter to regulate the voltage of the compressor motor, speeding it up or slowing it down, so doing changing the speed of the compressor rotors. This control allows the compressor to operate at speeds exceeding the rotating speed rating whereas the rotor speed does not change for slide valve control. A study conducted by Sun *et al.* (2021) compared the pressure fluctuation characteristics and thermodynamic performance of a twin screw refrigeration compressor with slide valve and variable speed control at the same volume flowrate (Sun *et al.*, 2021). Although the pressure fluctuation amplitude in the gas vent port area of the slide valve was significantly reduced, the maximum pressure fluctuation amplitude of the compressor with variable speed control was generally lower

at the same flowrate due to the severe under-compression of the compressor with the slide valve (Sun *et al.*, 2021). The compressor with variable speed control had better thermodynamic performance under part-load conditions than the compressor with a slide valve, implying a better COP and total efficiency for variable speed twin screw refrigeration compressors (Sun *et al.*, 2021).

Up to date, various mathematical models have been developed attempting to accurately simulate the refrigeration screw compressor. The majority of these mathematical models are algorithms applied to Computational Fluid Dynamic (CFD) models aiming to achieve more accurate simulations in a shorter time. Stošić and Hanjalic (1997) created a geometrical and thermodynamic model for screw machine simulation. They provided a method for creating rotor profiles, which are utilized to mimic the thermodynamic compression process and establish design parameters (Hanjalic & Stošić, 1997). Spille-Kohoff *et al.* (2015) published a 3-D CFD simulation of a dry screw compressor to examine leakage flows and rotor heating, whereas Ding and Young (2017) offered a 3-D transient CFD model of a screw compressor with oil injection to explore its influence on compressor performance. The findings of another study indicated that the fuzzy beamlet finite element model accounts for the twin rotor suction end clearance and its effect on the efficiency of the compressor (Zhao *et al.*, 2021). The model mitigates inaccuracies considering fuzzy affecting parameters and taking beamlet transformation as interpolating function of elements (Zhao *et al.*, 2021). This describes the slide effect on the Bitzer compact screw compressor, implying that the model is capable of simulating a compressor with infinite capacity control. The model claims to effectively reduce computation time and obtain accurate results compared to existing data (Zhao *et al.*, 2021). It must be noted that CFD models need a significant amount of computing effort, first for building machine geometry or grid to mesh, and secondly for solving differential equations, making them unsuitable for whole-system simulations (Di Mattia *et al.*, 2021).

The alternative is a simplified one dimensional empirical model that requires data from compressor maps or more complex models. Di Mattia *et al.* (2021) developed a Simulink model of a twin-screw compressor using differential equations for mass and energy balance to simulate the compression, suction and discharge phases (Di Mattia *et al.*, 2021). Computational time limits the use of the model in a refrigeration system, thus, the result of the detailed model is used to generate compressor performance maps that are applied to a simplified model to calculate mass flowrate, shaft power and the fluid discharge temperature. The calculation results for both models were validated by comparing them to performance datasets provided by the compressor manufacturer, with suction mass flowrate calculated with an absolute error of less than 1% and mechanical compression power calculated with an absolute error of less than 4%. The simplified model employs non-linear algebraic equations that require very little computational time, making it reliable and suitable for simulation use in an entire refrigeration system (Di Mattia *et al.*, 2021).

Chen *et al.* (2011) describe an integrated mathematical model to simulate the oil injected, twin-screw compressor with slide valve capacity control (Chen *et al.*, 2011). The valve slides, allowing it to change its effective length within the rotor length range to regulate the built-in volume ratio of full load, functioning on the same principle as the ACU compressor (Bitzer, 2012). In their paper, the effective bypass area and radial discharge area as functions of the rotation angle of the male rotor and the slide valve position were firstly calculated, and then a model for calculation of the gas-oil mixture flowrate through the bypass port was proposed. The model was verified through an experimental investigation carried out at part load to compare results to the bypass flowrate obtained from the flow coefficient, calculated from measured data. The simulation results were in accordance with the measured data, with the model underpredicting shaft power by 9.5% and cooling capacity by 8.3%, both at the 95% slide valve position (Chen *et al.*, 2011).

Chamoun *et al.* (2013) proposed a mathematical model of the compression cycle for twin screw compressors that use spatial discretisation based on the rotating angle of the male rotor to examine the compressor's dynamic behaviour (Chamoun *et al.*, 2013). Giuffrida has proposed a new simulation procedure combining the data obtained from detailed mathematical models with that of simplified models to achieve accurate transient simulation of a compressor as a component of a refrigeration system (Giuffrida, 2016). One dimensional empirical model using compressor performance maps proved to be appealing, delivering accurate results and leaving little room for error.

2.3.3 Condenser Modelling

The condenser of a vapour compression cycle is a heat exchanger that condenses the refrigerant passing through it from a superheated gas to a subcooled liquid (Hundy *et al.*, 2008). The ACU is equipped with a tube-in-tube refrigerant-to-water counterflow condenser with an intricate, compact design.

Over the last 50 years, an important group of correlated and validated mathematical models based on existing experimental data has been developed. These models are reliable, but a uniform criterion to determine what dimensionless groups should lead model development of heat transfer in tube-in-tube condensers does not yet exist (Camaraza-Medina *et al.*, 2019).

Fayssal and Moukalled presented four numerical algorithms for predicting the hydrodynamic and heat transfer characteristics of one-dimensional, two-phase flow problems (Fayssal & Moukalled, 2012). The models are distinctly derived from conservation of energy of a one-dimensional differential element and the conservation of mass based on the variations of the mean liquid and vapour phase velocities. The models operate with different assumptions considering the sensible heat transfer at the liquid–vapour interface to be zero or considering heat transfer to occur at the

interface between the two phases. All model variables are functionally linked to two parameters, the refrigerant saturation pressure and the vapour phase-velocity, according to the mathematical formulation. The average standard deviation of saturation temperature prediction of the model was 0.012%, pressure was 0.07%, flow quality was 4.16% and wall temperature was 0.04%, compared to experimental data. The models are verified only for pure refrigerants making its reliability unclear for the zeotropic refrigerant R407C (Fayssal & Moukalled, 2012).

Garcia *et al.* (2018) developed a model to predict the pressure-drop of R407C during two-phase flows, in horizontal tubes with the aid of artificial neural networks. The tubes used are relatable to the geometry of the ACU condenser tubes. The pressure-drop is determined using hydraulic diameter, mass flux, saturation pressure and vapour quality as inputs. The neural network was calibrated using data from 127 experiments, yielding a mean absolute percentage error of 6.11% (Garcia *et al.*, 2018).

Camaraza-Medina *et al.* (2019) developed a mathematic model calculating the heat transfer of condensing refrigerant in tubes. This method is based on the Gaussian Equation for an infinite straight line, accounting for this relation with the differential equations that govern the heat transfer process. This method was verified with experimental data of 22 different substances including R32, R125 and R134a. The method is applicable to the ACU tube geometries as it accommodates tube diameter ranging from 2 to 50 mm. The method accommodates horizontal or inclined tubes allowing condenser simulation in situations where the ACU is not level. The mean deviations of this method for horizontal pipes were considered satisfactory at 11.8% and for inclined tubes at 13% (Camaraza-Medina *et al.*, 2019).

2.3.4 Expansion Valve Modelling

The expansion valve of a vapour compression cycle is an air conditioning throttling device that controls the amount of refrigerant released into the evaporator. Flow restriction allows expansion that releases pressure resulting in vaporisation of the liquid refrigerant. This flow regulation allows the evaporator to maintain consistent superheating of refrigerant passing through it (Eames *et al.*, 2014).

The ACU is equipped with straight charged, externally equalized TEVs. Straight charged expansion valves use the same refrigerant as the system. An externally equalized valve monitors pressure at the evaporator outflow through an external equalization tube that runs to the same location as the temperature sensor bulb of the valve, correcting for any pressure dips caused by the evaporator, refrigerant distributor, and distributor tubes (Eames *et al.*, 2014).

The expansion valves used in commercial refrigeration systems are thermostatic or electrical. TEVs use pressure and temperature to regulate flow whereas electrical expansion valves often only use temperature. Their performance is relatable to each other, as suggested by the study conducted by Aprea and Mastrullo (2002).

Eames *et al.* developed generalised steady state and transient mathematical models specifically for TEVs used in commercial refrigeration systems (Eames *et al.*, 2014). The models use a series of standardised equations describing the physics of an expansion valve to solve for conservation of mass. The models yield accurate reliable results compared to experimental data. The models were incorporated in a vapour compression cycle and tested under transient conditions. Comparing the resulting evaporator saturation temperature to experimental data yielded a maximum error of 15.2%, underpredicting saturation temperature. Both models are useful when constructing generalised transient or steady state models of vapour compression refrigerator systems since they do not require manufacturer's data (Eames *et al.*, 2014).

More recently the expansion valve has been modelled in CFD using homogeneous equilibrium models and homogeneous relaxation models. The first drastically underestimate mass flowrate, due to the failure of the model to handle the sonic velocity values for the two-phase area near the liquid saturation line. The thermodynamic non-equilibrium effects are too significant for this model to be used for a variety of flow conditions. The latter model delivered satisfactory simulation results of 5.3% average relative difference, making it applicable to a variety of different expansion devices. The model is an attractive novel alternative solution for refrigeration modelling (Loska *et al.*, 2021).

2.3.5 Evaporator Modelling

The evaporator of a vapour compression cycle is a heat exchanger that evaporates the refrigerant passing through from a two-phase mixture to a superheated vapour (Trott & Welch, 2000). The ACU utilizes wavy aluminium-finned, copper-coiled, air-to-refrigerant crossflow evaporators.

Sripattrapan and Wongwises (2005) developed a one-dimensional flow model of the evaporation of refrigerants under constant heat flux in a horizontal tube to serve as a tool for evaporator design. It can be assumed the tubes of the ACU evaporators experience constant heat flux as ambient air is drawn in across the entire effective length of its tubes. The model's governing equations abide to the conservation of mass and momentum. The model predicts variation in temperature heat transfer coefficients and pressure-drop of various refrigerants. The model found that the evaporation rate decreases for increased pipe length due to diminishing latent heat transfer through the liquid–vapour interface. This model was validated using experimental data and yielded a pressure drop prediction within 30% of the experimental results. The Comparison of the heat transfer coefficient was within 10% (Sripattrapan & Wongwises, 2005).

Zabet *et al.* (2012) developed a steady state EES-based mathematical model for an aluminium finned copper coil air-to-refrigerant crossflow evaporator functioning with R407C refrigerant. The model requires air inlet-to-outlet temperatures, refrigerant outlet temperature, atmospheric pressure, the air mass flowrate, compressor displacement volume as well as evaporator geometry. The most relevant results generated include the refrigerant inlet temperature, refrigerant mass flowrate and the pressure-drop on the refrigerant-side. The model was validated using experimental data generated using an existing evaporator, fitting the description profile. The refrigerant mass flowrate error was around 2% and the logarithmic difference temperature has an error of 2% (Zabet *et al.*, 2012). Key features of the model include the use of the Lockhart-Martinnelli correlation along with an adapted approach by Wang *et al.*, (2010) to determine the pressure-drop in two-phase flow.

Taler *et al.* (2021) presented a new semi-analytical, thermal calculation method applicable to multi-pass, multi-row, wavy finned, cross-counterflow heat exchangers. Unlike the popular finite difference method, this method allows the temperature to be determined at any point within the control volume. The evaporator is divided into finite volumes, within which the fluid temperatures are given by analytical formulas. The method was verified for a single-pass, double-row, crossflow tube heat exchanger using a developed analytical model and validated experimentally for a two-pass car radiator. Comparing the exact solution to the presented method for steam and mean flue gas temperatures presented percentage errors within 0.387%. This method obtains accurate results in short calculation time when applied to complicated flow systems (Taler *et al.*, 2021).

2.3.6 Refrigerant R407C Modelling

The 1987 Montreal protocol was a global agreement to protect the stratospheric ozone layer by phasing out the production and use of ozone-depleting substances like chloroflourocarbons and hydrochloroflourocarbons. This protocol restricted the use of R22 refrigerant, which was widely used at the time due to its performance and affordability. R407C, the ACU refrigerant, was developed in the early 1990s as a substitute for R22. R407C is a zeotropic mixture of R32/R125/R143a with a mass fraction of 23/25/52%. Zeotropic mixtures experience a temperature glide in excess of 2.8°C during evaporation and condensation due to the diverse properties of their blended refrigerants. Azeotropic refrigerants on the other hand, perform like near homogeneous substances and experience a temperature glide of less than 2.8°C (Gabriellii & Vamling, 1997). Using zeotropic mixtures to replace azeotropic refrigerants is favoured for the following reasons:

1. If the temperature glides match that of the heat transfer fluid in the condenser, an entropy reduction will occur (Venkatarathnam *et al.*, 1996).

2. Zeotropic mixtures achieve a higher COP than azeotropic refrigerants if the temperature glides' capability is exploited correctly, avoiding pinching in the heat exchangers (Gabrieli & Vamling, 1997).
3. The capacity of the refrigeration system can be regulated by changing the composition of the zeotropic mixture (Venkatarathnam et al., 1996).

The unfavoured result of using zeotropic mixtures is the occurrence of pinch points in heat exchangers. This occurs as a result of the large non-linear variation in two-phase enthalpy with temperature during phase change in some zeotropic mixtures (Venkatarathnam *et al.*, 1996).

2.4 Summarising the Literature Study

This literature study investigates the source of heat in the underground mining environment, paying special attention to the Witwatersrand Basin and the Bushveld Complex due to the significance of these regions to the future of the South African mining industry.

Deep level mining refrigeration systems is briefly addressed in a broad sense with the main focus on the refrigeration goals of cooling and heat rejection. The dynamics of the vapour compression cycle is discussed as a means of achieving cooling whilst the role of water as a heat rejection medium is investigated for various mine refrigeration systems.

The bulk of the literature study is made up of a thorough review of mathematical modelling approaches for the components of the ACU. Modelling the vapour compressor cycle as an entire system is investigated with literature suggesting all VCS models are defined between physics- and data-based approaches (Rasmussen, 2012). A more detailed investigation follows where component specific modelling approaches are reviewed. This includes reviewing the compressor, condenser, expansion valve, evaporator and R407C refrigerant. Valuable insight from the literature review will contribute to the model development process of the ACU simulation.

The Flownex ACU model will make use of both physics- and data-based approaches, similar to most VCS models. Reviewing relevant literature, the ACU compressor can be interpreted as a one dimensional empirical model that makes use of compressor maps like the approaches by Di Mattia *et al.* (2021) and Giuffrida (2016). Using compressor maps would produce an accurate compressor model with a limited operational range (Rasmussen, 2012). The consistent geothermal gradients over the Witwatersrand Basin and the Bushveld Complex respectively indicates that the compressor would operate within a specific and predictable range, justifying this compressor modelling approach (Boudreau, 20/01/2022; Rasmussen, 2012) (Jones, 2018). The ACU condenser model can be considered a physics-based model applying Flownex relations for heat transfer coefficients and pressure-drop in two phase flow similarly to the approach used by Fayssal

and Moukalled (2012) for heat transfer coefficients and Garcia *et al.* (2018) for pressure-drop. The ACU expansion valve can be considered a physics-based model operating on the basis of conservation of mass relatable to the approach suggested by Eames *et al.* (2014). The ACU evaporator can be considered a physics-based model, relatable to the Zabet *et al.* approach, which applies the Lockhart-Martinelli method for two-phase flow (Zabet *et al.*, 2012). The literature study is concluded with a brief background of the R407C refrigerant and its zeotropic nature. This theoretical foundation laid by the literature review is applied in the model development chapter that follows.

CHAPTER 3: MODEL DEVELOPMENT

The previous chapter investigated various means of modelling the vapour compression cycle as a system as well as its individual components. In this chapter the main components of the ACU consisting of the compressor, condenser, expansion valve, evaporator, fan and interconnecting piping are each individually developed within the Flownex domain.

The chapter first provides an introductory overview of basic Flownex terminology and concepts to ensure a better understanding of the software-specific aspects of model development. The chapter proceeds to provide background on the details of each component, discuss model development within Flownex and verify the accuracy of developed models. The individually developed models are interconnected to create a flow network of the entire ACU. The chapter is concluded with an integrated ACU model with conditions set in place to achieve transient simulation.

3.1 Introductory Overview of Flownex Terminology and Concepts

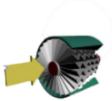
This section provides a background on Flownex terminology and concepts to help the reader better interpret the content of Chapter 3. Flownex is the commercial thermal fluid modelling software that was used to develop models of the components discussed in this chapter. The Flownex simulation environment contains elements that encompasses the fundamental theory for a wide range of thermodynamic processes, eliminating the need to do model development de novo. The software significantly simplifies the model development process. However, it's important to emphasize that despite the simplification, a fundamental understanding of the process being modelled is required to accurately model the components of the ACU. Moreover, Flownex allows for transient simulation, which is crucial to achieve the set goals of this study.

3.1.1 *Flownex Elements Used in this Study*

The Table 8 below contains a description of all the flow network elements used in this study.

Table 8: Flownex Elements Used in Model Development (Flownex, 2022)

Icon	Element	Description
	Basic Centrifugal Pump	Where pump/fan curves are unknown, the Basic Centrifugal Pump element is best suited for preliminary design of centrifugal pumps/fans.

Icon	Element	Description
	Bend	The Bend element is useful for simulating flow across short circular arc bends.
	Boundary Condition	The Boundary Condition element can be used to specify a network's boundary conditions.
	Composite Heat Transfer	The Composite Heat Transfer element is used to model heat transfer to or from solid structures, simulating conduction, convection, and radiation.
	Compressor	A Compressor element adds energy to a gas stream by increasing the pressure.
	Flow Resistance	The Flow Resistance element can be used to model components that have limited geometrical information or to shorten the time it takes to solve complex networks.
	Restrictor with Discharge Coefficient	A throttling process can be represented by the Restrictor with Discharge Coefficient element.
	Finned-Tube Heat Exchanger - Fin Side	Conventional and complex finned-tube heat exchangers can be simulated using the Finned-Tube Heat Exchanger element.
	Finned-Tube Heat Exchanger - Tube Side	Conventional and complex finned-tube complex heat exchangers can be simulated using the Finned-Tube Heat Exchanger element.

Icon	Element	Description
	Node	Nodes are the elements' end points.
	Pipe	The Flownex Pipe element is a versatile tool for modelling fluid flow in pipes. It accurately accounts for friction and secondary losses in pipe networks, providing an accurate representation of fluid behaviour.
	Psychrometric Boundary Condition	The Psychrometric Boundary Condition element can be used to specify a flow network's humid air boundary conditions.
	Quick Script	Quick Script element is a C# scripting tool that greatly simplifies coding when required.
	Variable Speed Pump	A hydraulic pump and a fan can be modelled using the Variable Speed Pump/Fan element.
	Data Transfer Link	The link serves to transfer an input value from one element to another, allowing it to be utilized as input for the latter. Additionally, it can manipulate the transferred value through addition, subtraction or multiplication of a constant value.
	Steady State Controller Script	The script provides C# syntax code that may be used as a controller for steady state conditions.

3.1.2 Conservation Equations

The fundamental operation of the Flownex elements can be captured within the three conservation equations. The assumption was made to disregard the effect of elevation change since there is no significant elevation differential in the ACU vapour compression cycle. For this reason, elevation change is not accounted for in the three conservation equations below.

Conservation of mass (Flownex, 2022).

$$\underbrace{\rho \frac{\partial \forall}{\partial t}}_{\text{Accumulation term}} = \underbrace{\sum_i^n \rho v_{in,i} A - \sum_j^m \rho v_{in,j} A}_{\text{Advection terms}} \quad [3.1]$$

ρ	=	Density [kg/m ³],
$\forall$	=	Volume [m ³],
t	=	Time [s],
v	=	Velocity [m/s],
A	=	Area [m ²],
i, j	=	Increment number [-],
n, m	=	Final increment [-],
in	=	Inlet flow to increment [-],

- The 'Accumulation term' in the conservation of mass equation represents the contribution of changes in mass within a control volume over time. This term accounts for how mass can accumulate or deplete within the control volume due to various processes.
- The 'Advection terms' in the conservation of mass equation represent the contributions from the convective transport or movement of mass within a fluid. These terms account for the transport of mass by the bulk motion of the fluid.

Conservation of energy (Flownex, 2022).

$$\underbrace{\dot{Q}}_{\text{Heat input}} - \underbrace{\dot{W}}_{\text{Work}} = \underbrace{\frac{\partial}{\partial t} [\rho h_0 \forall]}_{\text{Energy accumulation term}} - \underbrace{\frac{\partial p}{\partial t} \forall}_{\text{Energy change due to pressure}} + \underbrace{\frac{1}{A} \frac{\partial}{\partial s} [\rho v A h_0 \forall]}_{\text{Advection terms}} \quad [3.2]$$

$\dot{Q}$	=	Heat transfer rate [W],
$\dot{W}$	=	Rate of work performed [W],
h_0	=	Stagnation enthalpy [J/kg],
p	=	Pressure [Pa],
s	=	Distance [m],
A	=	Heat transfer surface area [m ²],

- The 'Energy accumulation term' typically represents the rate of change of internal energy with respect to time within a control volume or system. This term accounts for the change in the system's internal energy, often due to processes like heat transfer and work done on or by the system.
- The 'Energy change due to pressure' term in the conservation of energy equation often refers to the work done or energy exchanged with the surroundings due to changes in pressure within a system.
- The 'Advection terms' typically refer to the convective transport or movement of energy within a fluid or material. These terms account for the transport of energy by the bulk motion of the fluid or material.

Conservation of momentum (Flownex, 2022).

$$\underbrace{\frac{\partial}{\partial t}(\rho v)}_{\text{Inertial term}} + \underbrace{\frac{1}{A} \frac{\partial}{\partial s}(\rho v^2 A)}_{\text{Acceleration term}} + \underbrace{\left(\frac{f}{D} + \frac{K}{\partial s}\right) \frac{1}{2} \rho |v| v}_{\text{Wall shear stress term}} = - \frac{\partial p}{\partial s} \quad [3.3]$$

f = Friction factor [-],
 D = Pipe diameter [m],
 K = K-value for secondary losses [-],

- The 'Inertial term' refers to the contribution of the inertia of a fluid to changes in momentum within a system. This term is crucial when considering the motion of fluids and how their velocity changes over time.
- The 'Acceleration term' in the conservation of momentum equation represents the contribution of acceleration to changes in momentum within a fluid. This term accounts for how the velocity of the fluid changes over time, affecting its momentum.
- The 'Wall shear stress term' in the conservation of momentum equation represents the contribution of shear stress at the boundary or interface between the fluid and a solid surface (such as a wall). This term accounts for the frictional forces exerted by the fluid on the surface and how these forces influence the momentum of the fluid near the boundary.

3.1.3 The Use of Parameter Tables

All Flownex flow networks discussed in this study operate using parameter tables since they are a convenient way to simultaneously vary multiple input conditions and record their effects. For each row of the parameter table, a steady state calculation is done, applying input conditions to elements of the model, recording their effects.

To better understand this concept, consider an example where R407C refrigerant enters a flow network at a fixed temperature and absolute pressure and is exposed to a constant heat input. The

outlet temperature is recorded for different flowrates. Assume this is a real process taking place at a plant with measured data available for the outlet temperature. A Flownex flow network is created to model this process. Figure 9 below illustrates the flow network to parameter table interface.

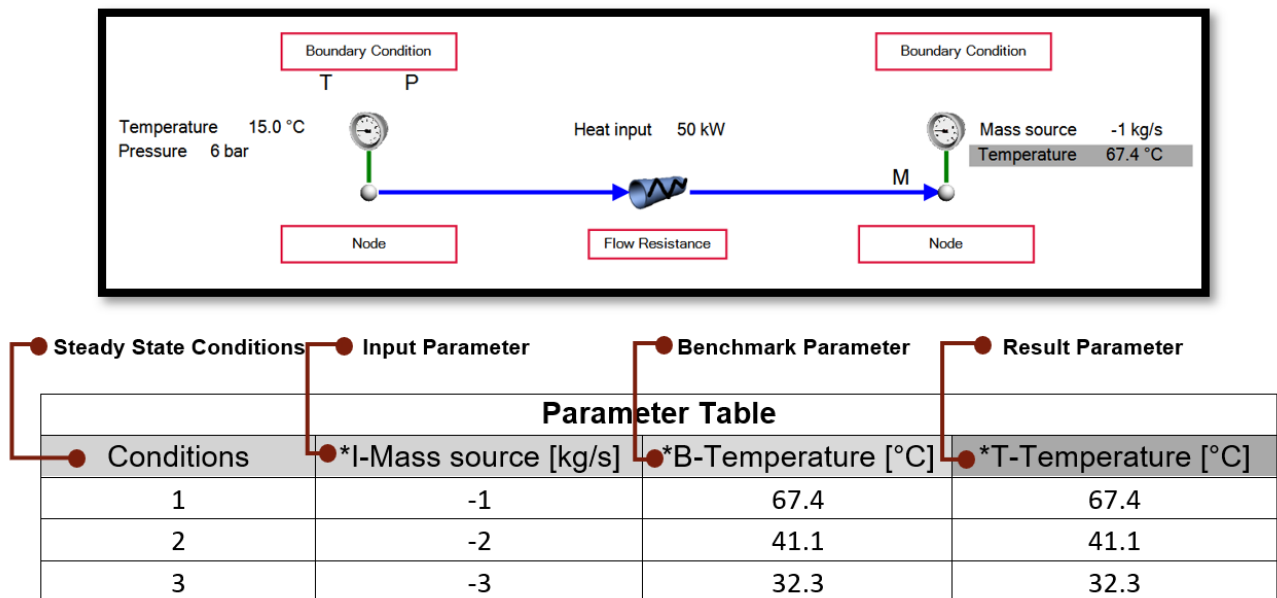


Figure 9: Flownex flow network example with parameter table interface

The black-bordered diagram displays a Flow Resistance element which receives a constant heat input of 50 kW, conveying R407C refrigerant. The Boundary Condition elements on either side are denoted by letters such as 'T' for temperature, 'P' for pressure, 'Q' for quality, or 'M' for mass source, indicating the applied boundary input. The Boundary Condition element to the left receives a temperature input of 15°C and absolute pressure of 6 bar. The Boundary Condition element to the right receives a mass source input of -1 kg/s. The negative value indicates the fluid flows to the Boundary Condition element and not away from it. To prevent confusion, the parameter tables in the appendices have no negative signs associated with flow direction. The dark grey temperature of 67.4°C indicates the resulting temperature recorded on the Boundary Condition element to the right as the fluid exits the Flow Resistance element.

The parameter table beneath the black-bordered diagram in Figure 9, features four columns and three rows. Each row represents a unique steady state calculation. In this example there are four input parameters, but only the mass source parameter in light grey, is varied in the parameter table.

The 'Benchmark Parameter' column in light grey contains the recorded plant data for outlet temperature and this column would be used to validate the accuracy of generated results. If there was reliable outlet temperature data available from alternative software modelling this process, that data would have been used to verify the accuracy of the Flownex model. Verification data would have been plotted in a 'Reference Parameter' column. Reference and benchmark parameter

columns are included in the parameter tables since it is a convenient way to compare generated results to verification or validation data. In Chapter 3 reference parameters are retrieved using independent software to verify the accuracy of the developed Flownex models. The same procedure is implemented in Chapter 4 for validation with reference parameters referred to as benchmarks.

In the parameter table above, the Flownex generated outlet temperatures are recorded in the dark grey 'Results parameter' column for the varied input parameters. The parameter table illustrates the effect of the increased mass source on the outlet temperature of the refrigerant. The ability of parameter tables to simultaneously vary multiple input conditions and record their effects makes it a powerful tool for model development.

Throughout all parameter tables of this study, the light grey columns are associated with input parameters denoted by the prefix, '*I-', reference parameters denoted by the prefix, '*R-' or benchmark parameters denoted by the prefix, '*B-'. The dark grey columns in parameter tables are result parameters the Flownex flow network solved for denoted by the prefix, '*T-', for 'test'. The following section provides an overview of the main ACU components that were modelled.

3.2 Overview of the Main Components of the ACU

The location of the main components of the ACU that were modelled is illustrated in Figure 10 below.

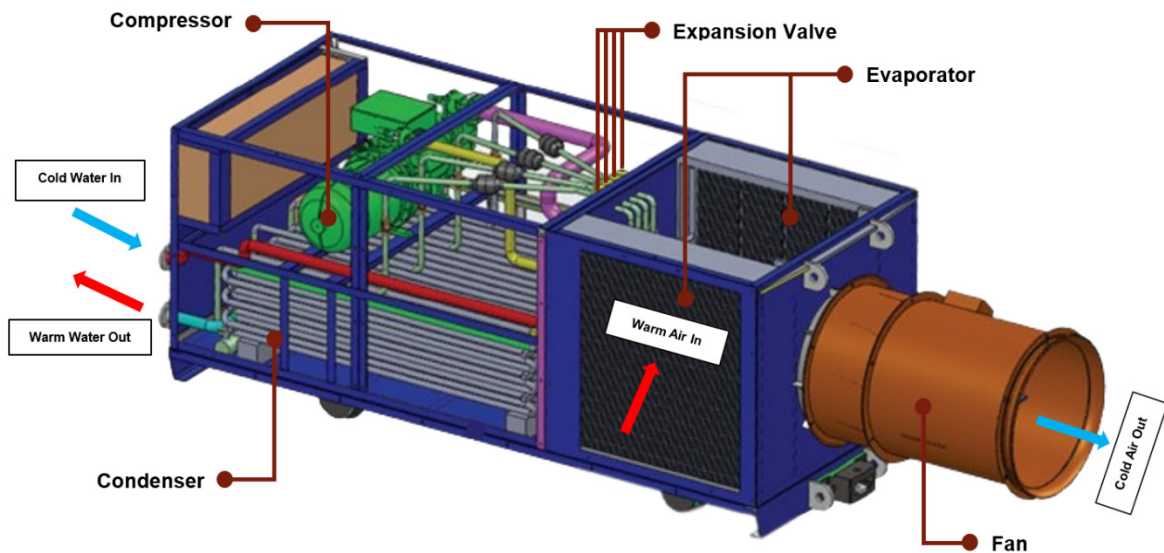


Figure 10: Isometric illustration of the ACU displaying components that were modelled and the direction and temperature differentials of fluid streams on the inlet and outlet of the heat exchangers (M-Tech Industrial, 2022).

In Figure 10 above, the ACU is depicted with its covers removed, revealing the placement of the main components that were modelled, such as the compressor, condenser, expansion valve,

evaporator, fan and interconnecting piping. Additionally, the diagram shows the entry and exit points for water and air in the ACU. The cooling process involves cold water extracting heat from warm air, using the vapour compression cycle as discussed in Chapter 2: Literature Study, Section [2.2.1 Vapour Compression Cycle](#). The following section provides a detailed discussion of each modelled component.

3.3 Compressor Model

This section describes the development of the compressor model.

3.3.1 Compressor Component Overview

Figure 11 below illustrates a view inside the Bitzer manufactured, compact screw compressor, used in the ACU.

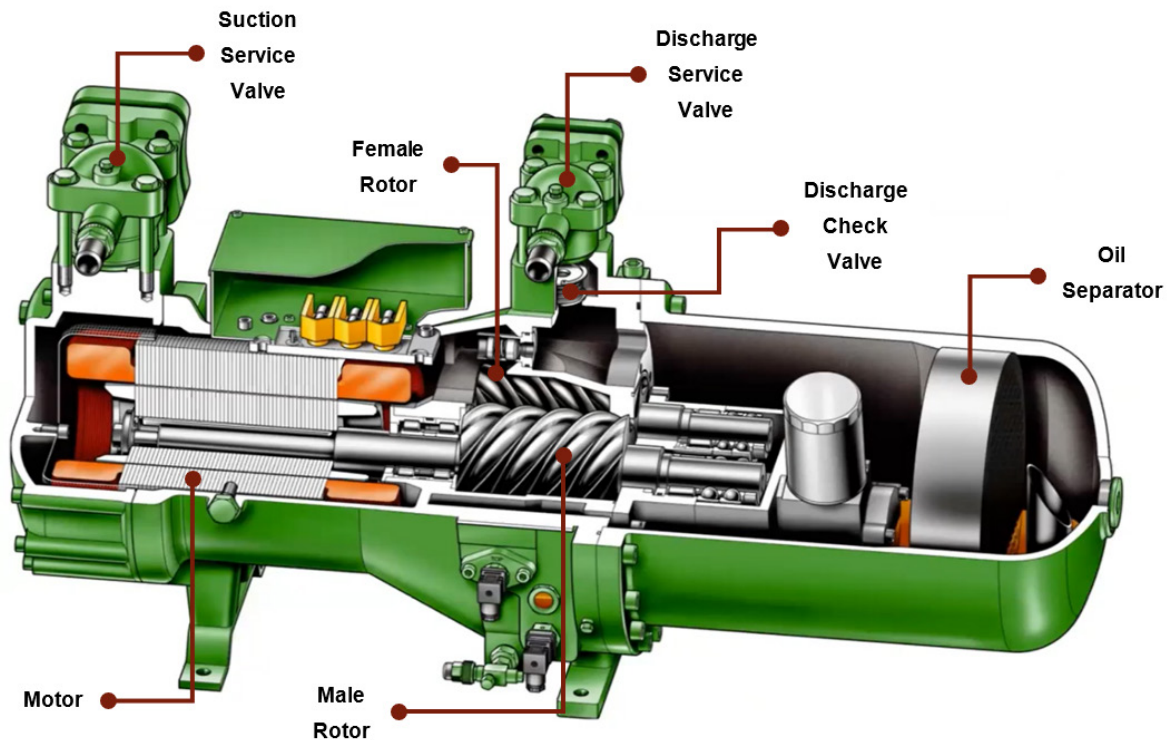


Figure 11: Inside the Bitzer compact screw compressor (Boudreau, 2022)

Refrigerant enters the compressor suction service valve and passes over the motor, extracting heat from the motor windings. A male rotor is driven by the motor and guides an adjacent female rotor (Boudreau, 2022). Drilled bores in the male rotor allow refrigerant to pass through the motor (Bitzer, 2012). Refrigerant then enters the screws. The screws rotate in opposite directions, creating a gradual compression of the refrigerant within continuous pockets. This compression process persists until the refrigerant reaches the desired discharge pressure. Subsequently, it is efficiently discharged through an oil separator and exits the compressor via the discharge service valve (Boudreau, 2022). Screw compressors do not have active working valves, necessitating the use of

a discharge check valve to prevent high side refrigerant from traveling back to the suction line, equalizing pressure when the compressor is turned off (Bitzer, 2012). Figure 12 below illustrates the hydraulic slide valve system that controls the compressor capacity.

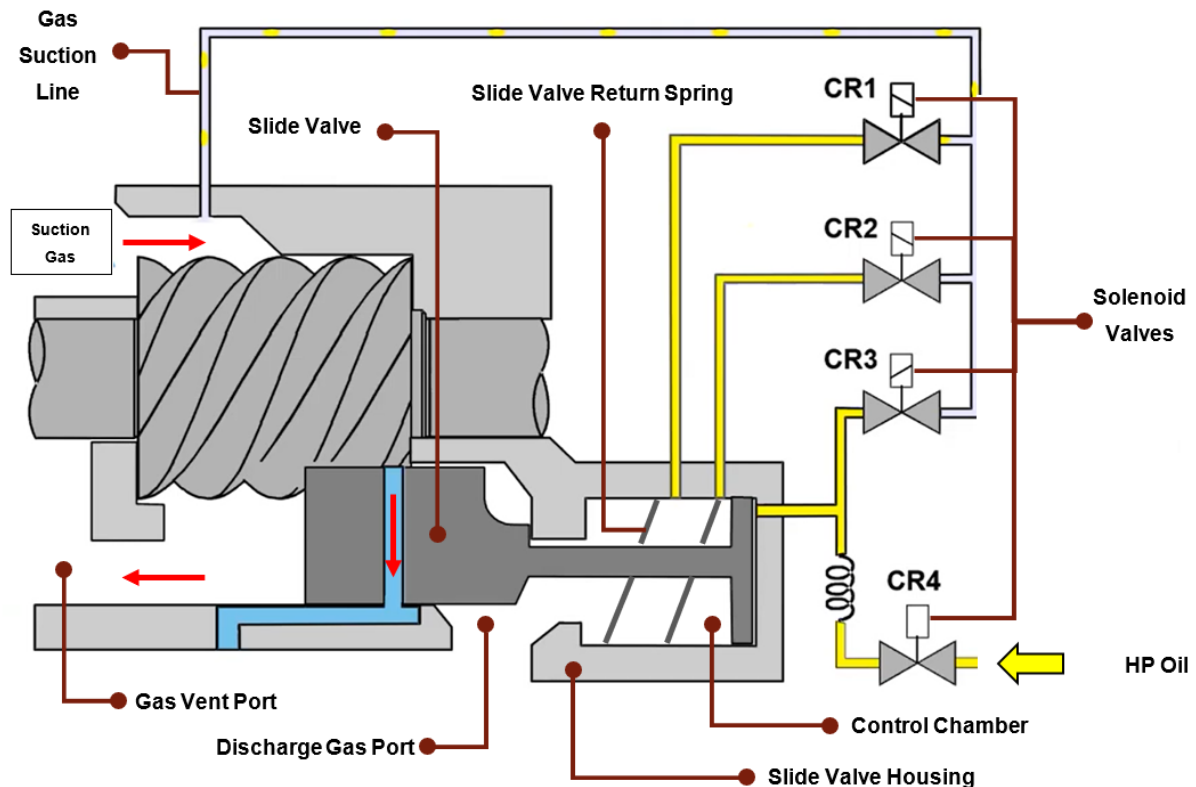
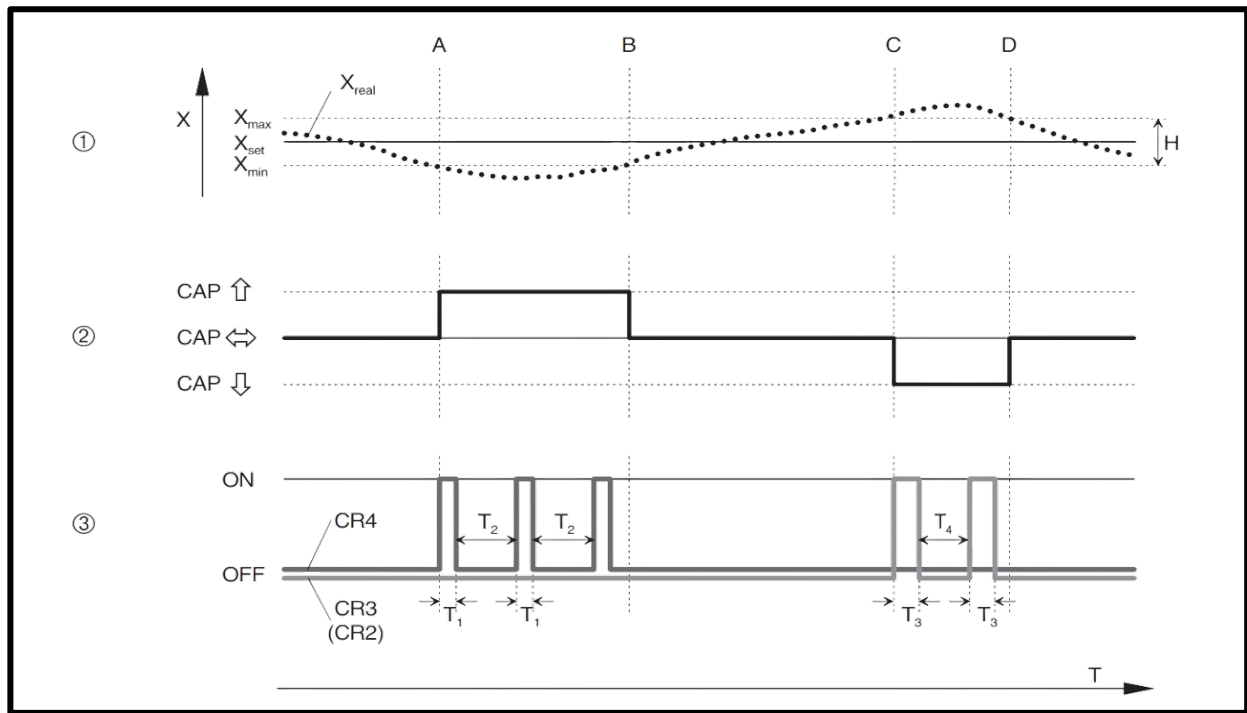


Figure 12: Bitzer compact screw compressor hydraulic slide valve system (Boudreau, 2022)

The capacity is regulated by a hydraulic slide valve system. Four solenoid valves load or unload oil into a control chamber within the slide valve housing, moving the slide valve. At full capacity, the loading solenoid valve (CR4) injects oil into the control chamber. The slide valve closes a gas vent port, allowing the use of the full effective length of the twin compressor rotors. At reduced capacity, one of the three unloading solenoid valves (CR1-3) vent oil into the gas suction line, allowing the slide valve return spring to move the slide valve, opening the gas vent port, so doing decreasing the effective length of the twin rotor screws. When the compressor is turned off, CR3 is opened to relieve the pressure in the hydraulic cylinder, allowing the return spring to push the slide valve back to the 25% minimum capacity position (Bitzer, 2012). Viewing Figure 12, the system is at 25% capacity indicated by the far-right position of the slide valve. Following the red arrows, the majority of the suction gas is directed towards the gas vent port, but a small amount passes to the discharge gas port through the blue channel.

Figure 13 below illustrates the dynamics of infinite capacity control.



①	Control input implemented as compressor discharge gauge pressure.
②	Control thermostat, signalling an output to the oscillator.
③	CR solenoid valves being energized by the oscillator.
A..D	Operating points
X	Control input
X _{set}	Set point
X _{max}	Upper break point
X _{min}	Lower break point
X _{real}	Actual value
H	Set control range
CAP ↑	Increased demand
CAP ↔	Unchanged demand
CAP ↓	Decreased demand
ON	Cr solenoid valve open
OFF	Cr solenoid valve closed
T ₁ , T ₃	Pulse period (approx. 0.5 s .. max 1 s)
T ₂ , T ₄	Pulse period
T	Time

Figure 13: Illustrating the functionality of the Infinite capacity control strategy of the Bitzer compact screw compressor (Bitzer, 2012)

Figure 13 above illustrates how the infinite capacity control system reacts when the control input moves outside the set limits. If the monitored control input (compressor discharge gauge pressure) lies within the set control range (H), no solenoid valves are activated and the slide valve stands still, thus the operating capacity is maintained at its current value.

If the compressor discharge pressure falls below the lower break point (X_{min}) of 19 bar, the operating capacity increases (operating point A in Figure 13). The loading solenoid valve, CR4 is opened for short intervals until the discharge pressure rises within the limits of H, where the operating capacity is maintained at the higher value (operating point B in Figure 13). If the compressor discharge pressure exceeds the upper break point (X_{max}) of 23 bar, the operating capacity decreases

(operating point C in Figure 13). The unloading solenoid valve, CR3 or CR2 is opened for short intervals until the discharge pressure-drops within the limits of H, where the operating capacity is maintained at the lower value (operating point D in Figure 13). Furthermore, if the compressor discharge pressure exceeds 26 bar or the compressor suction pressure falls below 2 bar, high pressure and low-pressure switches trip the compressor to prevent damage. Solenoid valves CR3/CR4 controls compressor capacity between 25% and nominal 100%. Alternatively, valves CR2/CR4 can be energized, if compressor capacity is to be limited between 50% and nominal 100%.

At start-up, the control input is below the set control range, thus capacity increases from 25% until the control input is within the set control range. If the supply of feedwater is sufficient to remove heat from the vapour compression cycle, the ACU compressor is expected to operate at 100% capacity and be within the set control range.

3.3.2 *Developed Compressor Model*

Based on the overview done on the Bitzer compressor and the nature of the Flownex simulation environment, the compressor was modelled using a Compressor element. The compressor model only applies conservation of energy and momentum equations since the element assumes negligible fluid mass. Furthermore, the compressor element assumes negligible thermal inertia for the fluid and negligible heat loss to the environment.

The ACU has benchmark data available for surface and underground conditions. Since it can be validated for both strata, the Flownex model is developed to operate in both surface and underground conditions. The surface benchmark data was collected from the Factory Acceptance Test (FAT) station. The FAT is an in-factory quality assurance protocol where the unit operates in a controlled environment to determine among other things, if the newly manufactured unit achieves an acceptable COP. This test has limited scope to recreate the operating conditions expected underground. These limitations include:

1. The surface has less dense, less moist, cooler air compared to underground conditions. Elevated air is less dense, has less heat capacity due to its dryness and carries less heat since it is cooler compared to underground conditions, thus it has less potential for heat transfer in the evaporator.
2. The electrical supply at the FAT station is insufficient to operate the ACU at full capacity, resulting in reduced refrigerant mass flowrate compared to underground conditions.
3. In the FAT, water is supplied at a reduced flowrate compared to that expected in the mining environment, thus the waterside has less potential to remove heat in the condenser.

The literature, as discussed in Chapter 2: Literature Study, Section [2.1 Sources of Heat in Mines](#) suggests that underground temperature, humidity and pressure are notably higher compared to surface conditions. The underground ACU refrigeration cycle circulates more refrigerant, has greater feedwater supply, the air is denser and carries more heat, thus greater heat transfer occurs compared to surface operation. Since the underground ACU operates at an elevated state, Superheat (SH), Subcool (SC), Condensing Temperature (TC) and Evaporating Temperature (TO) is elevated as well, compared to surface conditions. Since the operational ranges are different for surface and underground conditions, separate compressor charts are generated. Using Bitzer software, output data for compressor properties are generated over a realistic range of TCs and TOs associated with surface and underground conditions. The inputs defined between the Minimum (Min) and Maximum (Max) values in Table 9 were used to generate output data for Surface (Surf) and Underground (Und) compressor properties.

Table 9: Bitzer Software Inputs Used for Compressor Chart Generation

Strata		TO [°C]	TC [°C]	Compressor Capacity [%]	SC [°C]	SH [°C]
Surf	Min	-6	40	25	10.5	5.0
	Max	5	55	100		
Und	Min	0	40	25	11.5	7.5
	Max	12	60	100		

Although the TO and TC inputs were varied extensively across the stated ranges, the software only generates data for the four compressor capacities of 25/50/75/100%. The Bitzer software applies these inputs to a simple vapour compression cycle as illustrated Figure 14 below.

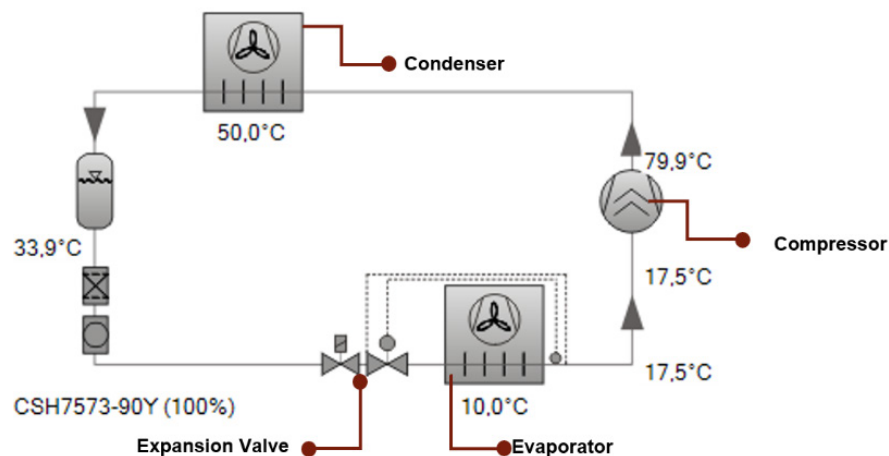


Figure 14: The Bitzer software vapour compression cycle illustrating applied inputs (Bitzer, 2022)

The generated compressor data is available in Appendix A, Section [A.1.1 Bitzer Compressor Data](#), containing compressor properties like evaporator heat transfer (Q) [kW], compressor electrical power (P) [kW] and refrigerant mass flowrate (m) [kg/s].

The chart used by the Flownex Compressor element contains maps requiring pressure ratio and efficiency data, which is not amongst the generated Bitzer data. The Flownex flow network in Figure 15 below was created to retrieve this data using the Bitzer generated compressor electrical power and refrigerant mass flowrate data along with the inputs of Table 9.

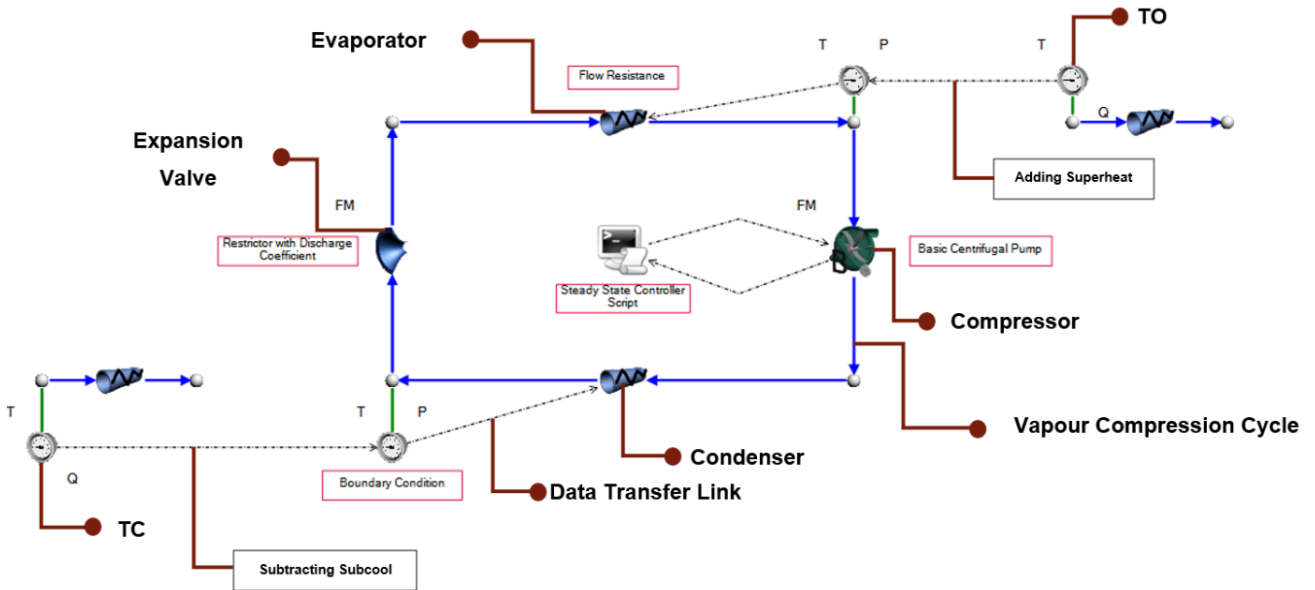


Figure 15: Flownex flow network used to retrieve compressor efficiency and pressure ratio data

The flow network in Figure 15 depicts a typical vapour compression cycle with the refrigerant flow direction indicated by blue arrows. A Basic Centrifugal Pump element (compressor) pumps refrigerant to a Flow Resistance element (condenser), responsible for maintaining a constant outlet temperature provided by a data transfer link, condensing the refrigerant. There is no pressure-drop through either of the heat exchangers in the flow network.

The refrigerant is then passed through a Restrictor with Discharge Coefficient element (expansion valve), which throttles the refrigerant to maintain the desired mass flowrate set by the pump in an isenthalpic process. The refrigerant subsequently moves through another Flow Resistance element (evaporator) that maintains a constant outlet temperature provided by a data transfer link.

The TC- and TO labelled Boundary Condition elements to the side of the vapour compression cycle are used to achieve SC and SH. The SC-side boundary is fixed at the TC and saturation quality of 0. A data transfer link subtracts the degree of SC from the TC and applies the subcooled temperature along with the relevant pressure to the respective Boundary Conditions element in the vapour compression cycle. Similarly, the SH-side boundary is fixed at the TO and saturation quality of 1. A data transfer link adds the degree of SH to the TO and applies the superheated temperature along with the relevant pressure to the respective Boundary Conditions element in the vapour compression cycle.

A parameter table communicates input values of TC, TO, compressor capacity, refrigerant mass flowrate and compressor electrical power to the flow network. The Steady State Controller Script varies the Basic Centrifugal Pump element's efficiency at Best Efficiency Point (BEP) to match the Bitzer compressor electrical power value. The Bitzer refrigerant mass flowrate value is applied as an input to the Basic Centrifugal Pump element which then changes design to match the applied mass flowrate and efficiency requirements.

Steady state calculations are performed using parameter tables for both surface and underground conditions, as illustrated in Appendix A, Section [A.1.2 Generating Compressor Charts](#). Within the table, corrected mass flowrate, corrected speed and pressure ratio are calculated using the equations below.

Corrected mass flowrate:

$$CMF = \frac{\dot{m}\sqrt{T_{0i}}}{p_{0i}} \quad [3.4]$$

Corrected speed:

$$CS = \frac{N}{\sqrt{T_{0i}}} \quad [3.5]$$

Pressure ratio:

$$PR = \frac{p_{0e}}{p_{0i}} \quad [3.6]$$

- $\dot{m}$ = Mass flowrate through compressor [kg/s],
- T_{0i} = Upstream total temperature [K],
- p_{0i} = Upstream total pressure [bar],
- N = Rotational speed of compressor [rev/s],
- p_{0e} = Downstream total pressure [bar],

The Flownex compressor charts can each contain multiple curves for multiple compressor rotational speeds, implying the charts are capable of modelling variable speed control. The variable speed control capability of the charts is used, treating the four step capacities of 25/50/75/100% as if they were compressor rotational speeds in order to achieve infinite capacity control. Pressure ratio and efficiency at BEP values are then plotted as functions of corrected mass flowrate. Second order polynomials are generated for these plots, and they are applied within Flownex charts to be used by the Compressor element. Table 10 and Table 11 below illustrates the generated polynomials for surface and underground compressor charts.

Table 10: Second Order Polynomials for Pressure Ratio as a Function of Corrected Mass Flow

Capacity [%]	Surface PR = f(CMF) [-]	R ²	Underground PR = f(CMF) [-]	R ²
25	PR = 4.343(CMF) ² – 20.064(CMF) + 23.046	1.00	PR = –17.195(CMF) ² + 43.478(CMF) – 23.917	0.83
50	PR = –6.380(CMF) ² + 22.748(CMF) – 15.314	0.83	PR = –4.262(CMF) ² + 14.766(CMF) – 8.360	0.85
75	PR = –1.510(CMF) ² + 4.690(CMF) + 4.141	0.86	PR = –0.550(CMF) ² – 0.381(CMF) + 10.091	0.88
100	PR = 0.649(CMF) ² – 9.244(CMF) + 32.092	0.82	PR = 0.266(CMF) ² – 5.226(CMF) + 21.213	0.83

Table 11: Second Order Polynomials for Efficiency at BEP as a Function of Corrected Mass Flow

Capacity [%]	Surface Eta = f(CMF) [-]	R ²	Underground Eta = f(CMF) [-]	R ²
25	Eta = 20.477(CMF) ² – 1.941(CMF) + 11.793	1.00	Eta = 34.998(CMF) ² – 53.224(CMF) + 56.365	0.97
50	Eta = –16.589(CMF) ² + 106.061(CMF) – 89.603	0.98	Eta = –35.519(CMF) ² + 180.002(CMF) – 159.835	0.98
75	Eta = 21.385(CMF) ² + 152.928(CMF) – 201.053	0.99	Eta = –31.348(CMF) ² + 206.290(CMF) – 271.118	0.95
100	Eta = –13.516(CMF) ² + 138.996(CMF) – 275.496	0.99	Eta = –18.970(CMF) ² + 177.449(CMF) – 339.968	0.96

Inspecting the R² values in the tables above, the trendline fits in Table 10 may seem low, but the verification of the developed charts would be the true indication of their reliability.

3.3.3 Compressor Model Verification

The following equations are used for verification and validation purposes:

Comparing the value of a single test point to a single reference point:

$$error = \frac{|Value_{reference} - Value_{test}|}{Value_{reference}} \quad [3.7]$$

Comparing the change in test data to the change in reference data:

$$error = \frac{|(Value_{out} - Value_{in})_{reference} - (Value_{out} - Value_{in})_{test}|}{|(Value_{out} - Value_{in})_{reference}|} \quad [3.8]$$

error = The deviation of the test value from the reference value [-],

Value_{reference} = The verification standard of comparison, obtained using well-established mathematical equations and principles relevant to the specific engineering scenario [-],

Value_{test} = The Flownex-generated value obtained through simulation [-],

Value_{benchmark} = The validation standard of comparison, obtained from experimental data [-],

Both Equations [3.7] and [3.8] are used for validation by simply substituting reference values with benchmark values. Equation [3.8] is essential for assessing the accuracy of temperature or enthalpy changes across heat exchangers since simply comparing the outlet results will not suffice in these instances. All Flownex-generated results (test values) are obtained within a convergence criterion

of 1E-04, implying that all flow network results have to converge within this criterion for steady state solver status to occur.

Figure 16 below contains the flow network used to generate test values for the Flownex compressor model verification.

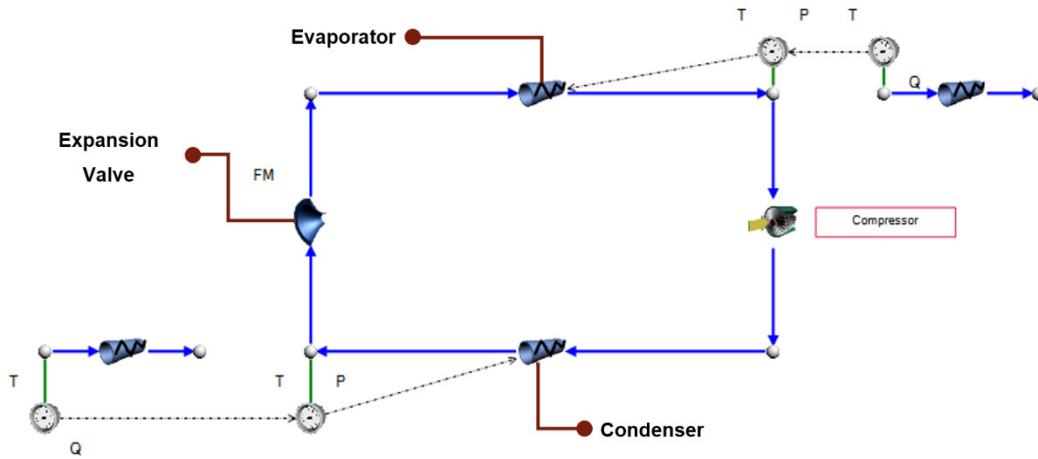


Figure 16: Flownex flow network used for compressor model accuracy verification

The flow network uses the same range of inputs as the flow network used to generate compressor charts in Figure 15 of Section [3.3.2 Developed Compressor Model](#). The only difference between the flow networks is that the Steady State Controller Script and Basic Centrifugal Pump elements in Figure 15 have been replaced with a Compressor element in Figure 16. The Compressor element uses the generated compressor charts.

For verification, the Flownex model results for the compressor electrical power [kW] and refrigerant mass flowrate [kg/s] were tested against Bitzer reference values. As stated in Equations [3.7] and [3.8], in the context of verification, reference results refer to the accurate standard of comparison against which test values are compared.

Verification of the compressor electrical power tested whether the developed model obeyed the energy conservation equation and the refrigerant mass flowrate tested whether the developed model obeyed the momentum conservation equation, so doing accounting for all conservation equations applicable to the compressor model.

For the parameter tables of the verification flow network, refer to Appendix A, Section [A.1.4 Compressor Verification Results](#). The Average (Avg) and maximum errors for the verification of the accuracy of the developed compressor model for surface and underground conditions are shown in Table 12, along with the references to Equations (Eq.) used to calculate the errors for each tested property.

Table 12: Compressor Model Verification Results

Strata		Compressor Electrical Power [%]	Refrigerant Mass Flowrate [%]
		Eq. [3.7]	Eq. [3.7]
Surf	Avg Error	1.08	1.28
	Max Error	4.36	4.71
Und	Avg Error	1.23	1.68
	Max Error	7.17	6.59

The Flownex compressor model results show a good relation to the Bitzer software results for compressor electrical power and refrigerant mass flowrate each exhibiting an average error margin within 1.68%. The surface verification was conducted over 126 points and the underground verification over 196 points. Of the 200 operating points used for underground chart generation, four were disregarded in this verification since they lie at low capacity values of 25 and 50% for low TO values ranging from 0 to 2°C and high TC values ranging from 40 to 50°C. This combination falls outside the anticipated operating range of the underground compressor. These operating points were only important for compressor chart generation since they improve the overall accuracy of the generated trendlines in Table 10 and Table 11.

On average, the underground refrigerant mass flowrate is the most inaccurate amongst the compared results. The high inaccuracies mainly occurred at 25 and 50% operating capacity for combinations of low TO values ranging from 0 to 5°C and high TC values ranging from 45 to 50°C. These operating points fall on the edge of the anticipated operating range of the underground compressor.

3.4 Condenser Model

This section describes the development of the condenser model.

3.4.1 Condenser Component Overview

Figure 17 below illustrates the tube-in-tube, refrigerant-to-water, counterflow ACU condenser, indicating the inlet and outlet points of the working fluids.

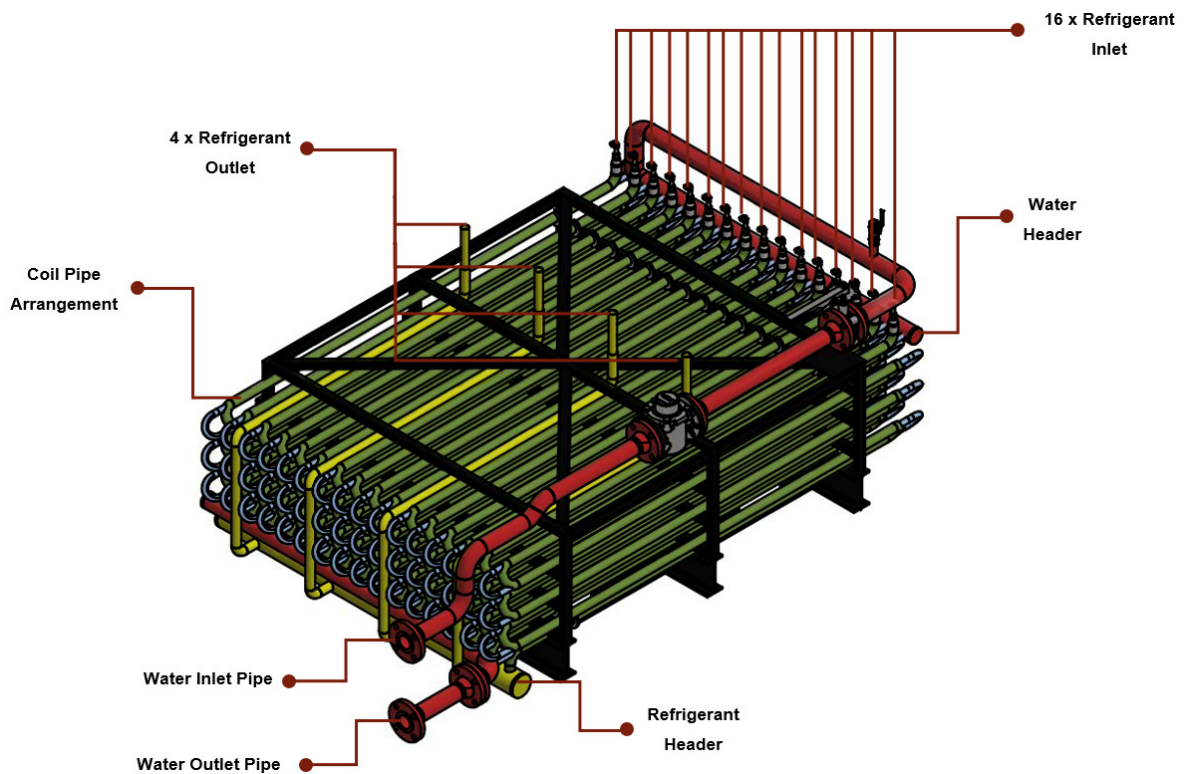


Figure 17: Isometric illustration of the ACU condenser pack (M-Tech Industrial, 2022)

On the waterside, the condenser has a single inlet pipe feeding a header that branches into sixteen parallel circuits which rejoin into a single outlet pipe. On the refrigerant-side, the condenser has sixteen parallel flow circuits, that come together in a condensate collection header before exiting the condenser through four parallel circuits. To simplify discussion, these sixteen parallel tube-in-tube circuits are referred to as condensing circuits. Figure 18 below illustrates a single condensing circuit.

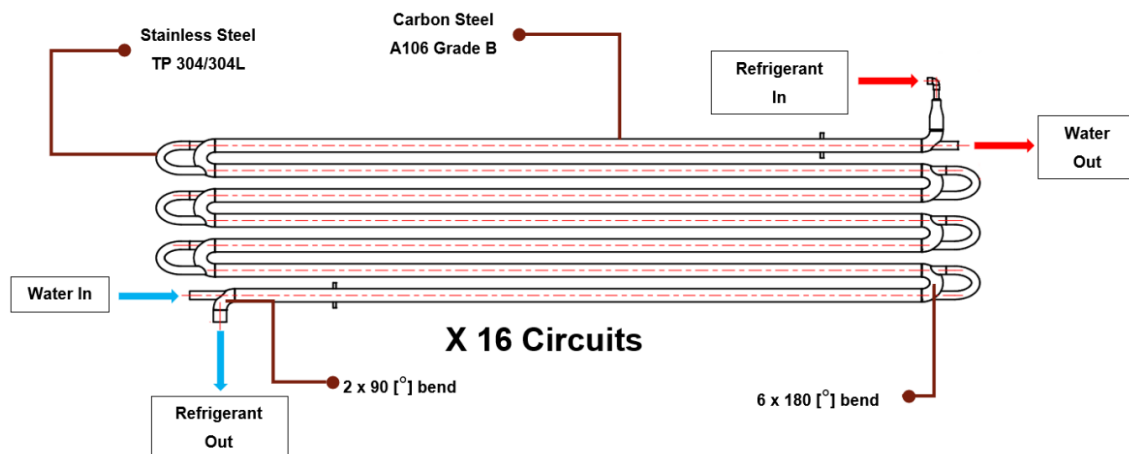


Figure 18: Illustration of an ACU condenser pack condensing circuit (M-Tech Industrial, 2022)

The complex condenser pack in Figure 17 could be simplified to sixteen parallel condensing circuits as illustrated above. These circuits are made up of a stainless steel inner pipe and carbon steel annulus. Each of these circuits contain seven stacked, tube-in-tube sections where the heat transfer takes place. Seven pipes are connected by six 180° bends and two 90° bends to form a single

continuous circuit on the refrigerant-side. On the waterside, seven stainless steel pipes are bent and connected, forming a single continuous circuit. This is a counter flow heat exchanger allowing the condensed refrigerant to collect at the bottom of the condenser.

3.4.2 Developed Condenser Model

Based on the overview done on the ACU condenser pack, key consideration for model development is the exchange of heat between two fluids, which are separated by a conductive pipe wall. Additionally, the presence of the inner pipe constrains the flow area of the annulus, and secondary losses occur due to bends in the circuit. The developed condenser model is based on the condensing circuit illustrated in Figure 18 in Section [3.4.1 Condenser Component Overview](#). The condenser model is illustrated in Figure 19 below.

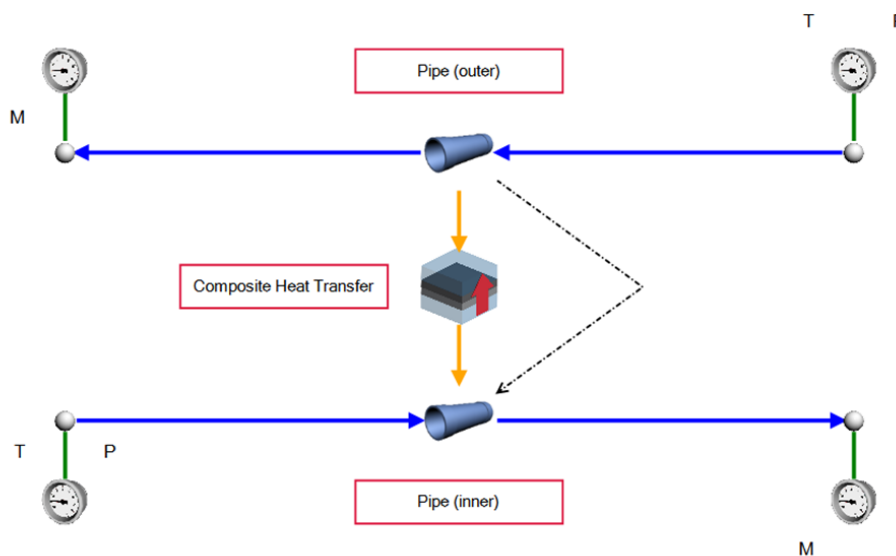


Figure 19: Developed Flownex condenser model

This Flownex flow network consists of two Pipe elements connected by a Composite Heat Transfer element, representing the heat transfer between the refrigerant and water streams in the condenser. The orange arrow indicates the heat transfer from the refrigerant in the outer pipe (annulus) to the water in the inner pipe. The blue arrows of both pipe sections indicate that flow originates from a single node and after passing through the condenser (divided flow between sixteen circuits), re-joins at a singular node. A single Pipe element can be used to represent sixteen parallel circuits as a Flownex feature. Thus, the single inner and outer pipes in Figure 19 convey only a sixteenth of the respective total fluid mass, but the nodes on either side register the total condenser fluid mass passing through.

The condenser flow network involves the representation of sixteen straight, parallel tubes. Upon inspection of Figure 18, it becomes evident there are straight tube sections on the waterside that are not receiving heat from the refrigerant-side, although still contributing to primary and secondary losses. The model is developed accordingly, accounting for the accurate heat transfer length and

the extended length over which losses occur. Roughness relevant to the pipe materials used are applied to both the inner pipe and annulus to account for primary losses. The roughness is retrieved from an internal Flownex material catalogue based on the material specified. In order to account for the secondary losses attributable to bends, K-factors are introduced to both the inner pipe and annulus. The K-factors for the simplified flow network are retrieved from a more complex condenser flow network presented in Figure 20 below.

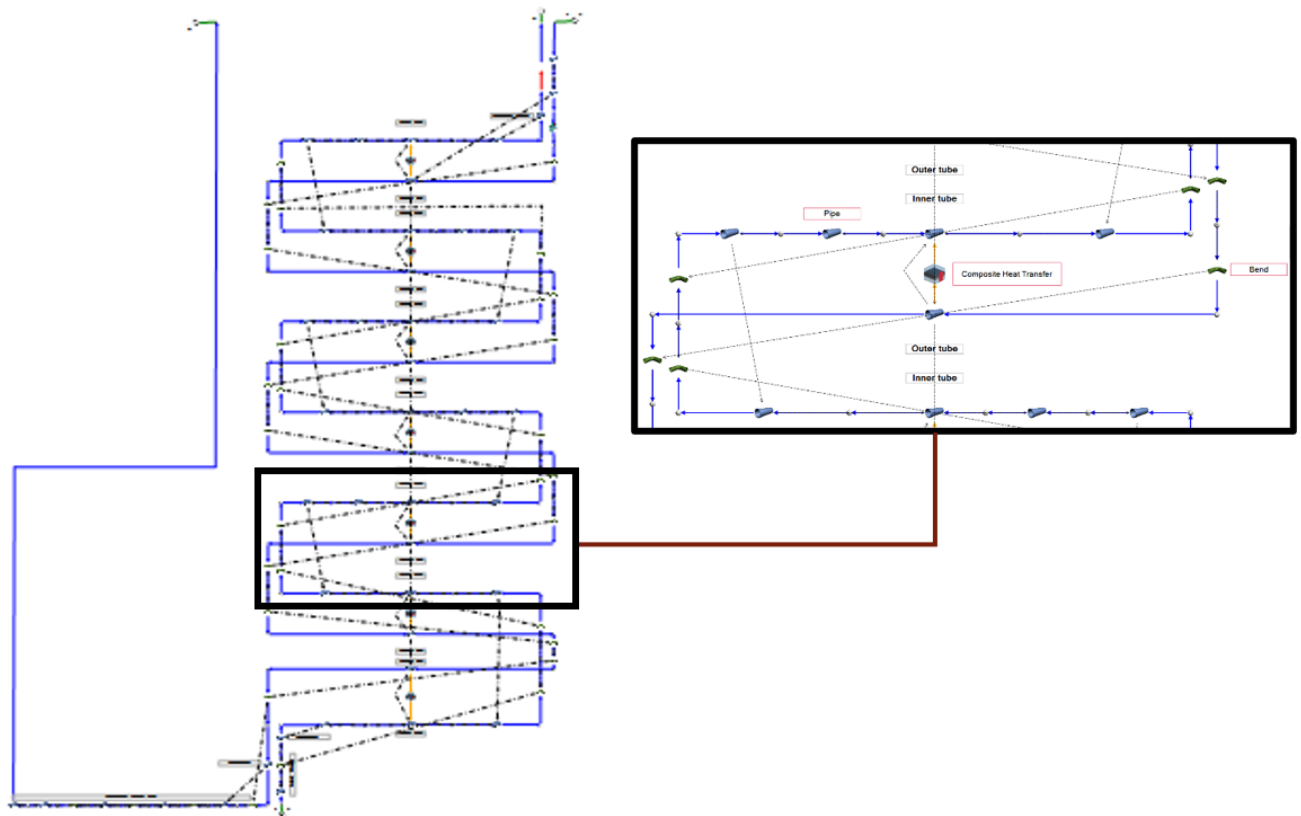


Figure 20: Complex developed Flownex condenser flow network along with an enlarged view

This complex flow network is a detailed representation of the condenser condensing circuit seen in Figure 18, accounting for heat transfer as well as the effect of roughness in pipes and bends. This is the more accurate model to determine the pressure-drop for both the inner pipe and annulus. This complex model is impractical for use in an integrated ACU network due to the significant solving time required. The simple and complex models have the same heat transfer areas and lengths of pipe for primary losses to occur. Implementing the K-values generated by the complex model in the simplified model ensures that secondary losses between the two models are the same, allowing the simplified model to be a likeness of the complex model without compromising on solving time. The roughness and K-values used are available in Appendix A, Section [A.2.1 Condenser Roughness and K-Values](#). A comparison between the results of the simple and complex flow networks is available in Appendix A, Section [A.2.2 Results Comparison of Simple and Complex Condenser Models](#).

3.4.3 Condenser Model Verification

There was no commercial package available to verify the accuracy of the Flownex condenser model, necessitating the development of a verification model. This model was developed using EES and applies the effectiveness-NTU method (Bergman *et al.*, 2011) to calculate heat transfer through a tube-in-tube section. Figure 21 below provides a visual illustration of the operation of the EES model.

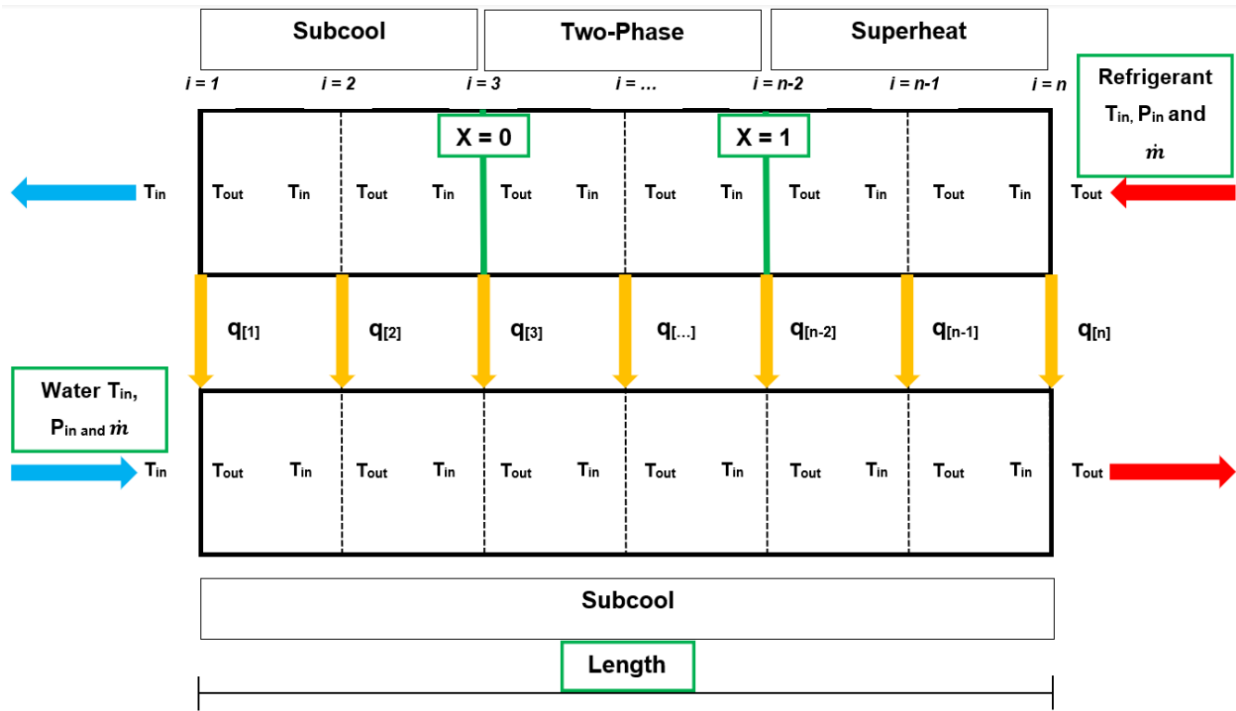


Figure 21: Visual interpretation of the EES condenser model (Source: Author's own depiction)

The red and blue arrows indicate the flow direction and temperature gradient experienced through both tubes. Both tubes are segmented using nodes (i). The condition within an individual segment is assumed to be uniform, thus heat transfer and pressure-drop only take place between segments, where the nodes are located. Similar to the Flownex condenser model, the model assumes that the temperatures as well as liquid and gas velocities over the cross-sectional area of a tube section are distributed equally.

The nine inputs required to solve the system are indicated in green. The purpose of the model is not to design a condenser, thus, since the geometry will not change, geometry was not treated as an input. The refrigerant-side requires the most inputs due to the complicated nature of phase change. The two nodes with quality inputs control where phase change occurs and they therefore adjust the length of superheated, two-phase and subcooled regions accordingly.

Like in Flownex, the Dittus-Boelter empirical correlation was used to estimate the convective heat transfer coefficient for both subcooled and superheated flows. The empirical correlation used in

Flownex to estimate the convective heat transfer coefficient of two-phase flow depends on the flow regime (e.g., bubbly flow, annular flow, slug flow, etc.), fluid properties and geometrical parameters. These correlations are complex, requiring an extensive series of flow maps to solve. Using these correlations may potentially result in a model that takes very long to solve or becomes unstable while solving. For this reason, a simplified approach is taken using a correlation presented by Shah (1979). When the ACU project was still in its early development, this correlation was used in the initial design of the condenser model and was later used again to simulate fluted tube refrigerant-to-water condensers when the ACU condenser packs were to be upgraded to fluted tubes (Rousseau *et al.*, 2003). The liquid heat transfer coefficient is calculated using Dittus-Boelter and is then adjusted using the two-phase quality of the refrigerant as indicated in the following equation:

$$h_{tp} = h_{liq} \left[(1 - x)^{0.8} + \frac{3.8x^{0.76}(1 - x)^{0.04}}{Pr^{0.38}} \right] \quad [3.9]$$

- h_{liq} = Liquid convective heat transfer coefficient [W-m²-K],
- h_{tp} = Two-phase convective heat transfer coefficient [W-m²-K],
- x = Fluid quality [-],
- Pr = Prandtl number [-],

Similar to Flownex, the Darcy-Weisbach equation was used to estimate the pressure-drop for all flow regions. The Moody chart was used to estimate the friction factor for subcooled and superheated while Lockhart-Martinelli accounted for two-phase flows.

Table 13 below summarises the scope of the inputs used for condenser model verification. The ‘State’ column in Table 13 indicates the thermal state of refrigerant in each verification case. The verification is performed for the refrigerant-side as superheated (SH), two-phase (TP) subcooled (SC) and a combination of all three states (All). The verification is conducted in this way since the heat transfer and pressure-drop correlations change for different thermal states and individually inspecting each state would expose errors.

Table 13: Inputs Used for Condenser Verification

State		Refrigerant				Water			Tube Length [m]
		Inlet Pressure [bar]	Inlet Temperature [°C]	Mass Flowrate [kg/s]	Quality [-]	Inlet Pressure [bar]	Inlet Temperature [°C]	Mass Flowrate [kg/s]	
SH	Min	15.90	80.0	0.024	x>=1	2.5	18.0	0.100	1.00
	Max	23.00	120.0	0.096			32.2	0.506	
TP	Min	19.40	48.3	0.033	1>x>0		24.0	0.100	4.00
	Max	20.00	49.3	0.050			36.0	0.400	
SC	Min	21.00	30.0	0.029	0>=x		18.0	0.100	7.55
	Max	23.00	47.0	0.096			32.2	0.506	

State		Refrigerant				Water			Tube Length [m]
		Inlet Pressure [bar]	Inlet Temperature [°C]	Mass Flowrate [kg/s]	Quality [-]	Inlet Pressure [bar]	Inlet Temperature [°C]	Mass Flowrate [kg/s]	
All	Min	21.00	80.0	0.041	$1 \geq x \geq 0$		18.0	0.225	12.55
	Max	23.00	120.0	0.060			32.2	0.506	

All inputs reflect realistic operating ranges for a single condenser tube in surface and underground conditions. The refrigerant inlet pressure ranges are based on the compressor break points discussed in Section [3.3.1 Compressor Component Overview](#). The refrigerant inlet temperature and mass flowrate ranges were based on observations made from compressor chart generation data used for compressor model development.

The water inlet pressure is a redundant input since the water never undergoes phase change, thus the specified minimum water inlet pressure of the manufactured condenser pack was used. The water inlet temperature and mass flowrate ranges were derived from FAT and mine data. Testing the three phases of flow separately requires the refrigerant flow to never change phase and so the choice of tube length for each state tested had to ensure this would be the case.

The verification of the two-phase and combined phase models was not performed over the entire anticipated refrigerant mass-flow range since the correlation by Shah (1979) for heat transfer coefficient for two-phase flow becomes inaccurate for greater mass flowrates. Despite this, the verification approach and applied ranges ensure for a thorough verification of the Flownex condenser model.

For verification the EES model results for fluid temperature changes [°C], pressure-drops [bar] for both fluid streams and the heat transfer [kW] were tested against Flownex reference values. The Flownex model is used as the reference as it relies on a more detailed fluid file.

Verification of the fluid temperature changes, and heat transfer tested if the developed models obeyed the energy conservation equation and the pressure-drops the momentum conservation equation. Mass conservation was achieved specifying mass flowrate as an input for both fluid streams.

The developed EES models are available in Appendix A, Section [A.2.3 Condenser EES Verification Models](#). A comprehensive parameter table containing input values for the Flownex and EES models, as well as their respective results, can be found in Appendix A, Section [A.2.4 Condenser Verification Results](#). The average and maximum errors for the verification of the accuracy of the developed condenser model for different thermal states of

refrigerant are shown in Table 14, along with the references to equations used to calculate the errors for each tested property.

Table 14: Condenser Model Verification Results

Model Phase		Refrigerant Temperature Change [%]	Water Temperature Change [%]	Water Pressure-Drop [%]	Refrigerant Pressure-Drop [%]	Heat Transfer [%]
		Eq. [3.8]	Eq. [3.8]	Eq. [3.7]	Eq. [3.7]	Eq. [3.7]
SH	Avg Error	1.21	0.81	0.13	1.86	0.84
	Max Error	1.89	1.56	0.27	3.29	1.54
TP	Avg Error	2.29	2.39	0.07	8.96	2.38
	Max Error	6.08	10.28	0.11	15.12	10.22
SC	Avg Error	0.17	0.16	0.09	0.07	0.14
	Max Error	0.26	0.70	0.11	0.36	0.18
All	Avg Error	0.65	2.70	2.06	4.18	2.76
	Max Error	2.21	5.75	3.90	11.97	5.75

The EES condenser model results show a good relation to the results of the Flownex model, with all compared results exhibiting average errors within 2.76%, excluding refrigerant pressure-drop. Each model was tested over 25 points, allowing five boundary conditions of each model to be varied individually for five times ($5 \times 5 = 25$). The entire verification was thus conducted over 100 points ($25 \times 4 = 100$). Varying one variable at a time made it simple to identify the cause of significant errors.

The two-phase model had the greatest heat transfer error, occurring at the highest refrigerant mass flow value tested, illustrating the inaccuracy of the correlation for heat transfer coefficient presented by Shah for high refrigerant mass flowrate (Shah, 1979). The two-phase model also had the greatest pressure-drop error occurring at the lowest refrigerant inlet temperature tested. The outlet quality was almost at the saturation point, thus the Flownex model possibly experienced some subcooling resulting in the discrepancy between the compared results. The high average refrigerant pressure-drop error is a result of a discrepancy between the Flownex and EES fluid files for R407C. It has been observed that the two software packages generate different refrigerant qualities for similar temperature and pressure inputs for the two-phase region. This results in different density and refrigerant velocity estimation ultimately resulting in the pressure-drop discrepancy.

3.5 Expansion Valve Model

This section describes the development of the expansion valve model.

3.5.1 Expansion Valve Component Overview

Figure 22 below illustrates the functionality of the orifice inside the straight charged, externally equalized expansion valves being used on the ACU.

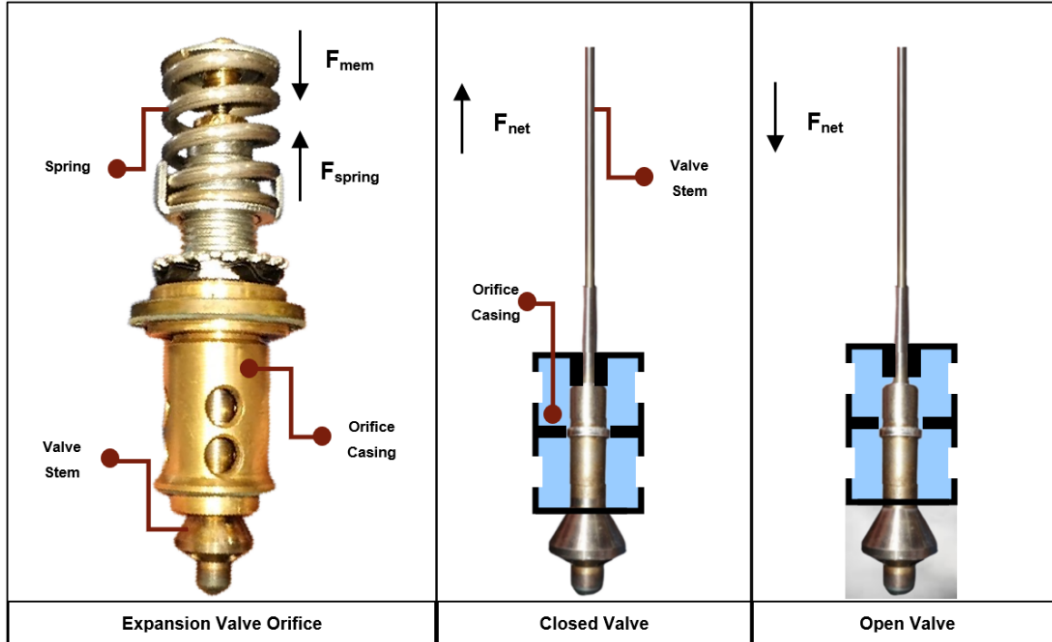


Figure 22: Illustrating the functionality of the ACU expansion valve orifice (Source: Author's own depiction)

In this control valve system, the movement of the valve stem is influenced by two main forces: The force applied by the membrane of the sensing bulb (F_{mem}) and the force applied by an adjustment spring (F_{spring}). The membrane of the sensing bulb displaces the valve stem, while the adjustment spring provides resistance against this displacement, resulting in the total force (F_{net}) experienced by the valve stem.

The black and blue diagram drawn over the valve stem in the 'Closed Valve' and 'Open Valve' subfigures of Figure 22 represents a clip section through the orifice casing and indicates how the valve stem seats on the casing. A clip section is where a portion of an object or assembly is illustrated using a cutting-plane line to showcase internal details or features that would otherwise be hidden from view. When the adjustment spring exerts a greater force than the membrane, the valve is closed. In this position, the orifice casing prevents fluid flow. The sensing bulb is connected to the outlet of the evaporator in the vapour compression cycle and applies the pressure experienced at this point as the force F_{mem} . The greater this flow restriction, the greater the superheat achieved in the refrigeration cycle, increasing the force experienced by the membrane of the sensing bulb (Eames *et al.*, 2014).

As the superheat increases, the membrane eventually applies a greater force than the adjustment spring, overcoming the resistance and pressing the valve stem downward. As a result, flow channels are formed, vaporising the fluid under the immense pressure difference experienced. Greater fluid flow implies there is more refrigerant to absorb heat and thus the superheat achieved in the refrigeration cycle decreases (Eames *et al.*, 2014). In this way, the sensing bulb and spring regulates refrigerant flow through the vapour compression cycle.

3.5.2 Developed Expansion Valve Model

Based on the overview done on the ACU expansion valve and the nature of the Flownex simulation environment, the valve was modelled using a Restrictor with Discharge Coefficient element to represent a throttling process. The Restrictor with Discharge Coefficient element only applies conservation of energy and momentum equations since it assumes negligible fluid mass. The element is isenthalpic assuming negligible thermal inertia for the fluid in the valve and negligible heat loss to the surrounding environment. The total pressure-drop through the element is equivalent to the dynamic pressure in the throat area, with no static pressure recovery downstream of the throat. It is applicable to gas, liquid and two-phase flows, with the ability to handle choking in the element's throat area for gas and two-phase flows.

The implementation of the Restrictor with Discharge Coefficient element is different for steady state and transient simulations. Figure 23 below illustrates the use of the element in a simple vapour compression cycle under steady state and transient simulation conditions.

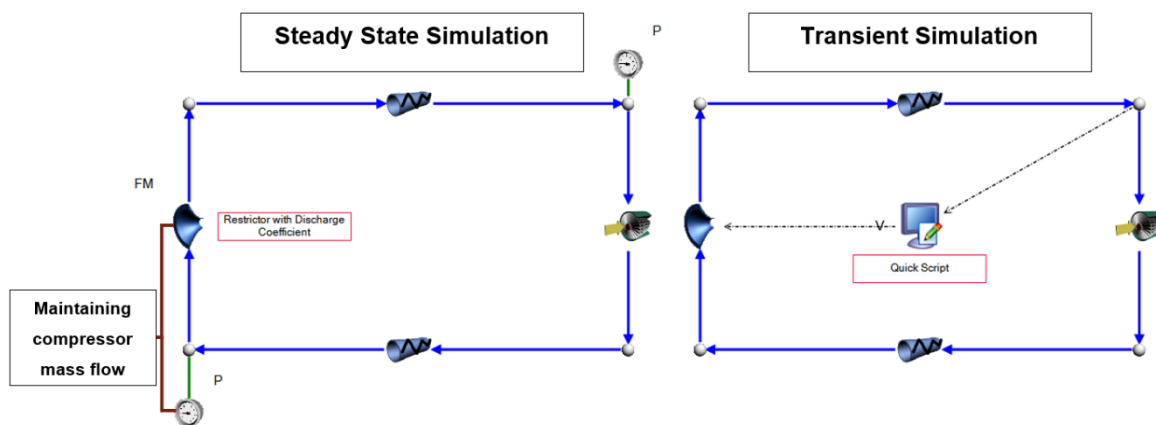


Figure 23: Implementation of the developed Flownex expansion valve model

Under steady state conditions the flow network uses two pressure boundaries to solve: the suction pressure of the compressor and the inlet pressure to the expansion valve. The opening of the Restrictor with Discharge Coefficient element is changed to satisfy the pressure exerted by its upstream Boundary Condition element and the mass flowrate passing through the compressor. The flow network has to converge under steady state solving before transient simulation is attempted.

No fixed boundaries are applied in transient simulation, to allow the flow network to regulate itself. A Quick script element controls the opening of the Restrictor with Discharge Coefficient element to maintain a set SH.

The complete ACU model contains four expansion valves in parallel receiving refrigerant from the four parallel condenser outlets. Assuming the expansion valve's throttling effect is isenthalpic, there is no need for verification as the component does not transfer heat to or from the refrigeration cycle.

3.6 Evaporator Model

This section describes the development of the evaporator model.

3.6.1 *Evaporator Component Overview*

Figure 24 below illustrates a single epoxy-coated, wavy aluminium-finned, rifled bore copper-coiled, air-to-refrigerant crossflow evaporator unit.

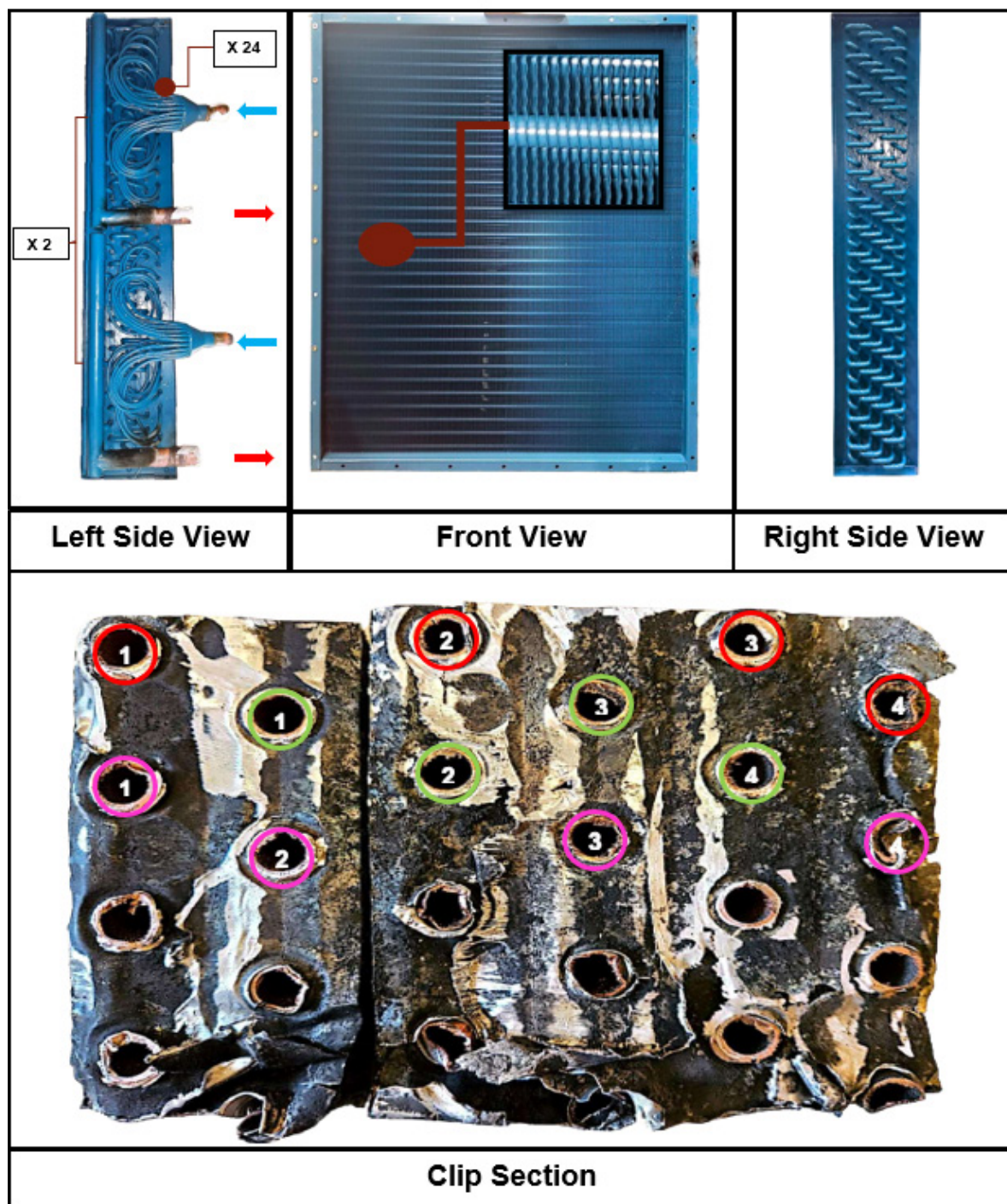


Figure 24: Side views, a front view and a clip section of the ACU evaporator unit (Source: Author's own depiction)

The ACU is equipped with two evaporator units, the second a mirrored image of the unit illustrated in Figure 24 above. The two evaporator units each contain two stacked evaporator packs. The blue arrows in the 'Left Side View' subfigure of Figure 24 indicates the inlet flow of two-phase refrigerant from the expansion valves into the evaporator packs. The refrigerant flow is subdivided into 24 circuits that pass through the identical evaporator packs before rejoining into headers as superheated refrigerant that exits the evaporator packs at the red arrows. The 'Front View' subfigure illustrates the vertical, epoxy coated, aluminium fins through which air is passed and

cooled. The 'Right Side View' subfigure illustrates how the passes of the 48 circuits of the evaporator unit are connected.

The 'Clip Section' subfigure illustrates a clip section through an old evaporator unit exposing the rifled bored copper tubing of the passes. Each of the 48 circuits contains four passes, arranged according to either the red, green or pink colour scheme. Refrigerant flows from Pass 1 to 4 and air moves from right to left to create a crossflow effect.

3.6.2 Developed Evaporator Model

Based on the overview done on the ACU evaporator and the nature of the Flownex simulation environment, the evaporator was modelled using a Finned-Tube Heat Exchanger element. This element is a generic tool for simulating heat transfer between two fluid streams, ideal for crossflow heat exchangers that have two-phase liquid flow on the tube-side and a gas on the fin-side. The element divides the heat exchanger into a two-dimensional array of smaller crossflow heat exchanger elements and calculates the heat transfer in each element. Figure 25 below illustrates the evaporator model.

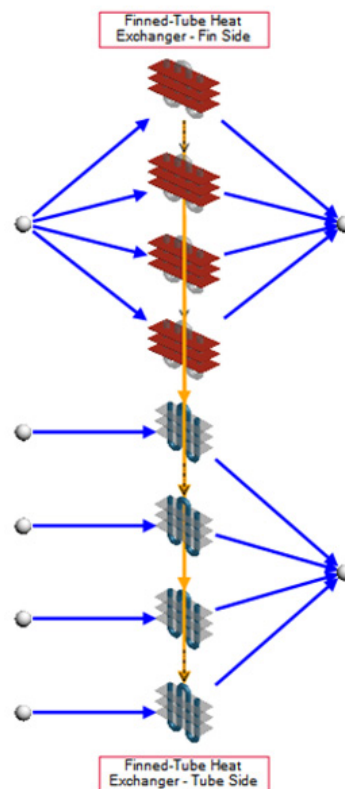


Figure 25: Developed Flownex evaporator model

The fins - highlighted in red - signify the fin-side, while the tubes - highlighted in blue - indicate the tube-side. The orange arrows represent the heat transfer from the fin-side of each evaporator pack to its corresponding tube-side.

On the fin-side, the blue arrows emanate from a common node due to the uniformity of ambient conditions. The blue arrows exiting the fin-side heat exchanger elements re-join, as the air downstream of the evaporator mixes in a chamber before the fan blows it out, resembling an isobaric process.

On the tube-side, the blue arrows emanate from separate nodes, since the flow is divided when exiting the expansion valves. The blue arrows exiting the tube-side heat exchanger elements re-join at a single node before entering the compressor.

The model relies on the Colburn J-factor to calculate the heat transfer coefficient between the fins. The Colburn J-factor is calculated as follows.

$$j = 0.14Re^{-0.328} \left(\frac{Pt}{Pl} \right)^{-0.502} \left(\frac{s}{d_o} \right)^{0.0312} \quad [3.10]$$

j	=	Colburn J-factor [-],
Re	=	Reynolds number of the air [-],
Pt	=	Lateral tube pitch [m],
Pl	=	Longitudinal tube pitch [m],
s	=	Gap between two fins [m],
d_o	=	Outside diameter of the tube [m],

The equation was used to generate a chart relating Reynolds number of the air to Colburn J-factor and was applied to the fin-side of the model.

Fin-side primary losses were calculated using a friction factor. The friction factor can be expressed as a function of air Reynolds number, typically resembling a logarithmic function. To obtain the specific function for the evaporator, five points of verification data - spanning over the entire range of anticipated air volume flow - was used. The source of this verification data is to be discussed in Section [3.6.3 Evaporator Model Verification](#). The resulting chart was applied to the fin-side.

On the tube-side primary losses are calculated using roughness related to rifflled bore copper tubing. Secondary losses were accounted for applying a K-value on the tube-side. The value used satisfied the refrigerant pressure-drop of a data point of the verification data. The calculations of all inputs to the Finned-Tube Heat Exchanger element are provided in Appendix A, Section [A.3.1 Evaporator Geometry Input Calculations](#).

3.6.3 Evaporator Model Verification

The ACU evaporators are designed and manufactured by HC Heat-Exchangers, who has also developed software for simulating evaporator performance in various scenarios. This software was used to generate reference results for verification.

Figure 26 below contains a flow network representing one of the dual evaporators of the ACU. This was used to generate test values for the Flownex evaporator model verification.

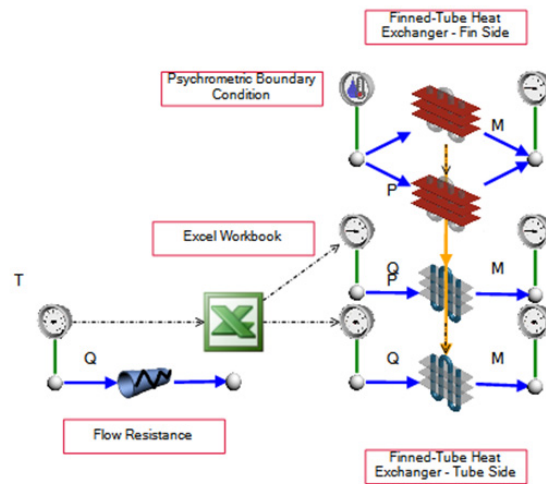


Figure 26: Flownex flow network used for evaporator model accuracy verification

In Figure 26, the tube-side pressure and quality are specified at the inlet boundaries, and the refrigerant mass flowrate is specified at the outlet boundary. The tube-side uses the same correlations for heat transfer coefficient and pressure-drop discussed for the developed condenser model's refrigerant-side in Section [3.4.3 Condenser Model Verification](#).

A Psychrometric Boundary Condition element was implemented at the inlet of the airside to set the pressure, dry bulb temperature and relative humidity. At the outlet boundary condition, air mass flowrate is specified.

The developed Flownex model and HC Heat-Exchangers software were subjected to the same input conditions. Table 15 below summarises the scope of the inputs used for evaporator model verification for surface and underground conditions.

Table 15: Inputs Used for Evaporator Verification

Strata		Air				Refrigerant			
		Inlet Pressure [bar]	Inlet Temperature [°C]	Inlet Relative Humidity [%]	Mass Flowrate [kg/s]	TO [°C]	SH [°C]	Inlet Quality [-]	Mass Flowrate [kg/s]
Surf	Min	0.8625	19.0	20	4.65	0.5	0	0.2	0.331
	Max	0.8625	28.0	40	5.25	5.0	7	0.2	0.605
Und	Min	1.1397	28.3	70	2.00	5.0	5	0.2	0.390
	Max	1.1397	38.0	90	2.80	12.5	12	0.2	1.021

The surface air inlet pressure relates to the ambient air pressure of Potchefstroom, where the surface validation data was collected. The underground air inlet pressure relates to the limit imposed by the HC Heat-Exchangers software at 1000 m below sea level. Considering that air cooling is essential in South African mines at depths exceeding 1500 m (Van Eldik, 2006) and that some mines are more than 4 km deep in the Witwatersrand area (Njini, 2021), an acceptable elevation verification range lies within 300 m above and 3800 m below sea level.

The ranges for underground inlet air temperature and humidity are higher compared to surface conditions as generally observed in these environments. Air mass flowrates were determined by relating known power consumption of the FAT fan and the underground fan to air mass flowrate using fan charts. Based on the fan power consumption data, the required underground air mass flowrate is lower than that at the surface. A possible explanation for this is that denser air underground enables more efficient heat transfer with less airflow.

The refrigerant's surface TO is lower compared to underground conditions since warmer, more humid air adds more energy to the refrigeration cycle and thus the ACU operates at elevated TO underground. The underground environment allows for greater SH due to better heat transfer, compared to surface conditions.

Refrigerant mass flowrates and inlet quality are the only two inputs that were not specifiable in the HC Heat-Exchangers software. The software returns a relevant refrigerant mass flowrate based on an internal energy balance and maintains an inlet quality of 0.2 for the specific evaporator model used on the ACU.

For verification, the Flownex model results for the heat transfer [kW], airside temperature change [°C] and pressure-drops [bar] for both fluid streams were tested against HC Heat-Exchangers software reference values. Verification of the heat transfer and airside temperature change tested whether the developed model obeyed the energy conservation equation, and the pressure-drops tested the momentum conservation equation. Mass conservation was achieved specifying mass flowrate as an input for both fluid streams in the Flownex model.

The parameter tables used are available in Appendix A, Section [A.3.2 Evaporator Verification Results](#). The average and maximum errors for the verification of the accuracy of the developed evaporator model for surface and underground conditions are shown in Table 16, along with the references to equations used to calculate the errors for each tested property.

Table 16: Evaporator Model Verification Results

Strata		Heat Transfer [%]	Airside Temperature Change [%]	Airside Pressure-Drop [%]	Refrigerant-Side Pressure-Drop [%]
		Eq. [3.7]	Eq. [3.8]	Eq. [3.7]	Eq. [3.7]
Surf	Avg Error	2.57	3.31	1.46	2.12
	Max Error	7.15	5.86	5.18	5.94
Und	Avg Error	4.42	3.26	1.99	2.30
	Max Error	9.07	7.52	8.50	8.49

The Flownex evaporator model results show a good relation to the results of the HC Heat-Exchangers program, with all compared results exhibiting average errors within 4.42%. The verification was performed over 51 points of which 22 related to surface and 29 to underground conditions. The evaporator verification followed the same approach as the condenser verification, varying one boundary condition at a time. Underground heat transfer exhibited the highest error occurring for very high SH. The high SH value is outside the anticipated SH range underground and is only included to illustrate the behaviour of the developed model.

3.7 Fan Model

This section describes the development of the fan model.

3.7.1 Fan Component Overview

The ACU makes use of an axial fan to facilitate the movement of air. Figure 27 illustrates the ACU in the FAT station, equipped with a 9.2 kW fan.



Figure 27: Axial fan fitted to the front of an ACU (Source: Author's own depiction)

The fan draws air in through the evaporator fins and subsequently propels the cooled air out through the front of the ACU.

3.7.2 Developed Fan Model

Based on the overview done on the ACU fan and the nature of the Flownex simulation environment, the fan was modelled using a Variable Speed Pump element linked to a Flow Resistance element. Figure 28 below illustrates the fan model incorporated with the evaporator model.

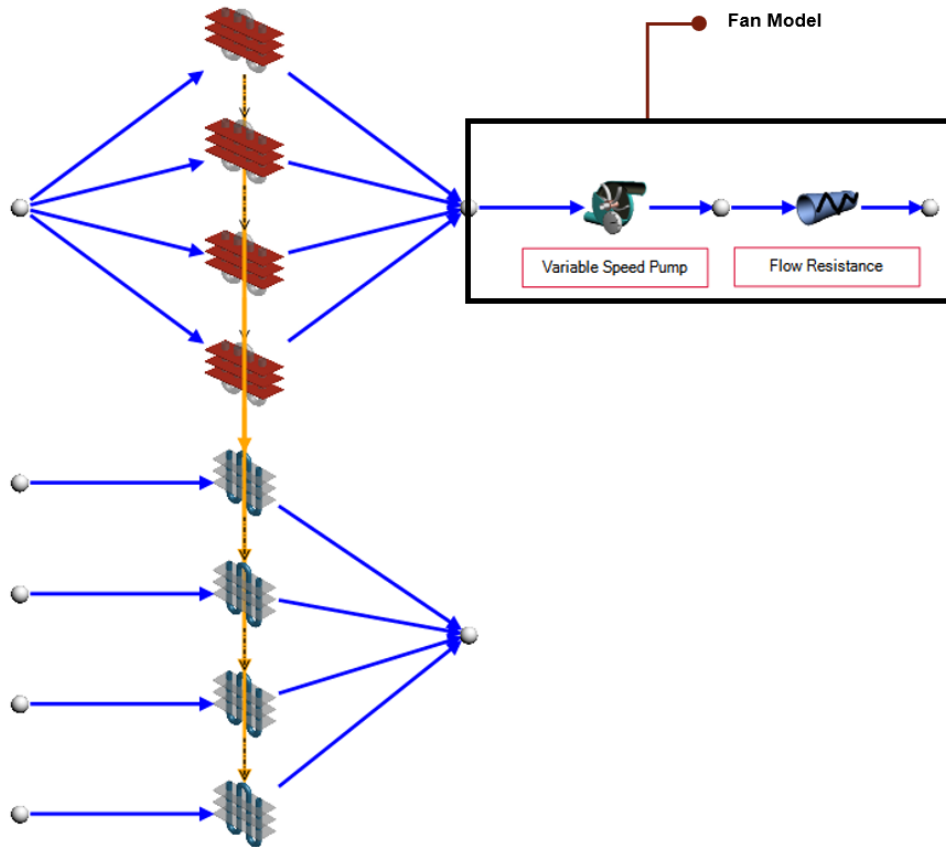


Figure 28: Developed Flownex fan model incorporated with the evaporator model

Similar to the Compressor element, the Variable Speed Pump element requires a chart to operate. Fan performance data is obtained from manufacturers to generate charts relating pressure rise and efficiency to volume flow at a reference density, in order to apply density scaling. The power consumption of the fan is dependent on the load experienced when pushing air, thus the volume flow is throttled to bring the power consumption of the fan model in line with the known power consumption using a Flow Resistance element.

The fan model uses interchangeable fan charts and flow admittance factors to facilitate modelling of different fans as required. The FAT station uses a 9.2 kW fan, while a 22 kW fan is utilized underground once the ACU is commissioned. The generated charts are available in Appendix A, Section [A.4.1 Fan Charts](#).

3.7.3 Fan Model Verification

Figure 29 below contains the flow network used to generate test values for the Flownex fan model verification.

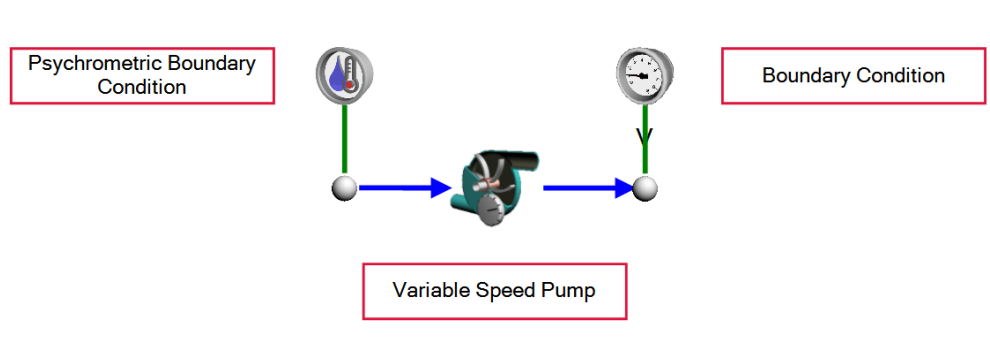


Figure 29: Flownex flow network used for fan model accuracy verification

The Psychrometric Boundary Condition element is assigned a temperature and relative humidity to satisfy the reference density specified by the manufacturer of the fan. The Variable Speed Pump element is assigned a pump chart, and the correct rotational speed as specified by the fan manufacturer.

The fan model is verified for both a 9.2 kW surface fan and a 22 kW underground mine fan over the entire volume flow range specified by the manufacturers. The surface fan model was verified for a volume flow ranging from 2.88 to 11.08 m³/s and the underground fan model from 1.04 to 11 m³/s.

For verification, the Flownex model results for the fan efficiency [%] and pressure-drop [bar] were tested against fan manufacturer reference values. Verification of the fan efficiency tested whether the developed model obeyed the energy conservation equation, and the pressure-drop tested whether the momentum conservation equation was obeyed.

Steady-state calculations are performed using parameter tables for both surface and underground conditions, as illustrated in Appendix A, Section [A.4.2 Fan Verification Results](#). The average and maximum errors for the verification of the accuracy of the developed fan model for surface and underground conditions are shown in Table 17, along with the references to equations used to calculate the errors for each tested property.

Table 17: Fan Model Verification Results

Strata		Efficiency [%]	Pressure-Drop [%]
		Eq. [3.7]	Eq. [3.7]
Surf	Avg Error	0.14	0.20
	Max Error	0.40	0.52
Und	Avg Error	0.68	0.81
	Max Error	1.54	2.38

The Flownex fan model results show a good relation to the results of the fan manufacturer data, with all compared results exhibiting average errors within 0.81%. The verification was performed over 29 points of which 11 relate to surface and 18 to underground conditions.

3.8 Interconnecting Piping Model

This section describes the development of the interconnecting piping model.

3.8.1 Interconnecting Piping Overview

The main components of the ACU refrigeration cycle are connected using copper piping and bends. Figure 30 below illustrates the various piping sizes used on the ACU.



Figure 30: Copper piping sizes used on the ACU (Source: Author's own depiction)

3.8.2 Developed Interconnecting Piping Model

Based on the overview done on the ACU interconnecting piping and the nature of the Flownex simulation environment, the piping system was modelled using Pipe elements and Bend elements that accurately divide and reconnect flow, mimicking the physical ACU layout.

The piping is included on the developed model to obtain realistic pressure-drops and maintain the appropriate refrigerant charge for the system. The effects of elevation differences between tube sections are assumed to be minimal, thus the effects thereof are neglected.

The pipes and bends used in the model are sized, modelled from appropriate materials, and scheduled in accordance with the physical model to achieve accuracy. Realistic roughness values, retrieved from an internal Flownex catalogue, are applied to each pipe section to further enhance the model's realism.

In a broad sense, the heat loss or gain from the interconnecting piping is insignificant. The interconnecting piping is enveloped in stagnant air that act as an isolator due to air's low heat capacity, consequently there is minimal heat loss or gain from the interconnecting piping. For this reason, the model is created with the assumption that interconnecting piping is isenthalpic, thus no verification is required.

3.9 Integrated ACU Model

The individually verified Flownex components are linked to complete the vapour compression cycle of the ACU. Figure 31 below illustrates the integrated ACU model.

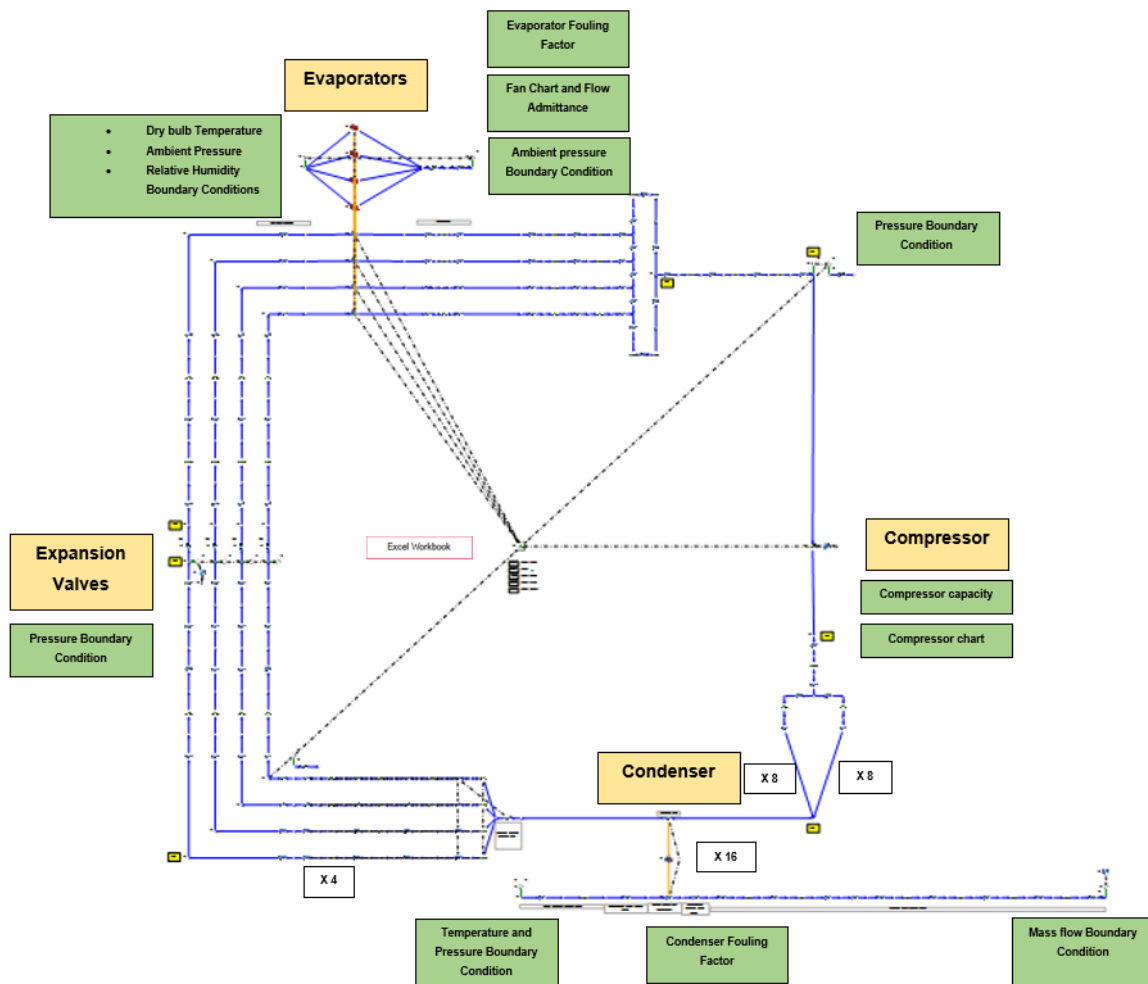


Figure 31: Integrated Flownex ACU model

Labelled in yellow, the compressor, condenser, expansion valves and evaporators form a complete vapour compression cycle. All the interconnecting piping of the system has been modelled in

accordance with the manufactured ACU. Note how the flow distribution changes throughout the system. The compressor discharge flow distributes into two circuits with headers that each divide into eight circuits that feed the sixteen refrigerant inlets on the condenser. Flow exits through the four outlets on the condenser, moving in four parallel circuits through the expansion valves and evaporators before they connect into a single circuit on the suction side of the compressor.

All variable inputs to the system required for the model to solve in steady state are labelled in green. On the refrigerant-side, the model relies on two pressure boundaries to solve, the first being the suction pressure of the compressor and the other, the upstream pressure that regulates expansion valve opening. Furthermore, the compressor charts, compressor capacity and fouling factors for the condenser and evaporators have to be specified. On the waterside of the condenser, a temperature and pressure boundary are specified on the inlet and a mass flowrate boundary is specified on the outlet. On the airside of the evaporator the dry bulb temperature, ambient pressure and relative humidity is specified along with the ambient pressure on the outlet side, since air is extracted and returned to the same body of air in an isobaric process. Furthermore, the fan chart and the flow admittance from the fan must be specified. An Excel Workbook element is introduced in the centre of the model to summate the total heat transfer taking place in the four evaporator packs.

3.9.1 *Transient Simulation*

To solve for transient simulation, the model must first have solved in steady state. For transient solving, all boundary conditions on the refrigeration cycle are removed, allowing the cycle to regulate itself based on the heat added and removed through the heat exchangers. For the cycle to regulate itself, the expansion valves have to maintain a constant SH by means of throttling the refrigerant flow and the compressor has to apply infinite capacity control.

These conditions are satisfied using Quick Script elements that contain custom C# codes that interpret inputs from the system and apply a response. These scripts are illustrated in Appendix C, Section [C1. Quick Scripts](#).

The expansion valve Quick Script element receives SH input from the compressor suction side which it compares against a setpoint value. When generated SH is less than the setpoint, the valve opening closes to reduce refrigerant mass flowrate. Less refrigerant has less capacity to absorb heat in the evaporators, thus this refrigerant becomes more superheated. Likewise increasing the valve opening would lead to reduced SH if the generated SH was higher than the setpoint value. An assumed gain value of 0.0001 is multiplied by the difference of the setpoint SH and the actual superheat. This value is added to or subtracted from the valve opening [0-1].

The compressor Quick Script element receives a compressor suction and discharge gauge pressure input and compares it to set limit values. These limits include an upper and a lower limit that represent the bounds of the set control range discussed in Figure 13 of Section [3.3.1 Compressor Component Overview](#). If the discharge gauge pressure exceeds the upper break point, compressor capacity decreases linearly over time until the discharge pressure is below the upper limit or the minimum compressor capacity of 25% is met. If the discharge gauge pressure falls below the lower break point, compressor capacity increases linearly over time until the discharge pressure is above the lower limit or the maximum compressor capacity of 100% is met. It was assumed the compressor slide valve takes 20 s to move from 25 to 100%. Furthermore, these limits include trip limits. If the compressor suction gauge pressure falls below the lower trip limit or the discharge gauge pressure above the higher trip limit, the transient simulation is terminated.

3.10 Summarising the Model Development Chapter

The model development chapter presents a detailed account of building a comprehensive simulation model for the ACU in the Flownex simulation environment. The chapter explains why Flownex was ideal to do model development, introducing fundamental Flownex terminology and concepts required to understand the development process that ensued.

The chapter continues providing an overview and modelling the main ACU components, providing and capturing essential insights into their characteristics and functionality. The compressor was modelled as a Compressor element that interprets data from a compressor chart. The section established the need for the developed ACU model to operate in both surface and underground conditions. The fundamental differences between these strata are highlighted introducing the need for developing separate compressor charts for these strata.

The condenser was modelled as two pipes exchanging heat through a conductive pipe wall, accounting for primary and secondary losses. The section further explained how the model relates to the ACU condenser pack with regards to split circuits, heat transfer, primary losses and in particular secondary losses due to bends.

The expansion valve was modelled as a Restrictor with Discharge Coefficient element, throttling refrigerant in an isenthalpic process. The section explained how the implementation of the model was different for steady state and transient simulations. For steady state solving, the model relies on an upstream pressure boundary condition to determine the valve opening, maintaining the mass flowrate set by the compressor. For transient solving no boundaries were applied and the model regulates itself freely.

The evaporator was modelled as a Finned-Tube Heat Exchanger element for simulating heat transfer between two fluid streams, ideal for crossflow heat exchangers that has two-phase liquid flow on the tube-side and a gas on the fin-side. The section further explained how the model relates to the ACU evaporator with regards to split circuits, heat transfer, primary losses and secondary losses.

The fan was modelled as a Variable Speed Pump element linked to a Flow Resistance element, operating on a fan chart and volume flow calibration. The section goes on to explain how the model relates to the ACU fan with regards to power consumption and volume flow.

The accuracy of all the individually developed models were verified using independent software before being joined using interconnecting piping. This piping is also considered to achieve realistic refrigerant charge and pressure-drop between components. Self-regulation of the refrigeration cycle in transient simulations is achieved using Quick Script elements containing custom coding.

The following chapter aims to validate the accuracy of the integrated ACU model for surface and underground conditions.

CHAPTER 4: INTEGRATED ACU MODEL VALIDATION

The previous chapter developed and verified the accuracy of individual models of the main ACU components and connected them to create an integrated ACU model. In this chapter, the accuracy of the integrated ACU model is validated for both surface and underground conditions.

4.1 Surface Validation

The surface validation of the Integrated ACU model was performed applying the variable inputs described in Section [3.9 Integrated ACU Model](#). Among these variable inputs, fouling factors were disregarded since the ACU is newly manufactured for the available surface validation data. Surface validation data supplied values to the required variable inputs of the integrated model. The results generated by the model were then compared to the validation data.

Over a series of three days, surface validation data was collected for three separate ACUs during FAT tests. In addition to the onboard logger data, the measurements illustrated in Figure 32 below were recorded manually during the FAT tests.

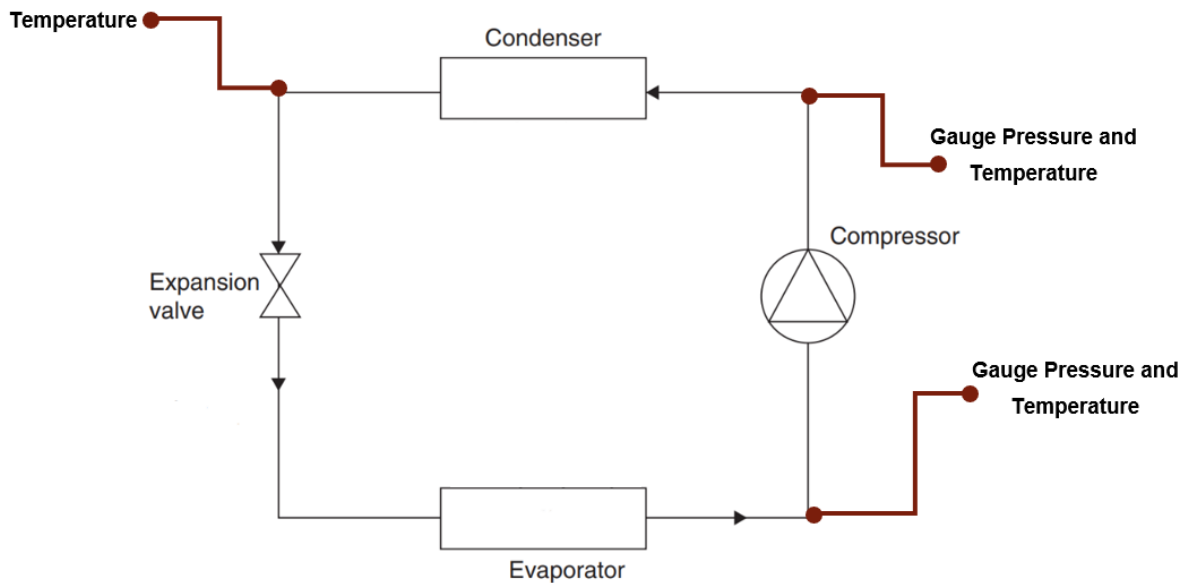


Figure 32: Points of measurement taken during the FAT tests.

The pressures were collected from the compressor suction and discharge gauges and the temperatures were measured using an infrared temperature gun. Information regarding the measurement instrumentation used during the FAT test is available in the Appendix B, Section [B.1 Measurement Instrumentation](#).

Pinpointing the compressor capacity for surface operation is challenging. The electrical supply at the FAT station is insufficient to fully power the ACU. Due to this, the ACU solenoid valves are reconnected in a manner to prevent reaching 100% operation. This reconfiguration of the solenoid valves is not a standard practice, and there is no documentation in Bitzer literature regarding its effects. Furthermore, the compressor has no physical indicators to indicate operating capacity. Consequently, the compressor capacity would be an unknown to solve for in the surface validation process.

To determine the operating capacity of the compressor during each of the FAT tests, the manually collected measurements of Figure 32 were applied to the flow network illustrated in Figure 33 below.

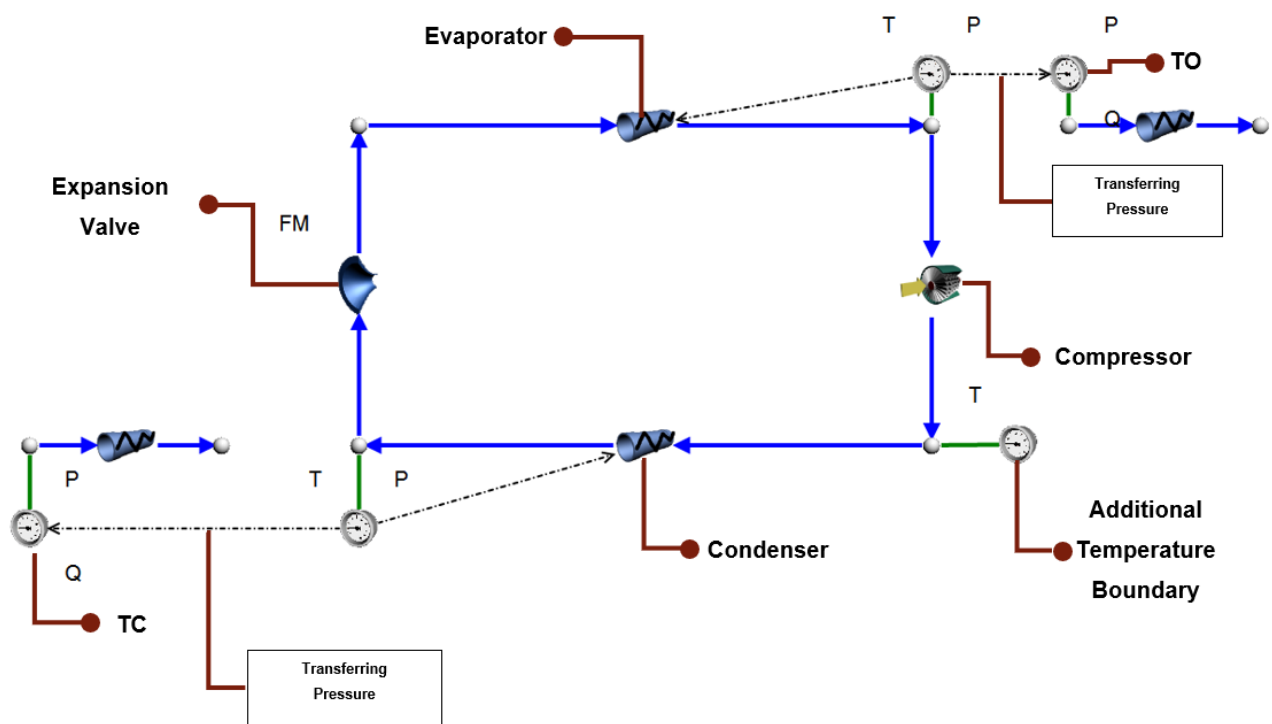


Figure 33: Flownex flow network used to determine compressor capacity to be used as a validation input

The flow network operates similarly to the compressor verification flow network of Figure 16 of Section [3.3.3 Compressor Model Verification](#), with the addition of a condenser inlet temperature boundary. The altitude and compressor gauge pressures were used to determine the absolute pressures applied on the compressor inlet of the model. The absolute compressor suction pressure, saturation quality and suction temperature were used to determine the TO and SH. Similarly, the absolute compressor discharge pressure, saturation quality and condenser discharge temperature were used to determine an estimate of the TC and SC. This is an estimate since in reality, the pressure-drop through the condenser results in a cooler TC. Like in Figure 16, the condenser in this flow network experiences no pressure-drop.

To determine the compressor capacities relevant to the FATs performed, the compressor capacity in the flow network of Figure 33 is varied until the compressor electrical power generated by the compressor model matches the benchmark compressor electrical power. A custom computer program uses the output from the data logger during the FAT to calculate system benchmark results like the compressor electrical power and condenser waterside heat transfer, respectively:

Compressor electrical power:

$$P = \sqrt{3}VI\cos(\theta) \quad [3.11]$$

- P = Three-phase power [W],
- V = Line-to-line voltage [V],
- I = Line current [A],
- θ = The phase angle between the voltage and current waveforms [°],

Condenser waterside heat transfer:

$$\dot{Q} = \dot{m}C_p\Delta T \quad [3.12]$$

- C_p = Specific heat capacity at constant pressure [J/kg-K],
- $\dot{m}$ = Mass flowrate [kg/s],
- ΔT = Temperature change [K],
- $\dot{Q}$ = Heat transfer rate [W],

Table 18 below contains the inputs applied to the flow network in Figure 33 as well as a comparison between the benchmark and generated values for temperature change over the compressor and condenser heat transfer for the three days of FAT data collection.

Table 18: Inputs and Results Used to Determine Compressor Capacity for Surface Validation

Day	*I-Compressor			*I-Condenser			Temperature Change Over Compressor			Condenser Heat transfer		
	Suction Pressure [bar]	Suction Temperature [°C]	Capacity [%]	Inlet Temperature [°C]	Outlet Pressure [bar]	Outlet Temperature [°C]	*B-[°C]	*T-[°C]	Error [%]	*B-[kW]	*T-[kW]	Error [%]
1	4.6	0.6	58.39	59.7	17.9	34.2	62.8	73.7	17.36	129.0	129.1	0.08
2	5.3	7.8	64.14	63.9	21.4	43.9	60.8	77.1	26.81	166.0	151.5	8.73
3	5.4	7.2	43.02	61.2	19.3	34.4	56.6	74.2	31.10	122.6	116.1	5.30

The connotations associated with light and dark grey columns as well as the meaning of the prefixes ‘*I-’, ‘*B-’ and ‘*T-’ has been stated previously in Section [3.1.3 The Use of Parameter Tables](#).

The compressor capacities applied to the flow network generated compressor electrical power values that matched the benchmark values for all three days. Applying the correct compressor capacities to the flow network ensures the correct amount of refrigerant is circulated through the

system for each day. To test whether this is the case, the generated and benchmark condenser heat transfer values could be compared. Since the temperature change across the condenser is known, as evident from Figure 33, the refrigerant mass flowrate would be the only unknown. In Table 18 above, the Flownex compressor model results show an acceptable relation to the benchmark results for condenser heat transfer with the greatest error being 8.73%. This error margin is acceptable considering the temperature measurements across the condenser were retrieved manually.

Table 18 contains a significant discrepancy of 17 to 31% between the generated and benchmark result for temperature change over the compressor. The verification process indicated a good correlation between the Flownex compressor model and the Bitzer software results. Since the energy equation over the compressor model has been verified, the temperature change over the compressor must be accurate. This is confirmed comparing the Flownex and Bitzer generated temperature change over the compressor. The results are available in Appendix A, Section [A.1.3 Small Compressor Verification Test](#).

The SH relating to the measured suction temperature and suction gauge pressure over the three days is extremely low, the highest SH being 3.7°C. The minimum specifiable SH input in the Bitzer software is 5°C, suggesting the ACUs in the FAT operated below the advised SH range of the compressor manufacturer. Since Bitzer software was used to generate compressor charts for the Flownex compressor model, the lowest SH the model can reliably operate under is 5°C. As the SH obtained in the FATs is lower than this, the compressor model generates higher compressor discharge temperatures compared to those measured.

Since the compressor model managed to generate the correct compressor electrical power and refrigerant mass flowrate required by the system, adding the measured compressor discharge temperature as a temperature boundary on the compressor discharge would allow the compressor model to function correctly allowing the validation to continue.

The Flownex generated results for air - and waterside temperature changes [°C], fan electrical power [kW], condenser heat transfer [kW], compressor electrical power [kW] and compressor discharge gauge pressure [bar] are compared to the known benchmark results of each dataset. The inputs to the flow network and retrieved results are available in Appendix B, Section [B.2 Validation Results](#). The average and maximum errors for the validation of the accuracy of the integrated ACU model for surface conditions are shown in Table 19, along with the references to equations used to calculate the errors for each tested property.

Table 19: Integrated ACU Model Surface Validation Results

Day		Airside Temperature Change [%]	Waterside Temperature Change [%]	Fan Electrical Power [%]	Condenser Heat Transfer [%]	Compressor Electrical Power [%]	Compressor Discharge Gauge Pressure [%]
		Eq. [3.8]	Eq. [3.8]	Eq. [3.7]	Eq. [3.7]	Eq. [3.7]	Eq. [3.7]
1	Avg	3.54	5.19	0.93	5.02	2.95	3.64
	Max	7.71	7.60	1.08	7.45	4.08	6.68
2	Avg	3.80	1.75	1.15	1.87	3.24	4.74
	Max	7.38	3.23	1.28	3.46	6.85	7.96
3	Avg	4.65	2.27	1.46	2.09	0.93	1.57
	Max	13.33	3.84	1.46	3.66	1.31	2.44

The integrated Flownex ACU model results show a good relation to the benchmark surface validation data, with each parameter exhibiting an average error margin within 5.19%. The verification was performed over eight data points for Day 1, eight data points for Day 2 and three for Day 3.

Inspecting Table 19, the compressor electrical power accuracy indicates that the correct compressor capacities were selected. This is confirmed by the results for condenser heat transfer since the refrigerant mass flowrate through the condenser is determined by the compressor capacity. The compressor discharge gauge pressure accuracy ensures that the correct TC temperatures are set. The result for the waterside temperature change reflects the heat transfer accuracy. The fan power results indicate that the mass flowrate of the air is correct and the results for the air temperature change indicate that the heat transfer on the airside is correct. The high air temperature change error is due to the small airside temperature change for Day 3 with the outlier underpredicting the temperature by less than 1°C. Overall, the Integrated ACU model is regarded to be a reliable tool for simulating surface FAT operations.

4.2 Underground Validation

The underground validation of the Integrated ACU model was performed applying the variable inputs described in Section [3.9 Integrated ACU Model](#). Underground mine logger data of an ACU supplied values to the required variable inputs of the integrated model. The results generated by the model were then compared to the underground logger data. The underground logger data satisfy most of the required variable inputs of the model, but some inputs had to be assumed. These assumptions are available in Table 20 below.

Table 20: Assumptions Made for Unknown Inputs to be Used in the Underground Validation

No.	Inputs	Assumption
1	Ambient air relative humidity	80%
2	Ambient absolute air pressure	1.20 [bar]

No.	Inputs	Assumption
3	Altitude	-1650 [m]
4	Water inlet pressure	2.5 [bar]
5	Compressor capacity	100%
6	Fouling factor evaporator airside	4.93E-03 [m ² -K /W]
7	Fouling factor condenser waterside	1.03E-04 [m ² -K/W]

The inputs used for relative humidity, air pressure and altitude reflect the deep-level and humid underground environmental conditions in which the ACU is expected to operate. The choice of water inlet pressure would have no significant impact on the results since there is no phase change, thus the minimum water inlet pressure of the condenser pack, as specified by the manufacturer, is applied. The compressor is expected to operate at full capacity in underground conditions if sufficient feedwater is supplied, as explained in Section [3.3.1 Compressor Component Overview](#). Fouling factors are considered due to the presence of fouling observed on decommissioned ACU evaporator fins and condenser water coils. This fouling is attributed to particle-laden air and hard water, which contribute to the fouling phenomenon. When applying the fouling values in Table 20 to the integrated model, solving for the initial operating point of the validation data yields a SH of 7.5°C and a SC of 11.5°C, the ranges used when generating the underground compressor chart.

The validation is performed using 70 data points of logged data for an ACU operating in deep-level mining conditions. Six data regions of stable ACU operation, each spanning an hour, is identified for validation. Table 21 below contains a summary of the known operational conditions for each hourly dataset.

Table 21: Known Operational Conditions of the Deep-level ACU Used for Validation

Dataset	*B-Compressor Electrical Power [kW]	*B-Condenser Heat Transfer [kW]	*I-Mass Flow Water [kg/s]	*I-Water Inlet Temperature [°C]	*B-Water Outlet Temperature [°C]	*I-Air Inlet Temperature [°C]	*B-Air Outlet Temperature [°C]
1	66.83	326.67	9.0	21.0	29.7	37.4	23.5
2	66.42	331.82	7.1	18.1	29.2	37.5	23.2
3	65.76	331.39	6.4	17.0	29.4	37.4	23.2
4	71.46	330.65	6.5	18.8	31.0	37.5	23.5
5	67.71	335.02	7.7	18.0	28.5	37.5	23.1
6	68.79	324.90	8.1	19.6	29.2	37.4	23.7

When selecting the six datasets from the available underground mine logger data, the focus was on ensuring diversity in operational conditions. This approach allows for validation across a broad spectrum of operating scenarios, ensuring the reliability and accuracy of integrated model testing.

The available underground logger data showed little diversity and so the ACU operational conditions in the six datasets of Table 21 reflect this.

The operational conditions are used as inputs to the integrated ACU model or as benchmark values for validation. The benchmark compressor electrical power and condenser heat transfer were calculated using logged data, while the water mass flowrate and temperatures in Table 21 were amongst available logged data.

The benchmark compressor electrical power is calculated subtracting the fan power and auxiliary power from the total power. The total power is calculated using Equation [3.11], with a power factor of 0.8 applied. The fan power is estimated by the Flownex underground mine fan model, based on the air volume flow. The auxiliary power is a summation of the remaining minor power drainages of the ACU excluding the compressor and fan. The auxiliary power drainage is assumed to mirror that observed during the FAT. In the FAT station the power drainage of the ACU was observed to be 0.57 kW when both the compressor and fan were turned off.

Calculations for waterside condenser heat transfer rely on Equation [3.12], assuming a specific heat capacity of $4.181 \text{ J/kg}\cdot^{\circ}\text{C}$. The water mass flowrate anticipated underground can potentially drop to 4.5 l/s, although this value falls below the range of available underground logger data. The inlet water temperature could rise to 30°C , yet the available data consistently presented lower temperatures. While the ambient air temperature in the data exhibited stability, it could dip as low as 25°C .

Setting the compressor suction pressure and condenser discharge pressure to the refrigeration cycle were the last inputs required to align the generated and benchmark results of each dataset. There are no temperature boundary conditions applied to the refrigeration cycle of the integrated ACU model, thus energy can only be added or removed through the heat exchangers and compressor.

The Flownex generated results for condenser heat transfer [kW], compressor electrical power [kW], as well as air - and waterside temperature changes [$^{\circ}\text{C}$] are compared to the known benchmark results of each dataset. The detailed validation results are available in Appendix B, Section [B.2 Validation Results](#). The average and maximum errors for the validation of the accuracy of the integrated ACU model for underground conditions are shown in Table 22, along with the references to equations used to calculate the errors for each tested property.

Table 22: Integrated ACU Model Underground Validation Results

Dataset		Condenser Heat Transfer [%]	Compressor Electrical Power [%]	Airside Temperature Change [%]	Waterside Temperature Change [%]
		Eq. [3.7]	Eq. [3.7]	Eq. [3.8]	Eq. [3.8]
1	Avg Error	0.55	0.45	0.55	0.87
	Max Error	1.04	0.94	1.15	1.37
2	Avg Error	0.50	0.90	1.00	0.46
	Max Error	1.32	1.89	1.27	1.19
3	Avg Error	1.22	1.53	0.56	1.32
	Max Error	1.84	2.71	0.91	1.95
4	Avg Error	0.41	5.51	0.81	0.42
	Max Error	1.06	6.59	1.32	0.93
5	Avg Error	1.52	1.28	0.47	1.77
	Max Error	2.01	3.07	1.07	2.25
6	Avg Error	2.94	1.00	3.58	3.16
	Max Error	3.96	3.35	4.22	4.19

The Flownex integrated model results show a good relation to the benchmark results, with all compared results exhibiting average errors within 5.51%. Each dataset spans over an hour and contains twelve operating points. Dataset 3 exhibited water mass flowrate fluctuations in the last two operating points of the dataset, thus the two unreliable datapoints were not included in the validation. The entire validation was conducted over 70 operating points and six hours of data. Dataset 4 had the greatest inaccuracy for compressor electrical power. Inspecting the dataset, the water mass flowrate increased by 0.6 kg/s over that hour. The applied pressure boundaries on the refrigeration cycle of the integrated ACU model over that hour catered to the final datapoint in the dataset. The operating conditions changed so much in that hour that the boundaries were not ideal for the first eleven datapoints in the dataset, resulting in the errors observed.

From the results in Table 22, it can be concluded that the integrated ACU model captures actual ACU performance accurately for the investigated operating range. For the generated condenser heat transfer to be accurate, the generated compressor discharge temperature and refrigerant mass flowrate had to be accurate. These results along with the accurately generated compressor electrical power result, validate the compressor model accuracy. The generated fan electrical power had to be accurate for the generated compressor electrical power to be accurate since the generated fan power and constant auxiliary power is subtracted from the total electrical power to determine the compressor electrical power benchmark. The fan is thus operating at the correct mass flowrate since this relates to fan electrical power through a chart. Considering that the evaporator model achieved accurate airside temperature change for the generated air mass flowrate, this validates the evaporator model for underground ACU operation.

At the end of this chapter, confidence in the integrated model is sustained for predicting ACU performance for surface and more importantly, underground conditions. The model proved to predict underground airside temperature change within a 4.22% error margin over the tested data

range. In the following chapter, the integrated ACU model is used to investigate the thermal fluid effects of varying environmental conditions on the refrigeration cycle through means of transient simulation.

CHAPTER 5: INVESTIGATION INTO THE TRANSIENT BEHAVIOUR OF THE ACU

The previous chapter validated the integrated Flownex ACU model for surface and underground conditions establishing confidence in the model's ability to realistically predict its performance. In this chapter, the integrated ACU model is used to investigate the effect of varying environmental conditions on the thermal fluid cycle by means of transient simulations. The following transients are used to investigate the thermal fluid response of the system:

1. Case 1: Feedwater supply gradually turned off.
2. Case 2: Feedwater fluctuations.
3. Case 3: Feedwater supply instantaneously turned off.
4. Case 4: Fan failure.
5. Case 5: Blocked-off air supply to half of the evaporators.

These transients would be used to identify the operational limits for feedwater and air supply, capture the response of compressor infinite capacity control to fluctuating feedwater supply and determine the obtainable cooling duty for limited feedwater availability.

The underground validation inputs of Dataset 1, row 1 was used to achieve the required steady state solution of the integrated ACU model before each transient simulation was performed. In the discussion that follows, 'baseline' refers to the inputs of Dataset 1, row 1. All inputs to this dataset and generated results are available in Appendix B, Section [B.2 Validation Results](#), in the first row of the 'Conditions' column in the 'Underground Validation'.

Below follows a discussion and illustration of the transient simulations of the abovementioned scenarios.

5.1 Case 1: Feedwater Supply Gradually Turned Off

- Case purpose: To investigate the system response to diminishing feedwater supply. This scenario is likely to occur at night when drilling occurs. Drilling demands significant amounts of feedwater, putting pressure on supply, thus as the night progresses ACU feedwater supply decreases.
- Procedure: The ACU operates at the baseline when feedwater supply decreases by 1 kg/s every 300 s.
- Time step size: 500 ms.
- Simulation duration: 2713.6 s.

The following four figures will be used to structure the discussion and illustrate the key impact of reduced feedwater supply on the thermal fluid cycle.

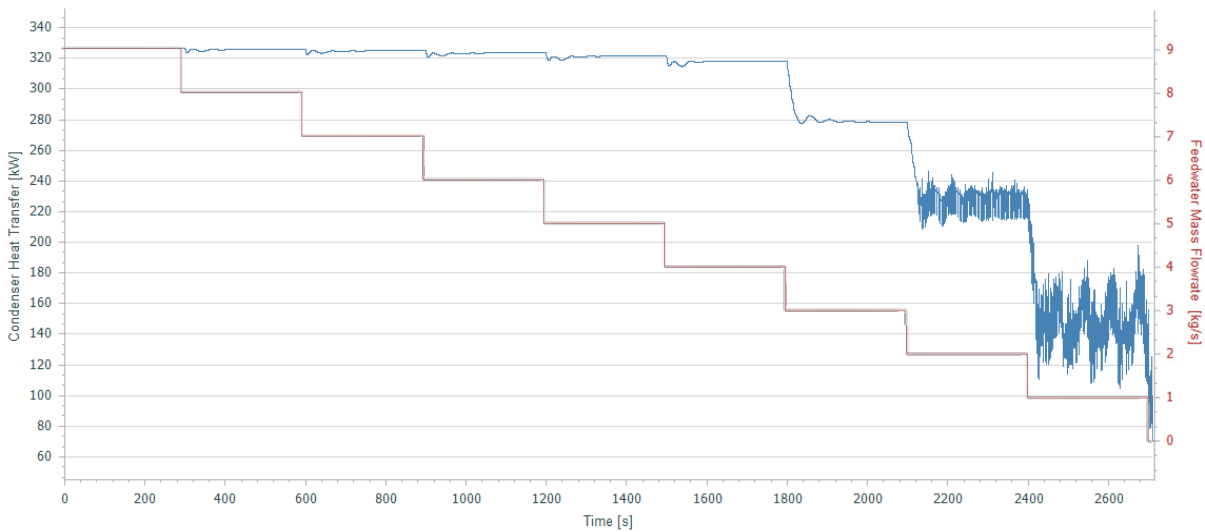


Figure 34: Variation in condenser heat transfer over time

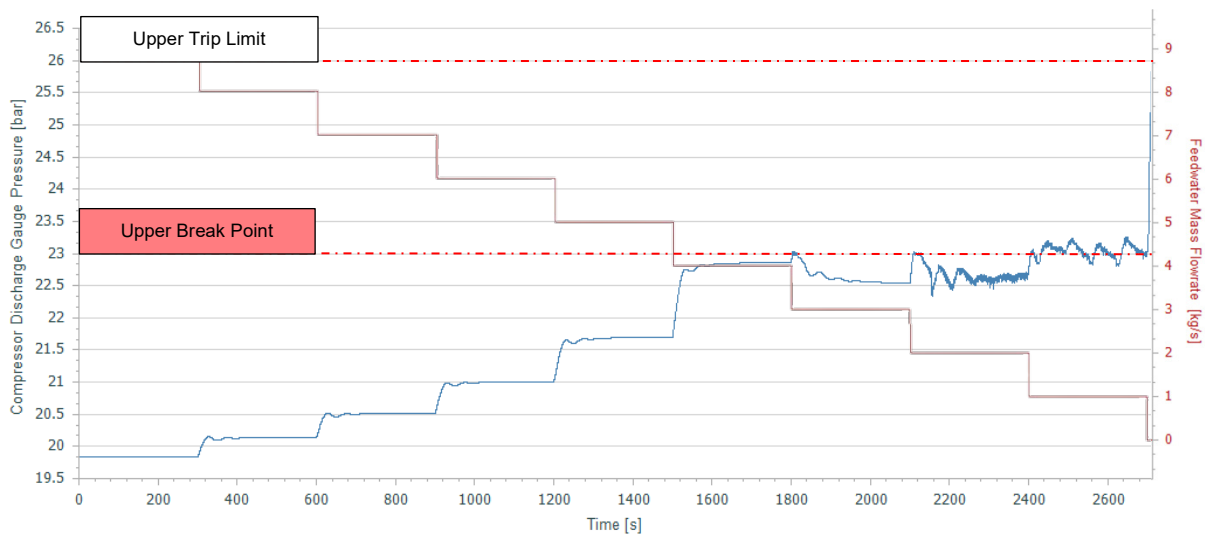


Figure 35: Variation in compressor discharge gauge pressure over time

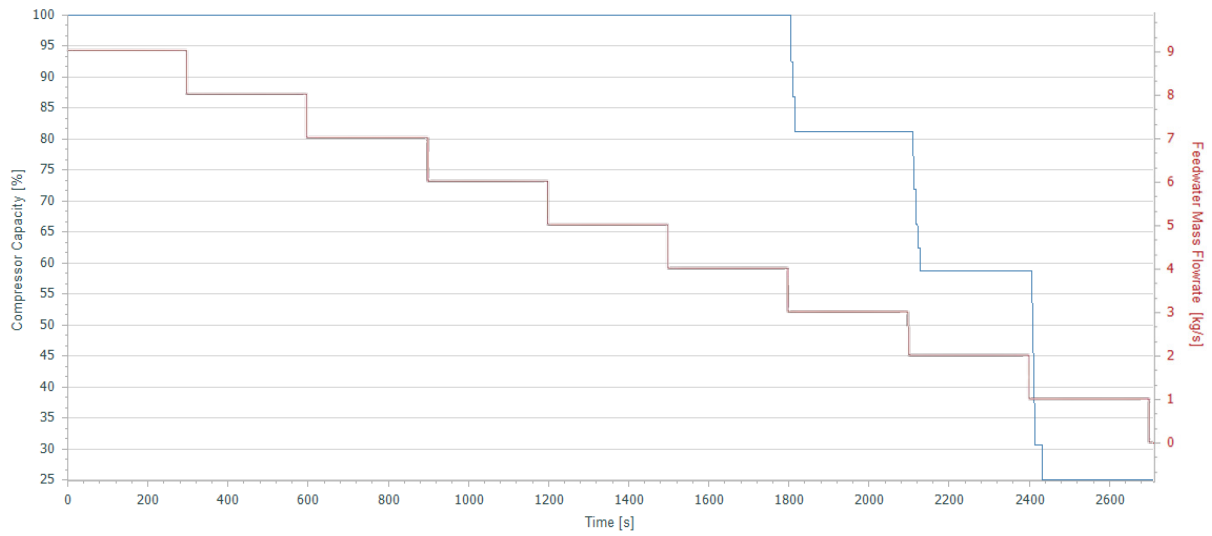


Figure 36: Variation in compressor capacity over time

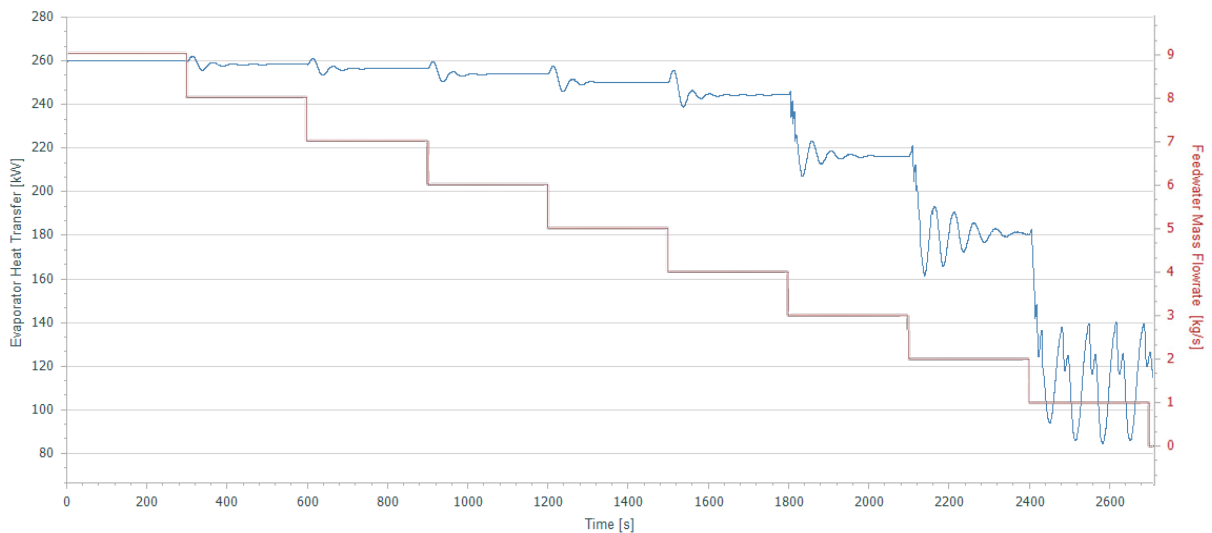


Figure 37: Variation in evaporator heat transfer over time

From 0 to 1800 s, the integrated ACU model maintained full compressor capacity (Figure 36) for a feedwater supply reducing from 9 to 4 kg/s. During this time, the condenser heat transfer decreased from 327 kW to 318 kW (Figure 34). Given that there was less feedwater available to remove heat in the condenser, the TC increased to maintain a temperature approach over which heat transfer could occur. The envelope of operations decreased as the TC increased, so there was less potential for the refrigerant to absorb heat from the air, and thus the evaporator heat transfer decreased from 260 kW to 244 kW (Figure 37). This particular ACU is clearly wasting water when operating at baseline since if the feedwater supply reduces by 5 kg/s the obtainable cooling only decreases by 6%. This oversupply of feedwater was an actual occurrence since the baseline operating point was retrieved from actual mine data.

The compressor discharge pressure (Figure 35) increased every time the feedwater supply was reduced but still remained within the capacity control limit. From 1800 s onward, the compressor upper break point limit of 23 bar was breached (Figure 35) every time the feedwater supply was stepwise reduced below 4 kg/s, prompting a compressor capacity decrease (Figure 36). This compressor reaction is expected as it is in line with the dynamics of infinite capacity control as illustrated in Figure 13 of Section [3.3.1 Compressor Component Overview](#).

For a 3 kg/s feedwater supply initiated at 1800 s, the compressor capacity settled at 81.2% (Figure 36), 18 s later, with the condensers rejecting 278 kW of heat (Figure 34) and the evaporators absorbing 216 kW of heat (Figure 37). For a 2 kg/s feedwater supply initiated at 2100 s, the compressor capacity settled at 58.8% (Figure 36), 35 s later, with the condensers rejecting 233 kW of heat (Figure 34) and the evaporators absorbing 182 kW of heat (Figure 37). For a 1 kg/s feedwater supply initiated at 2400 s, the compressor capacity settled at the minimum of 25% (Figure 36), 38 s later, with the condensers rejecting 151 kW of heat (Figure 34) and the evaporators absorbing 111 kW of heat (Figure 37).

The simulation ended 13.6 s after the feedwater supply reduced to zero, due to the compressor discharge pressure exceeding the upper trip limit of 26 bar (Figure 35). This transient simulation demonstrated how infinite capacity control ensures the ACU always delivers the greatest amount of cooling for diminishing feedwater supply, to the benefit of daily underground operations. Figures of relevant simulation results are available in Appendix C, Section [C.2.1 Case 1](#).

5.1.1 Comparison Between the Infinite and Step Capacity Control Strategies

To demonstrate the advantage of using the infinite capacity control strategy, the step capacity control strategy was applied to the integrated model and the Case 1 procedure repeated. Figure 38 below illustrates the evaporator heat transfer achieved applying step capacity control versus infinite capacity control.

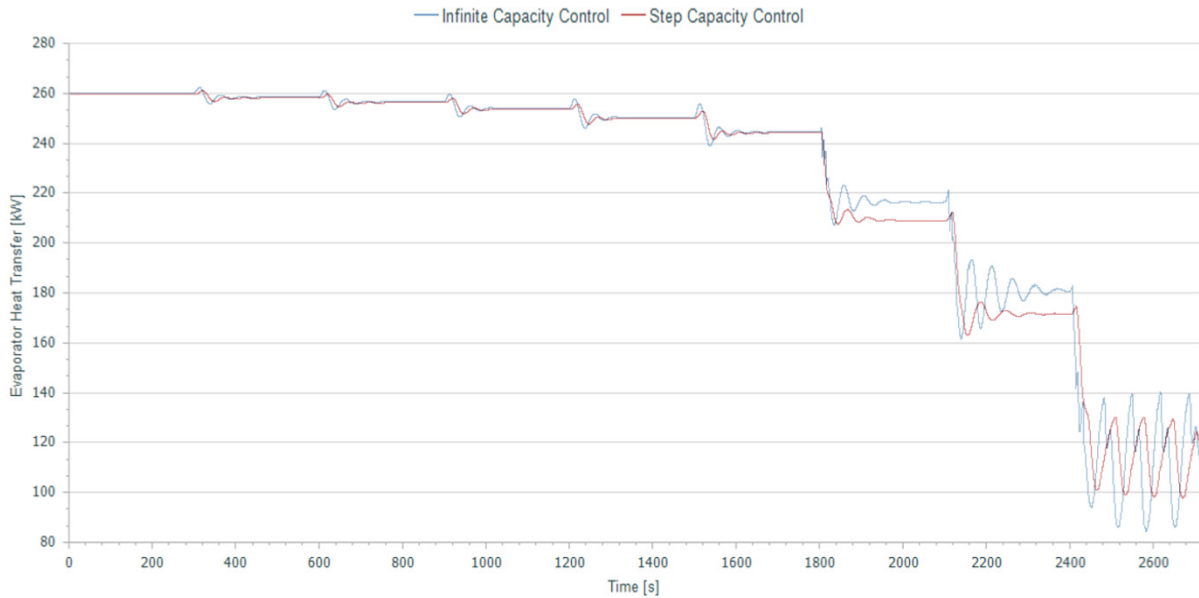


Figure 38: Variation in evaporator heat transfer over time when step capacity control is implemented as opposed to infinite capacity control under the Case 1 procedure.

Figure 38 illustrates the evaporator heat transfer achievable using infinite capacity control (blue line) and step capacity control (red line). Over the 1800 to 2100 s interval, for 3 kg/s feedwater supply, the infinite capacity settled at 81.2% and the step capacity at 75%, delivering 7 kW less cooling, which is a reduction of 3.2% in delivered cooling. Over the 2100 to 2400 s interval, for 2 kg/s feedwater supply, the infinite capacity settled at 58.8% and the step capacity at 50%, delivering 11 kW less cooling, which is a reduction of 6% in delivered cooling. The figures confirming all the results mentioned are available in Appendix C, Section [C.2.1.1 Step Capacity Control Under the Case 1 Procedure](#). This shows that step capacity control is a less effective means of operations, compared to infinite capacity control.

5.1.2 Comparison Between the Infinite Capacity Control Strategy and Continuous Full Capacity Operation

A similar investigation was done operating the unit at continuous full capacity instead of implementing step capacity control. Figure 39 below illustrates the evaporator heat transfer achieved operating the integrated model at continuous full capacity versus infinite capacity control.

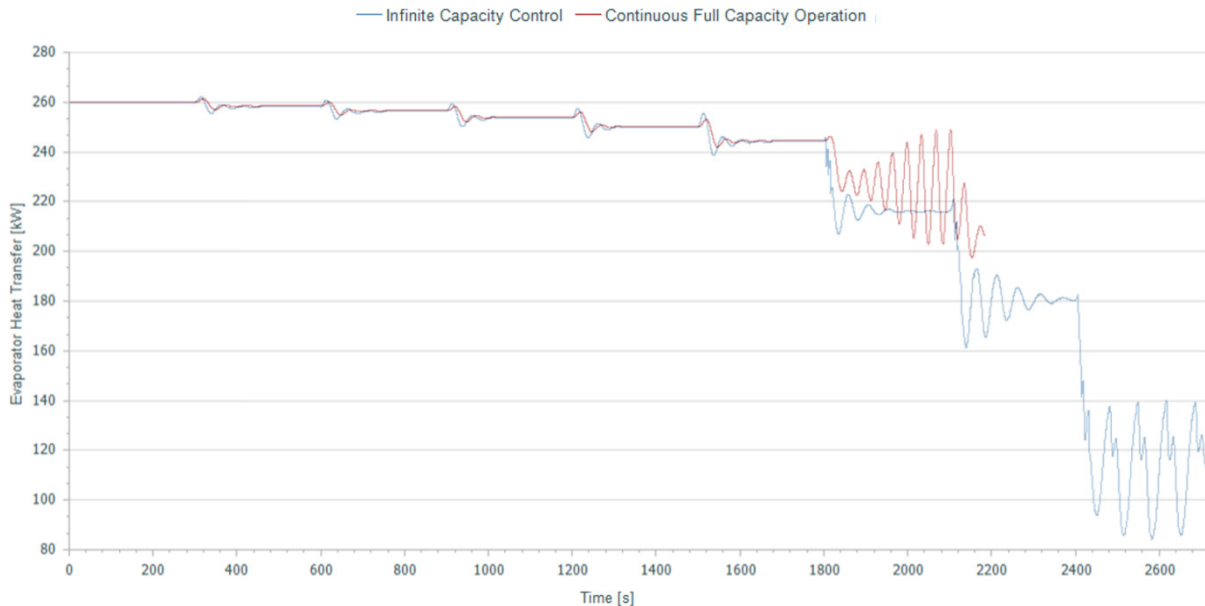


Figure 39: Variation in evaporator heat transfer over time when continuous full capacity operation is implemented as opposed to infinite capacity control under the Case 1 procedure.

Figure 39 illustrates the evaporator heat transfer achievable using infinite capacity control (blue line) and continuous full capacity operation (red line). Over the 1800 to 2100 s interval, for 3 kg/s feedwater supply, the infinite capacity settled at 81.2%. The continuous full capacity operation managed to deliver slightly more cooling in this interval, but the integrated model experienced expansion valve hunting, struggling to maintain the SH setpoint, resulting in the oscillation observed for the red line. Once the feedwater mass flowrate decreased to 2 kg/s, over the 2100 to 2400 s interval, the continuous full capacity operation model's compressor discharge pressure breached the upper trip limit of 26 bar, ending the simulation 83.6 s after the water reduction. Figure 39 clearly illustrates the amount of cooling lost during the final part of the runtime due to ACU downtime for continuous full capacity operation versus infinite capacity control. The figures confirming all the results mentioned are available in Appendix C, Section [C.2.1.2 Full Capacity Operation Under the Case 1 Procedure](#).

5.2 Case 2: Feedwater Fluctuations

- Case purpose: To investigate the system response to fluctuating feedwater supply. This scenario is likely to occur when feedwater supply valves are adjusted to redistribute flow.
- Procedure: The ACU operates at the baseline with a feedwater supply of 4.5 kg/s, then 2.25 kg/s and finally 9 kg/s for 300 s intervals. The feedwater flowrate transitions occur gradually rather than instantaneous as a point of realism.
- Time step size: 500 ms.
- Simulation duration: 967.6 s.

The following four figures will be used to structure the discussion and illustrate the key impact of fluctuating feedwater supply on the thermal fluid cycle.

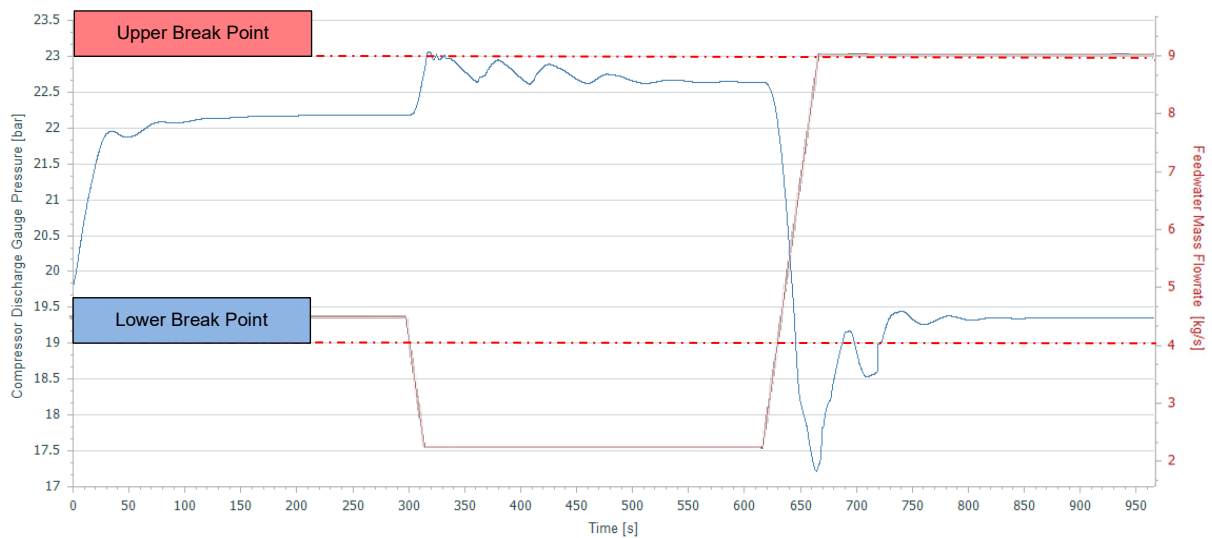


Figure 40: Variation in compressor discharge gauge pressure over time

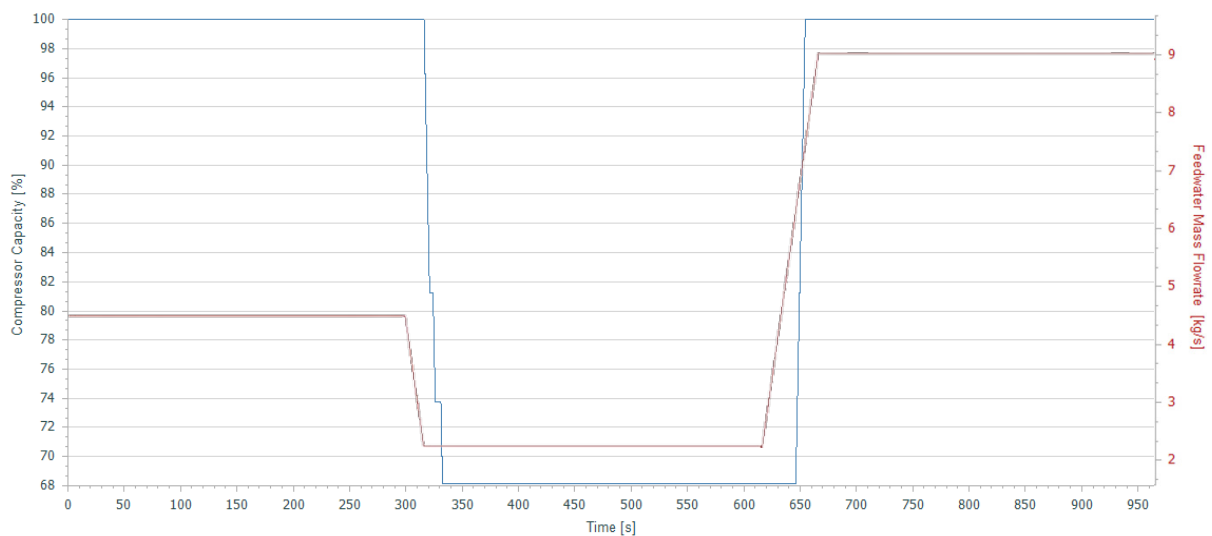


Figure 41: Variation in compressor capacity over time

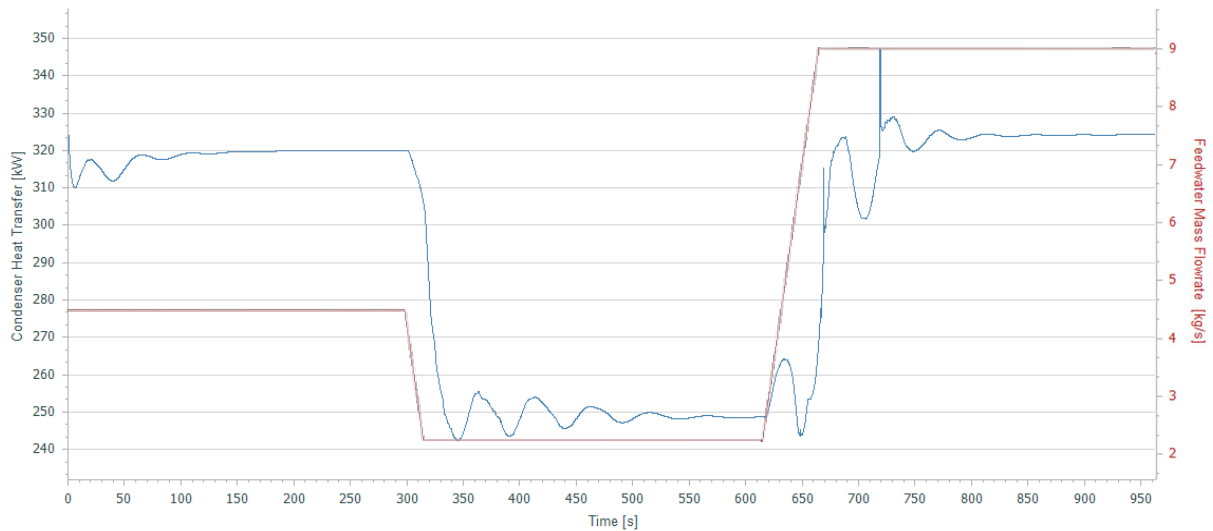


Figure 42: Variation in condenser heat transfer over time

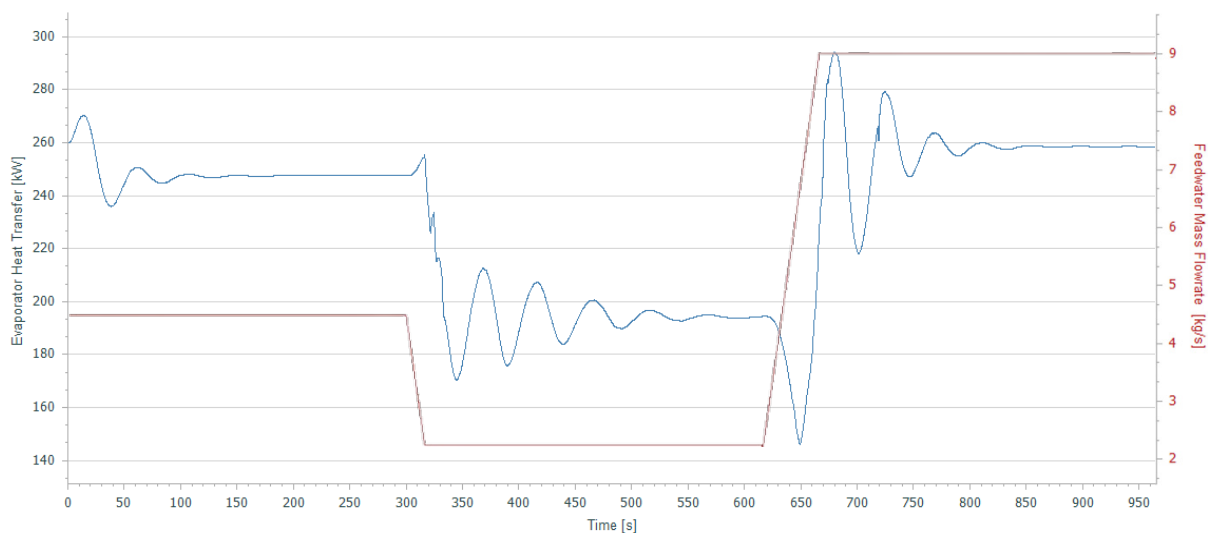


Figure 43: Variation in evaporator heat transfer over time

From 0 to 300 s, the integrated ACU model operated with a 4.5 kg/s feedwater supply. During this time, the model operated at a compressor discharge pressure of 22.2 bar (Figure 40). The condensing temperature and pressure was higher compared to that in baseline operation in order for the refrigerant-side to maintain a temperature approach suitable for heat transfer to occur in the condenser.

From 300 to 317 s, the feedwater supply gradually decreased to 2.25 kg/s. In this time, the compressor discharge pressure increased to exceed the upper break point of 23 bar (Figure 40), prompting a compressor capacity decrease to 68.1% (Figure 41), 19 s later. Following the capacity decrease, the compressor discharge pressure settled below the upper break point since less

refrigerant was in circulation. This reduction of refrigerant mass flowrate decreased the condenser heat transfer from 320 kW to 248 kW (Figure 42) and the evaporator heat transfer from 248 kW to 194 kW (Figure 43).

From 617 to 668 s, the feedwater supply gradually increased to 9 kg/s. The increased feedwater mass flowrate allowed for a smaller temperature approach across the condenser to achieve better heat transfer. This prompted a decrease of the compressor discharge pressure below the lower break point of 19 bar (Figure 40) at 645 s, allowing the compressor capacity to be restored to 100% (Figure 41), 12 s later. With both fluid streams increased, the condenser heat transfer increased to 324 kW (Figure 42) and the evaporator heat transfer to 259 kW (Figure 43).

This transient simulation demonstrated how infinite capacity control enables the ACU to restore compressor capacity once feedwater supply is restored, ensuring the ACU always delivers the greatest amount of cooling for the feedwater supply available, to the benefit of daily underground operations. Figures of relevant simulation results are available in Appendix C, Section [C.2.2 Case 2](#).

5.3 Case 3: Feedwater Supply Instantaneously Turned Off

- Case purpose: To investigate the system response if feedwater supply was to end instantaneously. This scenario is likely to occur when feedwater supply valves are adjusted to direct all feedwater towards drilling.
- Procedure: The ACU operates at the baseline for 5 s when the feedwater supply changes from 9 to 0 kg/s for the remainder of the simulation.
- Time step size: 100 ms.
- Simulation duration: 22 s.

The following four figures will be used to structure the discussion and illustrate the key impact of instantaneously turned off feedwater supply on the thermal fluid cycle.

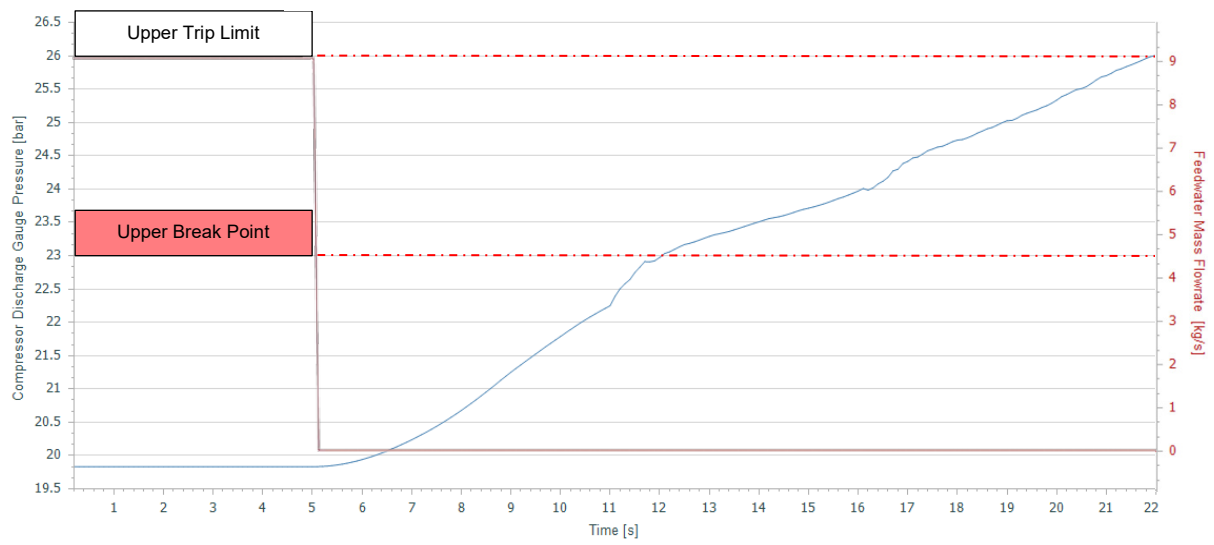


Figure 44: Variation in compressor discharge gauge pressure over time

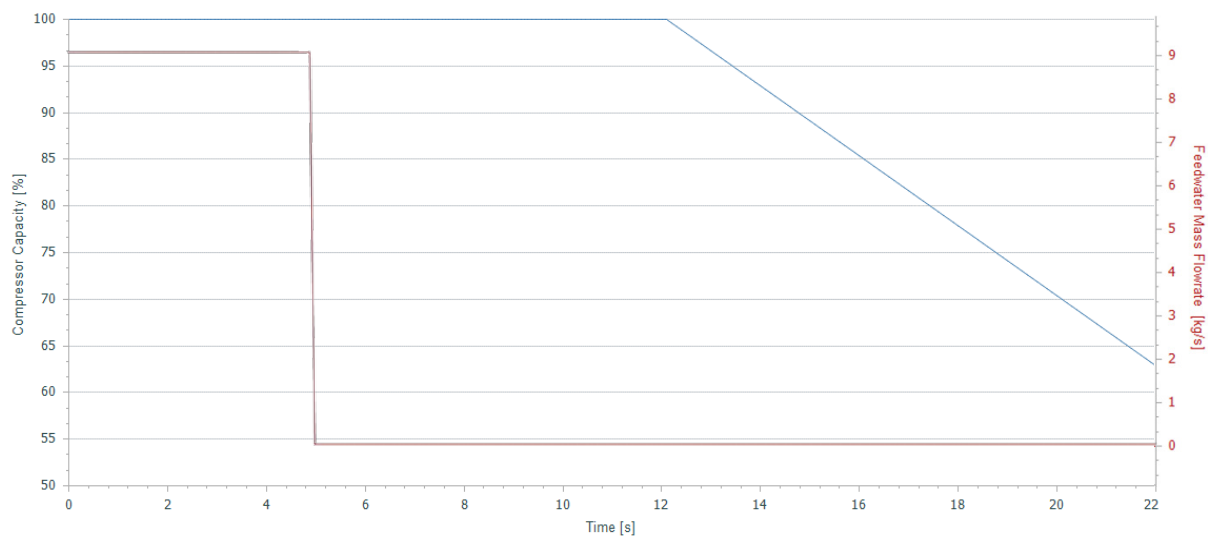


Figure 45: Variation in compressor capacity over time

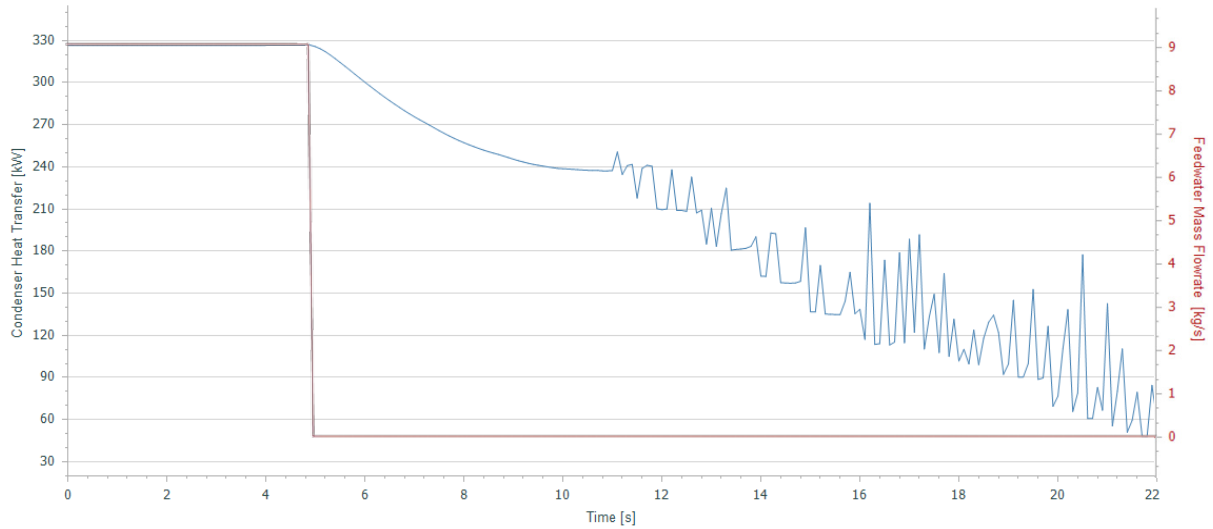


Figure 46: Variation in condenser heat transfer over time

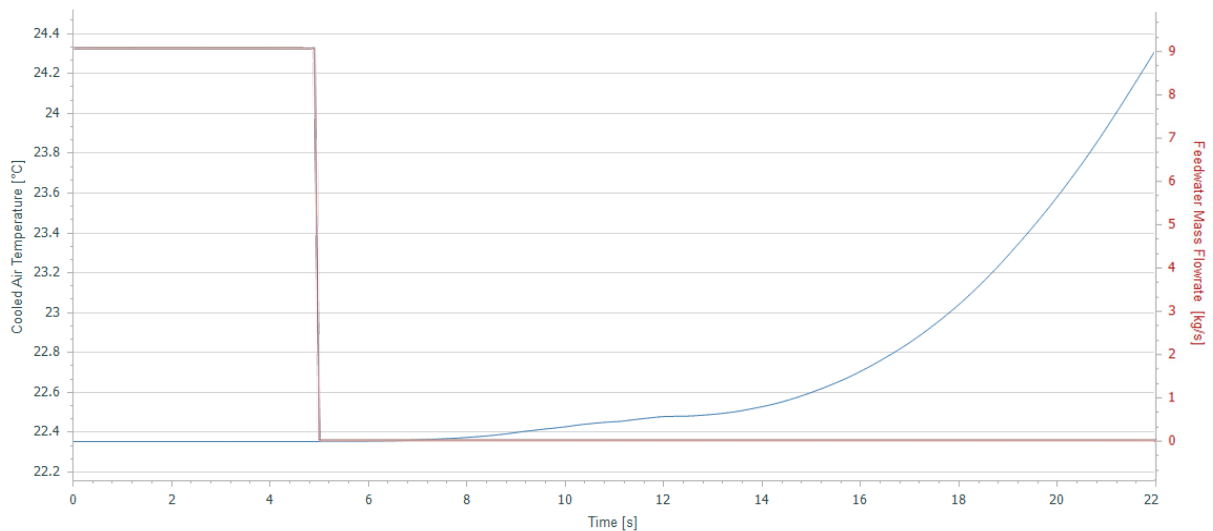


Figure 47: Variation in cooled air temperature over time

The instantaneous end of feedwater supply at 5 s, caused the water in the condenser to stagnate, severely limiting heat exchange potential. The condenser heat transfer was not lost instantaneously as the conductive pipe wall and stagnant water continued to absorb heat from the refrigerant (Figure 46). The condenser would continue to reject heat until the temperatures of the refrigerant, conductive pipe wall and water reaches equilibrium. That equilibrium is never met in this simulation, since as TC continued to rise, the high heat capacity of the stainless-steel ensured the pipe wall was always at a slightly lower temperature than the refrigerant, in the transient state.

Increased TC resulted in a corresponding increase in condensing pressure (Figure 44). The higher compressor discharge pressure throttled more refrigerant to a higher evaporation pressure through the expansion valves. The high TO that accompanied the rising evaporation pressure limited the

heat absorption potential in the evaporator, leading to an increase in cooled air temperature (Figure 47). The SH decreased as the evaporator heat transfer decreased prompting the expansion valves to close.

The compressor discharge pressure continued to increase, until the upper break point of 23 bar was breached, prompting a compressor capacity decrease at 12.3 s (Figure 45). Unlike for Case 1 and 2, the rising compressor discharge pressure did not decrease for reduced refrigerant mass flowrate. This indicates that an exceptionally high condensing pressure is required to maintain the condenser heat transfer at 12.3 s.

The simulation ended 17 s after the feedwater supply reduced to zero, due to the compressor discharge pressure exceeding the upper trip limit of 26 bar (Figure 44). The reduction in refrigerant mass flowrate due to the compressor capacity decrease, contributed to the decreased condenser and evaporator heat transfer towards the end of the simulation. This transient simulation demonstrated that the ACU cannot maintain operation if feedwater supply ends instantaneously. Figures of relevant simulation results are available in Appendix C, Section [C.2.3 Case 3](#).

5.4 Case 4: Fan Failure

- Case purpose: To investigate the system response if airflow supply was cut off instantaneously. This scenario is likely to occur when fan bearings seize due to rust in the humid underground environment.
- Procedure: The ACU operates at the baseline for 5 s when the fan rotation and thus the air mass flowrate through the evaporators is instantaneously stopped for the remainder of the simulation.
- Time step size: 100 ms.
- Simulation duration: 21.7 s.

The following three figures will be used to structure the discussion and illustrate the key impact of fan failure on the thermal fluid cycle.

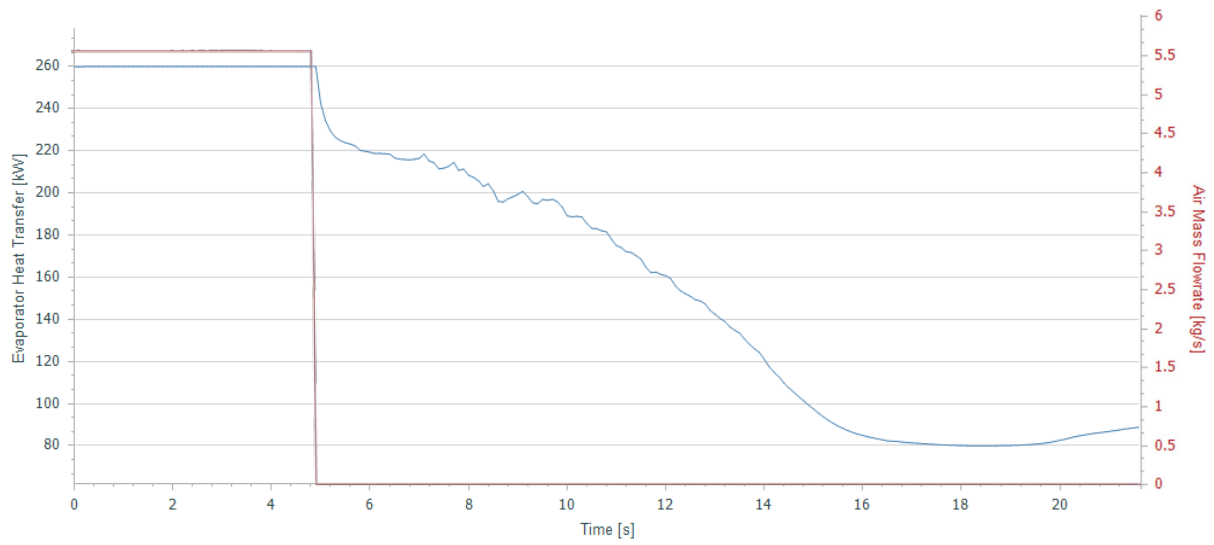


Figure 48: Variation in evaporator heat transfer over time

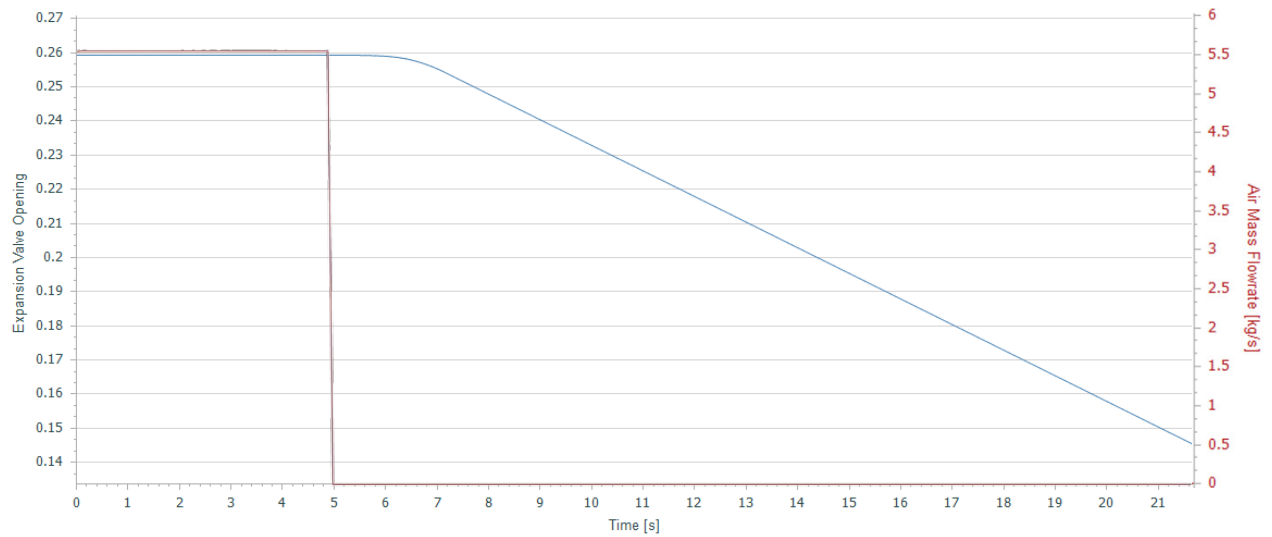


Figure 49: Variation in expansion valve opening over time

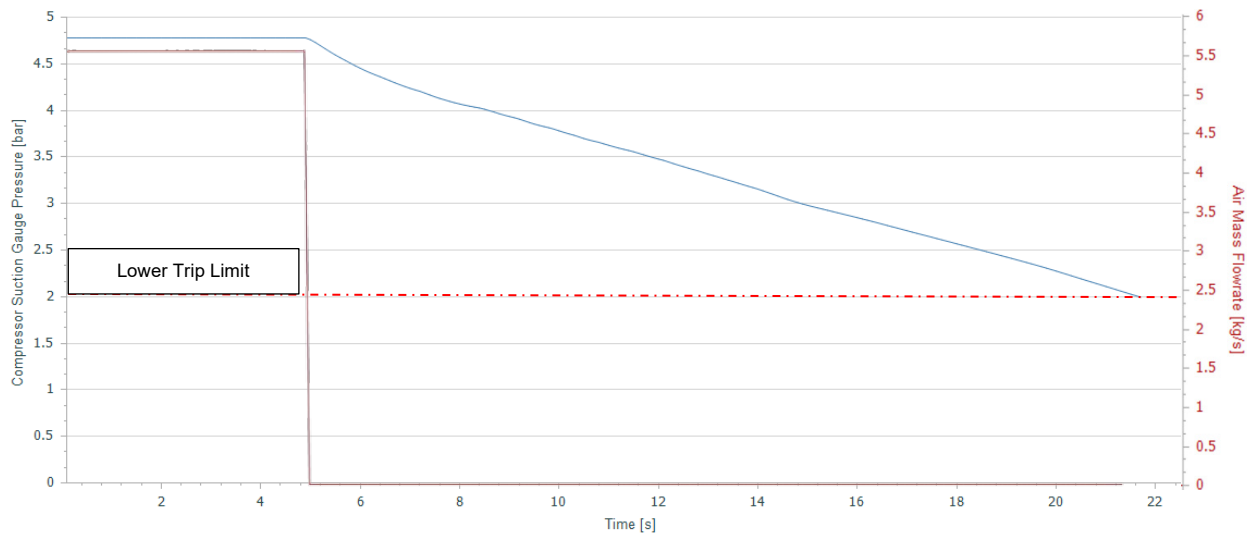


Figure 50: Variation in compressor suction gauge pressure over time

The instantaneous end of air supply at 5 s, caused the air in the evaporator to stagnate, drastically reducing the air convection heat transfer coefficient, in so doing limiting heat exchange potential. The evaporator heat transfer was not lost instantaneously as the conductive pipe wall, fins and stagnant air continued to transfer heat into the refrigerant (Figure 48). The evaporator would continue to absorb heat until the temperatures of the refrigerant, conductive pipe wall and air reaches equilibrium. That equilibrium is never met in this simulation, since as TO continued to decrease, the high heat capacity of the aluminium fins and copper pipe ensured the pipe wall was always at a slightly higher temperature than the refrigerant in this transient state. The decrease in evaporation temperature and pressure was a result of the expansion valve closing in an attempt to restore the decreasing SH.

The simulation ended once the lower trip limit of the compressor suction pressure was breached 16.7 s after the end of air supply (Figure 50). This transient simulation demonstrated that the ACU cannot maintain operation in the event of fan failure. Figures of relevant simulation results are available in Appendix C, Section [C.2.4 Case 4](#).

5.5 Case 5: Blocked-Off Air Supply to Half of the Evaporators

- Case purpose: To investigate the system response if the air supply to half of the face area of the evaporator packs was cut off. This scenario is likely to occur when the ACU is placed too close to a wall, forcing all the air sucked in by the fan through the unblocked evaporator packs.
- Procedure: The ACU operates at the baseline for 5 s when the air supply to half of the evaporator packs is cut-off, with the total fan suction drawing air through the unblocked side for the remainder of the simulation.

- Time step size: 100 ms.
- Simulation duration: 60 s.

The following five figures will be used to structure the discussion and illustrate the key impact of blocked-off air supply on the thermal fluid cycle.

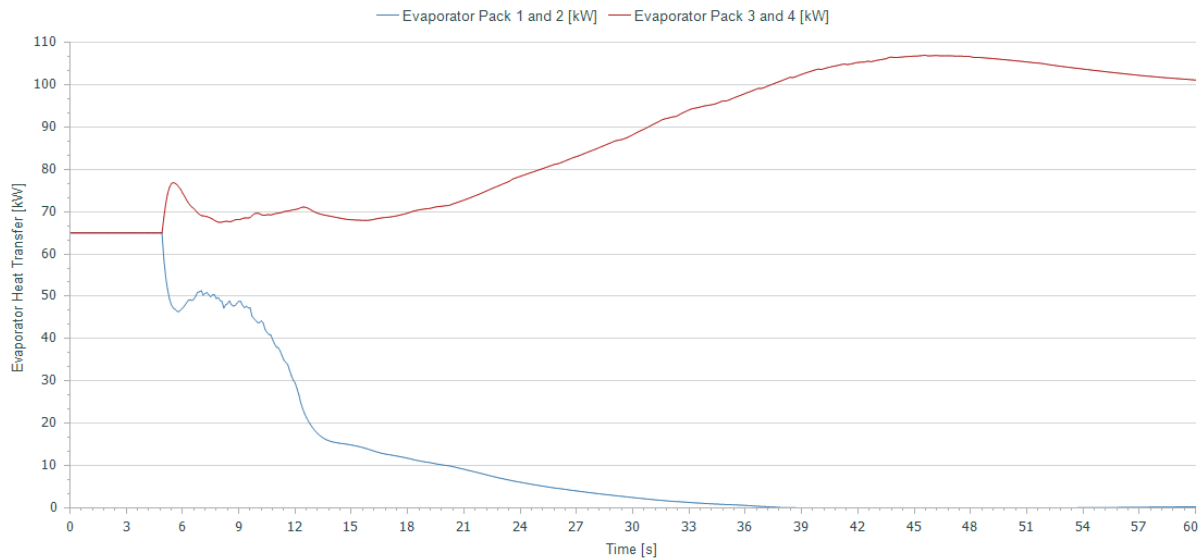


Figure 51: Variation in evaporator heat transfer over time

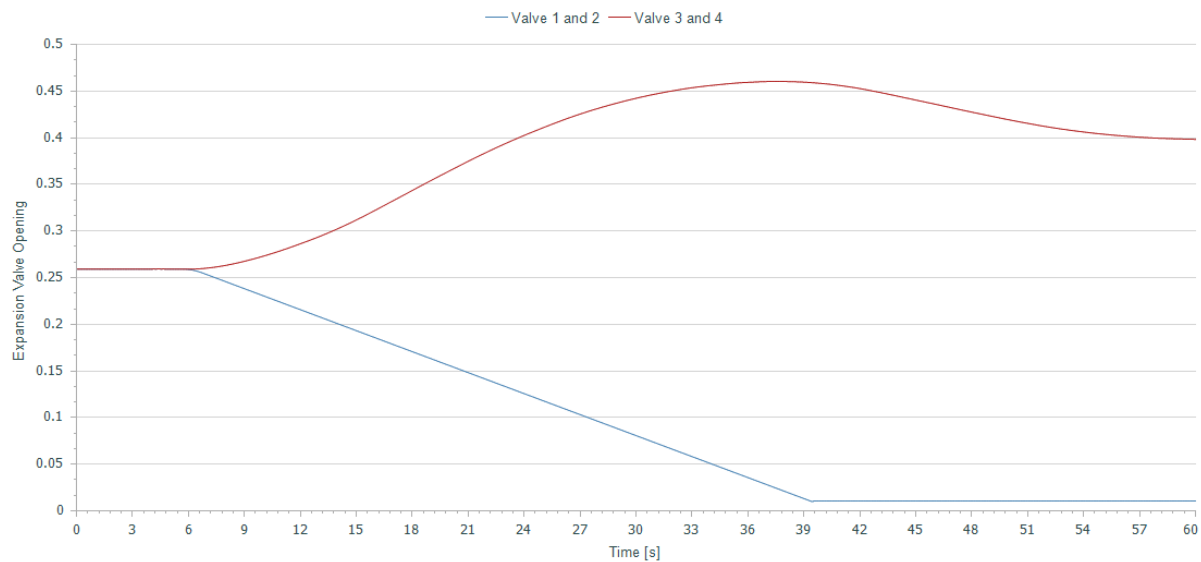


Figure 52: Variation in expansion valve opening over time

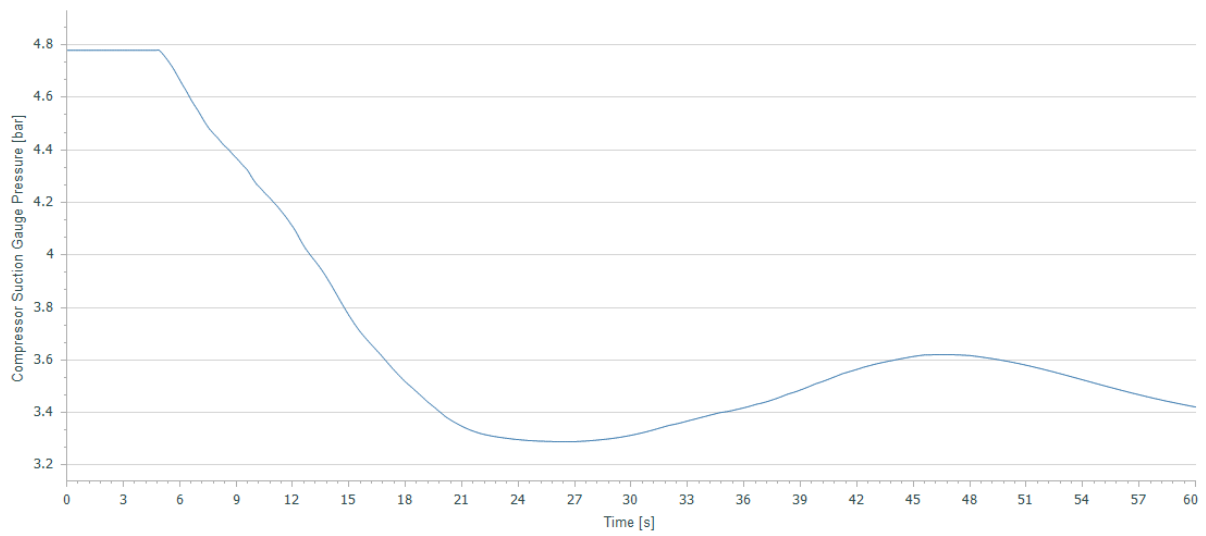


Figure 53: Variation in compressor suction gauge pressure over time

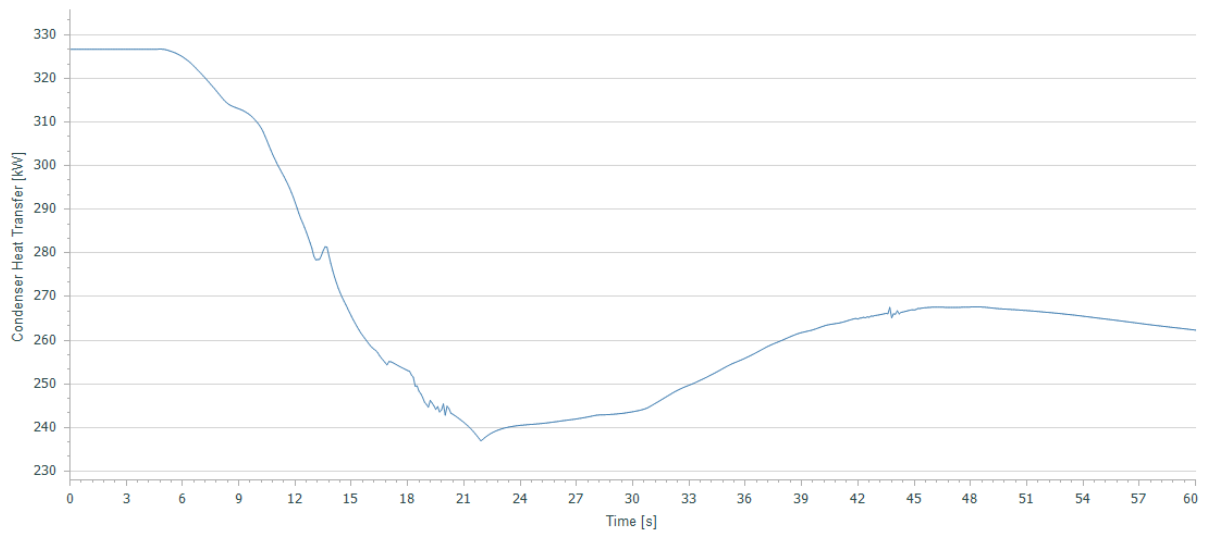


Figure 54: Variation in condenser heat transfer over time

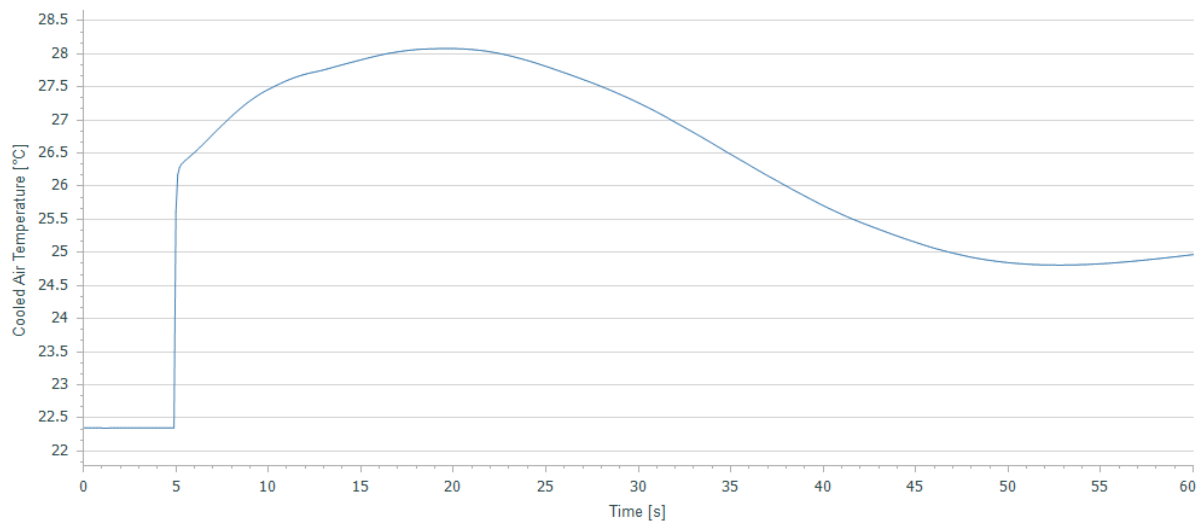


Figure 55: Variation in cooled air temperature over time

At 5 s, the total airflow was redirected through half of the evaporator packs, decreasing the convection heat transfer coefficient on one side and increasing it on the other. Subsequently, the blocked side experienced a decrease in evaporator heat transfer eventually leading up to a total heat transfer loss (Figure 51). The expansion valves of these evaporator packs closed in an attempt to restore the decreasing SH (Figure 52). The unblocked side experienced an increase in evaporator heat transfer, resulting in an increase of SH (Figure 51). The expansion valves of these evaporator packs opened further in an attempt to decrease the rising SH (Figure 52).

The system stabilised with the sum of the heat transfer of the four evaporator packs decreasing from 260 to 202 kW (Figure 51) and condenser heat transfer decreasing from 327 to 262 kW (Figure 54) over the duration of the simulation. The cycle increased the approach across the evaporators by reducing overall TO, in an attempt to increase evaporator heat transfer. For decreased TO, the overall expansion valve opening had to decrease, thus less refrigerant was circulated in the refrigeration cycle. The decreased refrigerant mass flowrate resulted in decreased condenser heat transfer. Since there was an oversupply of water available to remove heat from the reduced refrigerant stream, the refrigerant was condensed to a lower TC.

The cooled air temperature delivered by the ACU model increased from 22.4 to 25°C. This transient simulation demonstrated that the increased convection heat transfer coefficient on half of the evaporator packs compensated for the loss on the other half, allowing ACU operation to continue, albeit providing less cooling. Figures of relevant simulation results are available in Appendix C, Section [C.2.5 Case 5](#).

5.6 Summary

In this chapter, the integrated ACU model was used to investigate the effects of varying environmental conditions on the thermal fluid cycle through transient simulations. Practical environmental impacts such as decreased feedwater, fluctuating feedwater, fan failure, and evaporator blockage were investigated.

From the investigated cases it is evident the infinite capacity control strategy is well suited for an environment experiencing fluctuating feedwater supply. The control strategy managed to sustain ACU operation for reduced feedwater supply by reducing operational capacity. Even though less cooling is provided when the ACU operates at reduced capacity, the unit manages to restore operational capacity once feedwater supply increases again. This behaviour is very beneficial to the mining environment.

Furthermore, the investigated cases revealed the control strategy has minimal impact when air supply fluctuations occur. Unlike for feedwater supply, sustaining air supply presents minimal challenges and could be managed with good operating practice. The dissertation is concluded in the following chapter.

CHAPTER 6: CONCLUSION AND RECOMENDATIONS

With the depletion of surface ore bodies, the future of South-African mining is expanding deeper underground, creating a heightened demand for effective cooling. Traditional cooling methods, such as extensive chilled water networks with surface fridge plants, prove less efficient in the context of deep mining, especially in remote underground locations. Spot cooling in remote areas has become essential for compliance with mining regulations. The development of a modular and mobile ACU addressed this need for more effective spot cooling. The ACU uses a vapour compression cycle to extract energy from the air and expel it into service water. However, the ACU's reliance on a constant water flowrate for operation became challenging due to fluctuations in water supply, leading to safety mechanisms like tripping on overpressure. Implementing infinite capacity control of the compressor was a strategic measure to improve operations and minimize downtime in response to fluctuating water flow.

Since the impact of infinite capacity control on the thermal fluid dynamics of the ACU refrigeration cycle was not well known, this study aimed to investigate the cycle reaction when subjected to fluctuating feedwater and air supply through means of thermal fluid modelling.

The development of the thermal fluid ACU model involved a thorough literature review concentrated on modelling vapour compression cycles within the context of underground mining. Special emphasis was given to existing mathematical modelling approaches for essential refrigeration cycle components, including the compressor, condenser, expansion valve, and evaporator.

A thermal fluid ACU model was developed within the Flownex simulation environment. Individual models of the main ACU components were created after which they were integrated to form a complete cycle model. Custom coding within Quick Script elements was used to achieve appropriate transient system response for the compressor and expansion valves.

The accuracy of the component models was verified and the integrated ACU model validated. The accuracy of the component models was verified over anticipated operating ranges using third-party software that encompass well-established mathematical equations and principles relevant to the modelled components. To establish confidence in the model's ability to conduct the investigation, the accuracy of the integrated ACU model was validated over anticipated operating ranges for both surface and underground conditions.

Five transients were used to investigate the effect of varying environmental conditions on the thermal fluid cycle of the ACU. The study identified the operational limits for feedwater and air

supply, accurately captured the response of compressor infinite capacity control to fluctuating feedwater supply and determined the obtainable cooling duty for limited feedwater availability.

In terms of operational limits, the study revealed that an extremely restricted feedwater supply, less than 1 kg/s at 21°C, leads to ACU downtime. Additionally, it was observed that while the ACU cannot maintain operation in the event of fan failure, it can indeed maintain operation even if half of the evaporators are blocked.

The transients accurately captured the compressor response to fluctuating feedwater supply. The study revealed the compressor actively decreased capacity for reduced feedwater supply and increased capacity when feedwater supply was restored. This strategy allowed the cycle to scale its refrigerant mass flowrate in accordance with its ability to reject heat into feedwater, so doing maintaining continuous operation for fluctuating feedwater supply.

The study revealed oversupplying feedwater mass flow rate would not dramatically improve obtainable cooling duty. Undersupplying feedwater below 4 kg/s at 21°C resulted in a dramatic decrease in obtainable cooling due to compressor capacity decrease. Furthermore, the study revealed the infinite capacity control strategy proved to deliver better, continuous cooling than the step capacity control and the continuous full capacity operation strategies under the same operating conditions.

To conclude, this study investigated the thermal fluid response of the ACU to fluctuating environmental conditions through thermal fluid modelling, identifying operational limits, compressor response and obtainable cooling duty within these environmental constraints.

6.1 Recommendations for Further Study

Based on the findings of this study, further researched could be conducted into the following points:

- The current compressor has a 25% capacity limit. Further research is required to determine whether expanding the operating range to include the 0 to 25% range, using a variable speed controlled compressor, would improve the ACU's ability to maintain continuous operation with a limited feedwater supply.
- In the transient investigation, the feedwater flowrate, air and water temperatures, humidity and elevation could be varied to obtain more operational limits. The more operational limits that are known, the more effectively this information can be used to prevent ACU downtime.
- The transient study revealed feedwater oversupply to the ACU's takes place in the mines. The cause of this problem should be investigated and addressed.

The following improvements could be made within the developed ACU thermal fluid model:

- The evaporator model, that makes part of the integrated ACU model, assumes fins are dry. In the underground environment, condensation occurs on the fins leading to partially or fully wetted surfaces increasing the convection heat transfer coefficient. This is a limitation of the Finned-Tube Heat Exchanger Flownex element used for modelling and can be addressed in further studies.
- The code that controls the transient behaviour of the compressor and expansion valves rely on a few assumptions. Obtaining the actual compressor slide speed and expansion valve adjustment spring tension would improve transient simulation accuracy.

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APPENDICES

This section contains three appendices that are required to interpret the dissertation.

- Appendix A provides information on the development of individual ACU component models. This includes the base data used to create each component, as well as any verification models and results that were developed.
- Appendix B contains information on validating the accuracy of the integrated ACU model. This includes information about the measuring instruments used to collect the benchmark data, as well as the complete validation results.
- Appendix C provides details on the investigation into the ACU's transient behaviour. This includes the compressor and expansion valve model's operating logic during transient simulation, as well as all relevant ACU response figures for each transient case.

APPENDIX A: MODEL DEVELOPMENT

This appendix contains information on the development of the individual ACU component models.

A.1 Compressor Development

This section contains information regarding the development of the ACU compressor model.

A.1.1 Bitzer Compressor Data

The data was extracted from the Bitzer software using the appropriate compressor operating ranges for surface and underground conditions. This information was used to create compressor charts for the developed compressor model. This information is presented on the following page as Figure 56.

To continue reading from where you left off in the main dissertation, [click here](#).

[illegible]

Figure 56: Bitzer compressor data

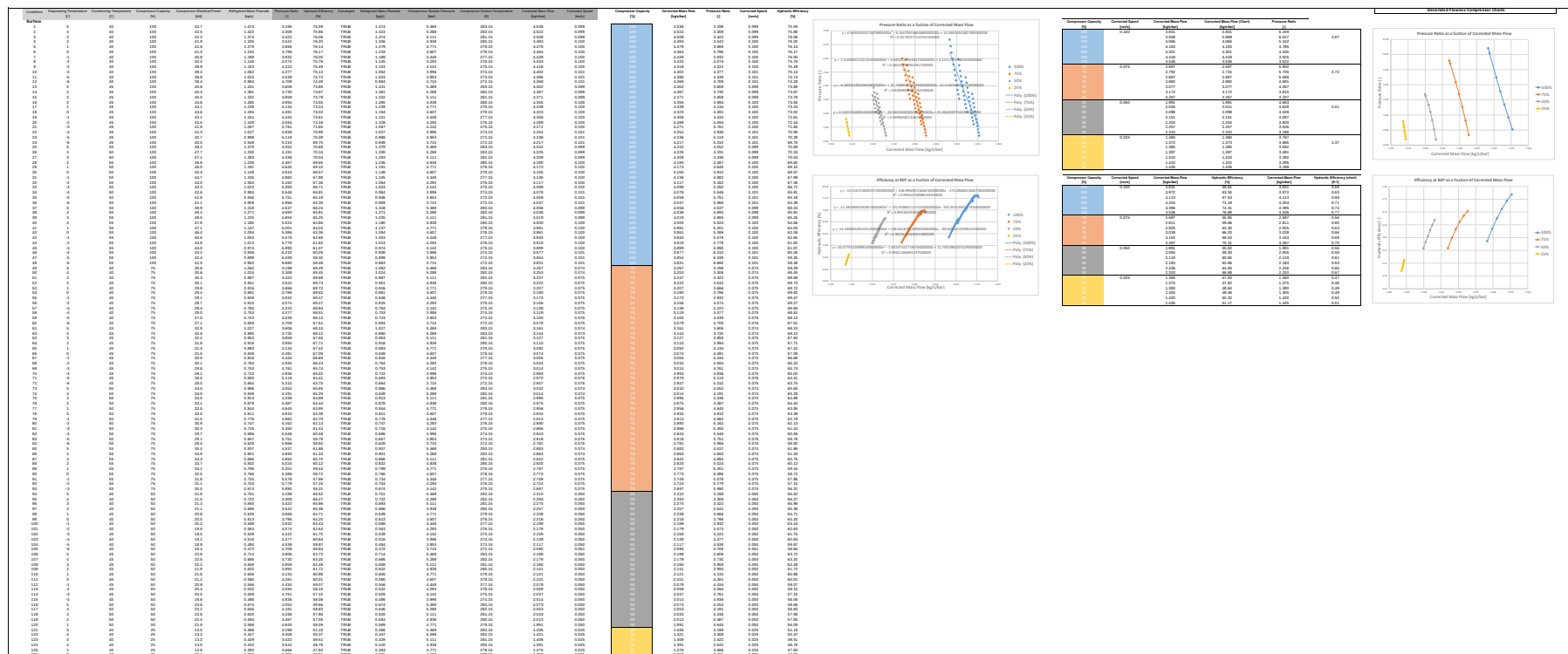
A.1.2 Generating Compressor Charts

The Bitzer data displayed on the previous page was processed into data that could be used in the Flownex compressor chart. This data set included corrected mass flow, corrected speed, pressure ratio, and hydraulic efficiency values. The developed Flownex compressor charts are included, as well as an R^2 fit of the chart lines to the original data.

A.1.2.1 Generating Surface Compressor Charts

The generated surface chart and supporting information is presented on the following page as Figure 57.

To continue reading from where you left off in the main dissertation, [click here](#).



A.1.2.2 Generating Underground Compressor Charts

The generated underground chart and supporting information is presented on the following page as Figure 58.

To continue reading from where you left off in the main dissertation, [click here](#).

[illegible]

Figure 58: Generating underground compressor charts

A.1.3 Small Compressor Verification Test

A small verification test was carried out, comparing the developed compressor model results to a Bitzer simulation in a spot-check comparison fashion. The comparison is presented on the following page as Figure 59.

To continue reading from where you left off in the main dissertation, [click here](#).

In addition to the compressor verification done, a small verification test was performed using the Bitzer software. The Bitzer software provides more detailed analysis of single setpoints that include additional refrigeration cycle results like temperature change over the compressor [°C] and condenser heat transfer [kW]. The specific operating points encountered in the FAT could not be tested in the Bitzer software since the SH recorded for all three days was below the minimum specifiable SH input of the Bitzer software. Instead, comparison between the Flownex compressor model and the Bitzer software were performed over several operating points that lie within the discussed compressor operating ranges. The table below illustrates the inputs and results for the additional results comparison.

Compressor Model Verification for Temperature Change Over the Compressor and Condenser Heat Transfer

Strata	*I-Capacity	*I-SH [°C]	*I-SC [°C]	*I-TO [°C]	*I-TC [°C]	Temperature Change Over Compressor [°C]			Condenser Heat Transfer [kW]		
						*R-Bitzer	*T-Flownex	Error [%]	*R-Bitzer	*T-Flownex	Error [%]
Und	100	7.5	11.5	8	60	84.7	86.2	1.8	294	290	1.4
Und	75			6	55	81.8	85	3.9	212	206	2.8
Und	50			7	45	64.4	68.3	6.1	171.3	167	2.5
Und	25			9	50	85.5	89.5	4.7	113.8	111	2.5
Surf	100	5	10.5	1	40	61	65	6.6	283	274	3.2
Surf	75			2	38	63.7	63.9	0.3	206	209	1.5
Surf	50			-1	35	63.2	67.6	7	142.8	137	4.1
Surf	25			1	41	87.1	90.2	3.6	95.8	93.5	2.4
							Avg Error	4.23		Avg Error	2.53
							Max Error	6.96		Max Error	4.06

The Flownex compressor model results show a good relation to the results of the Bitzer software, with temperature change over the compressor and condenser heat transfer both exhibiting average errors within 4.23%. Comparing the condenser heat transfer is a good verification test since the developed model relies on both the conservation of energy and momentum to be applied correctly by the compressor model in order to have a good comparison with the Bitzer reference results. Conservation of energy was tested since the inlet temperature to the condenser was generated by the compressor. Conservation of momentum was tested since the refrigerant mass flowrate through the compressor had to be correct in order for the model to generate the correct condenser heat transfer. The maximum temperature change over the compressor error of 7% is less than the minimum temperature change over the compressor error recorded in the FAT results. This confirms the verification accuracy of the compressor model for temperature change over the compressor.

Figure 59: Small compressor verification test

A.1.4 Compressor Verification Results

The accuracy of the developed compressor model was verified for surface and underground conditions.

A.1.4.1 Compressor Surface Verification Results

The applied input conditions, reference and test results for each data point of the surface verification is presented on the following page as Figure 60.

To continue reading from where you left off in the main dissertation, [click here](#).

Conditions	*H-Evaporating Temperature [C]	*H-Condensing Temperature [C]	*H-Compressor Capacity [m³]	*H-Compressor Electrical Power [kW]	*H-Refrigerant Mass Flowrate [kg/s]	*H-Compressor Electrical Power [kW]	*H-Refrigerant Mass Flowrate [kg/s]	Error Compressor Electrical Power	Error Refrigerant Mass Flowrate
Surface									
1	5	40	100	56.5	1.474	55.1	1.488	2.58%	0.31%
2	5	40	100	56.5	1.474	55.1	1.488	2.58%	0.31%
3	3	40	100	55.9	1.374	55.0	1.381	0.98%	0.52%
4	3	40	100	55.9	1.374	55.0	1.381	0.98%	0.52%
5	1	40	100	54.6	1.279	54.7	1.279	0.16%	0.09%
6	0	40	100	54.1	1.233	54.5	1.229	0.74%	0.29%
7	-1	40	100	53.7	1.188	54.3	1.181	1.07%	0.63%
8	-2	40	100	53.3	1.145	54.1	1.134	1.44%	0.97%
9	-3	40	100	52.9	1.103	53.9	1.088	1.87%	1.36%
10	-4	40	100	52.5	1.062	53.7	1.044	2.25%	1.76%
11	-5	40	100	52.1	1.023	53.5	1.000	2.60%	2.19%
12	-6	40	100	51.7	0.984	53.3	0.958	3.14%	2.63%
13	5	45	100	62	1.431	60.9	1.449	1.70%	1.30%
14	4	45	100	61.5	1.381	60.8	1.395	1.19%	1.06%
15	3	45	100	61	1.332	60.6	1.343	0.73%	0.78%
16	2	45	100	60.8	1.285	60.3	1.291	0.50%	0.47%
17	1	45	100	60.2	1.238	60.1	1.241	0.18%	0.18%
18	0	45	100	59.7	1.194	59.9	1.192	0.36%	0.21%
19	-1	45	100	59.3	1.151	59.7	1.144	0.65%	0.58%
20	-2	45	100	58.9	1.108	59.5	1.097	1.01%	0.97%
21	-3	45	100	58.5	1.067	59.4	1.052	1.47%	1.39%
22	-4	45	100	58.2	1.027	59.2	1.008	1.72%	1.65%
23	-5	45	100	57.8	0.988	59.1	0.965	2.21%	2.32%
24	-6	45	100	57.4	0.949	59.1	0.923	2.89%	2.82%
25	5	50	100	68.2	1.378	65.7	1.408	2.14%	1.53%
26	4	50	100	67.8	1.330	66.6	1.365	1.84%	1.85%
27	3	50	100	67.3	1.283	66.3	1.302	1.43%	1.53%
28	2	50	100	66.9	1.236	66.1	1.251	1.21%	1.19%
29	1	50	100	66.5	1.191	65.9	1.201	0.85%	0.82%
30	0	50	100	66.1	1.148	65.8	1.152	0.40%	0.41%
31	-1	50	100	65.7	1.105	65.6	1.105	0.09%	0.02%
32	-2	50	100	65.3	1.064	65.6	1.059	0.39%	0.43%
33	-3	50	100	64.9	1.023	65.5	1.014	0.99%	0.90%
34	-4	50	100	64.6	0.984	65.5	0.971	1.44%	1.38%
35	-5	50	100	64.2	0.946	65.6	0.928	2.25%	1.92%
36	-6	50	100	63.8	0.909	65.9	0.888	3.27%	2.46%
37	5	55	100	75.3	1.318	72.9	1.365	3.19%	3.57%
38	4	55	100	74.8	1.272	72.8	1.312	2.71%	3.22%
39	3	55	100	74.4	1.225	72.6	1.260	2.36%	2.87%
40	2	55	100	74	1.180	72.5	1.210	2.08%	2.48%
41	1	55	100	73.6	1.137	72.4	1.160	1.57%	2.05%
42	0	55	100	73.2	1.094	72.4	1.112	1.04%	1.64%
43	-1	55	100	72.8	1.053	72.4	1.065	0.49%	1.20%
44	-2	55	100	72.5	1.013	72.6	1.020	0.18%	0.71%
45	-3	55	100	72.1	0.974	72.9	0.975	1.12%	0.14%
46	-4	55	100	71.7	0.936	73.2	0.932	2.12%	0.43%
47	-5	55	100	71.4	0.899	73.6	0.889	3.66%	1.04%
48	-6	55	100	71	0.863	74.0	0.848	4.41%	1.09%
49	5	40	75	44.4	1.062	43.6	1.062	1.51%	0.99%
50	4	40	75	44	1.024	43.4	1.023	1.29%	0.96%
51	3	40	75	43.5	0.987	43.3	0.986	0.52%	0.11%
52	2	40	75	43.1	0.951	43.1	0.949	0.09%	0.21%
53	1	40	75	42.7	0.916	42.8	0.912	0.31%	0.34%
54	0	40	75	42.3	0.881	42.6	0.877	0.69%	0.46%
55	-1	40	75	41.9	0.846	42.3	0.842	0.86%	0.63%
56	-2	40	75	41.5	0.815	42.0	0.809	1.23%	0.82%
57	-3	40	75	41.2	0.784	41.7	0.778	1.30%	1.03%
58	-4	40	75	40.8	0.753	41.4	0.743	1.54%	1.32%
59	-5	40	75	40.4	0.723	41.2	0.711	1.89%	1.67%
60	-6	40	75	40.1	0.693	40.9	0.679	2.69%	2.63%
61	5	45	75	48.2	1.027	47.8	1.036	0.80%	0.83%
62	4	45	75	47.8	0.990	47.5	0.997	0.14%	0.74%
63	3	45	75	47.4	0.953	47.2	0.969	0.36%	0.62%
64	2	45	75	47	0.918	46.9	0.921	0.13%	0.44%
65	1	45	75	46.6	0.883	46.6	0.885	0.09%	0.23%
66	0	45	75	46.2	0.849	46.3	0.849	0.25%	0.69%
67	-1	45	75	45.8	0.814	46.0	0.814	0.38%	0.81%
68	-2	45	75	45.4	0.784	45.7	0.779	0.72%	0.61%
69	-3	45	75	45	0.753	45.5	0.745	1.04%	1.01%
70	-4	45	75	44.7	0.722	45.2	0.712	1.12%	1.47%
71	-5	45	75	44.3	0.693	45.1	0.679	1.73%	2.00%
72	-6	45	75	43.9	0.664	45.0	0.646	2.42%	2.71%
73	5	50	75	52.5	0.986	51.9	1.006	1.09%	2.08%
74	4	50	75	52.1	0.949	51.6	0.967	0.95%	1.93%
75	3	50	75	51.7	0.913	51.3	0.928	0.17%	1.68%
76	2	50	75	51.3	0.878	50.9	0.890	1.39%	1.94%
77	1	50	75	50.9	0.844	50.6	0.853	0.55%	1.04%
78	0	50	75	50.5	0.811	50.3	0.816	0.17%	0.69%
79	-1	50	75	50.1	0.778	50.1	0.780	0.08%	0.16%
80	-2	50	75	49.7	0.747	49.9	0.744	0.47%	0.27%
81	-3	50	75	49.3	0.716	49.8	0.708	1.07%	1.04%
82	-4	50	75	48.9	0.686	49.9	0.674	1.84%	1.81%
83	-5	50	75	48.6	0.657	50.1	0.640	2.61%	2.63%
84	-6	50	75	48.2	0.628	50.3	0.606	4.36%	3.61%
85	5	55	75	57.3	0.937	56.2	0.972	1.96%	3.72%
86	4	55	75	56.9	0.901	55.9	0.932	1.81%	3.40%
87	3	55	75	56.5	0.866	55.5	0.892	1.69%	2.98%
88	2	55	75	56.1	0.832	55.3	0.853	1.41%	2.53%
89	1	55	75	55.7	0.799	55.1	0.814	0.99%	1.93%
90	0	55	75	55.3	0.766	55.0	0.776	0.52%	1.23%
91	-1	55	75	54.9	0.734	55.1	0.738	0.43%	0.92%
92	-2	55	75	54.5	0.704	55.3	0.701	1.52%	0.31%
93	-3	55	75	54.1	0.674	55.6	0.665	2.86%	1.31%
94	5	40	50	32.7	0.751	32.5	0.746	0.71%	0.68%
95	4	40	50	32.5	0.722	32.4	0.718	0.28%	0.54%
96	3	40	50	32.3	0.693	32.3	0.690	0.11%	0.41%
97	2	40	50	32.2	0.666	32.2	0.664	0.12%	0.32%
98	1	40	50	32.1	0.639	32.1	0.637	0.11%	0.30%
99	0	40	50	31.9	0.613	32.0	0.611	0.39%	0.31%
100	-1	40	50	31.8	0.588	31.9	0.585	0.35%	0.47%
101	-2	40	50	31.7	0.563	31.8	0.559	0.33%	0.72%
102	-3	40	50	31.6	0.538	31.7	0.533	0.30%	1.15%
103	-4	40	50	31.5	0.516	31.6	0.507	0.38%	1.81%
104	-5	40	50	31.4	0.494	31.6	0.480	0.60%	2.64%
105	-6	40	50	31.3	0.472	31.6	0.452	1.04%	4.12%
106	5	45	50	35.9	0.714	35.9	0.724	0.08%	1.40%
107	4	45	50	35.7	0.686	35.8	0.696	0.32%	1.40%
108	3	45	50	35.6	0.658	35.7	0.668	0.23%	1.40%
109	2	45	50	35.5	0.632	35.5	0.639	0.13%	1.20%
110	1	45	50	35.4	0.606	35.4	0.611	0.04%	0.93%
111	0	45	50	35.3	0.580	35.3	0.583	0.00%	0.46%
112	-1	45	50	35.2	0.556	35.2	0.554	0.23%	0.93%
113	-2	45	50	35.1	0.532	35.2	0.524	0.41%	1.35%
114	-3	45	50	35	0.509	35.3	0.494	0.93%	2.90%
115	-4	45	50	34.9	0.486	35.5	0.463	1.99%	4.11%
116	5	50	50	39.5	0.674	39.5	0.697	0.08%	3.49%
117	4	50	50	39.4	0.646	39.3	0.667	0.17%	3.15%
118	3	50	50	39.3	0.620	39.2	0.636	0.19%	2.65%
119	2	50	50	39.2	0.594	39.2	0.604	0.08%	1.69%
120	1	50	50	39.1	0.569	39.2	0.572	0.20%	0.60%
121	5	40	25	26.4	0.466	26.4	0.465	0.13%	0.29%
122	4	40	25	26.5	0.447	26.5	0.446	0.08%	0.17%
123	3	40	25	26.6	0.429	26.6	0.428	0.11%	0.07%
124	2	40	25	26.6	0.410	26.6	0.411	0.16%	0.08%
125	1	40	25	26.7	0.392	26.7	0.393	0.22%	0.14%
126	0	40	25	26.7	0.376	26.7	0.376	0.52%	0.24%
								Average Error	1.08%
								Maximum Error	4.36%

Figure 60: Compressor surface verification results

A.1.4.2 Compressor Underground Verification Results

The applied input conditions, reference and test results for each data point of the underground verification are presented on the following page as Figure 61.

To continue reading from where you left off in the main dissertation, [click here](#).

Conditions	*1-Evaporating Temperature [°C]	*2-Condensing Temperature [°C]	*3-Compressor Capacity [m³/h]	*4-Compressor Electrical Power [kW]	*5-Refrigerant Mass Flowrate [kg/s]	*6-Refrigerant Mass Flowrate [kg/s]	*7-Compressor Electrical Power [kW]	*8-Refrigerant Mass Flowrate [kg/s]	Error Compressor Electrical Power	Error Refrigerant Mass Flowrate
Underground										
2	12	40	100	50.4	1.544	58.0	1.563	0.000	0.00%	0.00%
3	10	40	100	50.7	1.784	58.2	1.800	0.000	0.00%	0.00%
4	8	40	100	50.9	1.725	58.2	1.738	0.000	0.00%	0.00%
5	6	40	100	50.6	1.668	58.2	1.677	0.000	0.00%	0.00%
6	4	40	100	50.8	1.613	58.2	1.618	0.000	0.00%	0.00%
7	2	40	100	50.7	1.558	57.9	1.559	0.000	0.00%	0.00%
8	0	40	100	50.6	1.503	57.7	1.502	0.000	0.00%	0.00%
9	-2	40	100	50.5	1.448	57.4	1.446	0.000	0.00%	0.00%
10	-4	40	100	50.4	1.393	57.1	1.391	0.000	0.00%	0.00%
11	-6	40	100	50.5	1.338	56.8	1.337	0.000	0.00%	0.00%
12	-8	40	100	50.6	1.283	56.5	1.282	0.000	0.00%	0.00%
13	-10	40	100	50.7	1.228	56.2	1.224	0.000	0.00%	0.00%
14	-12	40	100	50.8	1.173	55.9	1.169	0.000	0.00%	0.00%
15	-14	40	100	50.9	1.118	55.6	1.113	0.000	0.00%	0.00%
16	-16	40	100	51.0	1.063	55.3	1.058	0.000	0.00%	0.00%
17	-18	40	100	51.1	1.008	55.0	1.003	0.000	0.00%	0.00%
18	-20	40	100	51.2	0.953	54.7	0.948	0.000	0.00%	0.00%
19	-22	40	100	51.3	0.898	54.4	0.893	0.000	0.00%	0.00%
20	-24	40	100	51.4	0.843	54.1	0.838	0.000	0.00%	0.00%
21	-26	40	100	51.5	0.788	53.8	0.783	0.000	0.00%	0.00%
22	-28	40	100	51.6	0.733	53.5	0.728	0.000	0.00%	0.00%
23	-30	40	100	51.7	0.678	53.2	0.673	0.000	0.00%	0.00%
24	-32	40	100	51.8	0.623	52.9	0.618	0.000	0.00%	0.00%
25	-34	40	100	51.9	0.568	52.6	0.563	0.000	0.00%	0.00%
26	-36	40	100	52.0	0.513	52.3	0.508	0.000	0.00%	0.00%
27	-38	40	100	52.1	0.458	52.0	0.453	0.000	0.00%	0.00%
28	-40	40	100	52.2	0.403	51.7	0.398	0.000	0.00%	0.00%
29	-42	40	100	52.3	0.348	51.4	0.343	0.000	0.00%	0.00%
30	-44	40	100	52.4	0.293	51.1	0.288	0.000	0.00%	0.00%
31	-46	40	100	52.5	0.238	50.8	0.233	0.000	0.00%	0.00%
32	-48	40	100	52.6	0.183	50.5	0.178	0.000	0.00%	0.00%
33	-50	40	100	52.7	0.128	50.2	0.123	0.000	0.00%	0.00%
34	-52	40	100	52.8	0.073	49.9	0.068	0.000	0.00%	0.00%
35	-54	40	100	52.9	0.018	49.6	0.013	0.000	0.00%	0.00%
36	-56	40	100	53.0	0.000	49.3	0.000	0.000	0.00%	0.00%
37	-58	40	100	53.1	0.000	49.0	0.000	0.000	0.00%	0.00%
38	-60	40	100	53.2	0.000	48.7	0.000	0.000	0.00%	0.00%
39	-62	40	100	53.3	0.000	48.4	0.000	0.000	0.00%	0.00%
40	-64	40	100	53.4	0.000	48.1	0.000	0.000	0.00%	0.00%
41	-66	40	100	53.5	0.000	47.8	0.000	0.000	0.00%	0.00%
42	-68	40	100	53.6	0.000	47.5	0.000	0.000	0.00%	0.00%
43	-70	40	100	53.7	0.000	47.2	0.000	0.000	0.00%	0.00%
44	-72	40	100	53.8	0.000	46.9	0.000	0.000	0.00%	0.00%
45	-74	40	100	53.9	0.000	46.6	0.000	0.000	0.00%	0.00%
46	-76	40	100	54.0	0.000	46.3	0.000	0.000	0.00%	0.00%
47	-78	40	100	54.1	0.000	46.0	0.000	0.000	0.00%	0.00%
48	-80	40	100	54.2	0.000	45.7	0.000	0.000	0.00%	0.00%
49	-82	40	100	54.3	0.000	45.4	0.000	0.000	0.00%	0.00%
50	-84	40	100	54.4	0.000	45.1	0.000	0.000	0.00%	0.00%
51	-86	40	100	54.5	0.000	44.8	0.000	0.000	0.00%	0.00%
52	-88	40	100	54.6	0.000	44.5	0.000	0.000	0.00%	0.00%
53	-90	40	100	54.7	0.000	44.2	0.000	0.000	0.00%	0.00%
54	-92	40	100	54.8	0.000	43.9	0.000	0.000	0.00%	0.00%
55	-94	40	100	54.9	0.000	43.6	0.000	0.000	0.00%	0.00%
56	-96	40	100	55.0	0.000	43.3	0.000	0.000	0.00%	0.00%
57	-98	40	100	55.1	0.000	43.0	0.000	0.000	0.00%	0.00%
58	-100	40	100	55.2	0.000	42.7	0.000	0.000	0.00%	0.00%
59	-102	40	100	55.3	0.000	42.4	0.000	0.000	0.00%	0.00%
60	-104	40	100	55.4	0.000	42.1	0.000	0.000	0.00%	0.00%
61	-106	40	100	55.5	0.000	41.8	0.000	0.000	0.00%	0.00%
62	-108	40	100	55.6	0.000	41.5	0.000	0.000	0.00%	0.00%
63	-110	40	100	55.7	0.000	41.2	0.000	0.000	0.00%	0.00%
64	-112	40	100	55.8	0.000	40.9	0.000	0.000	0.00%	0.00%
65	-114	40	100	55.9	0.000	40.6	0.000	0.000	0.00%	0.00%
66	-116	40	100	56.0	0.000	40.3	0.000	0.000	0.00%	0.00%
67	-118	40	100	56.1	0.000	40.0	0.000	0.000	0.00%	0.00%
68	-120	40	100	56.2	0.000	39.7	0.000	0.000	0.00%	0.00%
69	-122	40	100	56.3	0.000	39.4	0.000	0.000	0.00%	0.00%
70	-124	40	100	56.4	0.000	39.1	0.000	0.000	0.00%	0.00%
71	-126	40	100	56.5	0.000	38.8	0.000	0.000	0.00%	0.00%
72	-128	40	100	56.6	0.000	38.5	0.000	0.000	0.00%	0.00%
73	-130	40	100	56.7	0.000	38.2	0.000	0.000	0.00%	0.00%
74	-132	40	100	56.8	0.000	37.9	0.000	0.000	0.00%	0.00%
75	-134	40	100	56.9	0.000	37.6	0.000	0.000	0.00%	0.00%
76	-136	40	100	57.0	0.000	37.3	0.000	0.000	0.00%	0.00%
77	-138	40	100	57.1	0.000	37.0	0.000	0.000	0.00%	0.00%
78	-140	40	100	57.2	0.000	36.7	0.000	0.000	0.00%	0.00%
79	-142	40	100	57.3	0.000	36.4	0.000	0.000	0.00%	0.00%
80	-144	40	100	57.4	0.000	36.1	0.000	0.000	0.00%	0.00%
81	-146	40	100	57.5	0.000	35.8	0.000	0.000	0.00%	0.00%
82	-148	40	100	57.6	0.000	35.5	0.000	0.000	0.00%	0.00%
83	-150	40	100	57.7	0.000	35.2	0.000	0.000	0.00%	0.00%
84	-152	40	100	57.8	0.000	34.9	0.000	0.000	0.00%	0.00%
85	-154	40	100	57.9	0.000	34.6	0.000	0.000	0.00%	0.00%
86	-156	40	100	58.0	0.000	34.3	0.000	0.000	0.00%	0.00%
87	-158	40	100	58.1	0.000	34.0	0.000	0.000	0.00%	0.00%
88	-160	40	100	58.2	0.000	33.7	0.000	0.000	0.00%	0.00%
89	-162	40	100	58.3	0.000	33.4	0.000	0.000	0.00%	0.00%
90	-164	40	100	58.4	0.000	33.1	0.000	0.000	0.00%	0.00%
91	-166	40	100	58.5	0.000	32.8	0.000	0.000	0.00%	0.00%
92	-168	40	100	58.6	0.000	32.5	0.000	0.000	0.00%	0.00%
93	-170	40	100	58.7	0.000	32.2	0.000	0.000	0.00%	0.00%
94	-172	40	100	58.8	0.000	31.9	0.000	0.000	0.00%	0.00%
95	-174	40	100	58.9	0.000	31.6	0.000	0.000	0.00%	0.00%
96	-176	40	100	59.0	0.000	31.3	0.000	0.000	0.00%	0.00%
97	-178	40	100	59.1	0.000	31.0	0.000	0.000	0.00%	0.00%
98	-180	40	100	59.2	0.000	30.7	0.000	0.000	0.00%	0.00%
99	-182	40	100	59.3	0.000	30.4	0.000	0.000	0.00%	0.00%
100	-184	40	100	59.4	0.000	30.1	0.000	0.000	0.00%	0.00%
101	-186	40	100	59.5	0.000	29.8	0.000	0.000	0.00%	0.00%
102	-188	40	100	59.6	0.000	29.5	0.000	0.000	0.00%	0.00%
103	-190	40	100	59.7	0.000	29.2	0.000	0.000	0.00%	0.00%
104	-192	40	100	59.8	0.000	28.9	0.000	0.000	0.00%	0.00%
105	-194	40	100	59.9	0.000	28.6	0.000	0.000	0.00%	0.00%
106	-196	40	100	60.0	0.000	28.3	0.000	0.000	0.00%	0.00%
107	-198	40	100	60.1	0.000	28.0	0.000	0.000	0.00%	0.00%
108	-200	40	100	60.2	0.000	27.7	0.000	0.000	0.00%	0.00%
109	-202	40	100	60.3	0.000	27.4	0.000	0.000	0.00%	0.00%
110	-204	40	100	60.4	0.000	27.1	0.000	0.000	0.00%	0.00%
111	-206	40	100	60.5	0.000	26.8	0.000	0.000	0.00%	0.00%
112	-208	40	100	60.6	0.000	26.5	0.000	0.000	0.00%	0.00%
113	-210	40	100	60.7	0.000	26.2	0.000	0.000	0.00%	0.00%
114	-212	40	100	60.8	0.000	25.9	0.000	0.000	0.00%	0.00%
115	-214	40	100	60.9	0.000	25.6	0.000	0.000	0.00%	0.00%
116	-216	40	100	61.0	0.000	25.3	0.000	0.000	0.00%	0.00%
117	-218	40	100	61.1	0.000	25.0	0.000	0.000	0.00%	0.00%
118	-220	40	100	61.2	0.000	24.7	0.000	0.000	0.00%	0.00%
119	-222	40	100	61.3	0.000	24.4	0.000	0.000	0.00%	0.00%
120	-224	40	100	61.4	0.000	24.1	0.000	0.000	0.00%	0.00%
121	-226	40	100	61.5	0.000	23.8	0.000	0.000	0.00%	0.00%
122	-228	40	100	61.6	0.000	23.5	0.000	0.000	0.00%</	

A.2 Condenser Development

This section contains information regarding the development of the ACU condenser model.

A.2.1 Condenser Roughness and K-Values

The following page displays the roughness values used for the condenser piping, as well as the K-values as a result of bends Figure 62.

To continue reading from where you left off in the main dissertation, [click here](#).

Roughness Values Used		
Value	Unit	Description
90	μm	Commercial Steel (Inner Pipe)
30	μm	Stainless steel
97.35	μm	Combination (Annulus)

Secondary Loss Factors (K-Values) Outer Pipe	
Value	Description
0.32454	Bend 1
0.32434	Bend 2
0.32434	Bend 3
0.32489	Bend 4
0.32489	Bend 5
0.32563	Bend 6
0.32563	Bend 7
0.32666	Bend 8
0.32666	Bend 9
0.32817	Bend 10
0.32817	Bend 11
0.33060	Bend 12
0.33060	Bend 13
0.33493	Bend 14
4.58007	The 14, 90 deg bends I used to make 180 deg bends for the outer tube.

Secondary Loss Factors (K-Values) Inner Pipe	
Value	Description
0.36737	Bend 1
0.36737	Bend 2
0.36826	Bend 3
0.36827	Bend 4
0.36923	Bend 5
0.36923	Bend 6
0.37029	Bend 7
0.37029	Bend 8
0.37145	Bend 9
0.37145	Bend 10
0.37295	Bend 11
0.37296	Bend 12
4.43913	The 12, 90 deg bends I used to make 180 deg bends for the inner tube.

Figure 62: Condenser roughness and K-values

A.2.2 Results Comparison of Simple and Complex Condenser Models

The heat transfer and pressure drop results of the simple and complex condenser models were compared to ensure that model accuracy is not compromised by using the simplified model. For this comparison, the simple model results were tested against the complex model results, so the simple model results have the test suffix (*T-) and the complex model results have the reference suffix (*R). The comparison was performed using the same input range as the condenser verification process, with the addition of water inlet pressure variation to better test the water side pressure-drop comparison between the two models. The results are presented on the following page as Figure 63.

To continue reading from where you left off in the main dissertation, [click here](#).

Condition	% Refrigerant Inlet Pressure [bar]	% Refrigerant Inlet Temperature [C]	% Refrigerant Mass Flowrate [kg/s]	% Water Inlet Pressure [bar]	% Water Inlet Temperature [C]	% Water Mass Flowrate [kg/s]	*H Heat Transfer [kW]	*H Heat Transfer [kW]	% Refrigerant Pressure Drop [bar]	% Refrigerant Pressure Drop [bar]	% Water Pressure Drop [bar]	% Water Pressure Drop [bar]	Error Water Pressure Drop	Error Refrigerant Pressure Drop	Error Heat Transfer
1	21.00	100.0	0.800	2.50	25.0	3.600	196.068	196.041	0.1892	0.1825	0.1889	0.1891	0.11%	1.66%	0.24%
2	21.50	100.0	0.800	2.50	25.0	3.600	195.851	195.787	0.1789	0.1714	0.1869	0.1862	0.13%	4.39%	0.89%
3	22.00	100.0	0.800	2.50	25.0	3.600	195.583	195.501	0.1689	0.1616	0.1800	0.1803	0.16%	4.53%	0.11%
4	22.50	100.0	0.800	2.50	25.0	3.600	194.832	195.317	0.1536	0.1536	0.1990	0.1993	0.56%	4.73%	0.26%
5	23.00	100.0	0.800	2.50	25.0	3.600	194.380	194.969	0.1414	0.1445	0.1991	0.1994	0.22%	5.13%	0.23%
6	21.00	80.0	0.800	2.50	25.0	3.600	179.812	179.305	0.1537	0.1625	0.1993	0.1994	0.22%	5.43%	0.24%
7	21.00	90.0	0.800	2.50	25.0	3.600	187.142	187.241	0.1824	0.1732	0.1989	0.1992	0.14%	5.29%	0.05%
8	21.00	100.0	0.800	2.50	25.0	3.600	196.068	196.041	0.1892	0.1825	0.1989	0.1991	0.11%	1.66%	0.05%
9	21.00	110.0	0.800	2.50	25.0	3.600	204.716	204.733	0.1955	0.1906	0.1988	0.1990	0.50%	2.59%	0.04%
10	21.00	120.0	0.800	2.50	25.0	3.600	213.395	213.367	0.2094	0.1979	0.1988	0.1989	0.90%	2.96%	0.02%
11	21.00	100.0	0.658	2.50	25.0	3.600	162.123	162.166	0.2113	0.1955	0.1990	0.1990	0.19%	5.45%	0.08%
12	21.00	100.0	0.732	2.50	25.0	3.600	180.113	180.257	0.1495	0.1495	0.1995	0.1995	0.10%	4.99%	0.08%
13	21.00	100.0	0.808	2.50	25.0	3.600	197.812	197.898	0.1843	0.1876	0.1988	0.1991	0.11%	1.71%	0.06%
14	21.00	100.0	0.884	2.50	25.0	3.600	214.592	214.744	0.2494	0.2421	0.1985	0.1988	0.59%	3.00%	0.01%
15	21.00	100.0	0.962	2.50	25.0	3.600	230.658	230.838	0.3027	0.3081	0.1982	0.1982	0.96%	2.79%	0.04%
16	21.00	100.0	0.800	2.50	25.0	3.600	196.068	196.041	0.1892	0.1825	0.1989	0.1991	0.11%	1.65%	0.01%
17	21.00	100.0	0.800	6.81	25.0	3.600	196.060	196.034	0.1892	0.1826	0.1993	0.1993	0.11%	1.65%	0.01%
18	21.00	100.0	0.800	11.23	25.0	3.600	196.060	196.035	0.1893	0.1826	0.1990	0.1992	0.12%	1.65%	0.01%
19	21.00	100.0	0.800	15.63	25.0	3.600	196.053	196.057	0.1893	0.1827	0.1989	0.1992	0.11%	1.65%	0.01%
20	21.00	100.0	0.800	20.05	25.0	3.600	196.043	196.027	0.1894	0.1824	0.1991	0.1994	0.14%	1.65%	0.01%
21	21.50	100.0	0.800	18.87	25.0	3.600	205.722	205.431	0.1436	0.1536	0.2004	0.2003	0.50%	4.66%	0.14%
22	21.00	100.0	0.800	21.5	25.0	3.600	200.832	200.832	0.1743	0.1665	0.2008	0.2014	0.31%	3.88%	0.17%
23	21.00	100.0	0.800	25.0	25.1	3.600	195.950	195.955	0.1675	0.1675	0.1989	0.1990	0.11%	1.61%	0.01%
24	21.00	100.0	0.800	28.6	25.0	3.600	190.663	190.444	0.2072	0.2039	0.1970	0.1969	0.97%	1.89%	0.12%
25	21.00	100.0	0.800	32.2	25.0	3.600	185.312	185.656	0.2547	0.2399	0.1954	0.1954	0.21%	1.66%	0.01%
26	21.00	100.0	0.800	2.50	25.0	3.818	196.095	196.064	0.1889	0.1822	0.2008	0.2010	0.10%	1.69%	0.01%
27	21.00	100.0	0.800	4.738	25.0	4.738	196.849	197.050	0.1751	0.1676	0.2041	0.2041	0.20%	1.49%	0.01%
28	21.00	100.0	0.800	5.859	25.0	5.859	197.919	197.919	0.1594	0.1594	0.2000	0.1999	0.58%	1.81%	0.24%
29	21.00	100.0	0.800	6.980	25.0	6.980	197.063	197.063	0.1434	0.1434	0.2016	0.2016	0.88%	1.68%	0.12%
30	21.00	100.0	0.800	8.102	25.0	8.102	198.095	197.859	0.1411	0.1504	0.2073	0.2078	0.42%	1.48%	0.12%
Average Error													0.23%	4.97%	0.11%
Maximum Error													1.84%	7.87%	0.88%

Figure 63: Results comparison of simple and complex condenser models

A.2.3 Condenser EES Verification Models

The verification models for the condenser were developed in EES and are presented in this section.

To continue reading from where you left off in the main dissertation, [click here](#).

A.2.3.1 EES Code for Segmented Condenser Model: Superheated

\$Load Incompressible

\$Load NASA

{Fluid 1 = Water}

{Fluid 2 = R407C}

{Tube-in-tube refrigerant-to-water counterflow condenser}

{!*****Functions*****}

{!*****Nusselt Number*****}

Function mynuds_fluid_1(a,b)

If (a< 2300) **Then**

NuD_Fluid_1=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_1=0.023*a^(4/5)*b^{0.4} {Value in case of turbulent flow}{Heating coefficient = 0,4}

Endif

mynuds_fluid_1=NuD_Fluid_1

End

Function mynuds_fluid_2(x,y)

If (x< 2300) **Then**

NuD_Fluid_2=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_2=0.023*x^(4/5)*y^{0.3} {Value in case of turbulent flow}{Cooling coefficient = 0,3}

Endif

mynuds_fluid_2=NuD_Fluid_2

End

{!*****End*****}

{!*****Main Code*****}

{FLUID PROPERTIES: INLET TEMPERATURE}

T_out_Fluid_1[0] = 25 {Temperature of fluid 1 entering the inner tube}{Unit C}

T_out_Fluid_2[10] = 100 {Temperature of fluid 2 entering the outer tube}{Unit C}

{FLUID PROPERTIES: INLET PRESSURE}

P_Fluid_1[0] = 250000 {Pressure of fluid 1 entering the inner tube}{Unit Pa}

P_Fluid_2[10] = 2100000 {Pressure of fluid 2 entering the outer tube}{Unit Pa}

{MASS FLOW RATE}

m_dot_Fluid_1 = 0.225 {Mass flowrate of fluid 1 passing through inner tube}{Unit kg/s}

m_dot_Fluid_2 = 0.09 {Mass flowrate of fluid 2 passing through outer tube}{Unit kg/s}

{TUBE LENGTH}

L_tot = 1 {Total tube length}{Unit m}

{TUBE GEOMETRY: DIAMETER & WALL THICKNESS}

$D_{ii} = 0.01576$ {Inner diameter of the inner tube}{Unit m}
 $WT_{in_Fluid_1} = 0.00277$ {Wall thickness of the inner tube}{Unit m}
 $D_{io} = (D_{ii} + 2 * WT_{in_Fluid_1})$ {Outer diameter of the inner tube}{Unit m}
 $D_{h_Fluid_1} = D_{ii}$ {Hydraulic diameter of the inner tube}{Unit m}

$D_{oi} = 0.02664$ {Inner diameter of the outer tube}{Unit m}
 $WT_{in_Fluid_2} = 0.00338$ {Wall thickness of the outer tube}{Unit m}
 $D_{oo} = D_{oi} + 2 * WT_{in_Fluid_2}$ {Outer diameter of the outer tube}{Unit m}
 $D_{h_Fluid_2} = D_{oi} - D_{io}$ {Hydraulic diameter of the outer tube}{Unit m}

{TUBE GEOMETRY: FOULING}

$Epsilon_i = 100 * 10^{-6}$ {Roughness of inner tube} {Units m}
 $Epsilon_o = 100 * 10^{-6}$ {Roughness of outer tube} {Units m}

$RR_i = Epsilon_i / D_{h_Fluid_1}$ {Relative Roughness of inner tube} {Units -}
 $RR_o = Epsilon_o / D_{h_Fluid_2}$ {Relative Roughness of outer tube} {Units -}

$Rf_i = 0$ {Fouling factor for inner tube}{Units $m^2 \cdot K \cdot s / J$ }
 $Rf_o = 0$ {Fouling factor for outer tube}{Units $m^2 \cdot K \cdot s / J$ }

{TUBE GEOMETRY: AREA CALCULATIONS}

$A_{Fluid_1_cross} = \pi * (D_{ii} / 2)^2$ {Cross sectional area of inner tube}{Unit m^2 }
 $A_{Fluid_2_cross} = \pi * (D_{oi} / 2)^2 - \pi * (D_{io} / 2)^2$ {Cross sectional area of outer tube accounting for the presence of the inner tube}{Unit m^2 }

"!WATER: All regions"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{Fluid_1[i]} = \text{conductivity}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Conductivity of fluid 1}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

$\rho_{Fluid_1[i]} = \text{density}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Density of fluid 1}{Unit kg/m³}

{EFF-NTU METHOD: PR}

$Pr_{Fluid_1[i]} = \text{prandtl}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Prandtl number of fluid 1} {Units -}

{EFF-NTU METHOD: VISCOSITY}

$\mu_{Fluid_1[i]} = \text{viscosity}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Viscosity of fluid 1}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_{Fluid_1[i]} = \text{cp}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Specific heat of fluid 1} {Units J/kg-K}

End

"!REFRIGERANT: Superheated region"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{Fluid_2[i]} = \text{conductivity}(R407C, T=T_{Fluid_2_avg[i]}, P=P_{Fluid_2[i]})$ {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

$\rho_{Fluid_2[i]} = \text{density}(R407C, T=T_{Fluid_2_avg[i]}, P=P_{Fluid_2[i]})$ {Density of fluid 2}{Unit kg/m³}

{EFF-NTU METHOD: PR}

$Pr_{Fluid_2}[i] = \text{prandtl}(R407C, T=T_{Fluid_2_avg}[i], P=P_{Fluid_2}[i])$ {Prandtl number of fluid 2} {Units -}

{EFF-NTU METHOD: VISCOSITY}

$\mu_{Fluid_2}[i] = \text{viscosity}(R407C, T=T_{Fluid_2_avg}[i], P=P_{Fluid_2}[i])$ {Viscosity of fluid 2}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_{Fluid_2}[i] = \text{cp}(R407C, T=T_{Fluid_2_avg}[i], P=P_{Fluid_2}[i])$ {Specific heat of fluid 2} {Units J/kg-K}

End

"!REFRIGERANT: All regions"

Duplicate i = 0+1 , 10

$L[i] = L_{tot}/10$

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{stainless_steel}[i] = \text{conductivity}(\text{Stainless_AISI304}, T=((T_{Fluid_1_avg}[i]+T_{Fluid_2_avg}[i])/2))$ {Conductivity of the barrier separating the two fluids}{Units J/s-m-K}

{EFF-NTU METHOD: NUD}

$NuD_{Fluid_1}[i] = \text{mynuds_fluid_1}(Re_{Fluid_1}[i], Pr_{Fluid_1}[i])$ {Nusselt number of fluid 1} {Units -}

$NuD_{Fluid_2}[i] = \text{mynuds_fluid_2}(Re_{Fluid_2}[i], Pr_{Fluid_2}[i])$ {Nusselt number of fluid 2} {Units -}

{EFF-NTU METHOD: hD}

$hD_{Fluid_1}[i] = NuD_{Fluid_1}[i] * k_{Fluid_1}[i] / D_{h_Fluid_1}$ {Convection coefficient of fluid 1}{Unit J/s-m²-K}

$hD_{Fluid_2}[i] = NuD_{Fluid_2}[i] * k_{Fluid_2}[i] / D_{h_Fluid_2}$ {Convection coefficient of fluid 2}{Unit J/s-m²-K}

{EFF-NTU METHOD: C}

$C_{Fluid_1}[i] = m_{dot_Fluid_1} * cp_{Fluid_1}[i]$ {Heat capacity rate of fluid 1} {Units J/s-K}

$C_{Fluid_2}[i] = m_{dot_Fluid_2} * cp_{Fluid_2}[i]$ {Heat capacity rate of fluid 2} {Units J/s-K}

$C_{min}[i] = \min(C_{Fluid_1}[i], C_{Fluid_2}[i])$ {Minimum heat capacity rate of fluid 1 & fluid 2} {Units J/s-K}

$C_{max}[i] = \max(C_{Fluid_1}[i], C_{Fluid_2}[i])$ {Maximum heat capacity rate of fluid 1 & fluid 2} {Units J/s-K}

$C_r[i] = C_{min}[i] / C_{max}[i]$ {Ratio of the greatest fluctuation of heat capacity rate}{Unit -}

{EFF-NTU METHOD: NTU}

$NTU[i] = UA[i] / C_{min}[i]$ {Number of transfer units} {Units -}

{EFF-NTU METHOD: EFFICIENCY}

$eff[i] = (1 - \exp(-NTU[i] * (1 - C_r[i]))) / (1 - C_r[i] * \exp(-NTU[i] * (1 - C_r[i])))$ {Efficiency of heat transfer}{Unit -}

{EFF-NTU METHOD: SYSTEM RESISTANCE}

$1/UA[i]$ =
 $1/(A_{ii_Fluid_1}[i] * hD_{Fluid_1}[i] + Rf_i/A_{ii_Fluid_1}[i] + \ln(D_{io}/D_{h_Fluid_1})/(2 * \pi * k_{stainless_steel}[i] * L[i]) + Rf_o/A_{io_Fluid_2}[i] + 1/(A_{io_Fluid_2}[i] * hD_{Fluid_2}[i]))$ {Heat exchange heat transfer coefficient}{Units K-s/J}

$A_{ii_Fluid_1}[i] = \pi * D_{h_Fluid_1} * L[i]$ {Inner surface area of inner tube}{Unit m²}

$A_{io_Fluid_2}[i] = \pi * D_{io} * L[i]$ {Outer surface area of inner tube}{Unit m²}

{EFF-NTU METHOD: ENTHALPY}

$h_{out_Fluid_1[i]} = \text{enthalpy}(\text{Water}, T=T_{out_Fluid_1[i]}, P=P_{Fluid_1[i]})$ {Enthalpy of fluid 1}{Units J/kg}
 $h_{out_Fluid_2[i]} = \text{enthalpy}(R407C, T=T_{out_Fluid_2[i]}, P=P_{Fluid_2[i]})$ {Enthalpy of fluid 2}{Units J/kg}
 $h_{in_Fluid_1[i]} = \text{enthalpy}(\text{Water}, T=T_{in_Fluid_1[i]}, P=P_{Fluid_1[i]})$ {Enthalpy of fluid 1}{Units J/kg}
 $h_{in_Fluid_2[i]} = \text{enthalpy}(R407C, T=T_{in_Fluid_2[i]}, P=P_{Fluid_2[i]})$ {Enthalpy of fluid 2}{Units J/kg}

{EFF-NTU METHOD: ENTROPY}

$s_{Fluid_1[i]} = \text{entropy}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Entropy of fluid 1}{Units J/kg-K}
 $s_{Fluid_2[i]} = \text{entropy}(R407C, T=T_{Fluid_2_avg[i]}, P=P_{Fluid_2[i]})$ {Entropy of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: QUALITY}

$x_{Fluid1[i]} = \text{quality}(\text{Water}, T=T_{Fluid_1_avg[i]}, s=s_{Fluid_1[i]})$ {Quality of fluid 1 inaccurate in the saturated state}{Units -}
 $x_{Fluid2[i]} = \text{quality}(R407C, T=T_{Fluid_2_avg[i]}, s=s_{Fluid_2[i]})$ {Quality of fluid 2 inaccurate in the saturated state}{Units -}

{EFF-NTU METHOD: REYNOLDS NUMBER}

$Re_{Fluid_1[i]} = (\rho_{Fluid_1[i]} \cdot vel_{in_Fluid_1[i]} \cdot D_{h_Fluid_1}) / (\mu_{Fluid_1[i]})$ {Reynolds number of fluid passing through inner tube}{Unit -}
 $ReD_{Fluid_2[i]} = (\rho_{Fluid_2[i]} \cdot vel_{in_Fluid_2[i]} \cdot D_{h_Fluid_2}) / (\mu_{Fluid_2[i]})$ {Reynolds number of fluid passing through outer tube}{Unit -}

{EFF-NTU METHOD: FRICTION FACTOR}

$f_{i[i]} = \text{moodychart}(Re_{Fluid_1[i]}, RR_i)$ {Friction factor of fluid in inner tube}{Units -}
 $f_{o[i]} = \text{moodychart}(ReD_{Fluid_2[i]}, RR_o)$ {Friction factor of fluid in outer tube}{Units -}

{EFF-NTU METHOD: FLUID PROPERTIES: FLOW VELOCITY}

$vel_{in_Fluid_1[i]} = m_dot_Fluid_1 / (\rho_{Fluid_1[i]} \cdot A_{Fluid_1_cross})$ {Velocity of fluid passing through inner tube}{Unit m/s}
 $vel_{in_Fluid_2[i]} = m_dot_Fluid_2 / (\rho_{Fluid_2[i]} \cdot A_{Fluid_2_cross})$ {Velocity of fluid passing through outer tube}{Unit m/s}

{EFF-NTU METHOD: HEAT TRANSFER}

$q_{max[i]} = C_{min[i]} \cdot (T_{out_Fluid_2[i]} - T_{in_Fluid_1[i]})$ { op node warmste - koudste } {Greatest achievable heat transfer}{Unit J/s}

{EFF-NTU METHOD: PRESSURE DROP}

$P_{Fluid_1[i]} = P_{Fluid_1[i-1]} - \Delta P_{prim_i[i]}$ {Pressure relation between points in succession in the inner tube}{Unit Pa}
 $P_{Fluid_2[i-1]} = P_{Fluid_2[i]} - \Delta P_{prim_o[i]}$ {Pressure relation between points in succession in the outer tube}{Unit Pa}
 $\Delta P_{prim_i[i]} = f_{i[i]} \cdot L[i] / D_{h_Fluid_1} \cdot 1/2 \cdot \rho_{Fluid_1[i]} \cdot vel_{in_Fluid_1[i]}^2$ {Primary pressure loss in inner tube}{Unit Pa}
 $\Delta P_{prim_o[i]} = f_{o[i]} \cdot L[i] / D_{h_Fluid_2} \cdot 1/2 \cdot \rho_{Fluid_2[i]} \cdot vel_{in_Fluid_2[i]}^2$ {Primary pressure loss in outer tube}{Unit Pa}

{EFF-NTU METHOD: TEMPERATURES}

$T_{in_Fluid_1[i]} = T_{out_Fluid_1[i-1]}$ {Temperature relation between points in succession in the inner tube}{Unit C}
 $T_{in_Fluid_2[i]} = T_{out_Fluid_2[i-1]}$ {Temperature relation between points in succession in the outer tube}{Unit C}

$T_{Fluid_1_avg[i]} = (T_{in_Fluid_1[i]} + T_{out_Fluid_1[i]}) / 2$ {Average temperature at any given point along the inner tube}{Unit C}
 $T_{Fluid_2_avg[i]} = (T_{in_Fluid_2[i]} + T_{out_Fluid_2[i]}) / 2$ {Average temperature at any given point along the outer tube}{Unit C}

End

{HEAT TRANSFER EQUATIONS}

Duplicate i = 1 , 10

$q[i] = m_dot_Fluid_1 \cdot (h_{out_Fluid_1[i]} - h_{in_Fluid_1[i]})$ {Heat transfer over inner tube segment}{Unit J/s}
 $q[i] = m_dot_Fluid_2 \cdot (h_{out_Fluid_2[i]} - h_{in_Fluid_2[i]})$ {Heat transfer over outer tube segment}{Unit J/s}
 $q[i] = UA[i] \cdot (T_{Fluid_2_avg[i]} - T_{Fluid_1_avg[i]})$ {Heat transfer between tube segments}{Unit J/s}

End

q_tot = **sum**(q[i],i=1,10) {Total heat transfer}{Unit J/s}

{PRESSURE DROP OVER TUBE SECTIONS}

Pdrop_Fluid_1 = P_Fluid_1[0]-P_Fluid_1[10] {Pressure drop along inner tube}{Unit Pa}

Pdrop_Fluid_2 = P_Fluid_2[10]-P_Fluid_2[0] {Pressure drop along outer tube}{Unit Pa}

{!*****End*****}

A.2.3.2 EES Code for Segmented Condenser Model: Two-phase

\$Load Incompressible

\$Load NASA

{Fluid 1 = Water}

{Fluid 2 = R407C}

{Tube-in-tube refrigerant-to-water counterflow condenser}

{*****Functions*****}

{*****Nusselt Number*****}

Function mynuds_fluid_1(a,b)

If (a< 2300) **Then**

NuD_Fluid_1=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_1=0.023*a^(4/5)*b ^0.4 {Value in case of turbulent flow}{Heating coefficient = 0,4}

Endif

mynuds_fluid_1=NuD_Fluid_1

End

Function mynuds_fluid_2(x,y,a,b)

If (x< 2300) **Then**

NuD_Fluid_2=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_2=0.023*x^(4/5)*y ^0.3 *((1-a)^0.8+(3.8*a^0.76*(1-a)^0.04)/(b)) {Value in case of turbulent flow}{Cooling coefficient = 0,3}

Endif

mynuds_fluid_2=NuD_Fluid_2

End

{*****End*****}

{*****Friction Factor*****}

Procedure frictionfactor_singlephase(Re,D,epsilon:fric) {Calculation of homogeneous friction factor for single phase}

If (Re < 2300) **Then**

fric := 64 / Re

Else

If (Re < 5000) **Then**

f1 := 64 / 2300

f2:= 0.25/(log10(epsilon / 3.7 / D + 5.74 / 5000^0.9))^2 {For turbulent flow}

fric := (f2-f1)*(Re - 2300)/(5000-2300) + f1 {Interpolating on the Moody chart}

Else

fric := 0.25/(log10(epsilon / 3.7 / D + 5.74 / Re^0.9))^2

Endif

Endif

End

Procedure frictionfactor_twophase(p,h,mdot,A,D,epsilon:fric) {Calculation of friction factor for two phase}

x := quality(R407C,P=p,h=h)

rho := density(R407C,P=p,h=h)

G := mdot/A

If (x > 0) and (x < 1) **Then**

rho_L := density(R407C,P=p,x=0)

rho_G := density(R407C,P=p,x=1)

mu_L := viscosity(R407C,P=p,x=0) {Lockhart Martinelli}

```

mu_G := viscosity(R407C,P=p,x=1)

Re_L := G * (1 - x) * D / mu_L
Re_G := G * x * D / mu_G

Call frictionfactor_singlephase(Re_L,D,epsilon:f_L)
Call frictionfactor_singlephase(Re_G,D,epsilon:f_G)

lamda := rho_G/rho_L * (mu_L/mu_G)^0.2
t1 := exp(-(log10(lamda)+2.5)^2/(2.4 - G*1.0e-4))

C := -2 + (28-0.3*sqrt(G))*t1
If (C < 2) Then
    C := 2
Endif

fric := rho*(f_L*(1-x)*(1-x)/rho_L + C*x*(1-x)*sqrt(f_L*f_G/rho_L/rho_G) + f_G*x*x/rho_G)
Else
    mu := viscosity(R407C,P=p,x=x)
    Re := G * D / mu
    Call frictionfactor_singlephase(Re,D,epsilon:fric) {For single phase calculations call single phase function}
Endif

End
{!*****End*****}

{!*****Main Code*****}
{FLUID PROPERTIES: INLET TEMPERATURE}
T_out_Fluid_1[0] = 29 {Temperature of fluid 1 entering the inner tube}{Units C}
T_out_Fluid_2[10] = 49.2 {Temperature of fluid 2 entering the outer tube}{Units C}

{FLUID PROPERTIES: INLET PRESSURE}
P_Fluid_1[0] = 250000 {Pressure of fluid 1 entering the inner tube}{Units Pa}
P_Fluid_2[10] = 1960000 {Pressure of fluid 2 entering the outer tube}{Units Pa}

{MASS FLOW RATE}
m_dot_Fluid_1 = 0.225 {Mass flowrate of fluid 1 passing through inner tube}{Units kg/s}
m_dot_Fluid_2 = 0.04 {Mass flowrate of fluid 2 passing through outer tube}{Units kg/s}

{TUBE LENGTH}
L_tot = 4 {Total tube length}{Units m}

{TUBE GEOMETRY: DIAMETER & WALL THICKNESS}
D_ii = 0.01576 {Inner diameter of the inner tube}{Units m}
WT_in_Fluid_1 = 0.00277 {Wall thickness of the inner tube}{Units m}
D_io = (D_ii+2*WT_in_Fluid_1) {Outer diameter of the inner tube}{Units m}
D_h_Fluid_1 = D_ii {Hydraulic diameter of the inner tube}{Units m}

D_o_i = 0.02664 {Inner diameter of the outer tube}{Units m}
WT_in_Fluid_2 = 0.00338 {Wall thickness of the outer tube}{Units m}
D_oo = D_o_i+2*WT_in_Fluid_2 {Outer diameter of the outer tube}{Units m}
D_h_Fluid_2 = D_o_i-D_o_i {Hydraulic diameter of the outer tube}{Units m}

{TUBE GEOMETRY: FOULING}
Epsilon_i = 100*10^(-6) {Roughness of inner tube}{Units m}
Epsilon_o = 100*10^(-6) {Roughness of outer tube}{Units m}

```

RR_i = Epsilon_i/D_h_Fluid_1 {Relative Roughness of inner tube}{Units -}
 RR_o = Epsilon_o/D_h_Fluid_2 {Relative Roughness of outer tube}{Units -}

Rf_i = 0 {Fouling factor for inner tube}{Units m^2-K-s/J}
 Rf_o = 0 {Fouling factor for outer tube}{Units m^2-K-s/J}

{TUBE GEOMETRY: AREA CALCULATIONS}

A_Fluid_1_cross = pi*(D_ii/2)^2 {Cross sectional area of inner tube}{Units m^2}
 A_Fluid_2_cross = pi*(D_oi/2)^2-pi*(D_io/2)^2 {Cross sectional area of outer tube accounting for the presence of the inner tube}{Units m^2}

"!WATER: All regions"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_1[i] = **conductivity**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Conductivity of fluid 1}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_1[i] = **density**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Density of fluid 1}{Units kg/m^3}

{EFF-NTU METHOD: PR}

Pr_Fluid_1[i] = **prandtl**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Prandtl number of fluid 1}{Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_1[i] = **viscosity**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Viscosity of fluid 1}{Units N-s/m^2}

{EFF-NTU METHOD: CP}

cp_Fluid_1[i] = **cp**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Specific heat of fluid 1}{Units J/kg-K}

End

"!REFRIGERANT: Two-phase region"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_2_ff[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], x=0) {Conductivity of fluid 2 when saturated liquid}{Units J/s-m-K}
 k_Fluid_2_g[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], x=1) {Conductivity of fluid 2 when saturated vapour}{Units J/s-m-K}
 k_Fluid_2[i] = x_Fluid2[i]*(k_Fluid_2_g[i]-k_Fluid_2_ff[i])+k_Fluid_2_ff[i] {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_2[i] = **density**(R407C, T=T_Fluid_2_avg[i], x=x_Fluid2[i]) {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: PR}

Pr_Fluid_2_ff[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], x=0) {Prandtl number of fluid 2 when saturated liquid}{Units -}
 Pr_Fluid_2_g[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], x=1) {Prandtl number of fluid 2 when saturated vapour}{Units -}
 Pr_Fluid_2[i] = x_Fluid2[i]*(Pr_Fluid_2_g[i]-Pr_Fluid_2_ff[i])+Pr_Fluid_2_ff[i] {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_2_ff[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], x=0) {Viscosity of fluid 2 when saturated liquid}{Units N-s/m^2}
 mu_Fluid_2_g[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], x=1) {Viscosity of fluid 2 when saturated vapour}{Units N-s/m^2}
 mu_Fluid_2[i] = x_Fluid2[i]*(mu_Fluid_2_g[i]-mu_Fluid_2_ff[i])+mu_Fluid_2_ff[i] {Viscosity of fluid 2}{Units N-s/m^2}

{EFF-NTU METHOD: CP}

```

cp_Fluid_2_ff[i] = cp(R407C,T=T_Fluid_2_avg[i],x=0) {Specific heat of fluid 2 when saturated liquid}{Units J/kg-K}
cp_Fluid_2_g[i] = cp(R407C,T=T_Fluid_2_avg[i],x=1) {Specific heat of fluid 2 when saturated vapour}{Units J/kg-K}
cp_Fluid_2[i] = x_Fluid2[i]*(cp_Fluid_2_g[i]-cp_Fluid_2_ff[i])+cp_Fluid_2_ff[i] {Specific heat of fluid 2}{Units J/kg-K}
End

```

"!REFRIGERANT: All regions"

Duplicate i = 0+1 , 10

L[i] = L_tot/10

{EFF-NTU METHOD: CONDUCTIVITY}

k_stainless_steel[i] = **conductivity**(Stainless_AISI304, T=((T_Fluid_1_avg[i]+T_Fluid_2_avg[i])/2)) {Conductivity of the barrier separating the two fluids}{Units J/s-m-K}

{EFF-NTU METHOD: NUD}

NuD_Fluid_1[i] = **mynuds_fluid_1**(Re_Fluid_1[i],Pr_Fluid_1[i]) {Nusselt number of fluid 1}{Units -}

NuD_Fluid_2[i] = **mynuds_fluid_2**(ReD_Fluid_2[i],Pr_Fluid_2[i],x_Fluid2[i],**average**(P_Fluid_2[i],P_Fluid_2[i-1])/p_crit(R407C)) {Nusselt number of fluid 2}{Units -}

{EFF-NTU METHOD: hD}

hD_Fluid_1[i] = NuD_Fluid_1[i] *k_Fluid_1[i] /D_h_Fluid_1 {Convection coefficient of fluid 1}{Units J/s-m^2-K}

hD_Fluid_2[i] = NuD_Fluid_2[i]*k_Fluid_2[i]/ D_h_Fluid_2 {Convection coefficient of fluid 2}{Units J/s-m^2-K}

{EFF-NTU METHOD: C}

C_Fluid_1[i] = m_dot_Fluid_1*cp_Fluid_1[i] {Heat capacity rate of fluid 1}{Units J/s-K}

C_Fluid_2[i] = m_dot_Fluid_2*cp_Fluid_2[i] {Heat capacity rate of fluid 2}{Units J/s-K}

C_min[i] = **min**(C_Fluid_1[i] , C_Fluid_2[i]) {Minimum heat capacity rate of fluid 1 & fluid 2}{Units J/s-K}

C_max[i] = **max**(C_Fluid_1[i] , C_Fluid_2[i]) {Maximum heat capacity rate of fluid 1 & fluid 2}{Units J/s-K}

C_r[i] = C_min[i]/C_max[i] {Ratio of the greatest fluctuation of heat capacity rate}{Units -}

{EFF-NTU METHOD: NTU}

NTU[i] = UA[i]/C_min[i] {Number of transfer units}{Units -}

{EFF-NTU METHOD: EFFICIENCY}

eff[i] = (1-**exp**(-NTU[i]*(1-C_r[i])))/(1-C_r[i]***exp**(-NTU[i]*(1-C_r[i]))) {Efficiency of heat transfer}{Units -}

{EFF-NTU METHOD: SYSTEM RESISTANCE}

1/UA[i] = 1/(A_i_Fluid_1[i]*hD_Fluid_1[i])+Rf_i/A_i_Fluid_1[i]+ln(D_o/D_h_Fluid_1)/(2*pi*k_stainless_steel[i]*L[i])+Rf_o/A_o_Fluid_2[i]+1/(A_o_Fluid_2[i]*hD_Fluid_2[i]) {Heat exchange heat transfer coefficient}{Units K-s/J}

A_i_Fluid_1[i] = pi*D_h_Fluid_1*L[i] {Inner surface area of inner tube}{Units m^2}

A_o_Fluid_2[i] = pi*D_o_Fluid_2*L[i] {Outer surface area of inner tube}{Units m^2}

{EFF-NTU METHOD: ENTHALPY}

h_out_Fluid_1[i] = **enthalpy**(Water,T=T_out_Fluid_1[i],P=P_Fluid_1[i]) {Enthalpy of fluid 1}{Units J/kg}

h_out_Fluid_2[i] = **enthalpy**(R407C,T=T_out_Fluid_2[i],P=P_Fluid_2[i]) {Enthalpy of fluid 2}{Units J/kg}

h_in_Fluid_1[i] = **enthalpy**(Water,T=T_in_Fluid_1[i],P=P_Fluid_1[i]) {Enthalpy of fluid 1}{Units J/kg}

h_in_Fluid_2[i] = **enthalpy**(R407C,T=T_in_Fluid_2[i],P=P_Fluid_2[i]) {Enthalpy of fluid 2}{Units J/kg}

{EFF-NTU METHOD: ENTROPY}

s_Fluid_1[i] = **entropy**(*Water*, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Entropy of fluid 1}{Units J/kg-K}

s_Fluid_2[i] = **entropy**(*R407C*, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Entropy of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: QUALITY}

x_Fluid1[i] = **quality**(*Water*, T=T_Fluid_1_avg[i], s=s_Fluid_1[i]) {Quality of fluid 1 inaccurate in the saturated state}{Units -}

x_Fluid2[i] = **quality**(*R407C*, T=T_Fluid_2_avg[i], s=s_Fluid_2[i]) {Quality of fluid 2 inaccurate in the saturated state}{Units -}

{EFF-NTU METHOD: REYNOLDS NUMBER}

Re_Fluid_1[i] = (rho_Fluid_1[i]*vel_in_Fluid_1[i]*D_h_Fluid_1)/(mu_Fluid_1[i]) {Reynolds number of fluid passing through inner tube}{Units -}

ReD_Fluid_2[i] = (rho_Fluid_2[i]*vel_in_Fluid_2[i]*D_h_Fluid_2)/(mu_Fluid_2[i]) {Reynolds number of fluid passing through outer tube}{Units -}

{EFF-NTU METHOD: FRICTION FACTOR}

f_i[i] = **moodychart**(Re_Fluid_1[i], RR_i) {Friction factor of fluid in inner tube}{Units -}

Call frictionfactor_twophase(P_Fluid_2[i], h_in_Fluid_2[i], m_dot_Fluid_2, A_Fluid_2_cross, D_h_Fluid_2, Epsilon_o:f_o[i]) {Friction factor of fluid in outer tube}{Units -}

{EFF-NTU METHOD: FLUID PROPERTIES: FLOW VELOCITY}

vel_in_Fluid_1[i] = m_dot_Fluid_1/(rho_Fluid_1[i]*A_Fluid_1_cross) {Velocity of fluid passing through inner tube}{Units m/s}

vel_in_Fluid_2[i] = m_dot_Fluid_2/(rho_Fluid_2[i]*A_Fluid_2_cross) {Velocity of fluid passing through outer tube}{Units m/s}

{EFF-NTU METHOD: HEAT TRANSFER}

q_max[i] = C_min[i]*(T_out_Fluid_2[i]-T_in_Fluid_1[i]) { op node warmste - koudste}{Greatest achievable heat transfer}{Units J/s}

{EFF-NTU METHOD: PRESSURE DROP}

P_Fluid_1[i] = P_Fluid_1[i-1]-Delta_P_prim_i[i] {Pressure relation between points in succession in the inner tube}{Units Pa}

P_Fluid_2[i-1] = P_Fluid_2[i]-Delta_P_prim_o[i] {Pressure relation between points in succession in the outer tube}{Units Pa}

Delta_P_prim_i[i] = f_i[i]*L[i]/D_h_Fluid_1 *1/2*rho_Fluid_1[i]*vel_in_Fluid_1[i]^2 {Primary pressure loss in inner tube}{Units Pa}

Delta_P_prim_o[i] = f_o[i]*L[i]/D_h_Fluid_2 *1/2*rho_Fluid_2[i]*vel_in_Fluid_2[i]^2 {Primary pressure loss in outer tube}{Units Pa}

{EFF-NTU METHOD: TEMPERATURES}

T_in_Fluid_1[i] = T_out_Fluid_1[i-1] {Temperature relation between points in succession in the inner tube}{Units C}

T_in_Fluid_2[i] = T_out_Fluid_2[i-1] {Temperature relation between points in succession in the outer tube}{Units C}

T_Fluid_1_avg[i] = (T_in_Fluid_1[i] + T_out_Fluid_1[i])/2 {Average temperature at any given point along the inner tube}{Units C}

T_Fluid_2_avg[i] = (T_in_Fluid_2[i] + T_out_Fluid_2[i])/2 {Average temperature at any given point along the outer tube}{Units C}

End

{HEAT TRANSFER EQUATIONS}

Duplicate i = 1 , 10

q[i] = m_dot_Fluid_1*(h_out_Fluid_1[i]-h_in_Fluid_1[i]) {Heat transfer over inner tube segment}{Units J/s}

q[i] = m_dot_Fluid_2*(h_out_Fluid_2[i]-h_in_Fluid_2[i]) {Heat transfer over outer tube segment}{Units J/s}

q[i] = UA[i]*(T_Fluid_2_avg[i]-T_Fluid_1_avg[i]) {Heat transfer between tube segments}{Units J/s}

End

q_tot = **sum**(q[i], i=1, 10) {Total heat transfer}{Units J/s}

{PRESSURE DROP OVER TUBE SECTIONS}

Pdrop_Fluid_1 = P_Fluid_1[0]-P_Fluid_1[10] {Pressure drop along inner tube}{Units Pa}

Pdrop_Fluid_2 = P_Fluid_2[10]-P_Fluid_2[0] {Pressure drop along outer tube}{Units Pa}

{!*****End*****}

A.2.3.3 EES Code for Segmented Condenser Model: Subcool

\$Load Incompressible

\$Load NASA

{Fluid 1 = Water}

{Fluid 2 = R407C}

{Tube-in-tube refrigerant-to-water counterflow condenser}

{*****Functions*****}

{*****Nusselt Number*****}

Function mynuds_fluid_1(a,b)

If (a< 2300) **Then**

NuD_Fluid_1=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_1=0.023*a^(4/5)*b^{0.4} {Value in case of turbulent flow}{Heating coefficient = 0,4}

Endif

mynuds_fluid_1=NuD_Fluid_1

End

Function mynuds_fluid_2(x,y)

If (x< 2300) **Then**

NuD_Fluid_2=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_2=0.023*x^(4/5)*y^{0.3} {Value in case of turbulent flow}{Cooling coefficient = 0,3}

Endif

mynuds_fluid_2=NuD_Fluid_2

End

{*****End*****}

{*****Main Code*****}

{FLUID PROPERTIES: INLET TEMPERATURE}

T_out_Fluid_1[0] = 25 {Temperature of fluid 1 entering the inner tube}{Unit C}

T_out_Fluid_2[10] = 40 {Temperature of fluid 2 entering the outer tube}{Unit C}

{FLUID PROPERTIES: INLET PRESSURE}

P_Fluid_1[0] = 250000 {Pressure of fluid 1 entering the inner tube}{Unit Pa}

P_Fluid_2[10] = 2100000 {Pressure of fluid 2 entering the outer tube}{Unit Pa}

{MASS FLOW RATE}

m_dot_Fluid_1 = 0.225 {Mass flowrate of fluid 1 passing through inner tube}{Unit kg/s}

m_dot_Fluid_2 = 0.09 {Mass flowrate of fluid 2 passing through outer tube}{Unit kg/s}

{TUBE LENGTH}

L_tot = 7.55 {Total tube length}{Unit m}

{TUBE GEOMETRY: DIAMETER & WALL THICKNESS}

D_ii = 0.01576 {Inner diameter of the inner tube}{Unit m}

WT_in_Fluid_1 = 0.00277 {Wall thickness of the inner tube}{Unit m}

D_io = (D_ii+2*WT_in_Fluid_1) {Outer diameter of the inner tube}{Unit m}

D_h_Fluid_1 = D_ii {Hydraulic diameter of the inner tube}{Unit m}

D_oi = 0.02664 {Inner diameter of the outer tube}{Unit m}

WT_in_Fluid_2 = 0.00338 {Wall thickness of the outer tube}{Unit m}
D_oo = D_oi+2*WT_in_Fluid_2 {Outer diameter of the outer tube}{Unit m}
D_h_Fluid_2 = D_oi-D_io {Hydraulic diameter of the outer tube}{Unit m}

{TUBE GEOMETRY: FOULING}

Epsilon_i = 100*10⁻⁶ {Roughness of inner tube} {Units m}
Epsilon_o = 100*10⁻⁶ {Roughness of outer tube} {Units m}

RR_i = Epsilon_i/D_h_Fluid_1 {Relative Roughness of inner tube} {Units -}
RR_o = Epsilon_o/D_h_Fluid_2 {Relative Roughness of outer tube} {Units -}

Rf_i = 0 {Fouling factor for inner tube}{Units m²-K-s/J}
Rf_o = 0 {Fouling factor for outer tube}{Units m²-K-s/J}

{TUBE GEOMETRY: AREA CALCULATIONS}

A_Fluid_1_cross = pi*(D_ii/2)² {Cross sectional area of inner tube}{Unit m²}
A_Fluid_2_cross = pi*(D_oi/2)²-pi*(D_io/2)² {Cross sectional area of outer tube accounting for the presence of the inner tube}{Unit m²}

"!WATER: All regions"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_1[i] = **conductivity**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Conductivity of fluid 1}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_1[i] = **density**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Density of fluid 1}{Unit kg/m³}

{EFF-NTU METHOD: PR}

Pr_Fluid_1[i] = **prandtl**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Prandtl number of fluid 1} {Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_1[i] = **viscosity**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Viscosity of fluid 1}{Units N-s/m²}

{EFF-NTU METHOD: CP}

cp_Fluid_1[i] = **cp**(Water, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Specific heat of fluid 1} {Units J/kg-K}

End

"!REFRIGERANT: Subcooled region"

Duplicate i = 0+1 , 10

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_2[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_2[i] = **density**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Density of fluid 2}{Unit kg/m³}

{EFF-NTU METHOD: PR}

Pr_Fluid_2[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Prandtl number of fluid 2} {Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_2[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Viscosity of fluid 2}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_Fluid_2[i] = cp(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i])$ {Specific heat of fluid 2} {Units J/kg-K}

End

"!REFRIGERANT: All regions"

Duplicate i = 0+1 , 10

$L[i] = L_tot/10$

{EFF-NTU METHOD: CONDUCTIVITY}

$k_stainless_steel[i] = conductivity(Stainless_AISI304, T=((T_Fluid_1_avg[i]+T_Fluid_2_avg[i])/2))$ {Conductivity of the barrier separating the two fluids}{Units J/s-m-K}

{EFF-NTU METHOD: NUD}

$NuD_Fluid_1[i] = mynuds_fluid_1(Re_Fluid_1[i], Pr_Fluid_1[i])$ {Nusselt number of fluid 1} {Units -}

$NuD_Fluid_2[i] = mynuds_fluid_2(ReD_Fluid_2[i], Pr_Fluid_2[i])$ {Nusselt number of fluid 2} {Units -}

{EFF-NTU METHOD: hD}

$hD_Fluid_1[i] = NuD_Fluid_1[i] * k_Fluid_1[i] / D_h_Fluid_1$ {Convection coefficient of fluid 1}{Unit J/s-m^2-K}

$hD_Fluid_2[i] = NuD_Fluid_2[i] * k_Fluid_2[i] / D_h_Fluid_2$ {Convection coefficient of fluid 2}{Unit J/s-m^2-K}

{EFF-NTU METHOD: C}

$C_Fluid_1[i] = m_dot_Fluid_1 * cp_Fluid_1[i]$ {Heat capacity rate of fluid 1} {Units J/s-K}

$C_Fluid_2[i] = m_dot_Fluid_2 * cp_Fluid_2[i]$ {Heat capacity rate of fluid 2} {Units J/s-K}

$C_min[i] = \min(C_Fluid_1[i], C_Fluid_2[i])$ {Minimum heat capacity rate of fluid 1 & fluid 2} {Units J/s-K}

$C_max[i] = \max(C_Fluid_1[i], C_Fluid_2[i])$ {Maximum heat capacity rate of fluid 1 & fluid 2} {Units J/s-K}

$C_r[i] = C_min[i] / C_max[i]$ {Ratio of the greatest fluctuation of heat capacity rate}{Unit -}

{EFF-NTU METHOD: NTU}

$NTU[i] = UA[i] / C_min[i]$ {Number of transfer units} {Units -}

{EFF-NTU METHOD: EFFICIENCY}

$eff[i] = (1 - \exp(-NTU[i] * (1 - C_r[i]))) / (1 - C_r[i] * \exp(-NTU[i] * (1 - C_r[i])))$ {Efficiency of heat transfer}{Unit -}

{EFF-NTU METHOD: SYSTEM RESISTANCE}

$1/UA[i] = 1/(A_ii_Fluid_1[i] * hD_Fluid_1[i]) + Rf_i/A_ii_Fluid_1[i] + \ln(D_io/D_h_Fluid_1)/(2 * \pi * k_stainless_steel[i] * L[i]) + Rf_o/A_io_Fluid_2[i] + 1/(A_io_Fluid_2[i] * hD_Fluid_2[i])$ {Heat exchange heat transfer coefficient}{Units K-s/J}

$A_ii_Fluid_1[i] = \pi * D_h_Fluid_1 * L[i]$ {Inner surface area of inner tube}{Unit m^2}

$A_io_Fluid_2[i] = \pi * D_io * L[i]$ {Outer surface area of inner tube}{Unit m^2}

{EFF-NTU METHOD: ENTHALPY}

$h_out_Fluid_1[i] = enthalpy(Water, T=T_out_Fluid_1[i], P=P_Fluid_1[i])$ {Enthalpy of fluid 1}{Units J/kg}

$h_out_Fluid_2[i] = enthalpy(R407C, T=T_out_Fluid_2[i], P=P_Fluid_2[i])$ {Enthalpy of fluid 2}{Units J/kg}

$h_in_Fluid_1[i] = enthalpy(Water, T=T_in_Fluid_1[i], P=P_Fluid_1[i])$ {Enthalpy of fluid 1}{Units J/kg}

$h_in_Fluid_2[i] = enthalpy(R407C, T=T_in_Fluid_2[i], P=P_Fluid_2[i])$ {Enthalpy of fluid 2}{Units J/kg}

{EFF-NTU METHOD: ENTROPY}

s_Fluid_1[i] = **entropy**(*Water*, T=T_Fluid_1_avg[i], P=P_Fluid_1[i]) {Entropy of fluid 1}{Units J/kg-K}

s_Fluid_2[i] = **entropy**(*R407C*, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Entropy of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: QUALITY}

x_Fluid1[i] = **quality**(*Water*, T=T_Fluid_1_avg[i], s=s_Fluid_1[i]) {Quality of fluid 1 inaccurate in the saturated state}{Units -}

x_Fluid2[i] = **quality**(*R407C*, T=T_Fluid_2_avg[i], s=s_Fluid_2[i]) {Quality of fluid 2 inaccurate in the saturated state}{Units -}

{EFF-NTU METHOD: REYNOLDS NUMBER}

Re_Fluid_1[i] = (rho_Fluid_1[i]*vel_in_Fluid_1[i]*D_h_Fluid_1)/(mu_Fluid_1[i]) {Reynolds number of fluid passing through inner tube}{Unit -}

ReD_Fluid_2[i] = (rho_Fluid_2[i]*vel_in_Fluid_2[i]*D_h_Fluid_2)/(mu_Fluid_2[i]) {Reynolds number of fluid passing through outer tube}{Unit -}

{EFF-NTU METHOD: FRICTION FACTOR}

f_i[i] = **moodychart**(Re_Fluid_1[i], RR_i) {Friction factor of fluid in inner tube}{Units -}

f_o[i] = **moodychart**(ReD_Fluid_2[i], RR_o) {Friction factor of fluid in outer tube}{Units -}

{EFF-NTU METHOD: FLUID PROPERTIES: FLOW VELOCITY}

vel_in_Fluid_1[i] = m_dot_Fluid_1/(rho_Fluid_1[i]*A_Fluid_1_cross) {Velocity of fluid passing through inner tube}{Unit m/s}

vel_in_Fluid_2[i] = m_dot_Fluid_2/(rho_Fluid_2[i]*A_Fluid_2_cross) {Velocity of fluid passing through outer tube}{Unit m/s}

{EFF-NTU METHOD: HEAT TRANSFER}

q_max[i] = C_min[i]*(T_out_Fluid_2[i]-T_in_Fluid_1[i]) { op node warmste - koudste} {Greatest achievable heat transfer}{Unit J/s}

{EFF-NTU METHOD: PRESSURE DROP}

P_Fluid_1[i] = P_Fluid_1[i-1]-Delta_P_prim_i[i] {Pressure relation between points in succession in the inner tube}{Unit Pa}

P_Fluid_2[i-1] = P_Fluid_2[i]-Delta_P_prim_o[i] {Pressure relation between points in succession in the outer tube}{Unit Pa}

Delta_P_prim_i[i] = f_i[i]*L[i]/D_h_Fluid_1 *1/2*rho_Fluid_1[i]*vel_in_Fluid_1[i]^2 {Primary pressure loss in inner tube}{Unit Pa}

Delta_P_prim_o[i] = f_o[i]*L[i]/D_h_Fluid_2 *1/2*rho_Fluid_2[i]*vel_in_Fluid_2[i]^2 {Primary pressure loss in outer tube}{Unit Pa}

{EFF-NTU METHOD: TEMPERATURES}

T_in_Fluid_1[i] = T_out_Fluid_1[i-1] {Temperature relation between points in succession in the inner tube}{Unit C}

T_in_Fluid_2[i] = T_out_Fluid_2[i-1] {Temperature relation between points in succession in the outer tube}{Unit C}

T_Fluid_1_avg[i] = (T_in_Fluid_1[i] + T_out_Fluid_1[i])/2 {Average temperature at any given point along the inner tube}{Unit C}

T_Fluid_2_avg[i] = (T_in_Fluid_2[i] + T_out_Fluid_2[i])/2 {Average temperature at any given point along the outer tube}{Unit C}

End

{HEAT TRANSFER EQUATIONS}

Duplicate i = 1 , 10

q[i] = m_dot_Fluid_1*(h_out_Fluid_1[i]-h_in_Fluid_1[i]) {Heat transfer over inner tube segment}{Unit J/s}

q[i] = m_dot_Fluid_2*(h_out_Fluid_2[i]-h_in_Fluid_2[i]) {Heat transfer over outer tube segment}{Unit J/s}

q[i] = UA[i]*(T_Fluid_2_avg[i]-T_Fluid_1_avg[i]) {Heat transfer between tube segments}{Unit J/s}

End

q_tot = **sum**(q[i], i=1, 10) {Total heat transfer}{Unit J/s}

{PRESSURE DROP OVER TUBE SECTIONS}

Pdrop_Fluid_1 = P_Fluid_1[0]-P_Fluid_1[10] {Pressure drop along inner tube}{Unit Pa}

Pdrop_Fluid_2 = P_Fluid_2[10]-P_Fluid_2[0] {Pressure drop along outer tube}{Unit Pa}

{!*****End*****}

A.2.3.4 EES Code for Condenser Model: Combination of All Three States

\$Load Incompressible

\$Load NASA

{NOTES}

{Fluid 1 = Water}

{Fluid 2 = R407C refrigerant}

{Tube-in-tube refrigerant-to-water counterflow condenser}

{!*****Functions*****}

{!*****Nusselt Number*****}

Function mynuds_fluid_1(a,b)

If (a< 2300) **Then**

NuD_Fluid_1=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_1=0.023*a^(4/5)*b^{0.4} {Value in case of turbulent flow}{Heating coefficient = 0,4}

Endif

mynuds_fluid_1=NuD_Fluid_1

End

Function mynuds_fluid_2(x,y)

If (x< 2300) **Then**

NuD_Fluid_2=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_2=0.023*x^(4/5)*y^{0.3} {Value in case of turbulent flow}{Cooling coefficient = 0,3}

Endif

mynuds_fluid_2=NuD_Fluid_2

End

Function mynuds_tp_fluid_1(a,b)

If (a< 2300) **Then**

NuD_Fluid_1=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_1=0.023*a^(4/5)*b^{0.4} {Value in case of turbulent flow}{Heating coefficient = 0,4}

Endif

mynuds_tp_fluid_1=NuD_Fluid_1

End

Function mynuds_tp_fluid_2(x,y,a,b)

If (x< 2300) **Then**

NuD_Fluid_2=3.66 {Value in case of laminar flow}

Else

NuD_Fluid_2=0.023*x^(4/5)*y^{0.3} *(((1-a)^{0.8}+(3.8*a^{0.76}(1-a)^{0.04})/(b)) {Value in case of turbulent flow}{Cooling coefficient = 0,3}

Endif

mynuds_tp_fluid_2=NuD_Fluid_2

End

{!*****End*****}

{!*****Friction Factor*****}

Procedure frictionfactor_singlephase(Re,D,epsilon:fric) {Calculation of homogeneous friction factor for single phase}

If (Re < 2300) **Then**

fric := 64 / Re

Else

```

If (Re < 5000) Then

    f1 := 64 / 2300
    f2:= 0.25/((log10(epsilon / 3.7 / D + 5.74 / 5000^0.9))^2)    {For turbulent flow}
    fric := (f2-f1)*(Re - 2300)/(5000-2300) + f1    {Interpolating on the Moody chart}

Else

    fric := 0.25/((log10(epsilon / 3.7 / D + 5.74 / Re^0.9))^2)

Endif

Endif

End

Procedure frictionfactor_twophase(p,h,mdot,A,D,epsilon:fric) {Calculation of friction factor for two phase}
    x := quality(R407C,P=p,h=h)
    rho := density(R407C,P=p,h=h)
    G := mdot/A
    If (x > 0) and (x < 1) Then
        rho_L := density(R407C,P=p,x=0)
        rho_G := density(R407C,P=p,x=1)
        mu_L := viscosity(R407C,P=p,x=0)    {Lockhart Martinelli}
        mu_G := viscosity(R407C,P=p,x=1)
        Re_L := G * (1 - x) * D / mu_L
        Re_G := G * x * D / mu_G
        Call frictionfactor_singlephase(Re_L,D,epsilon:f_L)
        Call frictionfactor_singlephase(Re_G,D,epsilon:f_G)

        lamda := rho_G/rho_L * (mu_L/mu_G)^0.2
        t1 := exp(-(log10(lamda)+2.5)^2/(2.4 - G*1.0e-4))

        C := -2 + (28-0.3*sqrt(G))*t1
        If (C < 2) Then
            C := 2
        Endif
        fric := rho*(f_L*(1-x)*(1-x)/rho_L + C*x*(1-x)*sqrt(f_L*f_G/rho_L/rho_G) + f_G*x*x/rho_G)
    Else
        mu := viscosity(R407C,P=p,x=x)
        Re := G * D / mu
        Call frictionfactor_singlephase(Re,D,epsilon:fric) {For single phase calculations call single phase function}
    Endif
End

{!*****End*****}
{!*****Main Code*****}

{FLUID PROPERTIES: INLET TEMPERATURE}
T_out_Fluid_1[0] = 25 {Temperature of fluid 1 entering the inner tube}{Units C}
T_out_Fluid_2[30] = 100 {Temperature of fluid 2 entering the outer tube}{Units C}

{FLUID PROPERTIES: INLET PRESSURE}
P_Fluid_1[0] = 250000 {Pressure of fluid 1 entering the inner tube}{Units Pa}
P_Fluid_2[30] = 2100000 {Pressure of fluid 2 entering the outer tube}{Units Pa}

{FLUID PROPERTIES: BOUNDRY QUALITY}
x_Fluid2[10] = 0.01 {Quality of fluid 2 at the subcooled to two-phase transition node}{Units -}
x_Fluid2[20] = 0.99 {Quality of fluid 2 at the two-phase to superheat transition node}{Units -}

```

{MASS FLOW RATE}

$m_{\text{dot_Fluid_1}} = 0.225$ {Mass flowrate of fluid 1 passing through inner tube}{Units kg/s}

$m_{\text{dot_Fluid_2}} = 0.05$ {Mass flowrate of fluid 2 passing through outer tube}{Units kg/s}

{TUBE GEOMETRY: DIAMETER & WALL THICKNESS}

$D_{\text{ii}} = 0.01576$ {Inner diameter of the inner tube}{Units m}

$WT_{\text{in_Fluid_1}} = 0.00277$ {Wall thickness of the inner tube}{Units m}

$D_{\text{io}} = (D_{\text{ii}} + 2 * WT_{\text{in_Fluid_1}})$ {Outer diameter of the inner tube}{Units m}

$D_{\text{h_Fluid_1}} = D_{\text{ii}}$ {Hydraulic diameter of the inner tube}{Units m}

$D_{\text{oi}} = 0.02664$ {Inner diameter of the outer tube}{Units m}

$WT_{\text{in_Fluid_2}} = 0.00338$ {Wall thickness of the outer tube}{Units m}

$D_{\text{oo}} = D_{\text{oi}} + 2 * WT_{\text{in_Fluid_2}}$ {Outer diameter of the outer tube}{Units m}

$D_{\text{h_Fluid_2}} = D_{\text{oi}} - D_{\text{io}}$ {Hydraulic diameter of the outer tube}{Units m}

{TUBE GEOMETRY: FOULING}

$Epsilon_{\text{i}} = 100 * 10^{-6}$ {Roughness of inner tube}{Units m}

$Epsilon_{\text{o}} = 100 * 10^{-6}$ {Roughness of outer tube}{Units m}

$RR_{\text{i}} = Epsilon_{\text{i}} / D_{\text{h_Fluid_1}}$ {Relative Roughness of inner tube}{Units -}

$RR_{\text{o}} = Epsilon_{\text{o}} / D_{\text{h_Fluid_2}}$ {Relative Roughness of outer tube}{Units -}

$Rf_{\text{i}} = 0$ {Fouling factor for inner tube}{Units $m^2 \cdot K \cdot s / J$ }

$Rf_{\text{o}} = 0$ {Fouling factor for outer tube}{Units $m^2 \cdot K \cdot s / J$ }

{TUBE GEOMETRY: FLOW AREA }

$A_{\text{Fluid_1_cross}} = \pi * (D_{\text{ii}} / 2)^2$ {Cross sectional area of inner tube}{Units m^2 }

$A_{\text{Fluid_2_cross}} = \pi * (D_{\text{oi}} / 2)^2 - \pi * (D_{\text{io}} / 2)^2$ {Cross sectional area of outer tube accounting for the presence of the inner tube}{Units m^2 }

{TUBE GEOMETRY: LENGTH}

$L_{\text{total}} = 12.55$ {Total length of the heat exchanger}{Units m}

$L_{\text{total}} = L_{\text{sc}} + L_{\text{tp}} + L_{\text{sh}}$

"!WATER: All regions"

Duplicate i = 0+1 , 30

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{\text{Fluid_1}[i]} = \text{conductivity}(\text{Water}, T=T_{\text{Fluid_1_avg}[i]}, P=P_{\text{Fluid_1}[i]})$ {Conductivity of fluid 1}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

$\rho_{\text{Fluid_1}[i]} = \text{density}(\text{Water}, T=T_{\text{Fluid_1_avg}[i]}, P=P_{\text{Fluid_1}[i]})$ {Density of fluid 1}{Units kg/m³}

{EFF-NTU METHOD: PR}

$Pr_{\text{Fluid_1}[i]} = \text{prandtl}(\text{Water}, T=T_{\text{Fluid_1_avg}[i]}, P=P_{\text{Fluid_1}[i]})$ {Prandtl number of fluid 1}{Units -}

{EFF-NTU METHOD: VISCOSITY}

$\mu_{\text{Fluid_1}[i]} = \text{viscosity}(\text{Water}, T=T_{\text{Fluid_1_avg}[i]}, P=P_{\text{Fluid_1}[i]})$ {Viscosity of fluid 1}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_{\text{Fluid_1}[i]} = \text{cp}(\text{Water}, T=T_{\text{Fluid_1_avg}[i]}, P=P_{\text{Fluid_1}[i]})$ {Specific heat of fluid 1}{Units J/kg-K}

End

"!REFRIGERANT: Subcooled region"

Duplicate i = 0+1 , 9

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_2[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_2[i] = **density**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Density of fluid 2}{Units kg/m^3}

{EFF-NTU METHOD: PR}

Pr_Fluid_2[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_2[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Viscosity of fluid 2}{Units N-s/m^2}

{EFF-NTU METHOD: CP}

cp_Fluid_2[i] = **cp**(R407C, T=T_Fluid_2_avg[i], P=P_Fluid_2[i]) {Specific heat of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: NUD}

NuD_Fluid_1[i] = **mynuds_fluid_1**(Re_Fluid_1[i], Pr_Fluid_1[i]) {Nusselt number of fluid 1}{Units -}

NuD_Fluid_2[i] = **mynuds_fluid_2**(ReD_Fluid_2[i], Pr_Fluid_2[i]) {Nusselt number of fluid 2}{Units -}

{EFF-NTU METHOD: PRESSURE DROP}

P_Fluid_1[i] = P_Fluid_1[i-1]-Delta_P_prim_i[i] {Pressure relation between points in succession in the inner tube}{Units Pa}

P_Fluid_2[i-1] = P_Fluid_2[i]-Delta_P_prim_o[i] {Pressure relation between points in succession in the outer tube}{Units Pa}

Delta_P_prim_i[i] = f_i[i]*L[i]/D_h_Fluid_1 *1/2*rho_Fluid_1[i]*vel_in_Fluid_1[i]^2 {Primary pressure loss in inner tube}{Units Pa}

Delta_P_prim_o[i] = f_o[i]*L[i]/D_h_Fluid_2 *1/2*rho_Fluid_2[i]*vel_in_Fluid_2[i]^2 {Primary pressure loss in outer tube}{Units Pa}

L[i] = L_sc/9{Increment lengths of subcooled tube section}{Units m}

End

"!REFRIGERANT: Two-phase region"

Duplicate i = 10 , 20

{EFF-NTU METHOD: CONDUCTIVITY}

k_Fluid_2_ff[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], x=0) {Conductivity of fluid 2 when saturated liquid}{Units J/s-m-K}

k_Fluid_2_g[i] = **conductivity**(R407C, T=T_Fluid_2_avg[i], x=1) {Conductivity of fluid 2 when saturated vapour}{Units J/s-m-K}

k_Fluid_2[i] = x_Fluid2[i]*(k_Fluid_2_g[i]-k_Fluid_2_ff[i])+k_Fluid_2_ff[i] {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

rho_Fluid_2_ff[i] = **density**(R407C, T=T_Fluid_2_avg[i], x=0) {Prandtl number of fluid 2 when saturated liquid}{Units -}

rho_Fluid_2_g[i] = **density**(R407C, T=T_Fluid_2_avg[i], x=1) {Prandtl number of fluid 2 when saturated vapour}{Units -}

rho_Fluid_2[i] = **density**(R407C, T=T_Fluid_2_avg[i], x=x_Fluid2[i]) {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: PR}

Pr_Fluid_2_ff[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], x=0) {Prandtl number of fluid 2 when saturated liquid}{Units -}

Pr_Fluid_2_g[i] = **prandtl**(R407C, T=T_Fluid_2_avg[i], x=1) {Prandtl number of fluid 2 when saturated vapour}{Units -}

Pr_Fluid_2[i] = x_Fluid2[i]*(Pr_Fluid_2_g[i]-Pr_Fluid_2_ff[i])+Pr_Fluid_2_ff[i] {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: VISCOSITY}

mu_Fluid_2_ff[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], x=0) {Viscosity of fluid 2 when saturated liquid}{Units N-s/m^2}

mu_Fluid_2_g[i] = **viscosity**(R407C, T=T_Fluid_2_avg[i], x=1) {Viscosity of fluid 2 when saturated vapour}{Units N-s/m^2}

$\mu_{\text{Fluid}_2[i]} = x_{\text{Fluid}_2[i]}(\mu_{\text{Fluid}_2\text{g}[i]} - \mu_{\text{Fluid}_2\text{f}[i]}) + \mu_{\text{Fluid}_2\text{f}[i]}$ {Viscosity of fluid 2}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_{\text{Fluid}_2\text{f}[i]} = cp(R407C, T=T_{\text{Fluid}_2\text{avg}[i]}, x=0)$ {Specific heat of fluid 2 when saturated liquid}{Units J/kg-K}

$cp_{\text{Fluid}_2\text{g}[i]} = cp(R407C, T=T_{\text{Fluid}_2\text{avg}[i]}, x=1)$ {Specific heat of fluid 2 when saturated vapour}{Units J/kg-K}

$cp_{\text{Fluid}_2[i]} = x_{\text{Fluid}_2[i]}(cp_{\text{Fluid}_2\text{g}[i]} - cp_{\text{Fluid}_2\text{f}[i]}) + cp_{\text{Fluid}_2\text{f}[i]}$ {Specific heat of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: NUD}

$NuD_{\text{Fluid}_1[i]} = \text{mynuds_tp_fluid}_1(Re_{\text{Fluid}_1[i]}, Pr_{\text{Fluid}_1[i]})$ {Nusselt number of fluid 1}{Units -}

$NuD_{\text{Fluid}_2[i]} = \text{mynuds_tp_fluid}_2(ReD_{\text{Fluid}_2[i]}, Pr_{\text{Fluid}_2[i]}, x_{\text{Fluid}_2[i]}, \text{average}(P_{\text{Fluid}_2[i]}, P_{\text{Fluid}_2[i-1]}))/p_{\text{crit}}(R407C))$
{Nusselt number of fluid 2}{Units -}

{EFF-NTU METHOD: REYNOLDS NUMBER}

$ReD_{\text{Fluid}_2\text{f}[i]} = (\rho_{\text{Fluid}_2\text{f}[i]} \cdot vel_{\text{in_Fluid}_2[i]} \cdot D_{\text{h_Fluid}_2})/(\mu_{\text{Fluid}_2\text{f}[i]})$ {Reynolds number of a liquid passing through outer tube}{Units -}

$ReD_{\text{Fluid}_2\text{g}[i]} = (\rho_{\text{Fluid}_2\text{g}[i]} \cdot vel_{\text{in_Fluid}_2[i]} \cdot D_{\text{h_Fluid}_2})/(\mu_{\text{Fluid}_2\text{g}[i]})$ {Reynolds number of a gas passing through outer tube}{Units -}

{EFF-NTU METHOD: FRICTION FACTOR}

$f_{\text{g}[i]} = 0.25/(\log_{10}(\text{Epsilon}_o/(3.7 \cdot D_{\text{h_Fluid}_2}) + 5.74/(ReD_{\text{Fluid}_2\text{g}[i]}^{0.9})))^2$ {Swamee-Jain equation to determine the friction factor of a gas in outer tube}{Units -}

$f_{\text{f}[i]} = 0.25/(\log_{10}(\text{Epsilon}_o/(3.7 \cdot D_{\text{h_Fluid}_2}) + 5.74/(ReD_{\text{Fluid}_2\text{f}[i]}^{0.9})))^2$ {Swamee-Jain equation to determine the friction factor of a liquid in outer tube}{Units -}

$C[i] = -2 + (28 - 0.3 \cdot \sqrt{m_{\text{dot_Fluid}_2}/A_{\text{Fluid}_2\text{cross}}/UC[i]}) \cdot \exp(-(\log_{10}(\rho_{\text{Fluid}_2\text{g}[i]}/\rho_{\text{Fluid}_2\text{f}[i]} \cdot (\mu_{\text{Fluid}_2\text{f}[i]}/\mu_{\text{Fluid}_2\text{g}[i]})^{0.2}) + 2.5)^2/(2.4 - m_{\text{dot_Fluid}_2}/A_{\text{Fluid}_2\text{cross}} \cdot 1.0E-04/UC[i]))$ {Two-phase friction factor coefficient in outer tube}{Units -}

$UC[i] = 1$ [kg/s²m²]

$f_{\text{tp}[i]} = \rho_{\text{Fluid}_2[i]} \cdot ((f_{\text{f}[i]} \cdot (1 - x_{\text{Fluid}_2[i]})^2)/\rho_{\text{Fluid}_2\text{f}[i]} + C[i] \cdot x_{\text{Fluid}_2[i]} \cdot (1 - x_{\text{Fluid}_2[i]}) \cdot (f_{\text{f}[i]}/\rho_{\text{Fluid}_2\text{f}[i]} \cdot f_{\text{g}[i]}/\rho_{\text{Fluid}_2\text{g}[i]})^{1/2} + (f_{\text{g}[i]} \cdot (x_{\text{Fluid}_2[i]})^2)/\rho_{\text{Fluid}_2\text{g}[i]})$ {Lockhart-Martinelli equation used to determine the friction factor of two-phase flow in outer tube}{Units -}

{EFF-NTU METHOD: PRESSURE DROP}

$P_{\text{Fluid}_1[i]} = P_{\text{Fluid}_1[i-1]} - \Delta P_{\text{prim_i}[i]}$ {Pressure relation between points in succession in the inner tube}{Units Pa}

$P_{\text{Fluid}_2[i-1]} = P_{\text{Fluid}_2[i]} - \Delta P_{\text{prim_o}[i]}$ {Pressure relation between points in succession in the outer tube}{Units Pa}

$\Delta P_{\text{prim_i}[i]} = f_{\text{f}[i]} \cdot L[i]/D_{\text{h_Fluid}_1} \cdot 1/2 \cdot \rho_{\text{Fluid}_1[i]} \cdot vel_{\text{in_Fluid}_1[i]}^2$ {Primary pressure loss in inner tube}{Units Pa}

$\Delta P_{\text{prim_o}[i]} = f_{\text{tp}[i]} \cdot L[i]/D_{\text{h_Fluid}_2} \cdot 1/2 \cdot \rho_{\text{Fluid}_2[i]} \cdot vel_{\text{in_Fluid}_2[i]}^2$ {Primary pressure loss in outer tube}{Units Pa}

$L[i] = L_{\text{tp}}/10$ {Increment lengths of two-phase tube section}{Units m}

End

"!REFRIGERANT: Superheated region"

Duplicate i = 21 , 30

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{\text{Fluid}_2[i]} = \text{conductivity}(R407C, T=T_{\text{Fluid}_2\text{avg}[i]}, P=P_{\text{Fluid}_2[i]})$ {Conductivity of fluid 2}{Units J/s-m-K}

{EFF-NTU METHOD: FLUID DENSITY}

$\rho_{\text{Fluid}_2[i]} = \text{density}(R407C, T=T_{\text{Fluid}_2\text{avg}[i]}, P=P_{\text{Fluid}_2[i]})$ {Density of fluid 2}{Units kg/m³}

{EFF-NTU METHOD: PR}

$Pr_{\text{Fluid}_2[i]} = \text{prandtl}(R407C, T=T_{\text{Fluid}_2\text{avg}[i]}, P=P_{\text{Fluid}_2[i]})$ {Prandtl number of fluid 2}{Units -}

{EFF-NTU METHOD: VISCOSITY}

$\mu_{\text{Fluid}_2[i]} = \text{viscosity}(R407C, T=T_{\text{Fluid}_2_avg[i]}, P=P_{\text{Fluid}_2[i]})$ {Viscosity of fluid 2}{Units N-s/m²}

{EFF-NTU METHOD: CP}

$cp_{\text{Fluid}_2[i]} = cp(R407C, T=T_{\text{Fluid}_2_avg[i]}, P=P_{\text{Fluid}_2[i]})$ {Specific heat of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: NUD}

$NuD_{\text{Fluid}_1[i]} = \text{mynuds_fluid}_1(Re_{\text{Fluid}_1[i]}, Pr_{\text{Fluid}_1[i]})$ {Nusselt number of fluid 1}{Units -}

$NuD_{\text{Fluid}_2[i]} = \text{mynuds_fluid}_2(Re_{\text{Fluid}_2[i]}, Pr_{\text{Fluid}_2[i]})$ {Nusselt number of fluid 2}{Units -}

{EFF-NTU METHOD: PRESSURE DROP}

$P_{\text{Fluid}_1[i]} = P_{\text{Fluid}_1[i-1]} - \Delta P_{\text{prim}_i[i]}$ {Pressure relation between points in succession in the inner tube}{Units Pa}

$P_{\text{Fluid}_2[i-1]} = P_{\text{Fluid}_2[i]} - \Delta P_{\text{prim}_o[i]}$ {Pressure relation between points in succession in the outer tube}{Units Pa}

$\Delta P_{\text{prim}_i[i]} = f_{i[i]} * L[i] / D_{h_Fluid_1} * 1/2 * \rho_{\text{Fluid}_1[i]} * vel_{in_Fluid_1[i]}^2$ {Primary pressure loss in inner tube}{Units Pa}

$\Delta P_{\text{prim}_o[i]} = f_{o[i]} * L[i] / D_{h_Fluid_2} * 1/2 * \rho_{\text{Fluid}_2[i]} * vel_{in_Fluid_2[i]}^2$ {Primary pressure loss in outer tube}{Units Pa}

$L[i] = L_{sh}/10$ {Increment lengths of superheated tube section}{Units m}

End

"!REFRIGERANT: All regions"

Duplicate i = 0+1 , 30

{EFF-NTU METHOD: CONDUCTIVITY}

$k_{\text{stainless_steel}[i]} = \text{conductivity}(\text{Stainless_AISI304}, T=((T_{\text{Fluid}_1_avg[i]} + T_{\text{Fluid}_2_avg[i]})/2))$ {Conductivity of the barrier separating the two fluids}{Units J/s-m-K}

{EFF-NTU METHOD: hD}

$hD_{\text{Fluid}_1[i]} = NuD_{\text{Fluid}_1[i]} * k_{\text{Fluid}_1[i]} / D_{h_Fluid_1}$ {Convection coefficient of fluid 1}{Units J/s-m²-K}

$hD_{\text{Fluid}_2[i]} = NuD_{\text{Fluid}_2[i]} * k_{\text{Fluid}_2[i]} / D_{h_Fluid_2}$ {Convection coefficient of fluid 2}{Units J/s-m²-K}

{EFF-NTU METHOD: C}

$C_{\text{Fluid}_1[i]} = m_{\text{dot_Fluid}_1} * cp_{\text{Fluid}_1[i]}$ {Heat capacity rate of fluid 1}{Units J/s-K}

$C_{\text{Fluid}_2[i]} = m_{\text{dot_Fluid}_2} * cp_{\text{Fluid}_2[i]}$ {Heat capacity rate of fluid 2}{Units J/s-K}

$C_{\text{min}[i]} = \min(C_{\text{Fluid}_1[i]}, C_{\text{Fluid}_2[i]})$ {Minimum heat capacity rate of fluid 1 & fluid 2}{Units J/s-K}

$C_{\text{max}[i]} = \max(C_{\text{Fluid}_1[i]}, C_{\text{Fluid}_2[i]})$ {Maximum heat capacity rate of fluid 1 & fluid 2}{Units J/s-K}

$C_r[i] = C_{\text{min}[i]} / C_{\text{max}[i]}$ {Ratio of the greatest fluctuation of heat capacity rate}{Units -}

{EFF-NTU METHOD: NTU}

$NTU[i] = UA[i] / C_{\text{min}[i]}$ {Number of transfer Units}{Units -}

{EFF-NTU METHOD: EFFICIENCY}

$eff[i] = (1 - \exp(-NTU[i] * (1 - C_r[i]))) / (1 - C_r[i] * \exp(-NTU[i] * (1 - C_r[i])))$ {Efficiency of heat transfer}{Units -}

{EFF-NTU METHOD: SYSTEM RESISTANCE}

$1/UA[i]$

=

$1/(A_{i_Fluid_1[i]} * hD_{\text{Fluid}_1[i]} + R_{f_i/A_i_Fluid_1[i]} + \ln(D_{io}/D_{h_Fluid_1}) / (2 * \pi * k_{\text{stainless_steel}[i]} * L[i]) + R_{f_o/A_io_Fluid_2[i]} + 1/(A_{io_Fluid_2[i]} * hD_{\text{Fluid}_2[i]}))$ {Heat exchange heat transfer coefficient}{Units K-s/J}

$A_{i_Fluid_1[i]} = \pi * D_{h_Fluid_1} * L[i]$ {Inner surface area of inner tube}{Units m²}

$A_{io_Fluid_2[i]} = \pi * D_{io} * L[i]$ {Outer surface area of inner tube}{Units m²}

{EFF-NTU METHOD: ENTHALPY}

$h_{out_Fluid_1[i]} = \text{enthalpy}(\text{Water}, T=T_{out_Fluid_1[i]}, P=P_{Fluid_1[i]})$ {Enthalpy of fluid 1}{Units J/kg}
 $h_{out_Fluid_2[i]} = \text{enthalpy}(R407C, T=T_{out_Fluid_2[i]}, P=P_{Fluid_2[i]})$ {Enthalpy of fluid 2}{Units J/kg}
 $h_{in_Fluid_1[i]} = \text{enthalpy}(\text{Water}, T=T_{in_Fluid_1[i]}, P=P_{Fluid_1[i]})$ {Enthalpy of fluid 1}{Units J/kg}
 $h_{in_Fluid_2[i]} = \text{enthalpy}(R407C, T=T_{in_Fluid_2[i]}, P=P_{Fluid_2[i]})$ {Enthalpy of fluid 2}{Units J/kg}

{EFF-NTU METHOD: ENTROPY}

$s_{Fluid_1[i]} = \text{entropy}(\text{Water}, T=T_{Fluid_1_avg[i]}, P=P_{Fluid_1[i]})$ {Entropy of fluid 1}{Units J/kg-K}
 $s_{Fluid_2[i]} = \text{entropy}(R407C, T=T_{Fluid_2_avg[i]}, P=P_{Fluid_2[i]})$ {Entropy of fluid 2}{Units J/kg-K}

{EFF-NTU METHOD: QUALITY}

$x_{Fluid1[i]} = \text{quality}(\text{Water}, T=T_{Fluid_1_avg[i]}, s=s_{Fluid_1[i]})$ {Quality of fluid 1 inaccurate in the saturated state}{Units -}
 $x_{Fluid2[i]} = \text{quality}(R407C, T=T_{Fluid_2_avg[i]}, s=s_{Fluid_2[i]})$ {Quality of fluid 2 inaccurate in the saturated state}{Units -}

{EFF-NTU METHOD: REYNOLDS NUMBER}

$Re_{Fluid_1[i]} = (\rho_{Fluid_1[i]} * v_{in_Fluid_1[i]} * D_{h_Fluid_1}) / (\mu_{Fluid_1[i]})$ {Reynolds number of fluid passing through inner tube}{Units -}
 $ReD_{Fluid_2[i]} = (\rho_{Fluid_2[i]} * v_{in_Fluid_2[i]} * D_{h_Fluid_2}) / (\mu_{Fluid_2[i]})$ {Reynolds number of fluid passing through outer tube}{Units -}

{EFF-NTU METHOD: FRICTION FACTOR}

$f_{i[i]} = \text{moodychart}(Re_{Fluid_1[i]}, RR_i)$ {Friction factor of fluid in inner tube}{Units -}
 $f_{o[i]} = \text{moodychart}(ReD_{Fluid_2[i]}, RR_o)$ {Friction factor of fluid in outer tube}{Units -}

{EFF-NTU METHOD: FLUID PROPERTIES: FLOW VELOCITY}

$v_{in_Fluid_1[i]} = m_dot_Fluid_1 / (\rho_{Fluid_1[i]} * A_{Fluid_1_cross})$ {Velocity of fluid passing through inner tube}{Units m/s}
 $v_{in_Fluid_2[i]} = m_dot_Fluid_2 / (\rho_{Fluid_2[i]} * A_{Fluid_2_cross})$ {Velocity of fluid passing through outer tube}{Units m/s}

{EFF-NTU METHOD: HEAT TRANSFER}

$q_{max[i]} = C_{min[i]} * (T_{out_Fluid_2[i]} - T_{in_Fluid_1[i]})$ {Greatest achievable heat transfer}{Units J/s}

{EFF-NTU METHOD: TEMPERATURES}

$T_{in_Fluid_1[i]} = T_{out_Fluid_1[i-1]}$ {Temperature relation between points in succession in the inner tube}{Units C}
 $T_{in_Fluid_2[i]} = T_{out_Fluid_2[i-1]}$ {Temperature relation between points in succession in the outer tube}{Units C}

$T_{Fluid_1_avg[i]} = (T_{in_Fluid_1[i]} + T_{out_Fluid_1[i]}) / 2$ {Average temperature at any given point along the inner tube}{Units C}
 $T_{Fluid_2_avg[i]} = (T_{in_Fluid_2[i]} + T_{out_Fluid_2[i]}) / 2$ {Average temperature at any given point along the outer tube}{Units C}

End

{HEAT TRANSFER EQUATIONS}

Duplicate i = 1 , 30

$q[i] = m_dot_Fluid_1 * (h_{out_Fluid_1[i]} - h_{in_Fluid_1[i]})$ {Heat transfer over inner tube segment}{Units J/s}
 $q[i] = m_dot_Fluid_2 * (h_{out_Fluid_2[i]} - h_{in_Fluid_2[i]})$ {Heat transfer over outer tube segment}{Units J/s}
 $q[i] = UA[i] * (T_{Fluid_2_avg[i]} - T_{Fluid_1_avg[i]})$ {Heat transfer between tube segments}{Units J/s}

End

$q_{tot} = \text{sum}(q[i], i=1, 30)$ {Total heat transfer}{Units J/s}

{PRESSURE DROP OVER TUBE SECTIONS}

$P_{drop_Fluid_1} = P_{Fluid_1[1]} - P_{Fluid_1[30]}$ {Pressure drop along inner tube}{Units Pa}
 $P_{drop_Fluid_2} = P_{Fluid_2[30]} - P_{Fluid_2[1]}$ {Pressure drop along outer tube}{Units Pa}

{*****End*****}

A.2.4 Condenser Verification Results

The accuracy of the developed condenser model was verified using the developed verification models from the previous section. The applied input conditions, reference and test results for each data point of the surface and underground verification are presented on the following page as Figure 64.

To continue reading from where you left off in the main dissertation, [click here](#).

[illegible]

Figure 64: Condenser verification results

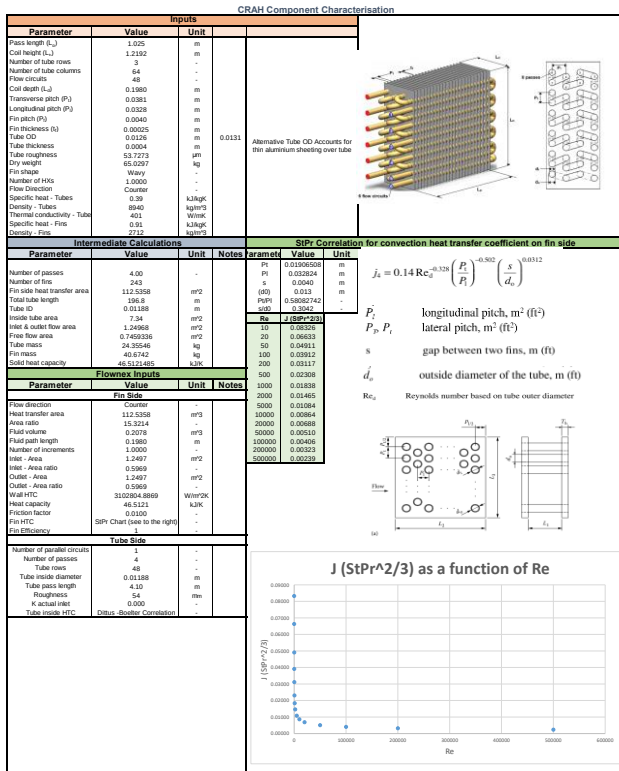
A.3 Evaporator Development

This section contains information regarding the development of the ACU evaporator model.

A.3.1 Evaporator Geometry Input Calculations

The processing of the evaporator geometry, applied to Flownex is presented on the following two pages as Figure 65.

To continue reading from where you left off in the main dissertation, [click here](#).



Multiple evaporator component layout is accordance with number of evaporators used											
14 evaporator				32 evaporator				64 evaporator			
3	-	rows deep	3	rows deep	3	rows deep	3	rows deep	3	rows deep	3
237.0732	m/s	face velocity	237.0732	m/s	face velocity	237.0732	m/s	face velocity	237.0732	m/s	face velocity
0.00025	m	fin thickness	0.00025	m	fin thickness	0.00025	m	fin thickness	0.00025	m	fin thickness
0.1980	m	fin length in direction of airflow	0.1980	m	fin length in direction of airflow	0.1980	m	fin length in direction of airflow	0.1980	m	fin length in direction of airflow
0.0328	m	Distance between two consecutive columns of the evaporator matrix.	0.0328	m	Distance between two consecutive columns of the evaporator matrix.	0.0328	m	Distance between two consecutive columns of the evaporator matrix.	0.0328	m	Distance between two consecutive columns of the evaporator matrix.
0.0381	m	Distance between two consecutive rows of the evaporator matrix.	0.0381	m	Distance between two consecutive rows of the evaporator matrix.	0.0381	m	Distance between two consecutive rows of the evaporator matrix.	0.0381	m	Distance between two consecutive rows of the evaporator matrix.
1.0250	m	face width	1.0250	m	face width	1.0250	m	face width	1.0250	m	face width
0.0096	m	face height	0.0096	m	face height	0.0096	m	face height	0.0096	m	face height
0.0131	m	tube outer diameter covered with aluminum	0.0131	m	tube outer diameter covered with aluminum	0.0131	m	tube outer diameter covered with aluminum	0.0131	m	tube outer diameter covered with aluminum
0.0119	m	tube inner diameter	0.0119	m	tube inner diameter	0.0119	m	tube inner diameter	0.0119	m	tube inner diameter
24	-	number of flow circuits	48	-	number of flow circuits	96	-	number of flow circuits	192	-	number of flow circuits
a) Total external surface area				a) Total external surface area				a) Total external surface area			
243	-	number of fins	243	-	number of fins	243	-	number of fins	243	-	number of fins
0.0417	m ²	Gross surface area of one fin	0.0417	m ²	Gross surface area of one fin	0.0417	m ²	Gross surface area of one fin	0.0417	m ²	Gross surface area of one fin
32	-	number of tubes per row	64	-	number of tubes per row	128	-	number of tubes per row	256	-	number of tubes per row
96.3840	-	Total number of tubes passing through fin plates	192.7680	-	Total number of tubes passing through fin plates	385.5360	-	Total number of tubes passing through fin plates	771.0720	-	Total number of tubes passing through fin plates
1.38E-04	m ²	External cross-sectional area of one tube	1.38E-04	m ²	External cross-sectional area of one tube	1.38E-04	m ²	External cross-sectional area of one tube	1.38E-04	m ²	External cross-sectional area of one tube
0.0417	m ²	Net surface area of one fin	0.0417	m ²	Net surface area of one fin	0.0417	m ²	Net surface area of one fin	0.0417	m ²	Net surface area of one fin
10.0442	-	Net surface area of all fins	10.0442	-	Net surface area of all fins	10.0442	-	Net surface area of all fins	10.0442	-	Net surface area of all fins
0.0412	m	External perimeter of one tube	0.0412	m	External perimeter of one tube	0.0412	m	External perimeter of one tube	0.0412	m	External perimeter of one tube
4.0486	-	Cross external surface area of all tubes	4.0486	-	Cross external surface area of all tubes	4.0486	-	Cross external surface area of all tubes	4.0486	-	Cross external surface area of all tubes
1.03E-05	m ²	Area of one tube obscured by one fin edge	1.03E-05	m ²	Area of one tube obscured by one fin edge	1.03E-05	m ²	Area of one tube obscured by one fin edge	1.03E-05	m ²	Area of one tube obscured by one fin edge
0.2400	-	Area of all tubes obscured by all fin edges	0.2400	-	Area of all tubes obscured by all fin edges	0.2400	-	Area of all tubes obscured by all fin edges	0.2400	-	Area of all tubes obscured by all fin edges
3.8096	-	Net area of all tubes	3.8096	-	Net area of all tubes	3.8096	-	Net area of all tubes	3.8096	-	Net area of all tubes
66.3559	m ²	Total external surface area	66.3559	m ²	Total external surface area	66.3559	m ²	Total external surface area	66.3559	m ²	Total external surface area
b) Total internal surface area				b) Total internal surface area				b) Total internal surface area			
0.0273	m	Internal perimeter of one tube	0.0273	m	Internal perimeter of one tube	0.0273	m	Internal perimeter of one tube	0.0273	m	Internal perimeter of one tube
3.8728	m ²	Total internal surface area of all tubes	3.8728	m ²	Total internal surface area of all tubes	3.8728	m ²	Total internal surface area of all tubes	3.8728	m ²	Total internal surface area of all tubes
c) Ratio of external to internal surface area				c) Ratio of external to internal surface area				c) Ratio of external to internal surface area			
15.3182	-	Ratio of external to internal surface area	15.3182	-	Ratio of external to internal surface area	15.3182	-	Ratio of external to internal surface area	15.3182	-	Ratio of external to internal surface area
d) Calculating weight of evaporator				d) Calculating weight of evaporator				d) Calculating weight of evaporator			
1.38E-05	m ²	Face area of a single Cu tube	1.38E-05	m ²	Face area of a single Cu tube	1.38E-05	m ²	Face area of a single Cu tube	1.38E-05	m ²	Face area of a single Cu tube
1.42E-05	-	Net volume of a single Cu tube	1.42E-05	-	Net volume of a single Cu tube	1.42E-05	-	Net volume of a single Cu tube	1.42E-05	-	Net volume of a single Cu tube
1.38E-03	-	Net volume of all Cu tubes	1.38E-03	-	Net volume of all Cu tubes	1.38E-03	-	Net volume of all Cu tubes	1.38E-03	-	Net volume of all Cu tubes
1.21E-01	m ²	Area of all Al fin without holes for tubes	1.21E-01	m ²	Area of all Al fin without holes for tubes	1.21E-01	m ²	Area of all Al fin without holes for tubes	1.21E-01	m ²	Area of all Al fin without holes for tubes
1.21E-02	-	Area of all the holes passing through a Al fin	1.21E-02	-	Area of all the holes passing through a Al fin	1.21E-02	-	Area of all the holes passing through a Al fin	1.21E-02	-	Area of all the holes passing through a Al fin
1.06E-01	-	Net face area of a single Al fin (a fin with holes)	1.06E-01	-	Net face area of a single Al fin (a fin with holes)	1.06E-01	-	Net face area of a single Al fin (a fin with holes)	1.06E-01	-	Net face area of a single Al fin (a fin with holes)
2.68E-05	-	Net volume of a single Al fin (a fin with holes)	2.68E-05	-	Net volume of a single Al fin (a fin with holes)	2.68E-05	-	Net volume of a single Al fin (a fin with holes)	2.68E-05	-	Net volume of a single Al fin (a fin with holes)
6.55E-03	-	Net volume of all Al fins (fins with holes)	6.55E-03	-	Net volume of all Al fins (fins with holes)	6.55E-03	-	Net volume of all Al fins (fins with holes)	6.55E-03	-	Net volume of all Al fins (fins with holes)
9.52E-04	-	Net volume of Al covering all copper tubes(excluding the fins)	9.52E-04	-	Net volume of Al covering all copper tubes(excluding the fins)	9.52E-04	-	Net volume of Al covering all copper tubes(excluding the fins)	9.52E-04	-	Net volume of Al covering all copper tubes(excluding the fins)
0.3900	-	Cu Cu	0.3900	-	Cu Cu	0.3900	-	Cu Cu	0.3900	-	Cu Cu
0.0000	-	Cu Al	0.0000	-	Cu Al	0.0000	-	Cu Al	0.0000	-	Cu Al
8940.0000	kg/m ³	Density of Cu	8940.0000	kg/m ³	Density of Cu	8940.0000	kg/m ³	Density of Cu	8940.0000	kg/m ³	Density of Cu
2712.0000	kg/m ³	Density of Al	2712.0000	kg/m ³	Density of Al	2712.0000	kg/m ³	Density of Al	2712.0000	kg/m ³	Density of Al
12.7777	kg	Weight Cu	12.7777	kg	Weight Cu	12.7777	kg	Weight Cu	12.7777	kg	Weight Cu
20.3371	kg	Weight Al	20.3371	kg	Weight Al	20.3371	kg	Weight Al	20.3371	kg	Weight Al
23.2561	kg	Solid heat capacity	23.2561	kg	Solid heat capacity	23.2561	kg	Solid heat capacity	23.2561	kg	Solid heat capacity
32.5148	kg	Total weight	32.5148	kg	Total weight	32.5148	kg	Total weight	32.5148	kg	Total weight
401.0000	W/mK	Thermal conductivity of Cu	401.0000	W/mK	Thermal conductivity of Cu	401.0000	W/mK	Thermal conductivity of Cu	401.0000	W/mK	Thermal conductivity of Cu
0.0004	-	Copper tube thickness	0.0004	-	Copper tube thickness	0.0004	-	Copper tube thickness	0.0004	-	Copper tube thickness
111388.8689	W/m ² K	Wall heat transfer coefficient of copper tube	111388.8689	W/m ² K	Wall heat transfer coefficient of copper tube	111388.8689	W/m ² K	Wall heat transfer coefficient of copper tube	111388.8689	W/m ² K	Wall heat transfer coefficient of copper tube
237.0000	W/mK	Thermal conductivity of Al	237.0000	W/mK	Thermal conductivity of Al	237.0000	W/mK	Thermal conductivity of Al	237.0000	W/mK	Thermal conductivity of Al
148000.0000	W/m ² K	Wall heat transfer coefficient of Al sheet covering copper tube	148000.0000	W/m ² K	Wall heat transfer coefficient of Al sheet covering copper tube	148000.0000	W/m ² K	Wall heat transfer coefficient of Al sheet covering copper tube	148000.0000	W/m ² K	Wall heat transfer coefficient of Al sheet covering copper tube
310288.8	-	Overall wall heat transfer coefficient of the composite	310288.8	-	Overall wall heat transfer coefficient of the composite	310288.8	-	Overall wall heat transfer coefficient of the composite	310288.8	-	Overall wall heat transfer coefficient of the composite
0.0009	-	i1	0.0009	-	i1	0.0009	-	i1	0.0009	-	i1
0.0063	-	i2	0.0063	-	i2	0.0063	-	i2	0.0063	-	i2
0.0066	-	i3	0.0066	-	i3	0.0066	-	i3	0.0066	-	i3
401.0000	W/mK	W1	401.0000	W/mK	W1	401.0000	W/mK	W1	401.0000	W/mK	W1
237.0000	W/mK	W2	237.0000	W/mK	W2	237.0000	W/mK	W2	237.0000	W/mK	W2
0.9643	m	L	0.9643	m	L	0.9643	m	L	0.9643	m	L
Fin (primary) side geometry				Fin (primary) side geometry				Fin (primary) side geometry			
Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559
Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182
Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078
Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980
Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237
Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001
Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001
Material Data				Material Data				Material Data			
Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248
Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938
total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248
total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836
Material Data				Material Data				Material Data			
Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8
Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182
Losses				Losses				Losses			
0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions
0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions
Heat Transfer				Heat Transfer				Heat Transfer			
SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-
Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1
Fin (tube) side geometry				Fin (tube) side geometry				Fin (tube) side geometry			
Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559
Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182
Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078
Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980
Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237
Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001
Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001	Volume of a single tube subtracting fins going through it	0.0001
Material Data				Material Data				Material Data			
Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248	Area	0.6248
Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938	Area ratio	0.2938
total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248	total frontal area	0.6248
total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836	total free flow area/cross-sectional area perpendicular to flow direction	0.1836
Material Data				Material Data				Material Data			
Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8	Wall heat transfer coefficient. Ek het die Al sheet wat die koper roer cover ingesluit	310288.8
Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182	Heat capacity	15.3182
Losses				Losses				Losses			
0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions	0.0098	-	Friciton factor for surface conditions
0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions	0.0128	-	Friciton factor for underground conditions
Heat Transfer				Heat Transfer				Heat Transfer			
SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-	SPH chart. Self generate.	-
Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1	Fin efficiency	1
Fin (tube) side geometry				Fin (tube) side geometry				Fin (tube) side geometry			
Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559	Heat transfer area	66.3559
Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182	Area ratio	15.3182
Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078	Fluid volume	0.2078
Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980	Fluid path length	0.1980
Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237	Volume without tubes or fins	0.1237
Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	0.0001	Volume of a single fin	

A.3.2 Evaporator Verification Results

The accuracy of the developed evaporator model was verified for surface and underground conditions. The applied input conditions, reference and test results for each data point of the surface and underground verification are presented on the following page as Figure 66.

To continue reading from where you left off in the main dissertation, [click here](#).

	Case	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)	Air Flow (m³/s)	Air Temp (°C)	Air Press (bar)	RH (%)
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Figure 66: Evaporator verification results

A.4 Fan Development

This section contains information regarding the development of the ACU fan model.

A.4.1 Fan Charts

The fan charts incorporated into Flownex for surface and underground conditions are presented on the following page as Figure 67.

To continue reading from where you left off in the main dissertation, [click here](#).

Surface						
Fan Size [kW]	9.2					
Air Density [kg/m^3]	1.184					
Fan Area [m^2]	0.442					
Fan Speed [rpm]	1440					
Air Volume Flow [m^3/s]	Static Pressure Drop [Pa]	Velocity [m/s]	Total Pressure Drop [Pa]	Useful Work [kW]	Shaft Power [kW]	Isentropic Efficiency
11.08	0	4.89	372	4.122	8.483	48.59%
10.31	134	4.55	456	4.701	8.756	53.69%
9.53	252	4.21	527	5.022	9.053	55.48%
8.75	346	3.87	578	5.058	9.263	54.60%
7.98	420	3.53	613	4.892	9.28	52.71%
7.2	408	3.18	565	4.068	9.485	42.89%
6.43	424	2.84	549	3.530	9.526	37.06%
5.65	449	2.50	545	3.079	9.04	34.06%
4.88	525	2.16	597	2.913	8.996	32.39%
3.99	650	1.76	698	2.785	9.368	29.73%
2.88	819	1.27	845	2.434	10.338	23.54%
0.11	1419	0.05	1419	0.156	16.375	0.95%

Underground						
Fan Size [kW]	22					
Air Density [kg/m^3]	1.2					
Fan Area [m^2]	0.456					
Fan Speed [rpm]	2980					
Air Volume Flow [m^3/s]	Static Pressure Drop [Pa]	Velocity [m/s]	Total Pressure Drop [Pa]	Useful Work [kW]	Shaft Power [kW]	Isentropic Efficiency
1.00	2350	0.46	2350	2.350	22.500	10.44%
2.00	2200	0.91	2200	4.401	20.500	21.47%
3.00	2050	1.37	2051	6.153	18.625	33.04%
4.00	1850	1.82	1852	7.408	17.000	43.58%
5.00	1680	2.28	1683	8.416	15.900	52.93%
5.50	1670	2.51	1674	9.206	16.250	56.65%
6.00	1675	2.74	1679	10.077	17.000	59.28%
6.50	1725	2.96	1730	11.247	18.000	62.48%
7.00	1800	3.19	1806	12.643	18.625	67.88%
7.25	1825	3.31	1832	13.279	18.875	70.35%
7.50	1800	3.42	1807	13.553	19.000	71.33%
8.00	1690	3.65	1698	13.584	19.100	71.12%
8.50	1500	3.88	1509	12.827	18.800	68.23%
9.00	1350	4.10	1360	12.241	18.500	66.17%
9.50	1100	4.33	1111	10.557	17.750	59.48%
10.00	900	4.56	912	9.125	17.125	53.28%
10.50	650	4.79	664	6.969	16.250	42.89%
11.00	400	5.02	415	4.566	15.000	30.44%

Figure 67: Fan charts

A.4.2 Fan Verification Results

The accuracy of the developed fan model was verified for surface and underground conditions. The applied input conditions, reference and test results for each data point of the surface and underground verification are presented on the following page as Figure 68.

To continue reading from where you left off in the main dissertation, [click here](#).

Conditions	*I-Volume Flow [m³/s]	*T-Efficiency [%]	*T-Pressure Drop [bar]	*R-Efficiency [%]	*R-Pressure Drop [bar]	Error Efficiency	Error Pressure Drop
Surface Fan							
1	11.08	48.48	0.003702	48.59	0.003720	0.23%	0.49%
2	10.31	53.55	0.004536	53.69	0.004560	0.27%	0.52%
3	9.53	55.42	0.005248	55.48	0.005270	0.10%	0.41%
4	8.75	54.63	0.005764	54.60	0.005780	0.05%	0.27%
5	7.98	52.77	0.006120	52.71	0.006130	0.10%	0.16%
6	7.20	43.06	0.005658	42.89	0.005650	0.40%	0.15%
7	6.43	37.12	0.005492	37.06	0.005490	0.17%	0.03%
8	5.65	34.08	0.005450	34.06	0.005450	0.06%	0.00%
9	4.88	32.39	0.005968	32.39	0.005970	0.02%	0.04%
10	3.99	29.73	0.006979	29.73	0.006980	0.01%	0.02%
11	2.88	23.50	0.008460	23.54	0.008450	0.17%	0.12%
Average Error						0.14%	0.20%
Maximum Error						0.40%	0.52%
Underground Fan							
1	1.04	10.46	0.023499	10.44	0.023501	0.15%	0.01%
2	2.00	21.32	0.022026	21.47	0.022005	0.71%	0.09%
3	3.00	33.13	0.020495	33.04	0.020511	0.27%	0.08%
4	4.00	43.81	0.018478	43.58	0.018520	0.53%	0.23%
5	5.00	53.21	0.016824	52.93	0.016831	0.53%	0.04%
6	5.50	56.88	0.016743	56.65	0.016738	0.41%	0.03%
7	6.00	59.60	0.016847	59.28	0.016795	0.55%	0.31%
8	6.50	63.13	0.017393	62.48	0.017303	1.03%	0.52%
9	7.00	68.59	0.018134	67.88	0.018061	1.04%	0.40%
10	7.25	70.65	0.018241	70.35	0.018316	0.42%	0.41%
11	7.50	71.30	0.017901	71.33	0.018070	0.05%	0.94%
12	8.00	70.68	0.016689	71.12	0.016980	0.63%	1.71%
13	8.50	67.93	0.014879	68.23	0.015090	0.43%	1.40%
14	9.00	65.30	0.013277	66.17	0.013601	1.32%	2.38%
15	9.50	58.84	0.010907	59.48	0.011113	1.08%	1.85%
16	10.00	52.46	0.008928	53.28	0.009125	1.54%	2.15%
17	10.50	42.35	0.006529	42.89	0.006638	1.26%	1.63%
18	11.00	30.37	0.004137	30.44	0.004151	0.23%	0.34%
Average Error						0.68%	0.81%
Maximum Error						1.54%	2.38%

Figure 68: Fan verification results

APPENDIX B: INTEGRATED ACU MODEL VALIDATION

This appendix contains information on the validation of the accuracy of the integrated ACU model.

B.1 Measurement Instrumentation

Table 23 below indicates the relevant measuring instruments as well as their exact use.

Table 23: Measurement Instrumentation Relevant to Validation

Instrument	Use
Fluke 375	Voltage and current measurement on the ACU during FAT tests.
Kestrel5000	Air temperature measurement on the ACU during FAT tests.
Burkert flow meter	Water flowrate measurement on the ACU during FAT tests.
PT 100	Water temperature measurement on the ACU during FAT tests.
62 Mini IR Thermometer	Hand temperature measurements of refrigeration cycle during FAT tests.
LTR pressure gauge EN 837-1	Compressor suction and discharge gauge measurement on the ACU during FAT tests.
Logger	On-board data logger that logs water flowrate, inlet and outlet temperatures on the air and waterside, total ACU current and voltage during underground operation.

To continue reading from where you left off in the main dissertation, [click here](#).

B.2 Validation Results

The accuracy of the integrated ACU model was validated using benchmark data. The applied input conditions, benchmark and test results for each data point of the surface and underground validation are presented on the following page as Figure 69.

To continue reading in the surface validation section, [click here](#).

To continue reading in the underground validation section, [click here](#).

To continue reading in the transient investigation section, [click here](#).

[illegible]

Figure 69: ACU validation results

APPENDIX C: TRANSIENT SIMULATION

This appendix contains information on the investigation into the transient behaviour of the ACU.

C.1 Quick Scripts

Quick Script elements were used to set the operating logic of the compressor and expansion valves during transient simulation. To continue reading from where you left off in the main dissertation, [click here](#).

C.1.1 Expansion Valve Quick Script

This section elaborates on the operating logic of the expansion valve quick script.

C.1.1.1 Inputs and Outputs

Script Inputs And Outputs			
T_superheat	7.1	°C	▼
T_setpointSH	7.5	°C	▼
opening	0.397457		
SHerror	0.4	°C	▼
Gain	0.0001		

Figure 70: Expansion valve quick script Input and output example

The code receives superheat (T_superheat) from the cycle during simulation. This value is compared against a setpoint (T_setpointSH). The difference is calculated as the error value (SHerror). Based on the error, the valve opens or closes (opening) to minimize the error. The Gain value is multiplied by the size of the error and that value is then added or subtracted from valve opening. Superheat was the only input with influence on expansion valve opening. The expansion valve opening (opening) is limited between 0.01 and 1.

C.1.1.2 Code

```
string state = Project.SimulationStateText;
if (state == "INACTIVE" || state == "SOLVINGSTEADYSTATE")
{
    double error = T_superheat - T_setpointSH;
    opening.Value = 0.259224900369873;
}
else
{
    SHerror.Value = T_superheat - T_setpointSH;
    if (SHerror <= 0)
    {
        opening.Value = opening + Gain*SHerror;
    }
    if (SHerror >= 0)
    {
        opening.Value = opening + Gain*SHerror;
    }
}
```

```

}
}

```

```

opening.Value = Math.Max(opening,0.01);
opening.Value = Math.Min(opening,1);

```

C.1.2 Compressor Quick Script

This section elaborates on the operating logic of the compressor quick script.

C.1.2.1 Inputs and Outputs

Script Inputs And Outputs			
Cap	100	rpm	▼
P_dis	19.8315	bar	▼
P_max	23	bar	▼
P_min	19	bar	▼
Cap_old	100	rpm	▼
Cap_step	0.375	rpm	▼
P_UL	26	bar	▼
P_LL	2	bar	▼
P_suc	4.77921	bar	▼

Figure 71: Compressor quick script Input and output example

The code receives suction and discharge gauge pressure inputs. If the suction pressure (P_suc) is less than the lower trip limit of 2 bar (P_LL), or the discharge pressure (P_dis) greater than the upper trip limit of 26 bar (P_UL), the simulation stops. P_dis is maintained within the ranges of the upper break point of 23 bar (P_max) and the lower break point of 19 bar (P_min). If P_dis is greater than P_max capacity (CAP) decreases until P_dis is within the set range again. Similarly, if P_dis is less than P_min, CAP increases to restore P_dis within the set range. CAP is limited between 25 and 100%.

C.1.2.2 Code

```
string state = Project.SimulationStateText;
if(state == "INACTIVE" || state == "SOLVINGSTEADYSTATE")

{
    Cap.Value = 100*(2*Math.PI)/60;
}
else
{
    if(P_dis > P_max)
    {
        Cap_old.Value = Cap;
        Cap.Value = Cap - Cap_step;
    }
    if(P_dis < P_min)
    {
        Cap_old.Value = Cap;
        Cap.Value = Cap + Cap_step;
    }

    if (P_dis > P_UL || P_suc < P_LL)
    {
        Project.DeactivateSimulation();
    }
    Cap.Value = Math.Max(Cap,25*(2*Math.PI)/60);
    Cap.Value = Math.Min(Cap,100*(2*Math.PI)/60);
}
```

C.2 Transient Behaviour Investigation

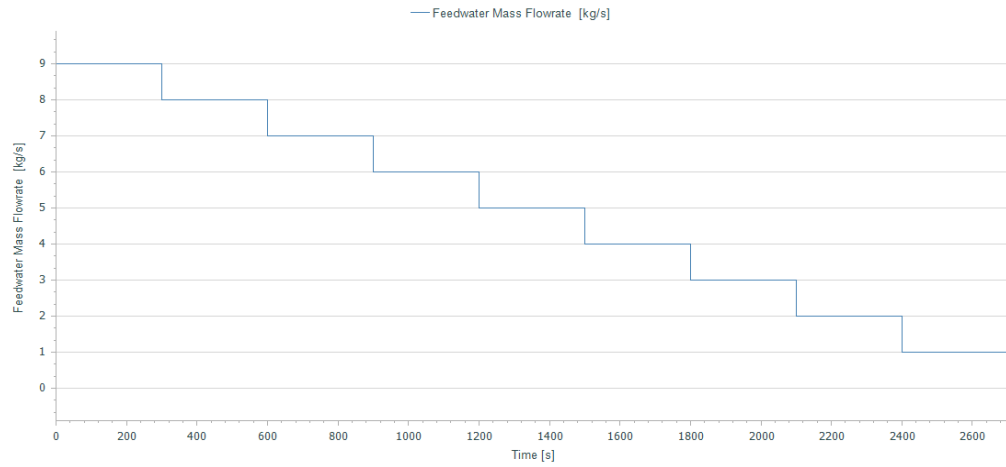
This section contains all relevant figures of ACU response to each of the investigated transient cases.

C.2.1 Case 1: Feedwater Supply Gradually Turned Off

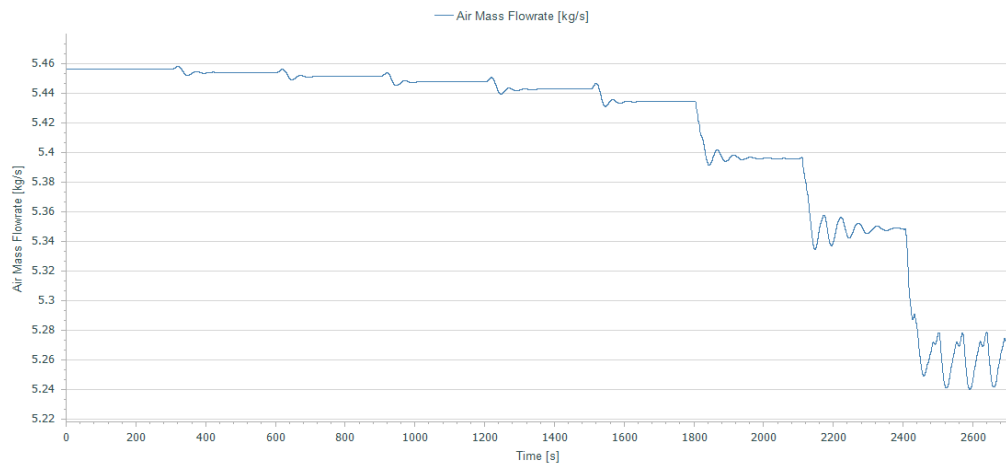
All relevant figures are presented on the following seven pages as Figure 72.

To continue reading from where you left off in the main dissertation, [click here](#).

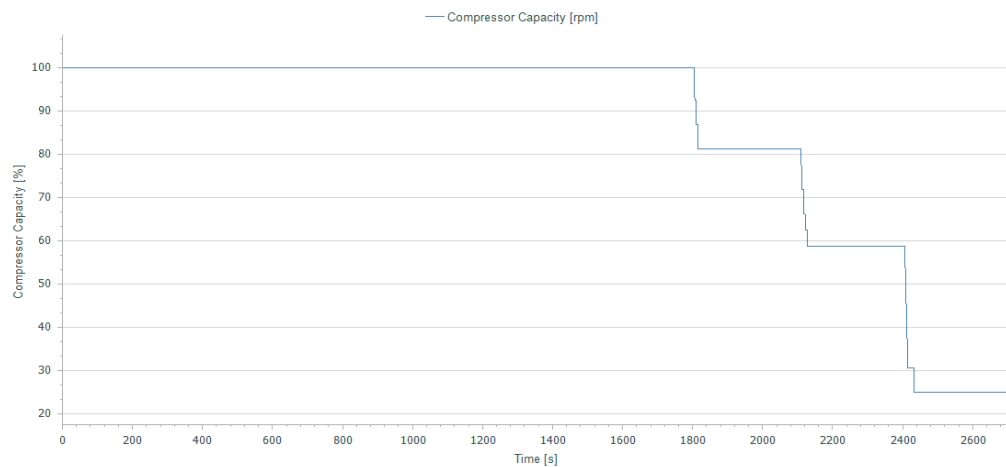
Variation in Feedwater Mass Flowrate over Time



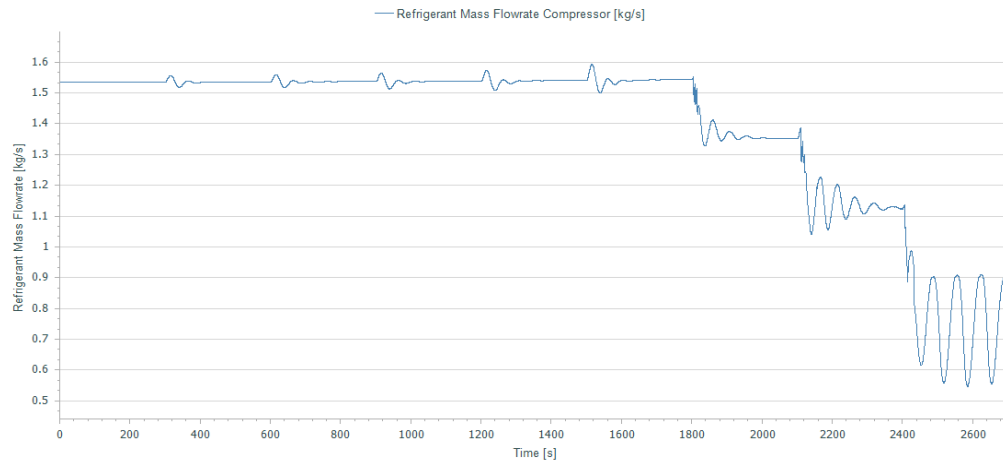
Variation in Air Mass Flowrate over Time



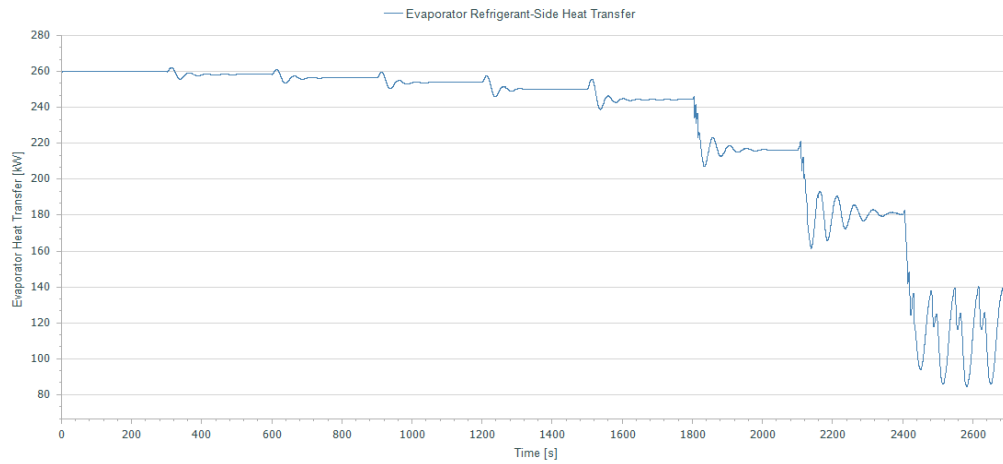
Variation in Compressor Capacity over Time



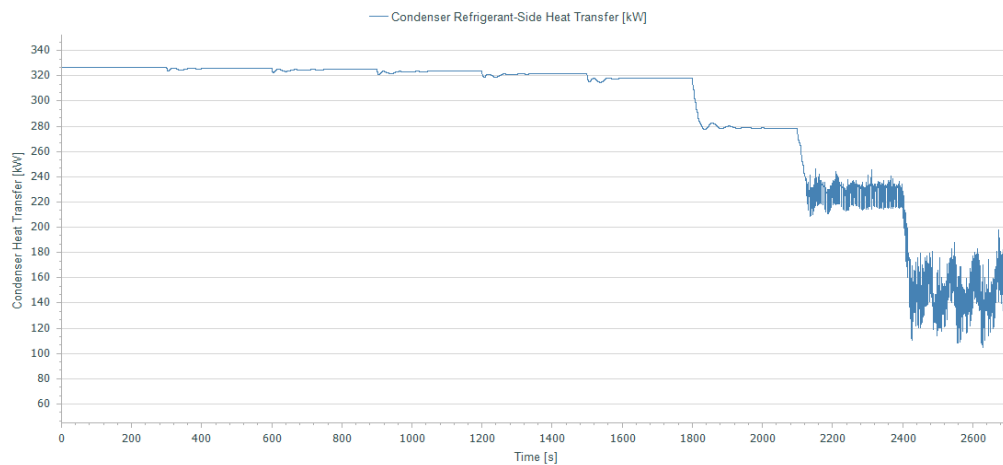
Variation in Refrigerant Mass Flowrate over Time



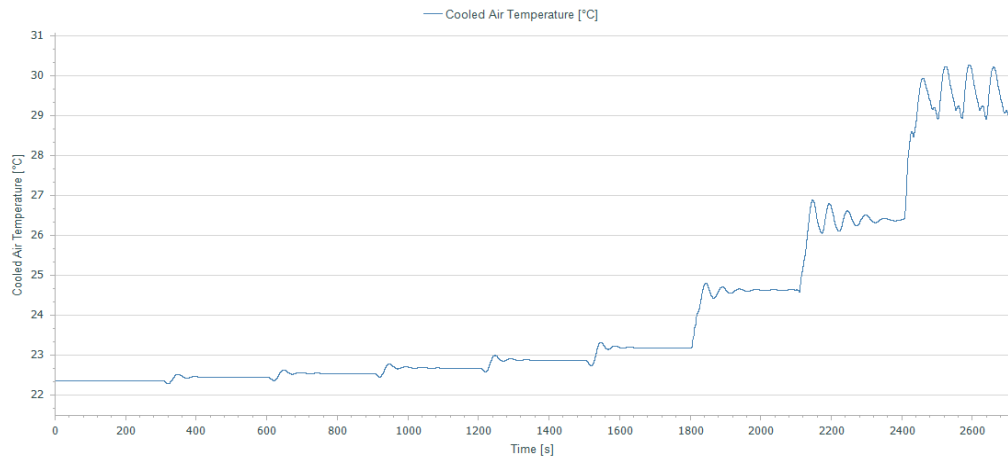
Variation in Evaporator Heat Transfer over Time



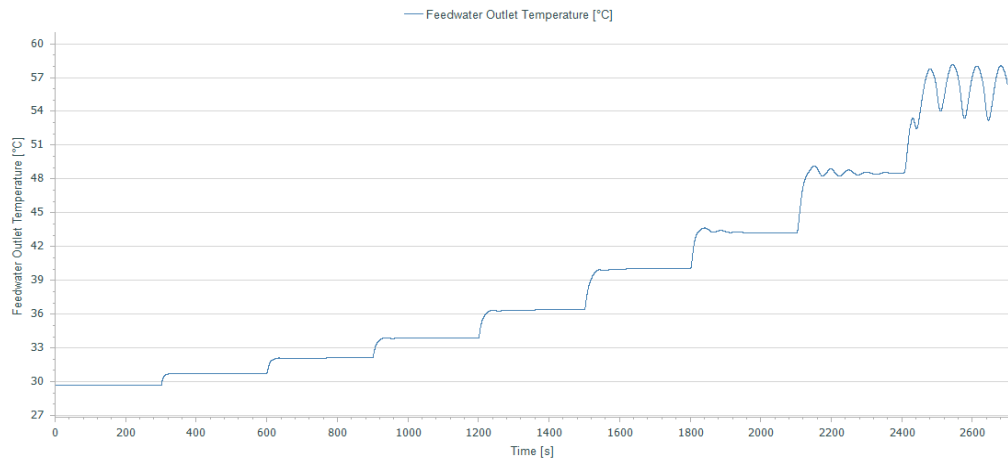
Variation in Condenser Heat Transfer over Time



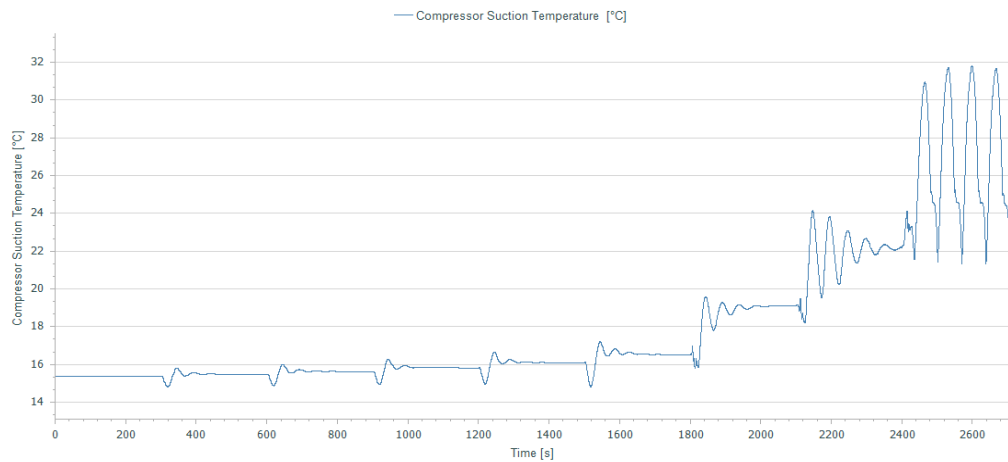
Variation in Cooled Air Temperature over Time



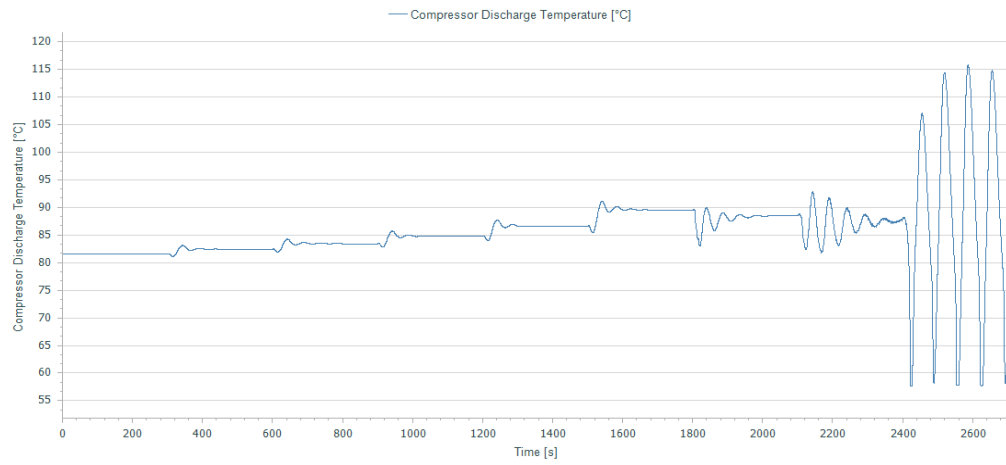
Variation in Feedwater Outlet Temperature over Time



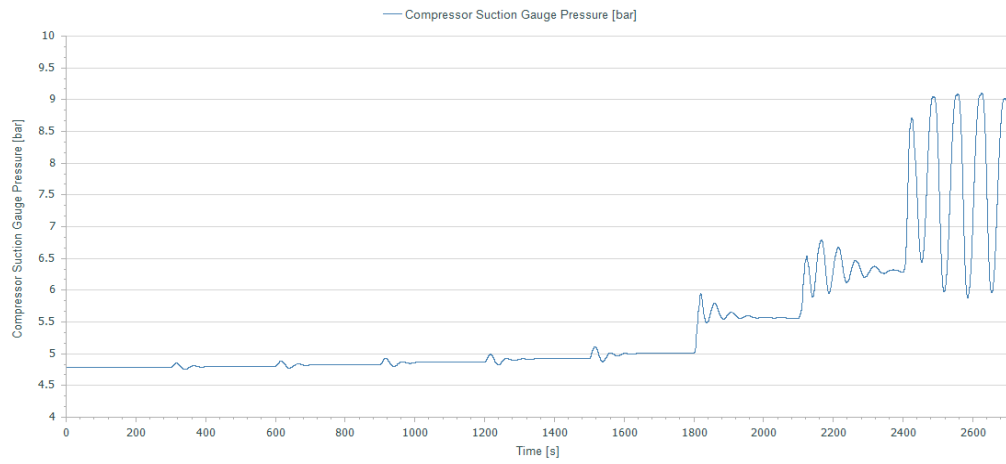
Variation in Compressor Suction Temperature over Time



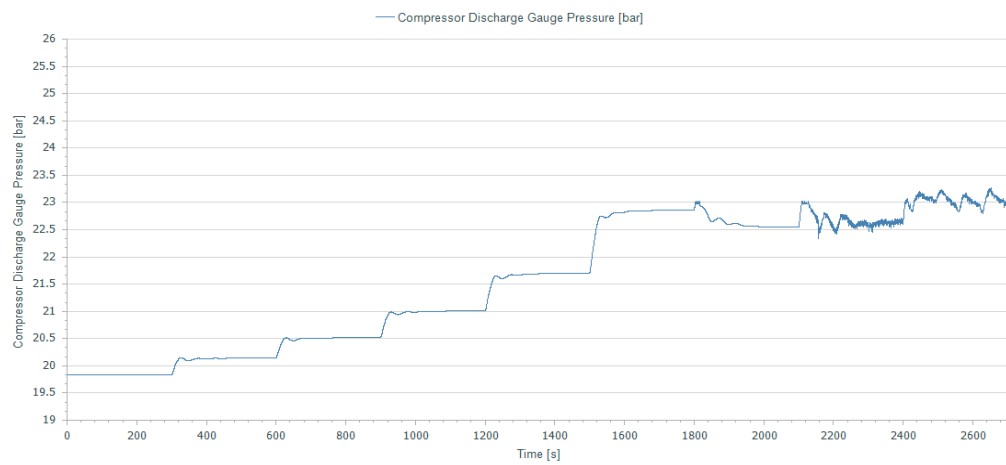
Variation in Compressor Discharge Temperature over Time

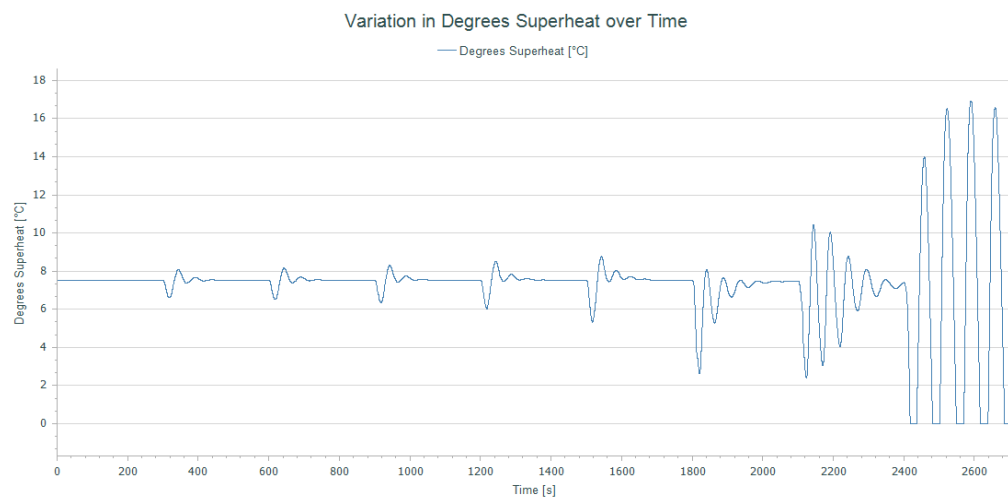
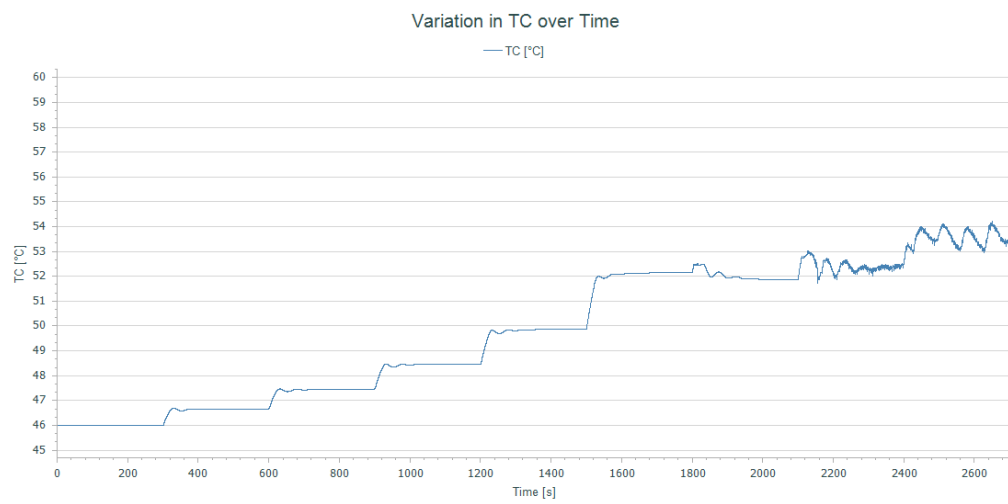
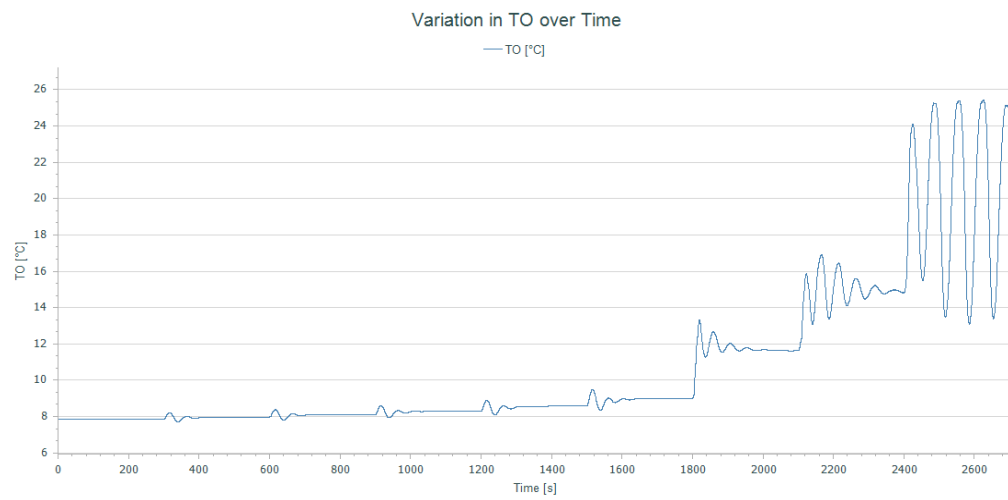


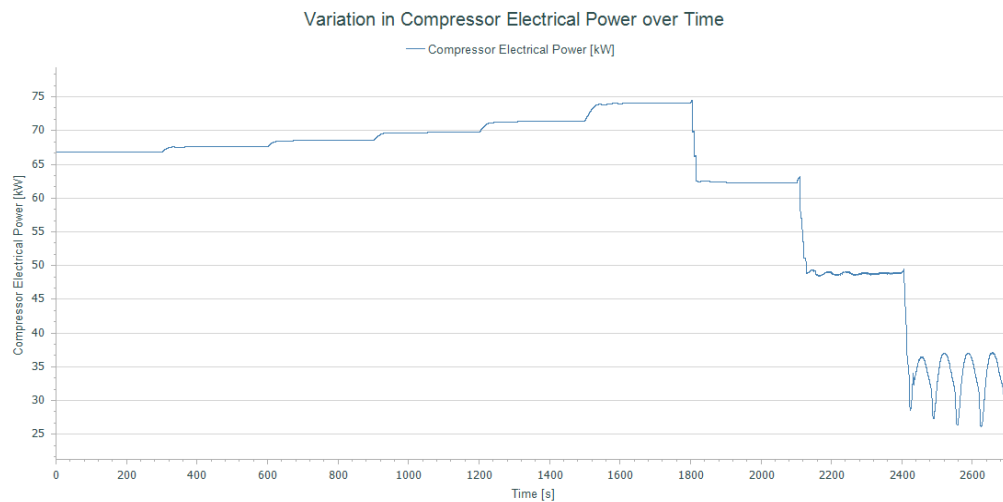
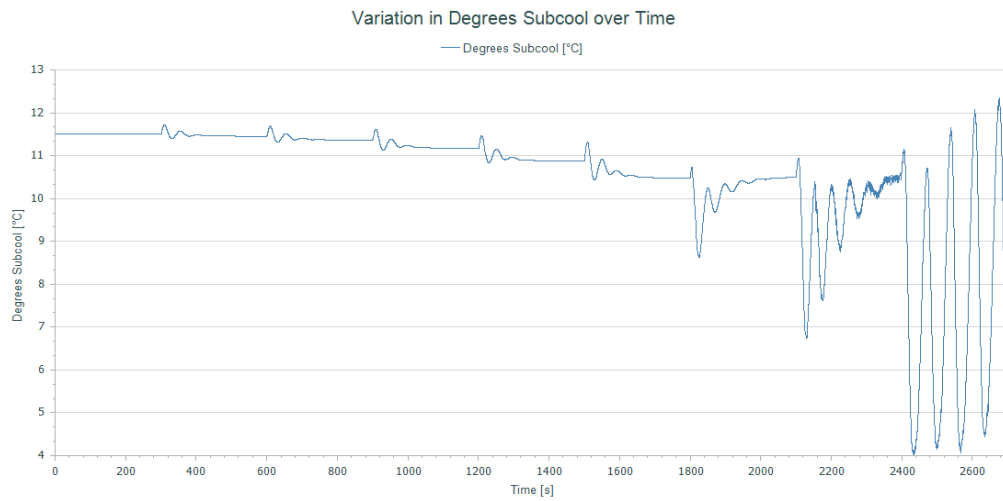
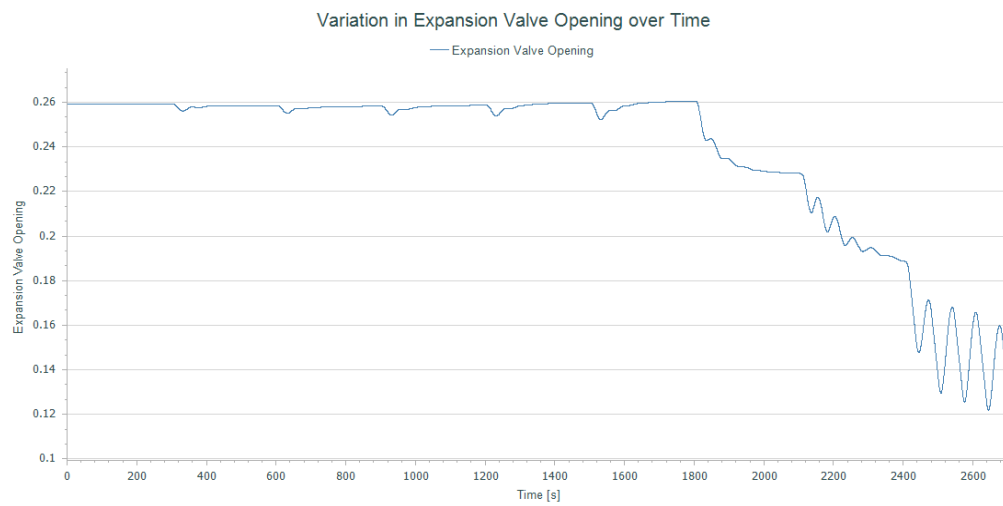
Variation in Compressor Suction Gauge Pressure over Time



Variation in Compressor Discharge Gauge Pressure over Time







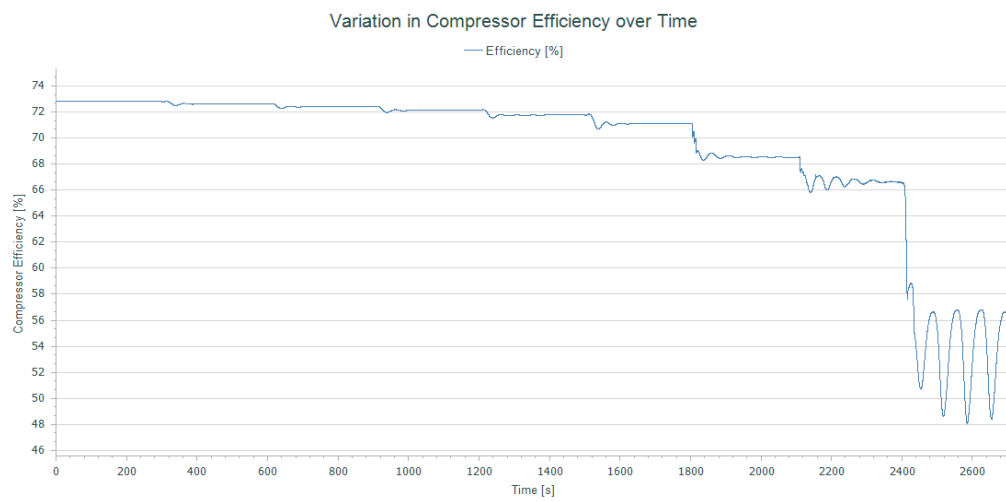


Figure 72: Case 1

C.2.1.1 Step Capacity Control Under the Case 1 Procedure

All relevant figures are presented on the following page as Figure73.

To continue reading from where you left off in the main dissertation, [click here](#).

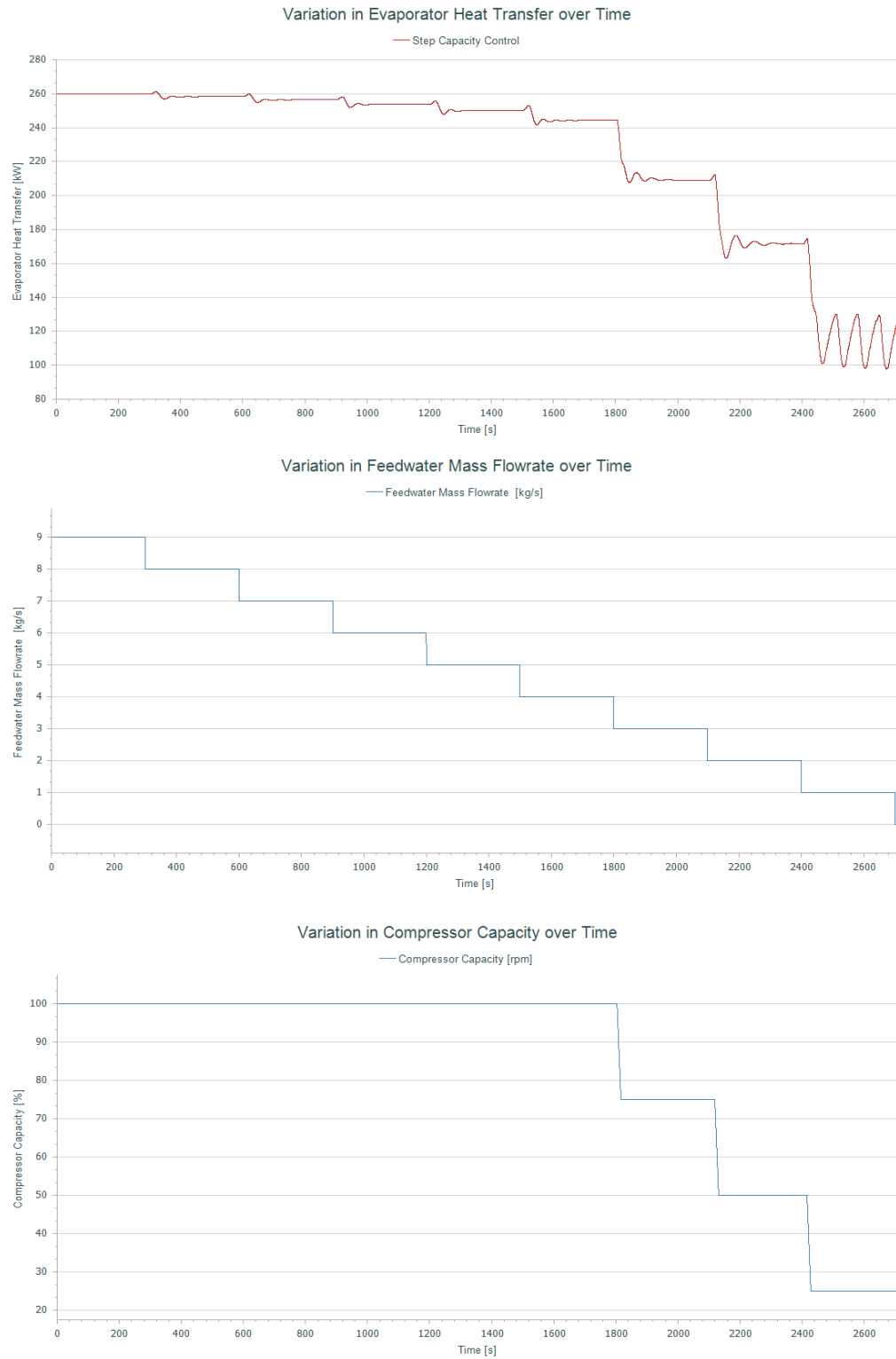


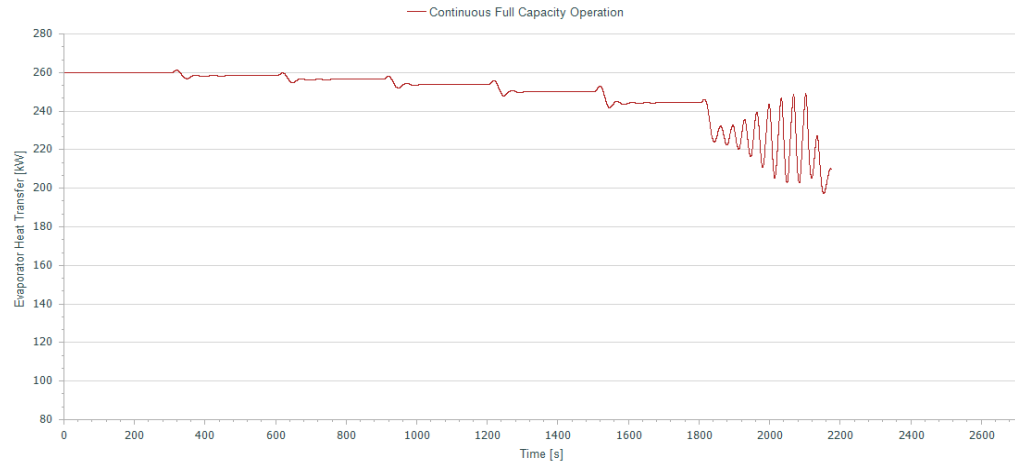
Figure 73: Case 1.1

C.2.1.2 Full Capacity Operation Under the Case 1 Procedure

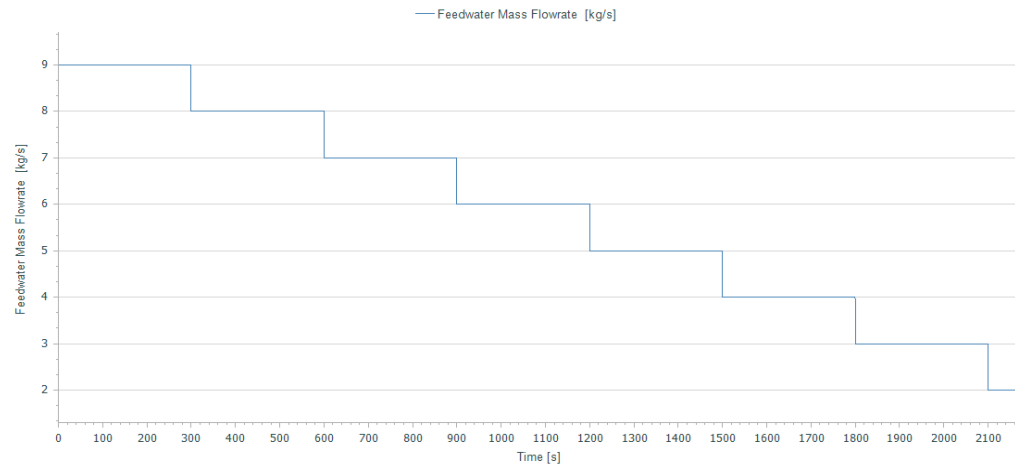
All relevant figures are presented on the following two pages as Figure 74.

To continue reading from where you left off in the main dissertation, [click here](#).

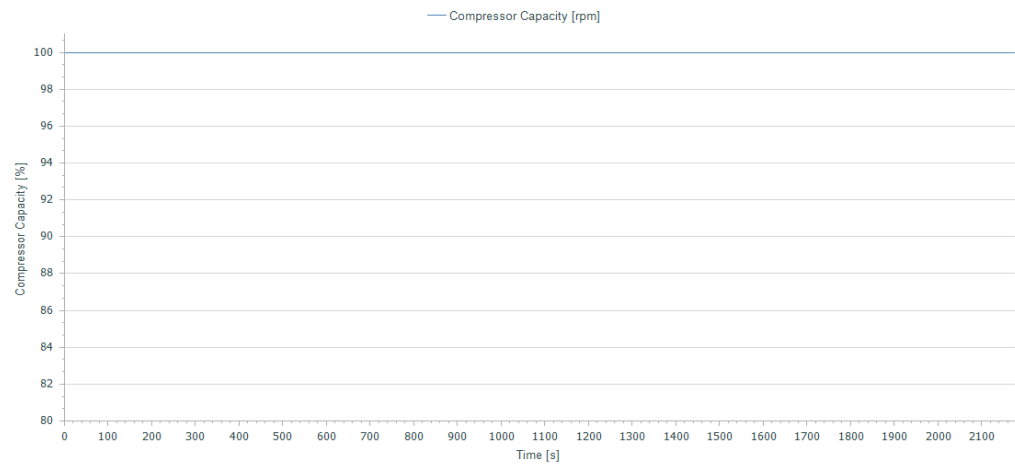
Variation in Evaporator Heat Transfer over Time



Variation in Feedwater Mass Flowrate over Time



Variation in Compressor Capacity over Time



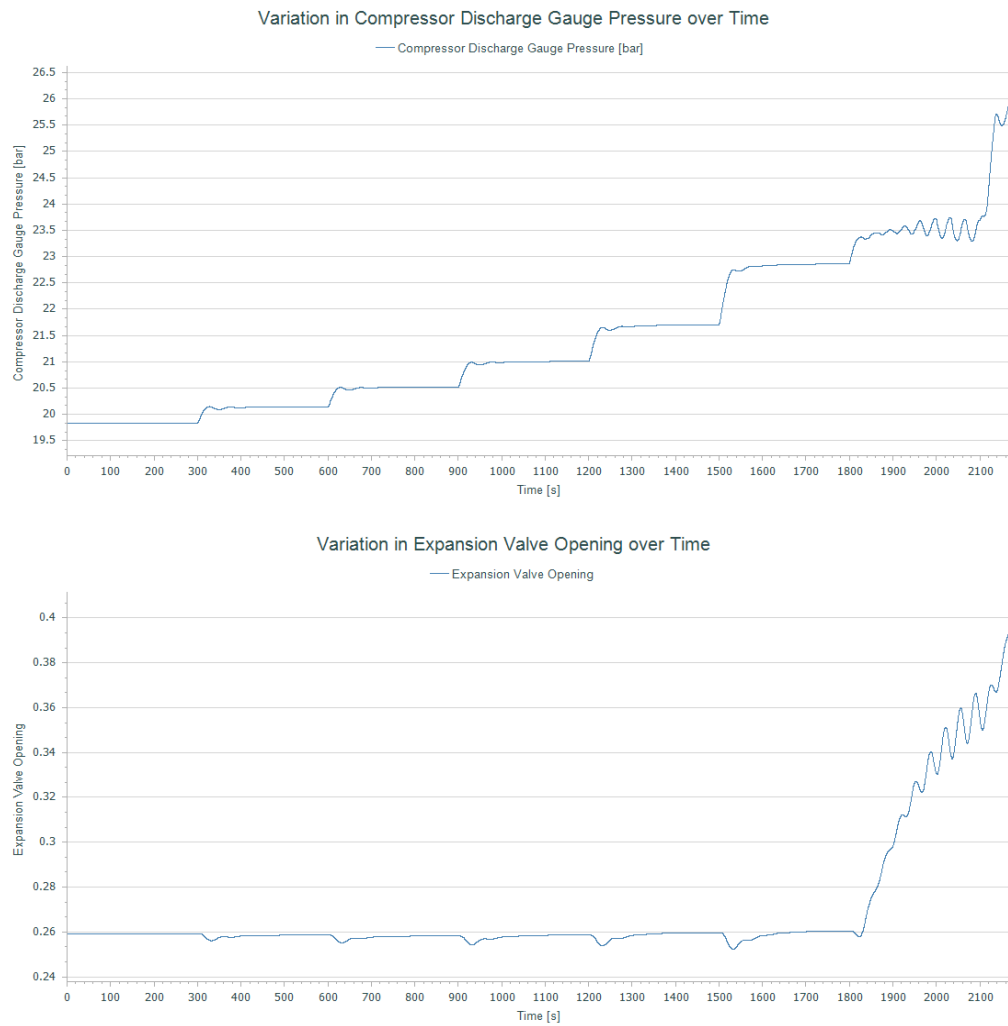


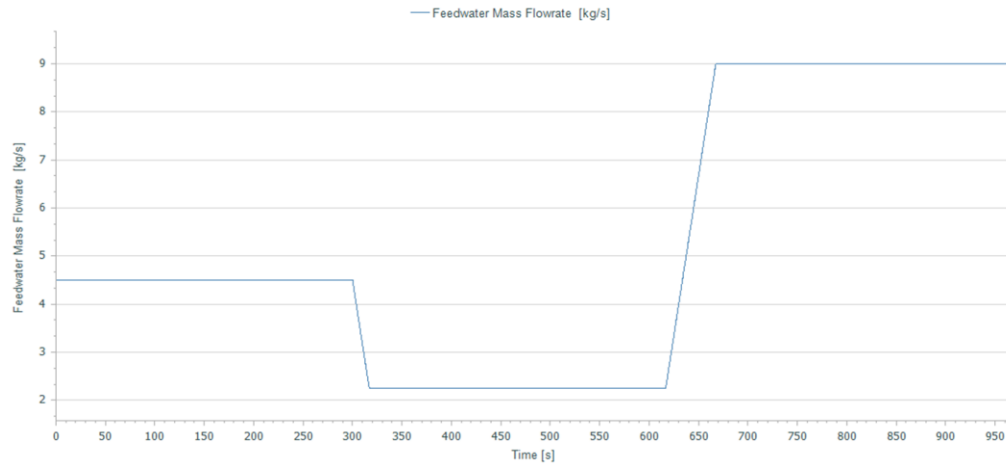
Figure 74: Case 1.2

C.2.2 Case 2: Feedwater Fluctuations

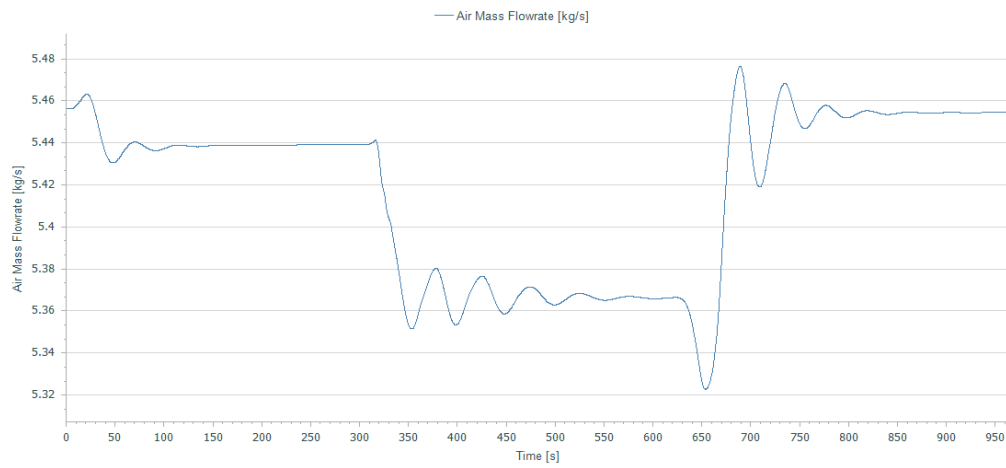
All relevant figures are presented on the following seven pages as Figure 75.

To continue reading from where you left off in the main dissertation, [click here](#).

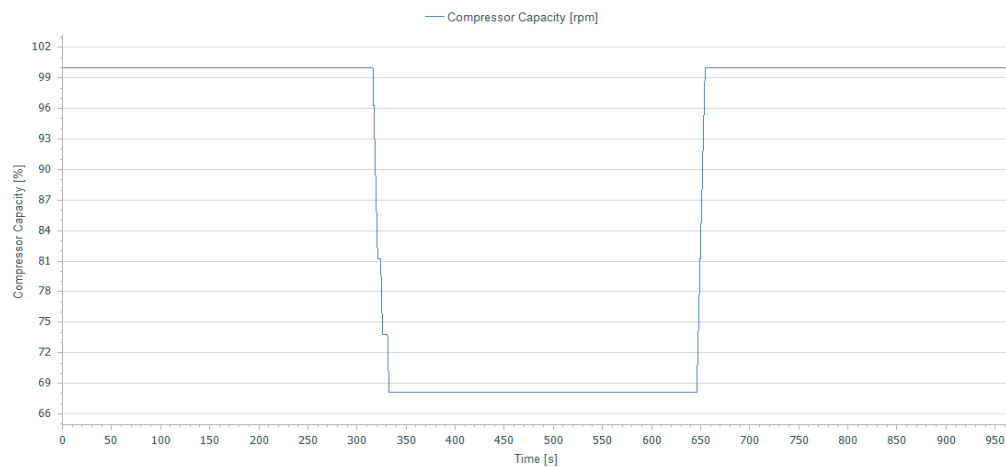
Variation in Feedwater Mass Flowrate over Time



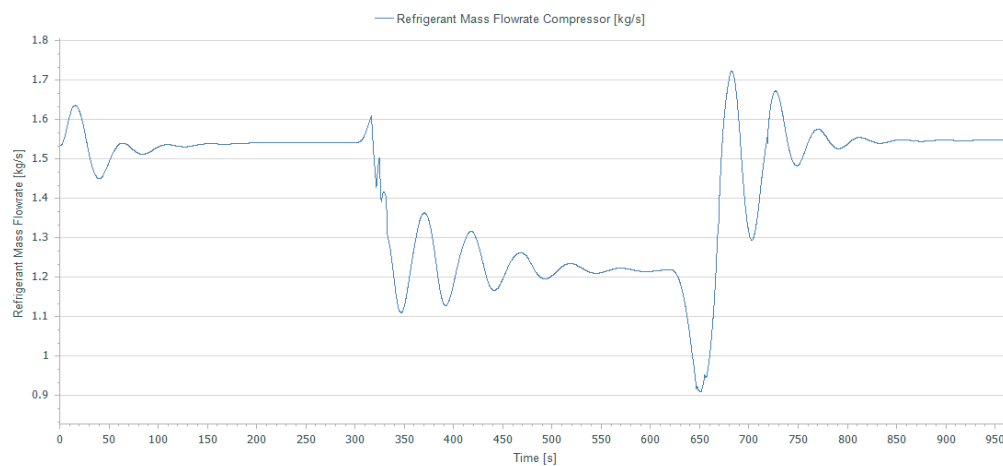
Variation in Air Mass Flowrate over Time



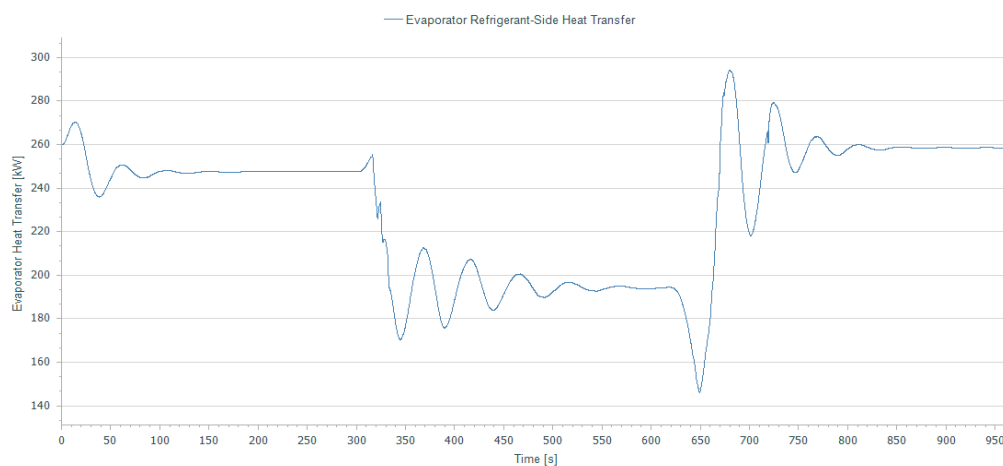
Variation in Compressor Capacity over Time



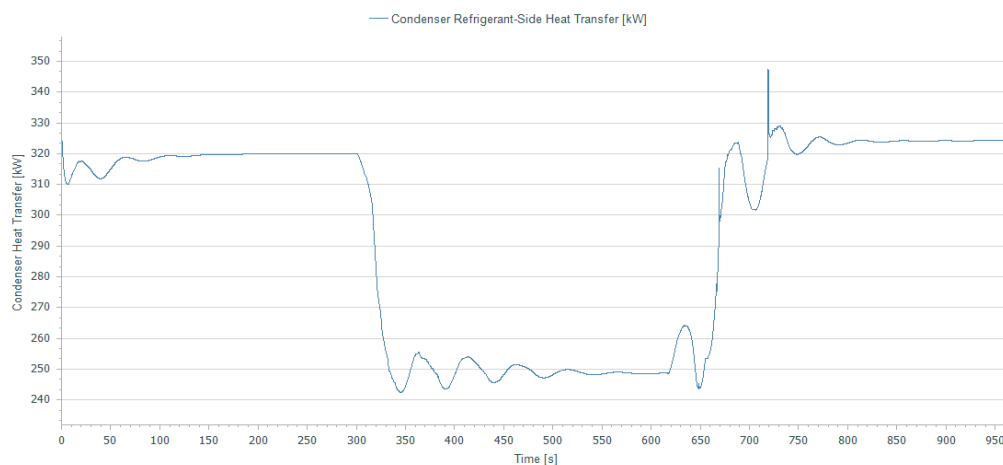
Variation in Refrigerant Mass Flowrate over Time

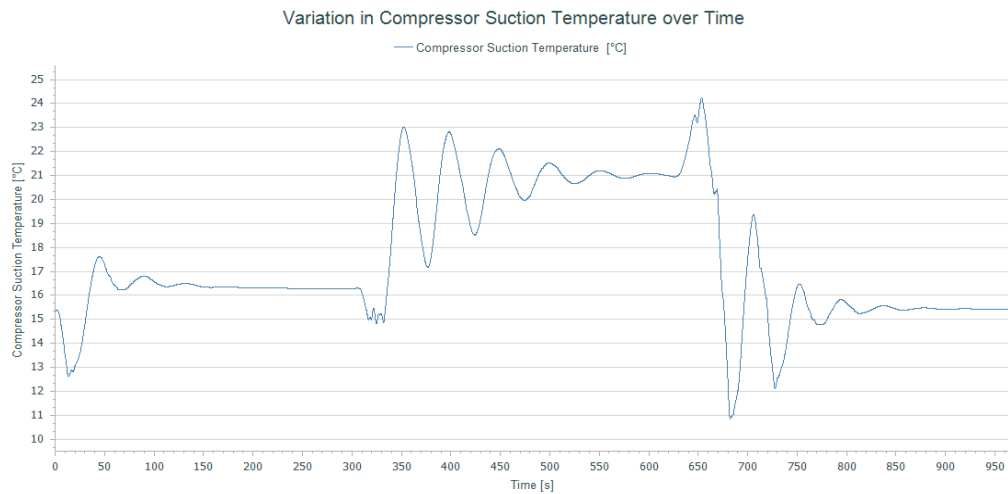
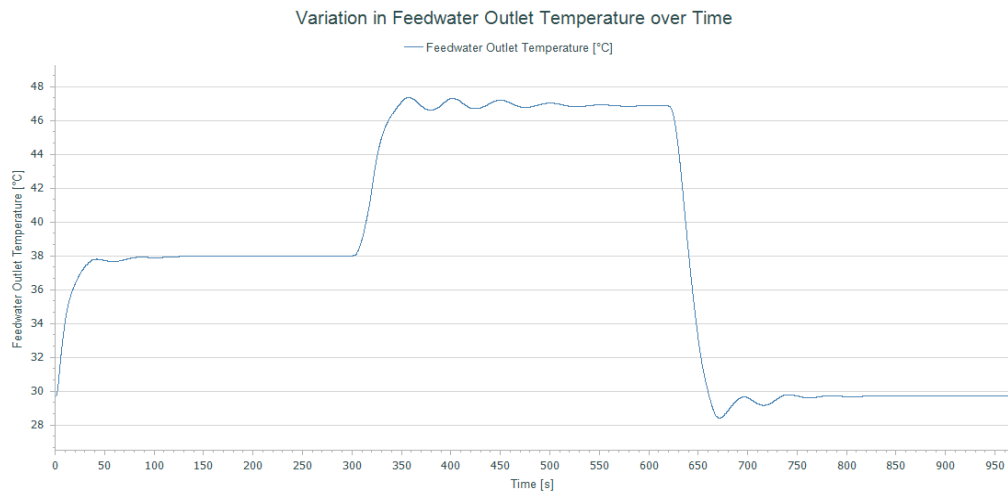
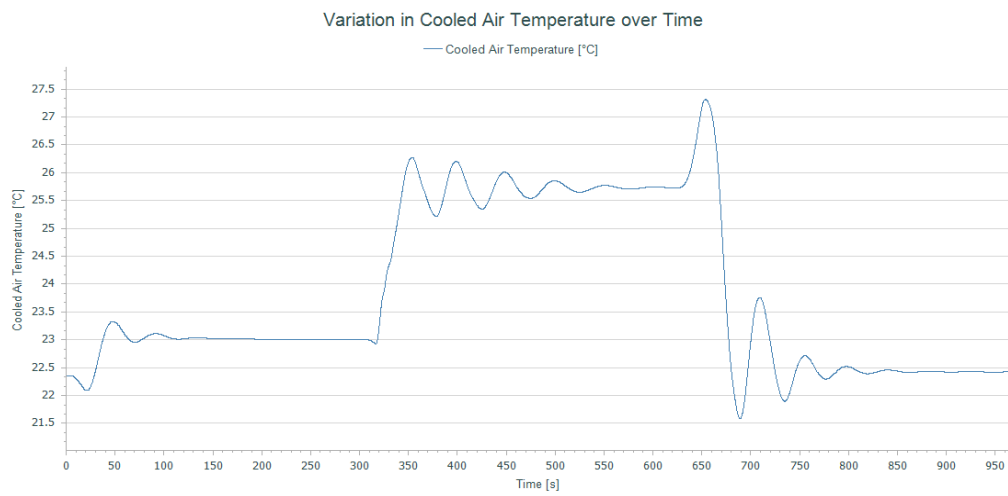


Variation in Evaporator Heat Transfer over Time

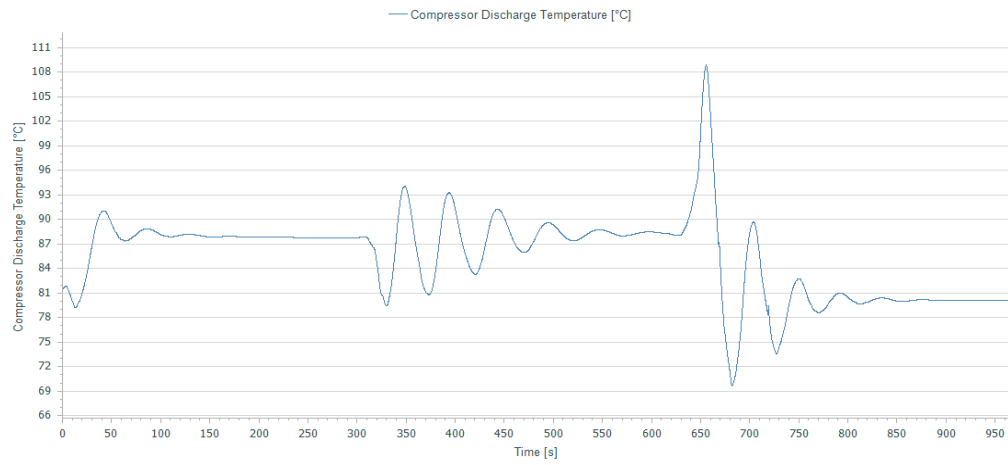


Variation in Condenser Heat Transfer over Time

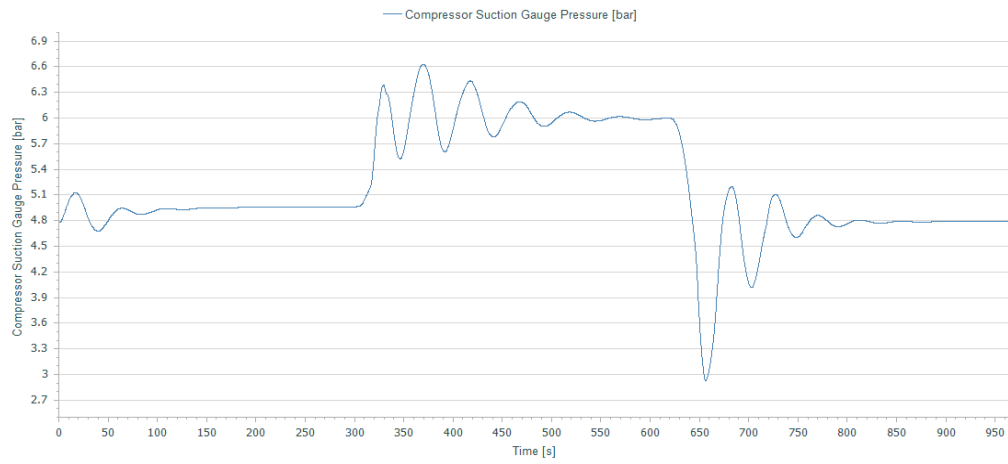




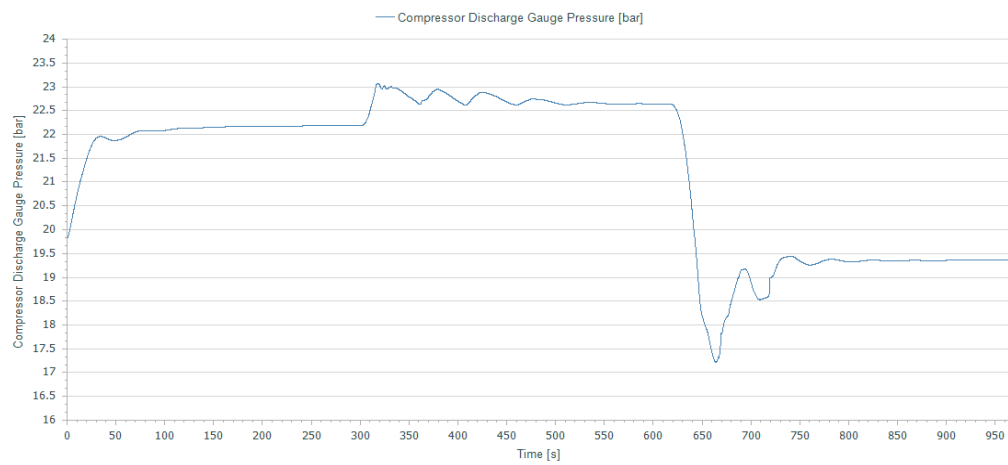
Variation in Compressor Discharge Temperature over Time

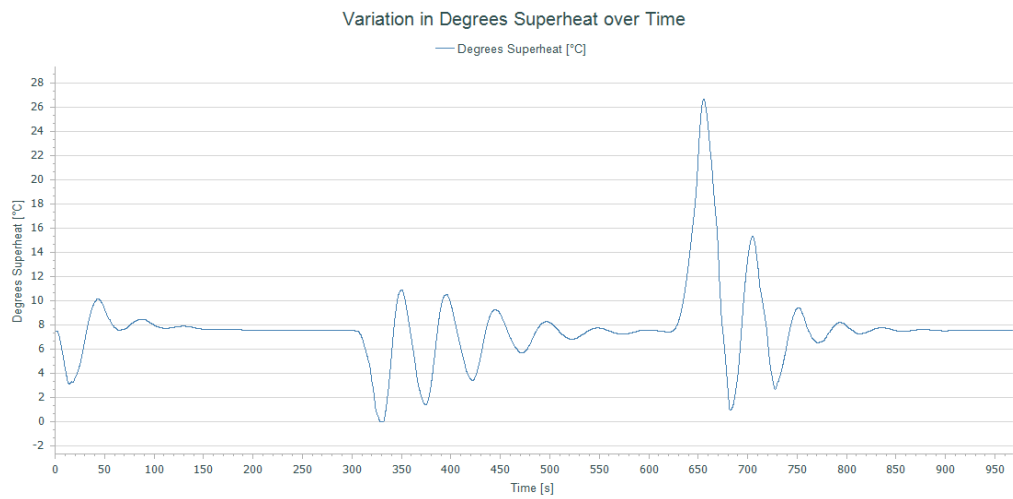
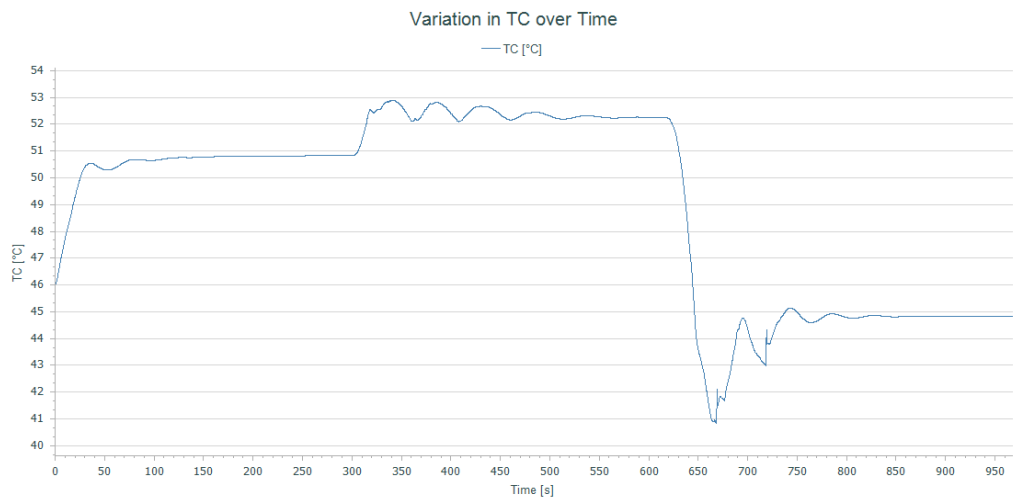
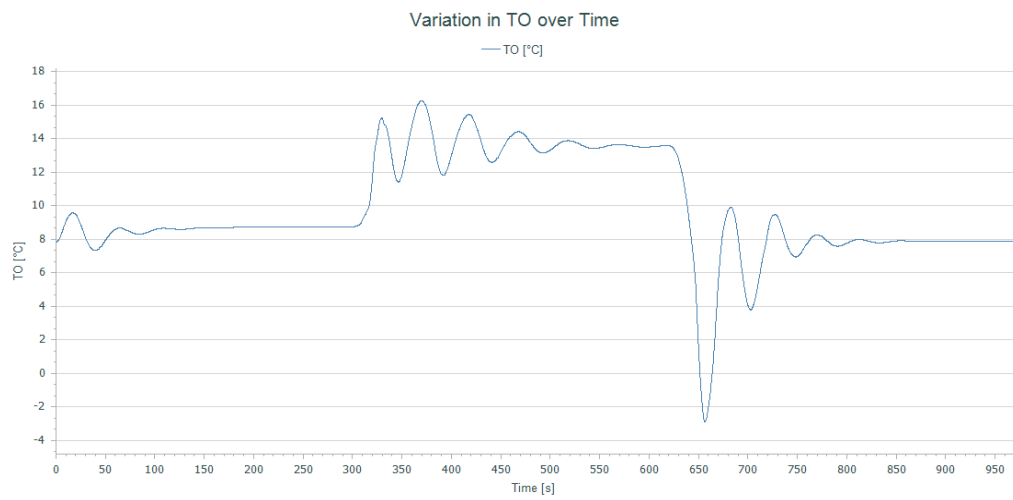


Variation in Compressor Suction Gauge Pressure over Time

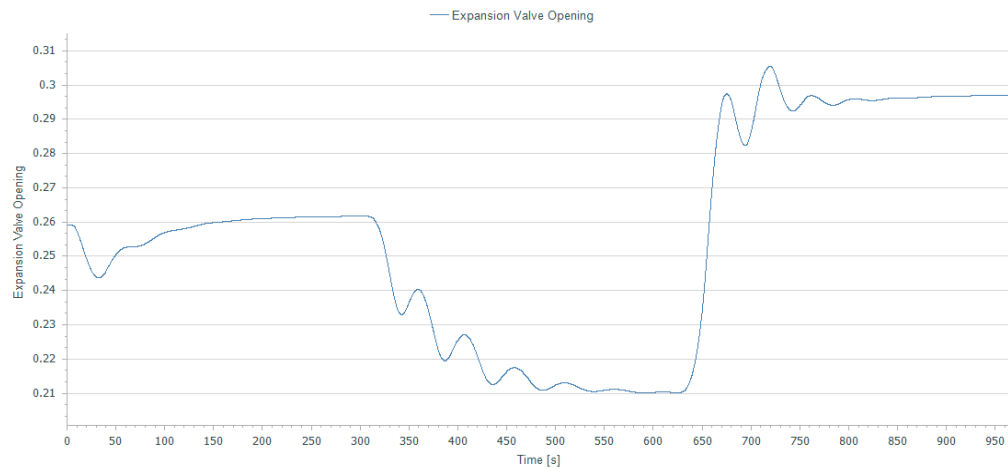


Variation in Compressor Discharge Gauge Pressure over Time

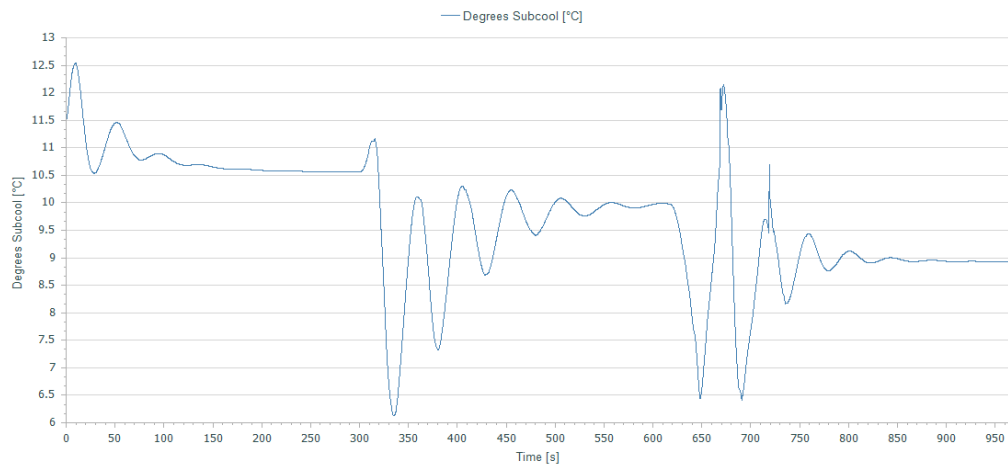




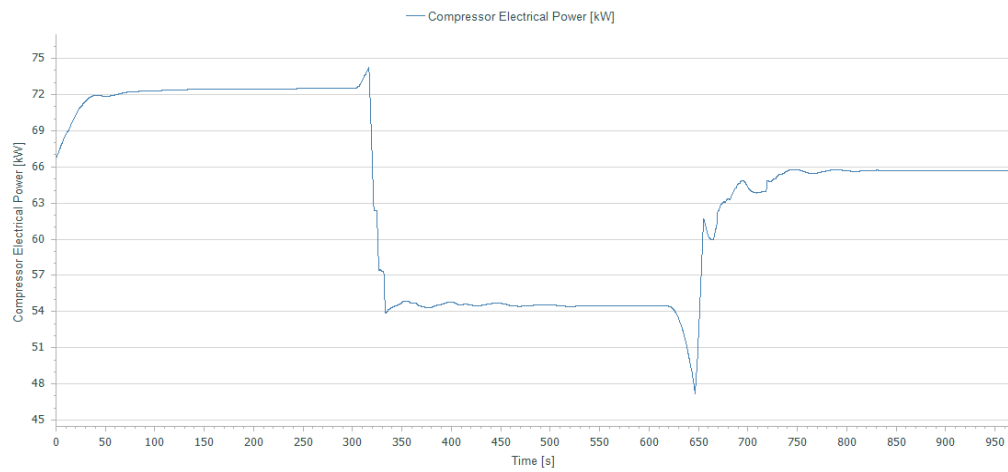
Variation in Expansion Valve Opening over Time



Variation in Degrees Subcool over Time



Variation in Compressor Electrical Power over Time



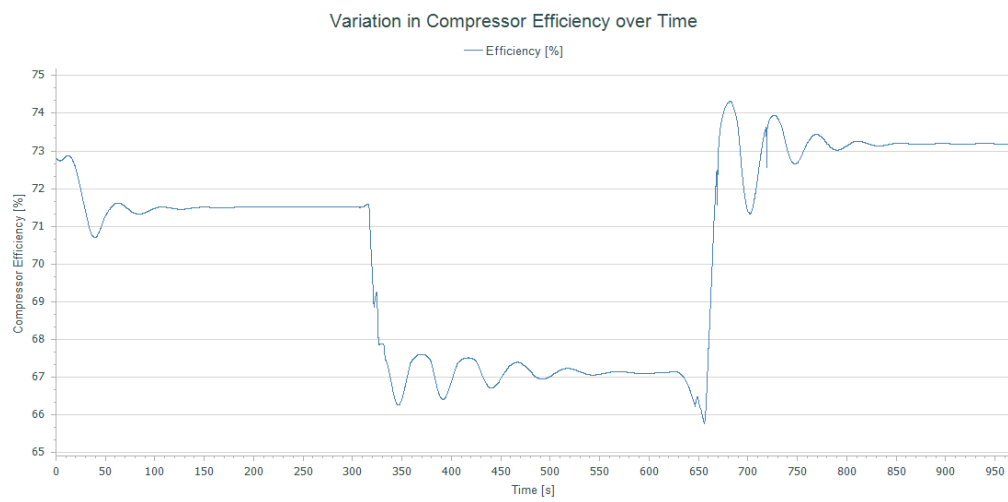


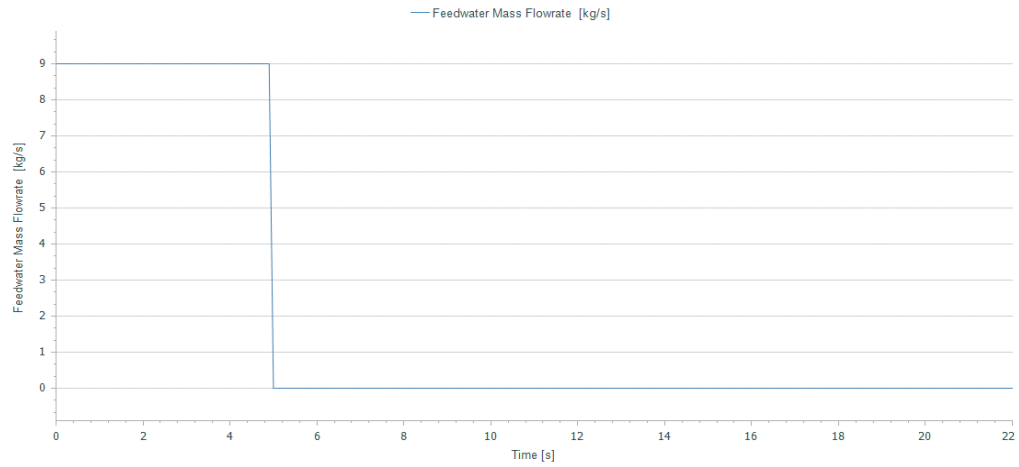
Figure 75: Case 2

C.2.3 Case 3: Feedwater Supply Instantaneously Turned Off

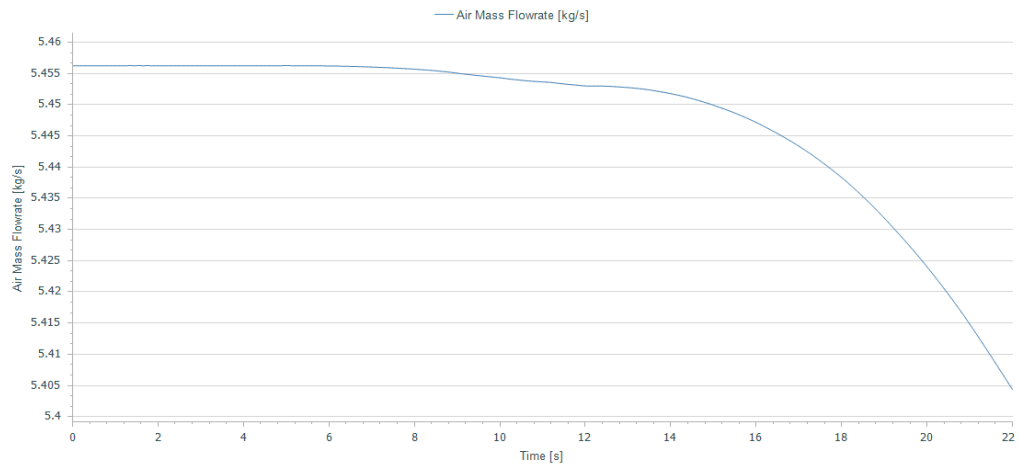
All relevant figures are presented on the following seven pages as Case 76.

To continue reading from where you left off in the main dissertation, [click here](#).

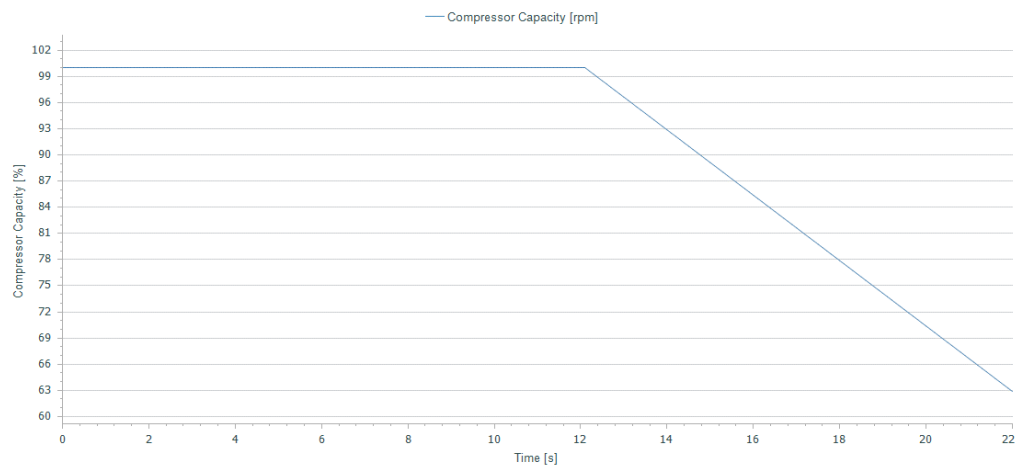
Variation in Feedwater Mass Flowrate over Time



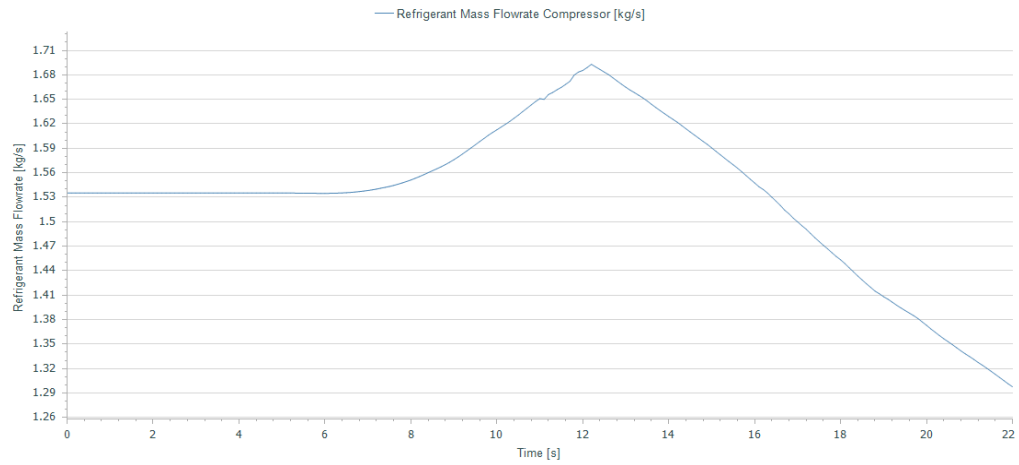
Variation in Air Mass Flowrate over Time



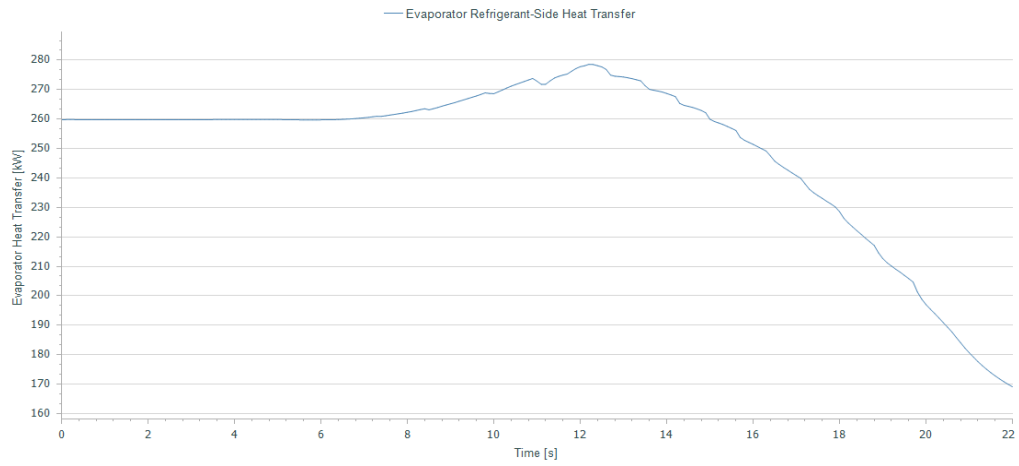
Variation in Compressor Capacity over Time



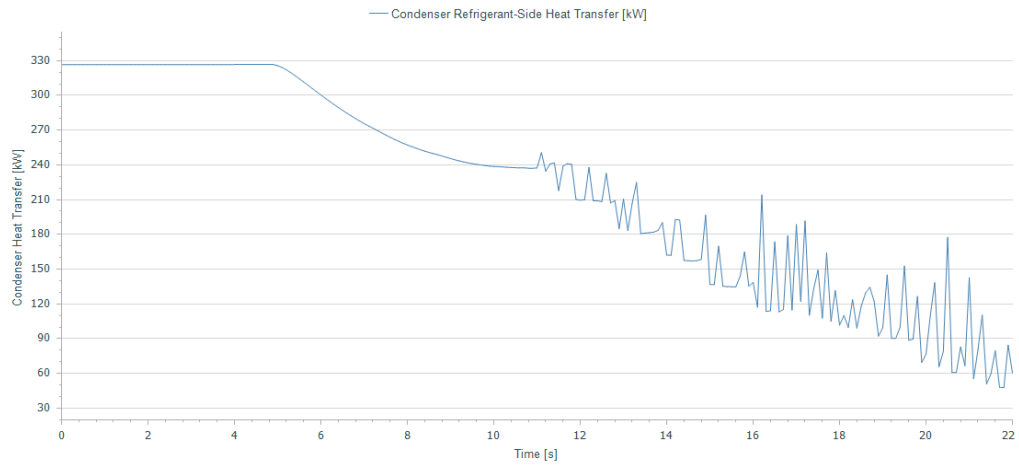
Variation in Refrigerant Mass Flowrate over Time



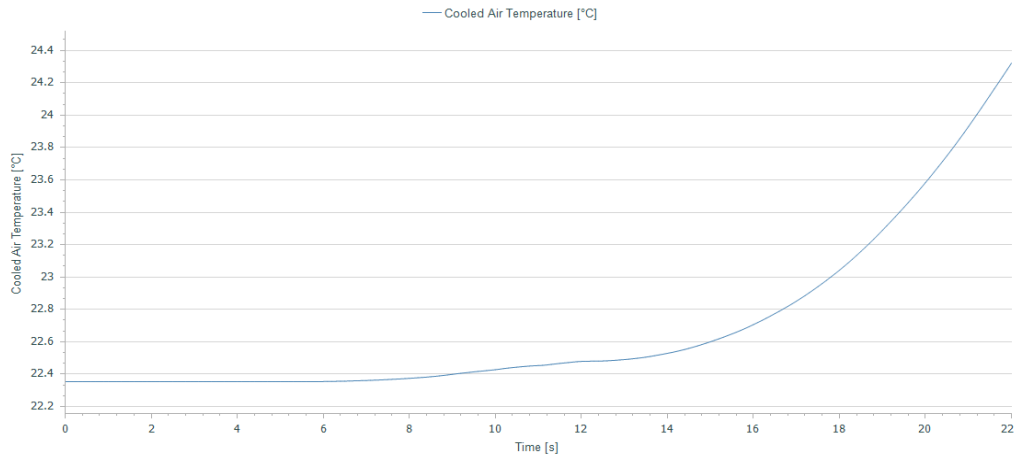
Variation in Evaporator Heat Transfer over Time



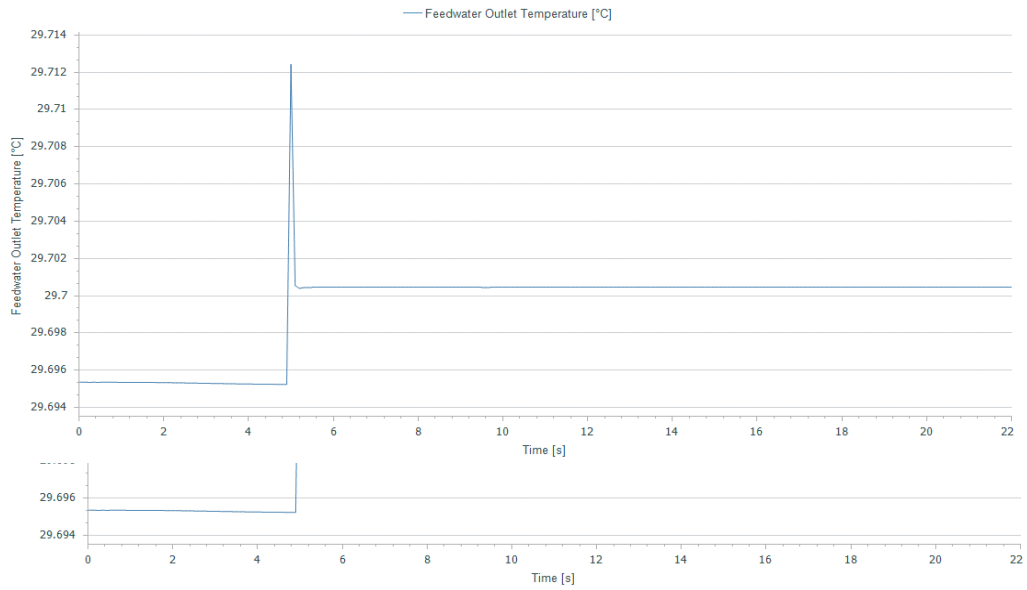
Variation in Condenser Heat Transfer over Time



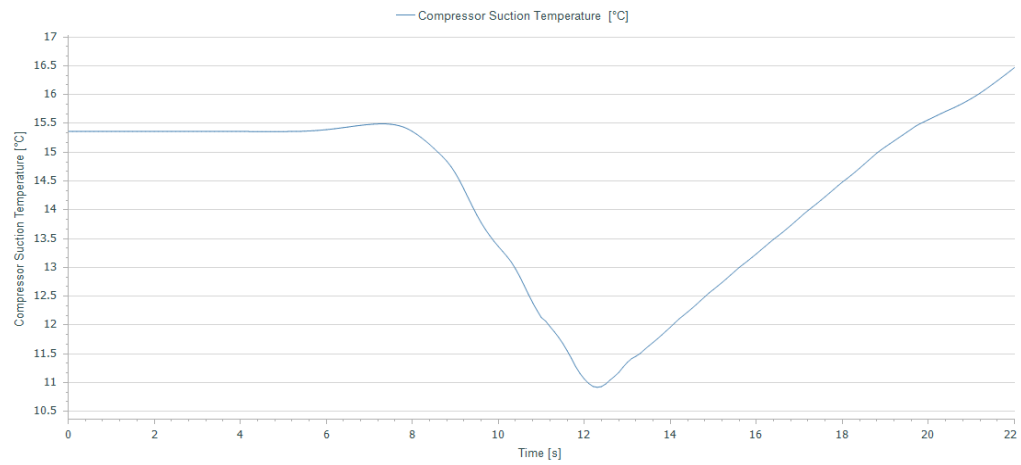
Variation in Cooled Air Temperature over Time

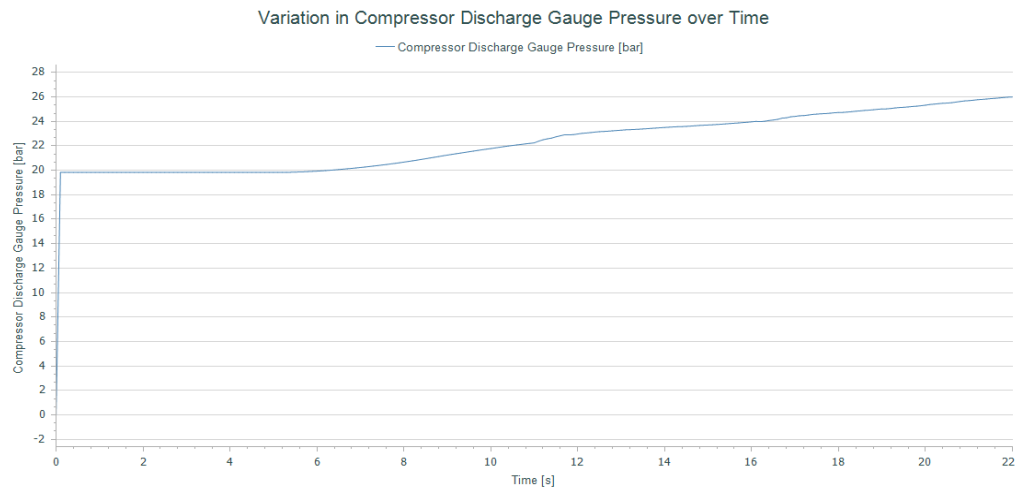
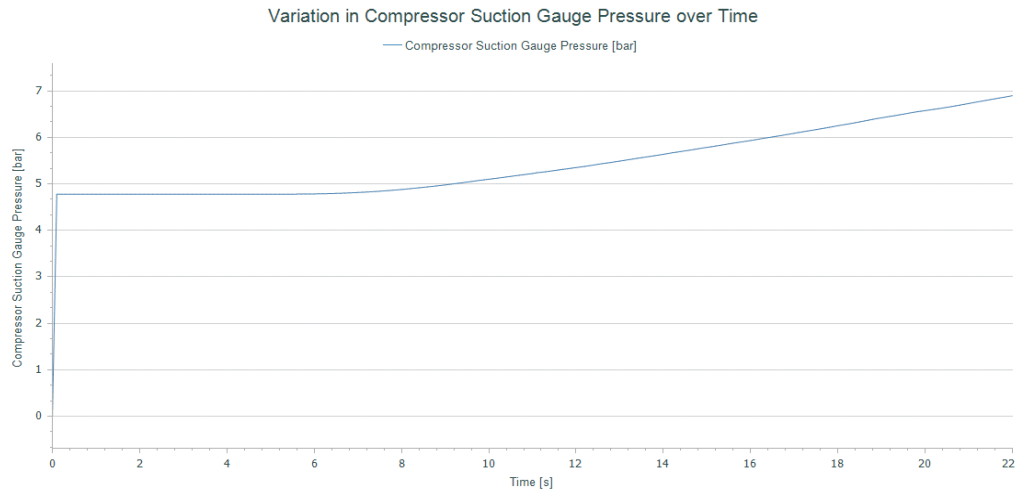
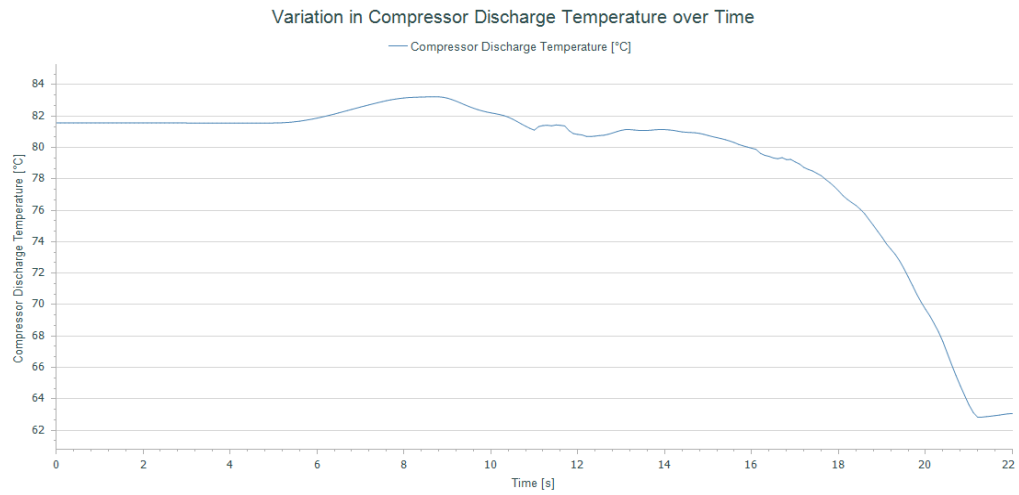


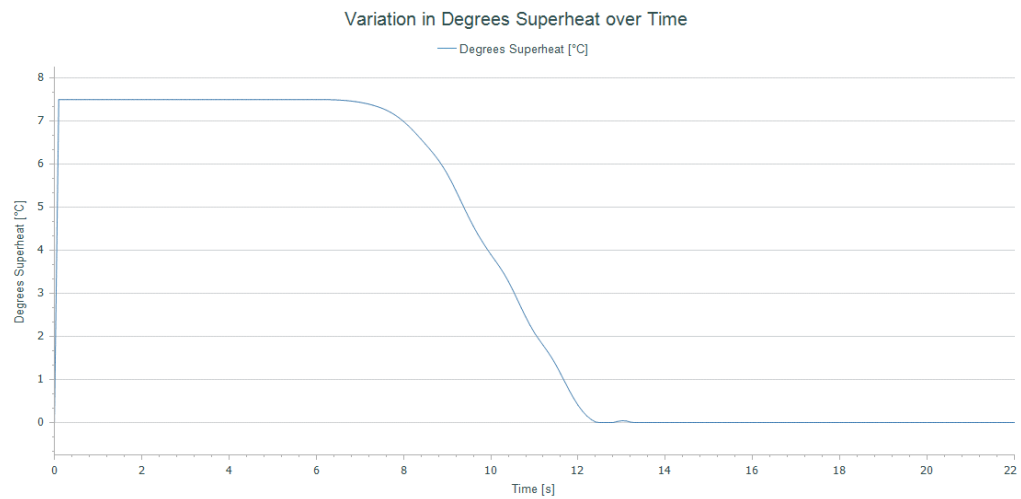
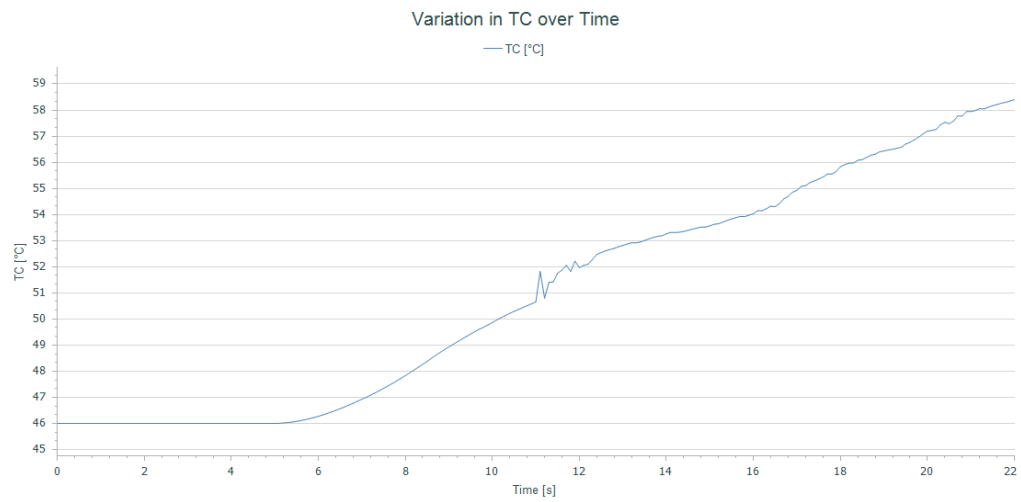
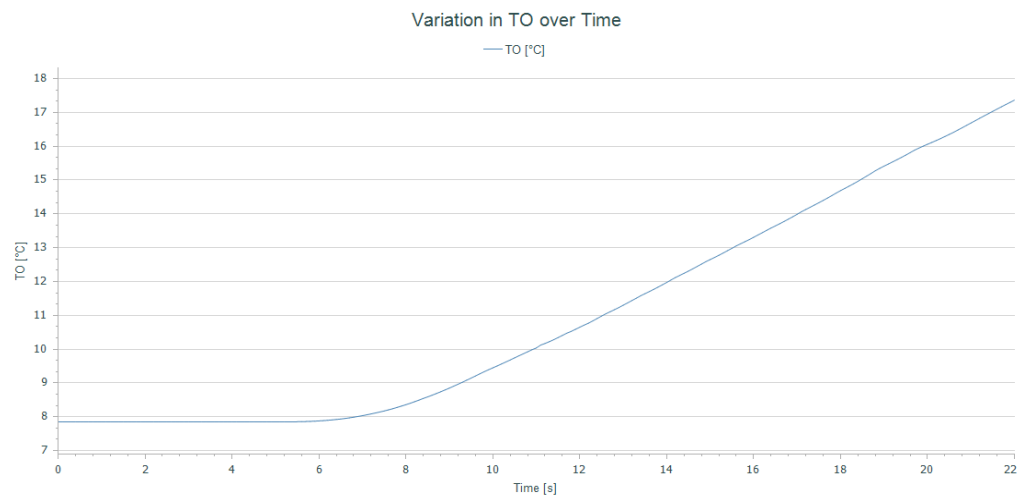
Variation in Feedwater Outlet Temperature over Time



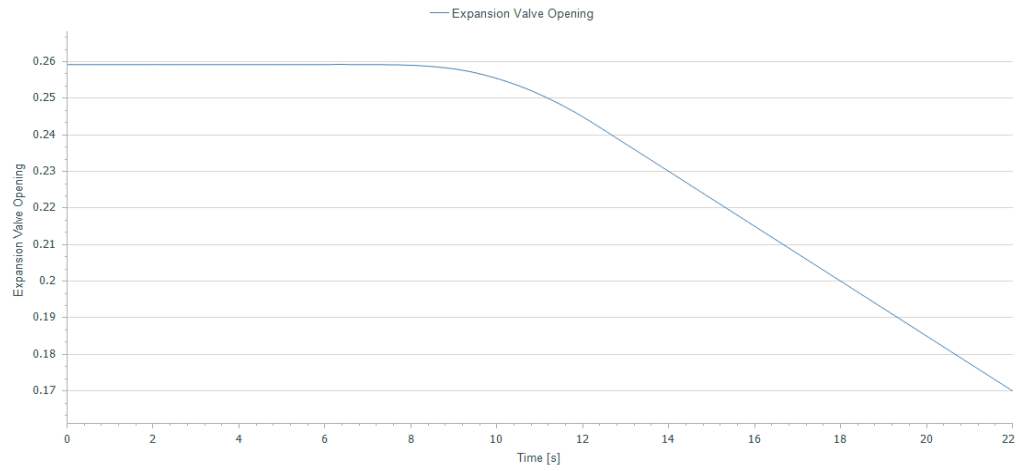
Variation in Compressor Suction Temperature over Time



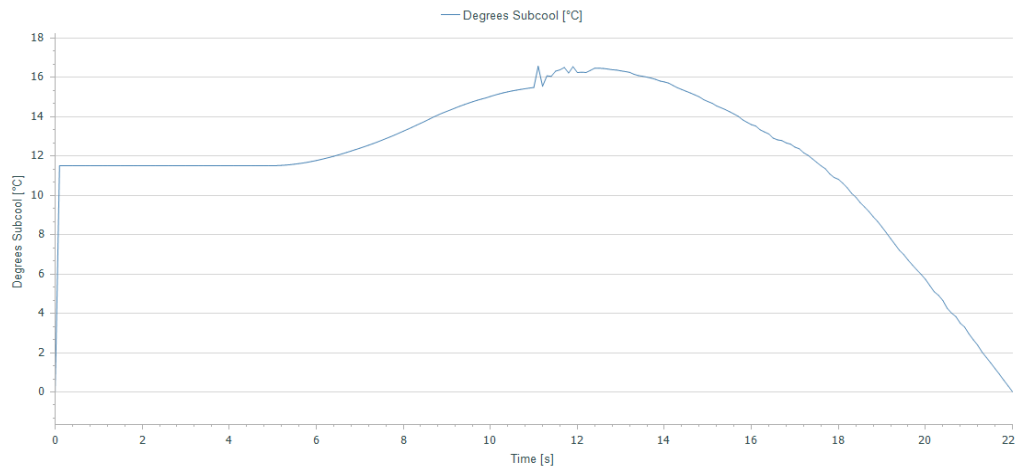




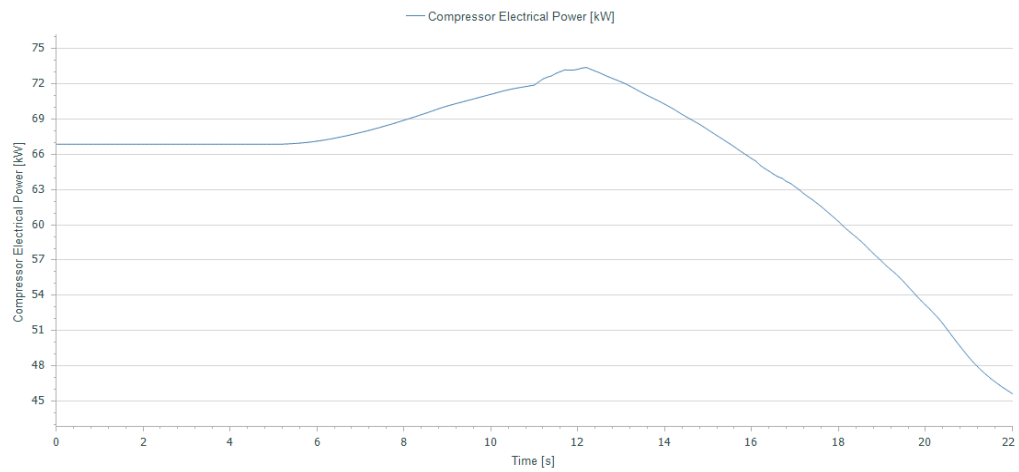
Variation in Expansion Valve Opening over Time



Variation in Degrees Subcool over Time



Variation in Compressor Electrical Power over Time



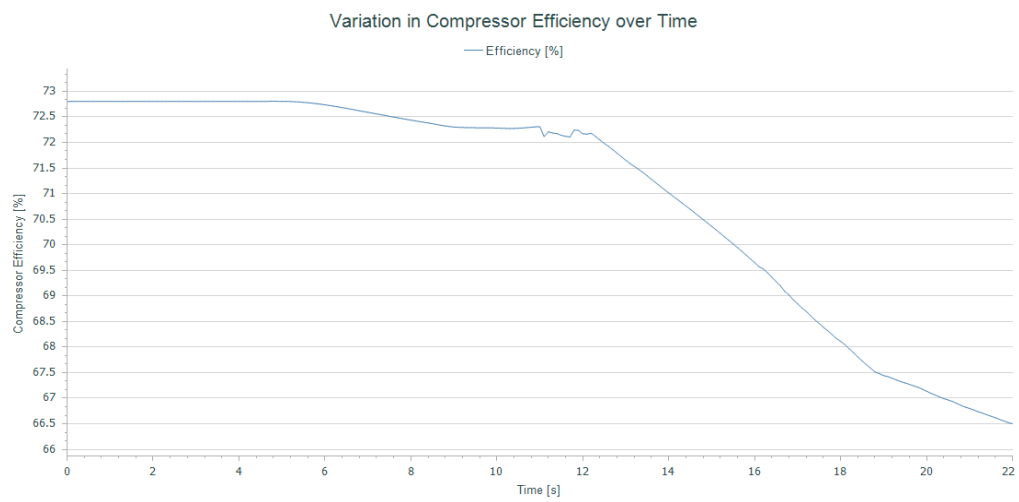


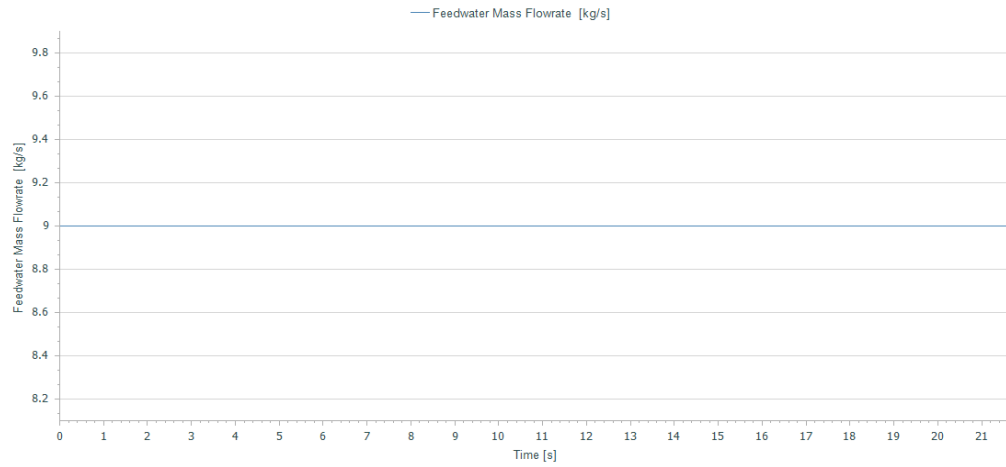
Figure 76: Case 3

C.2.4 Case 4: Fan Failure

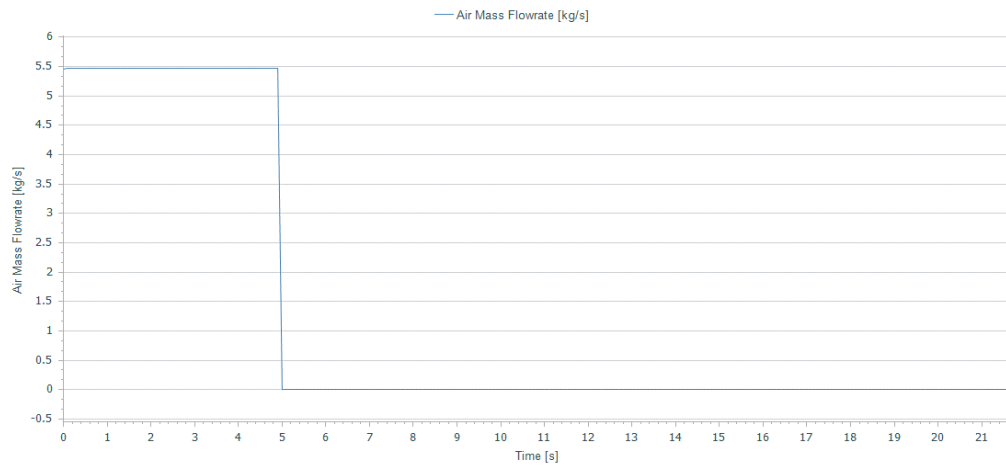
All relevant figures are presented on the following seven pages as Figure 77.

To continue reading from where you left off in the main dissertation, [click here](#).

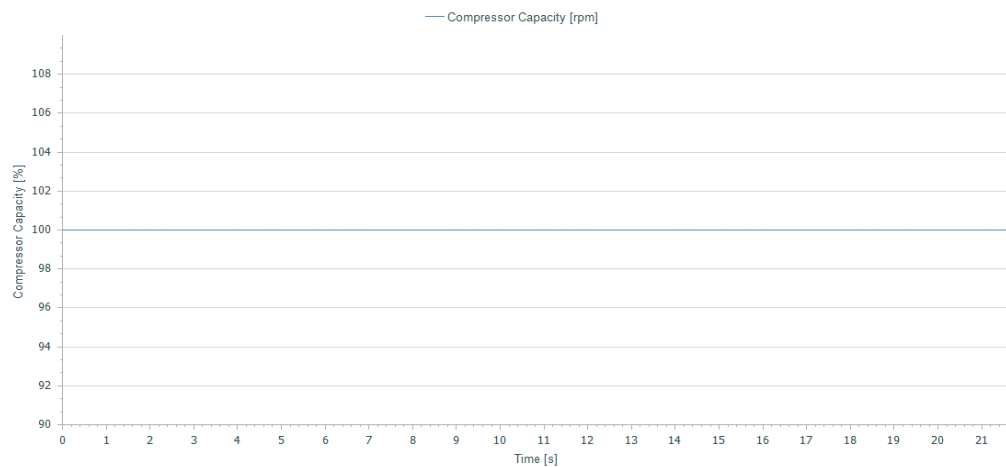
Variation in Feedwater Mass Flowrate over Time



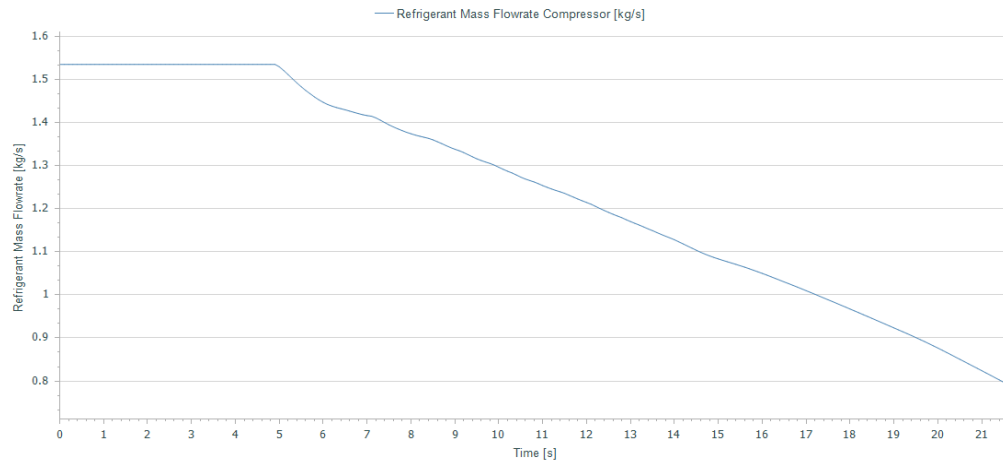
Variation in Air Mass Flowrate over Time



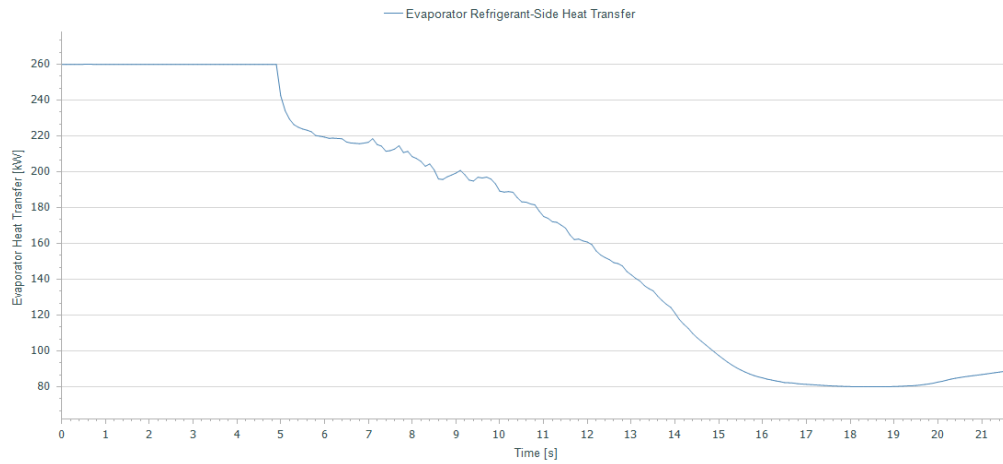
Variation in Compressor Capacity over Time



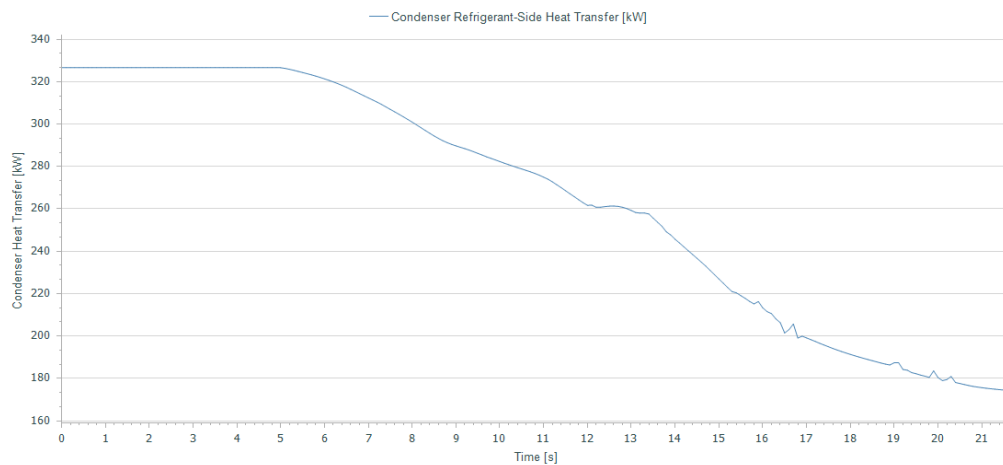
Variation in Refrigerant Mass Flowrate over Time

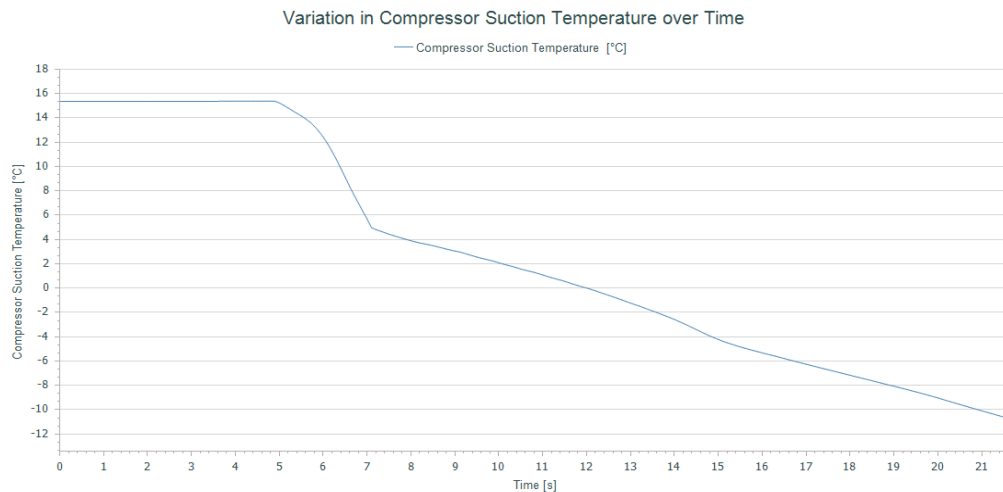
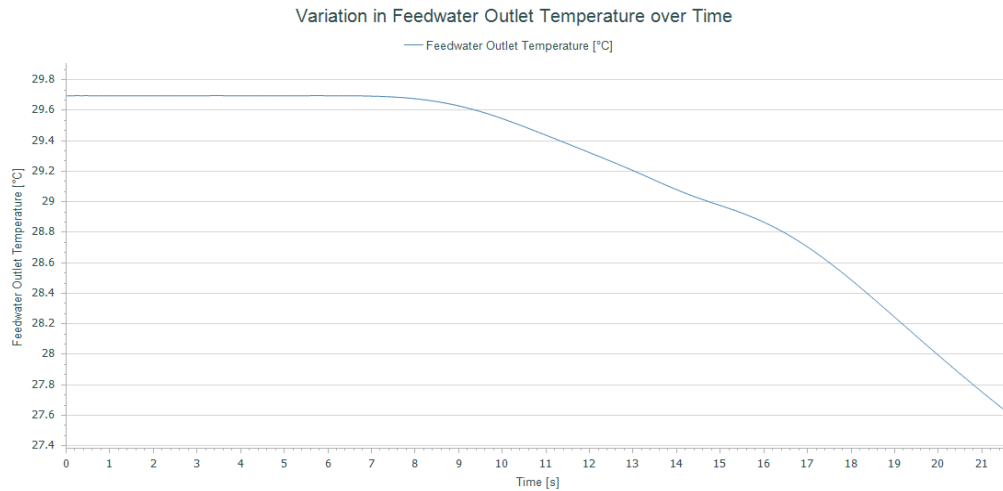
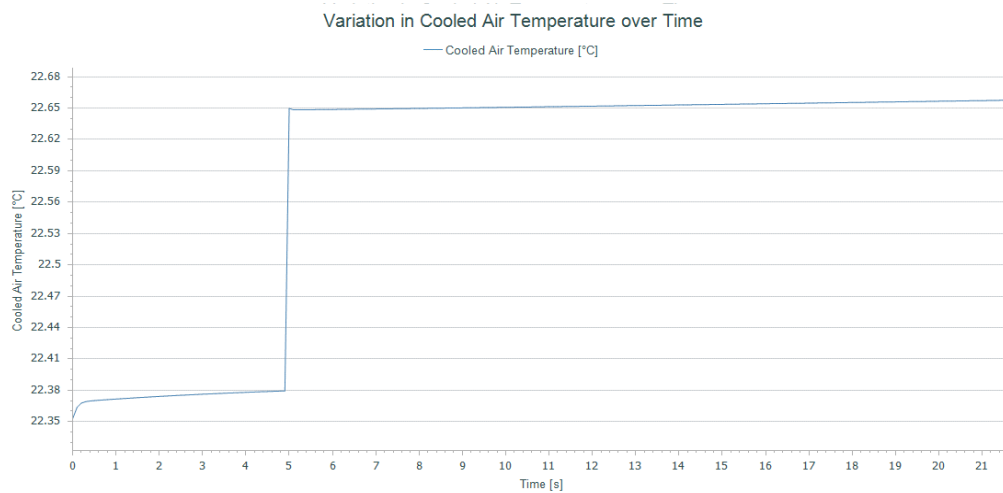


Variation in Evaporator Heat Transfer over Time

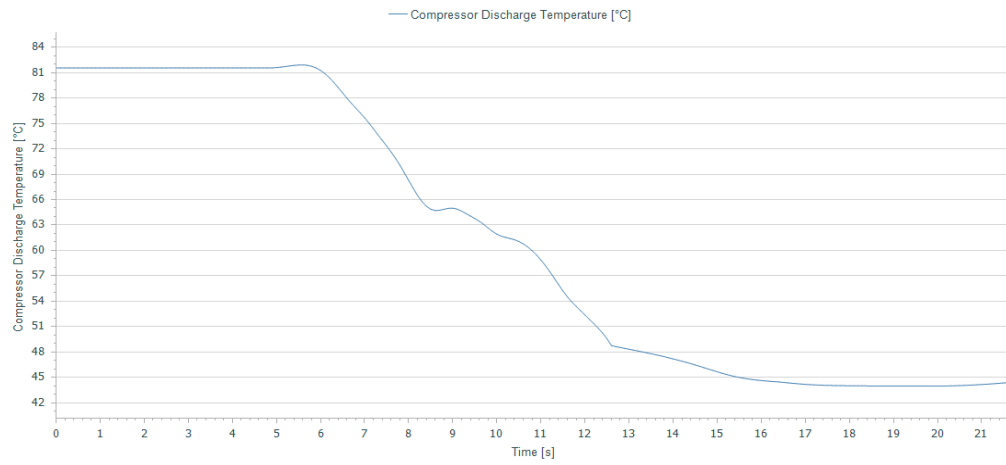


Variation in Condenser Heat Transfer over Time

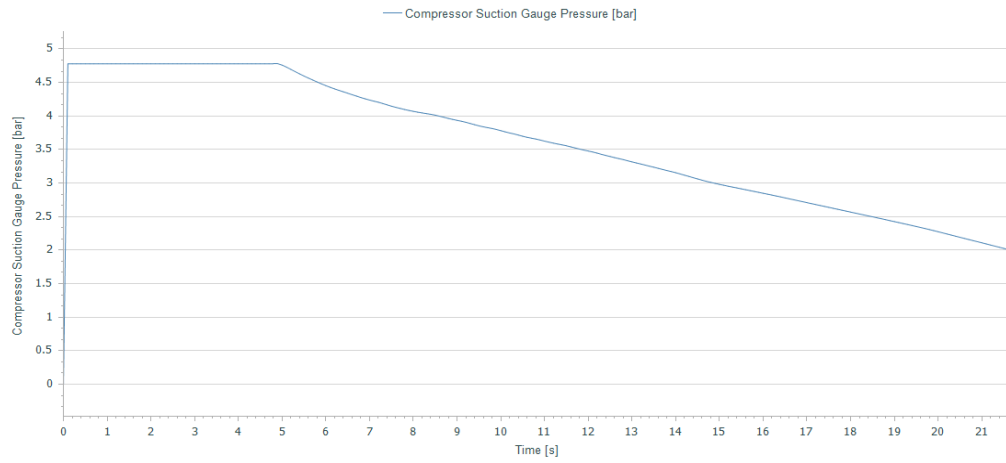




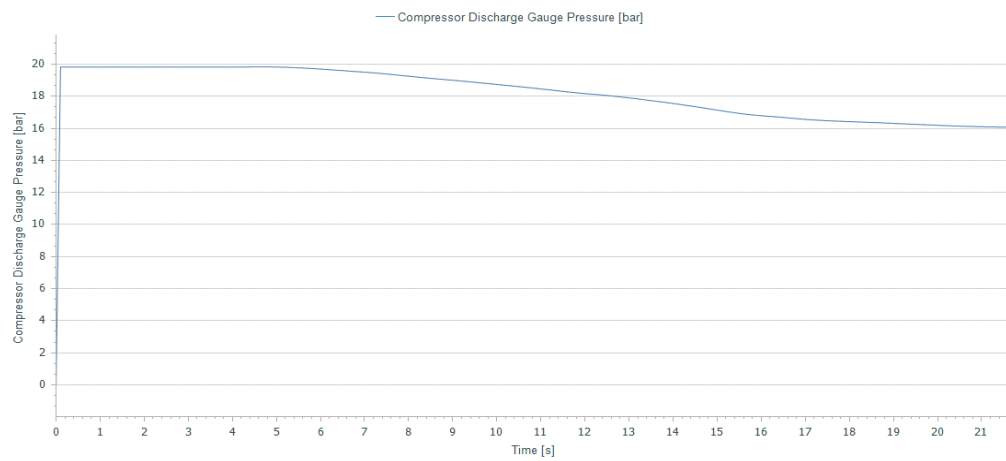
Variation in Compressor Discharge Temperature over Time

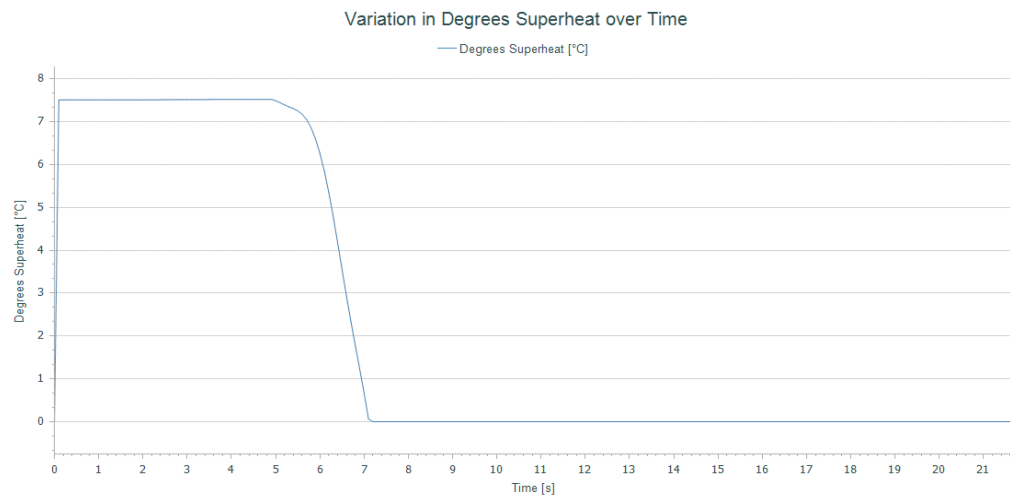
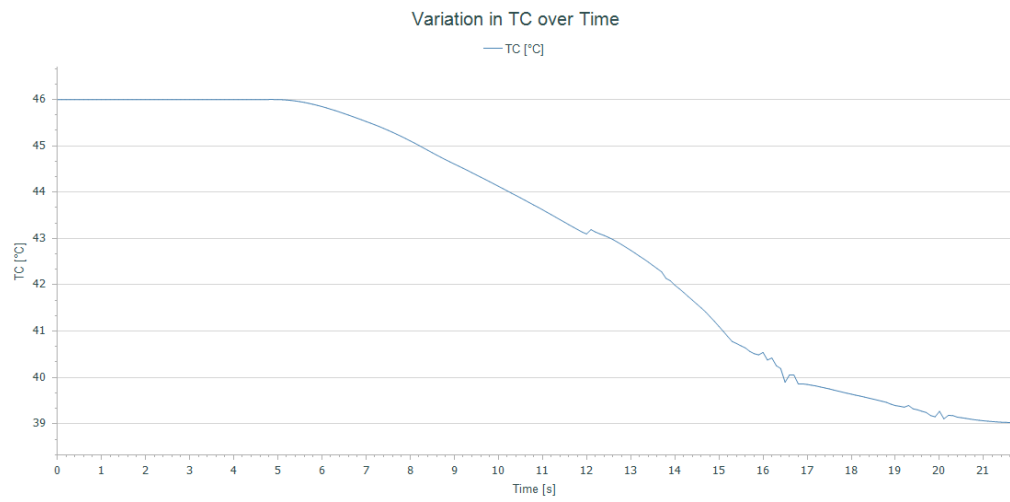
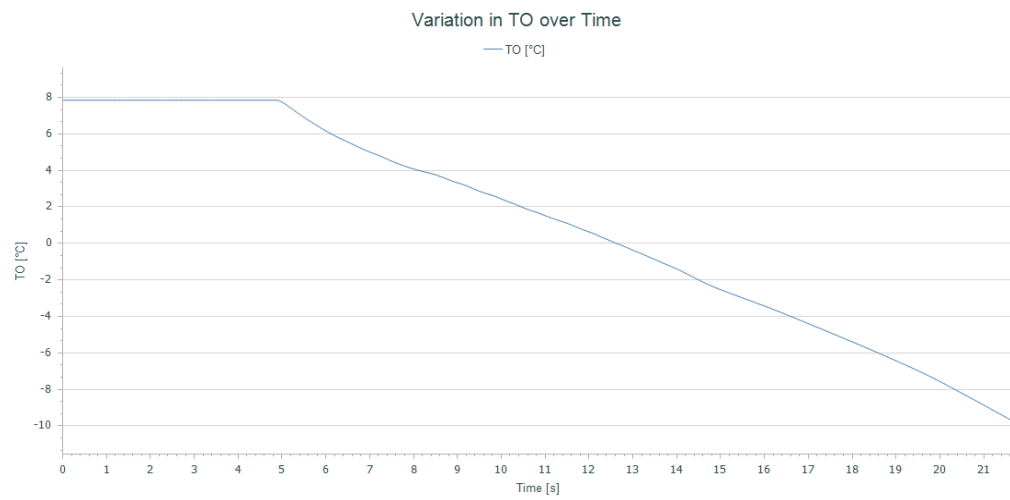


Variation in Compressor Suction Gauge Pressure over Time

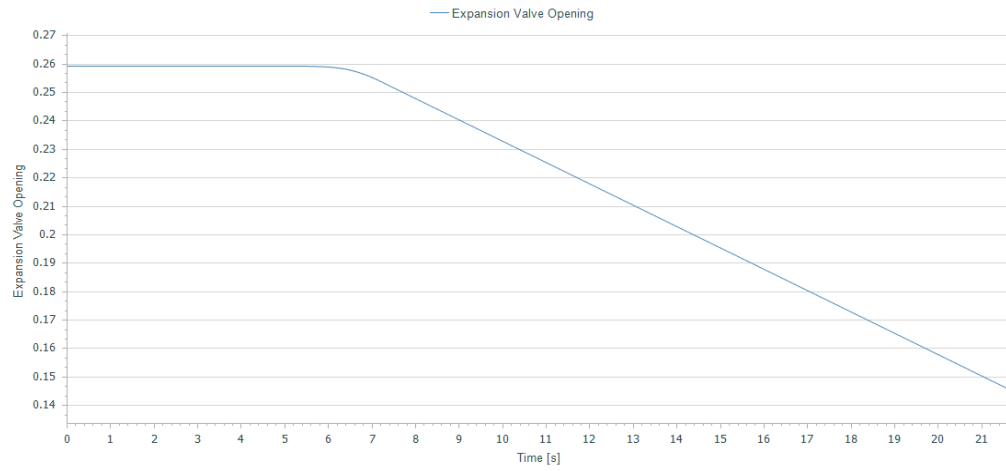


Variation in Compressor Discharge Gauge Pressure over Time

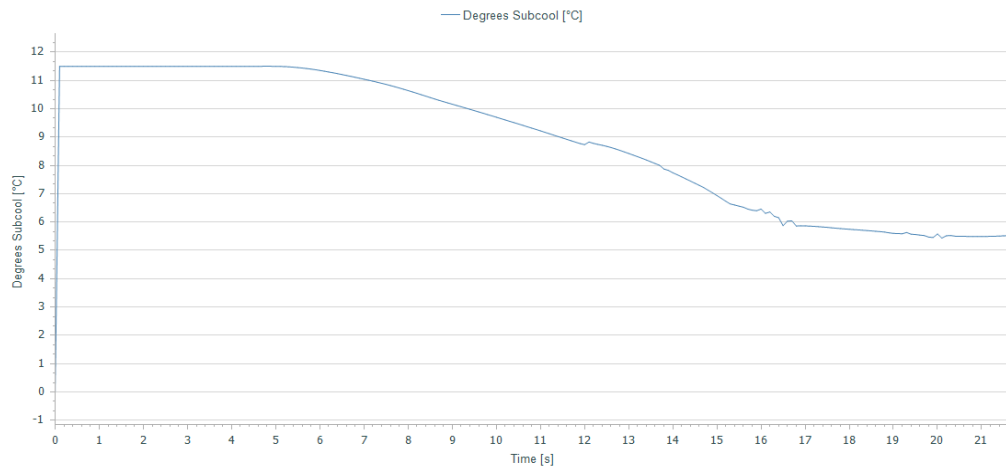




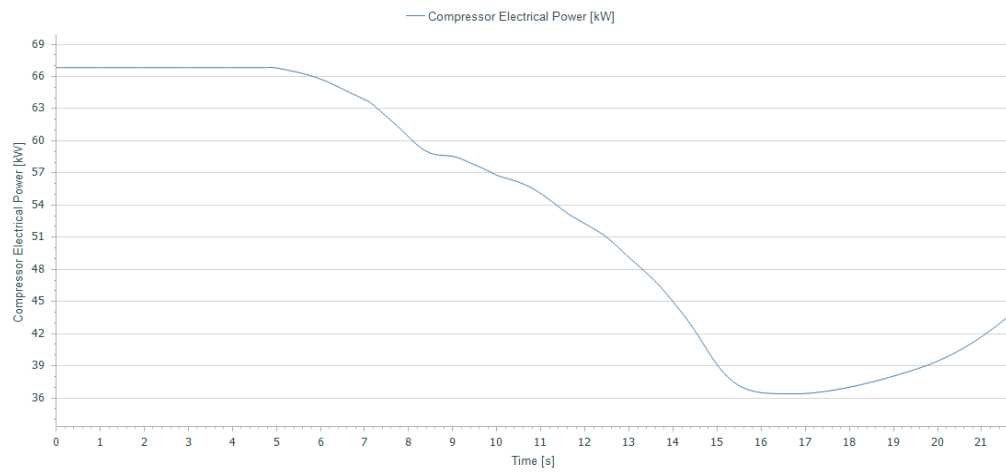
Variation in Expansion Valve Opening over Time



Variation in Degrees Subcool over Time



Variation in Compressor Electrical Power over Time



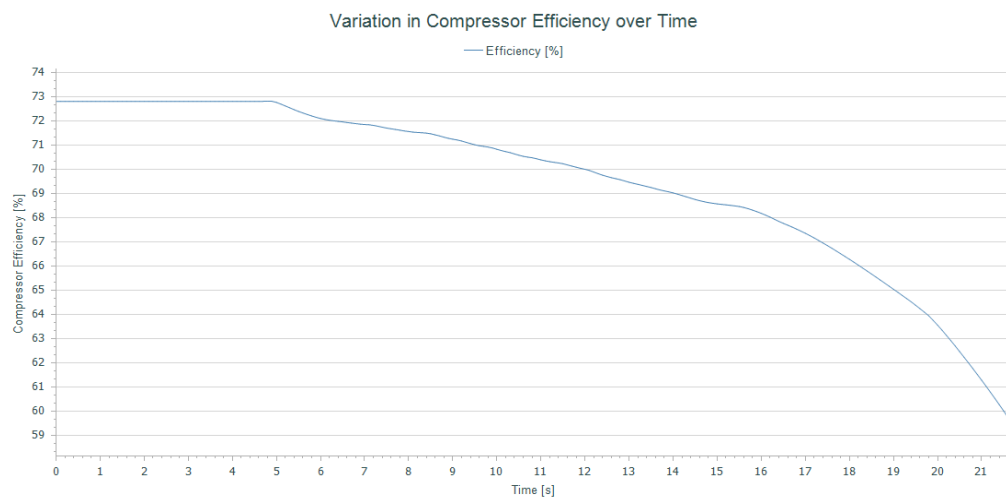


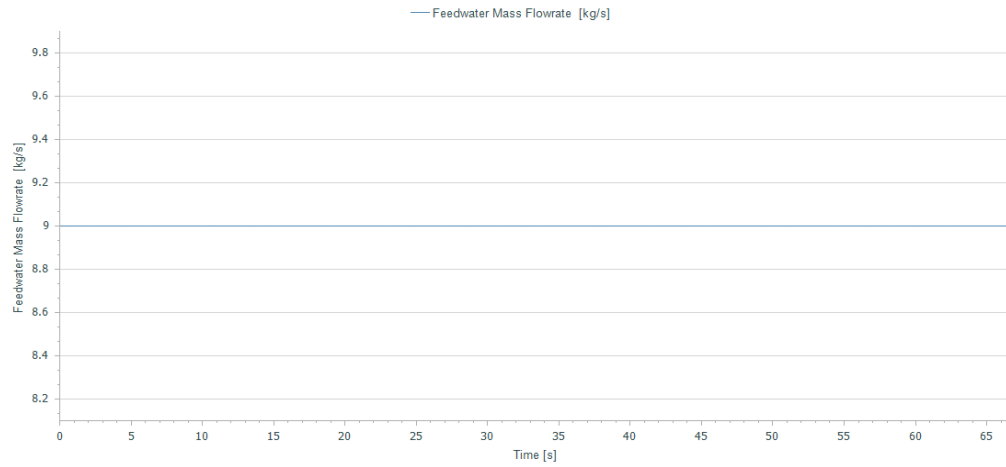
Figure 77: Case 4

C.2.5 Case 5: Blocked-Off Air Supply to Half of the Evaporators

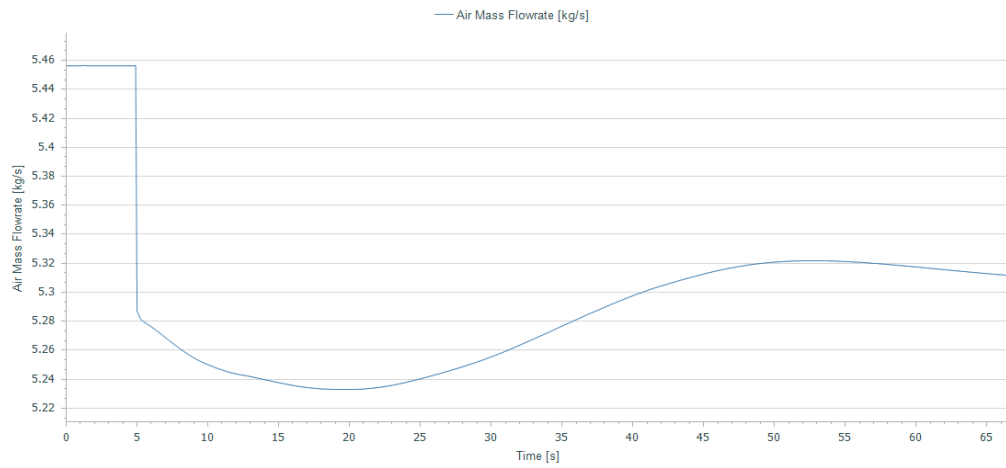
All relevant figures are presented on the following seven pages as Figure 78.

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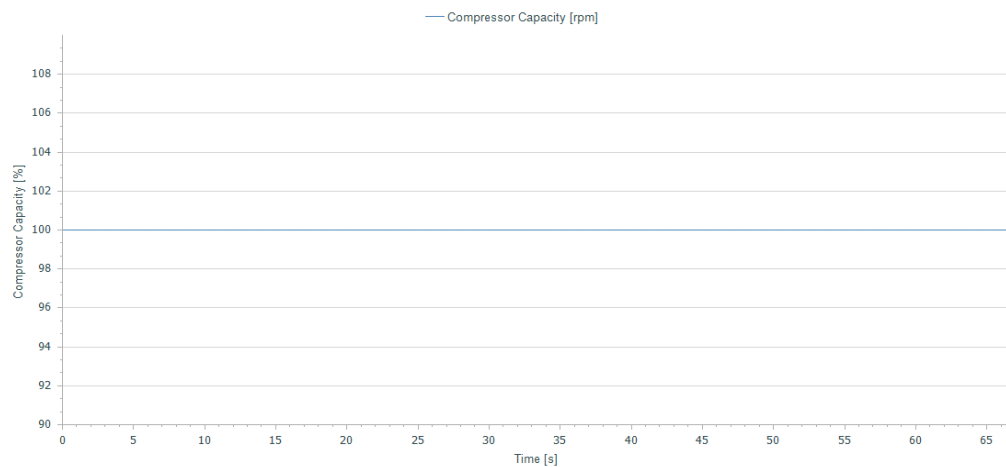
Variation in Feedwater Mass Flowrate over Time



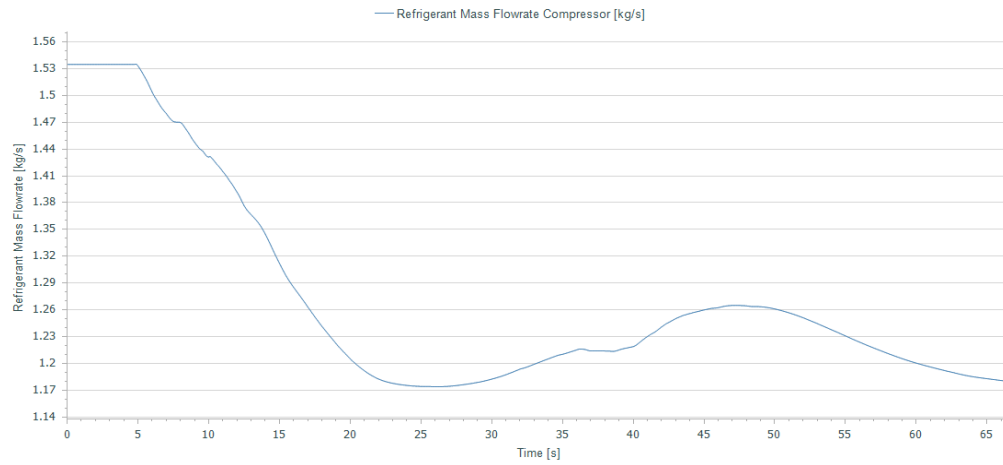
Variation in Air Mass Flowrate over Time



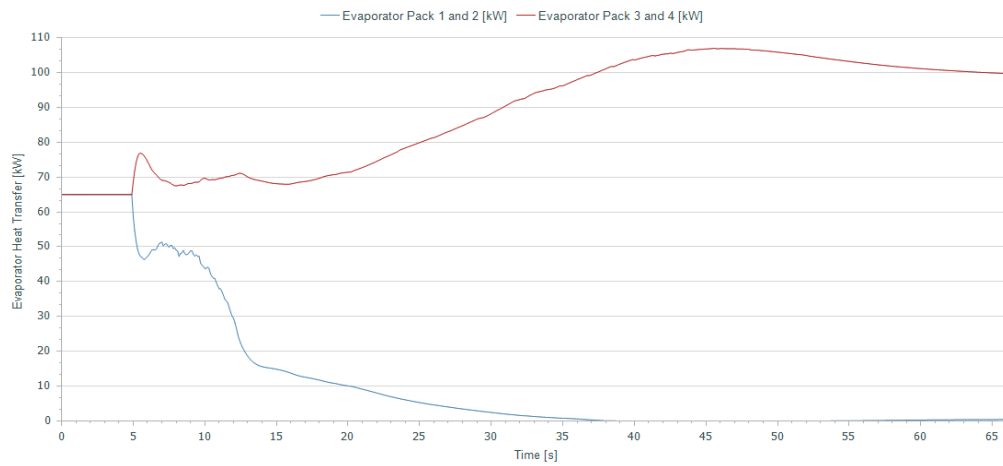
Variation in Compressor Capacity over Time



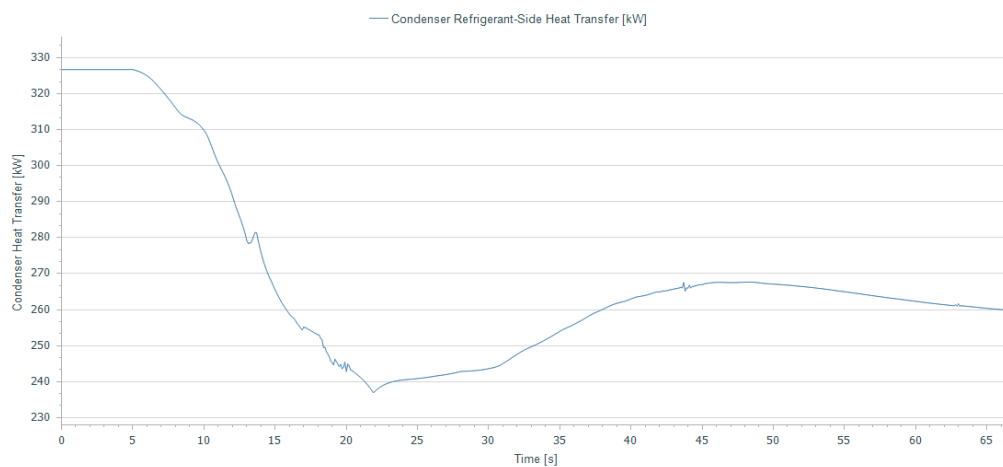
Variation in Refrigerant Mass Flowrate over Time

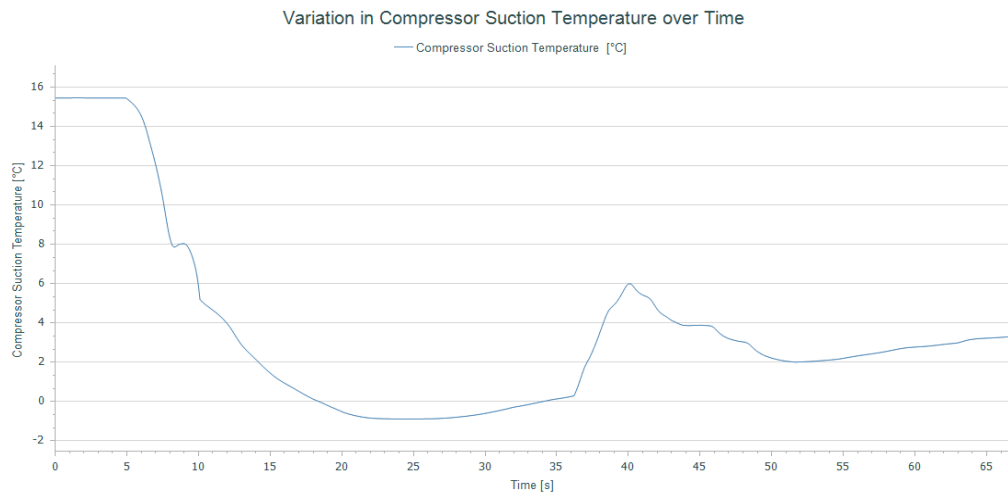
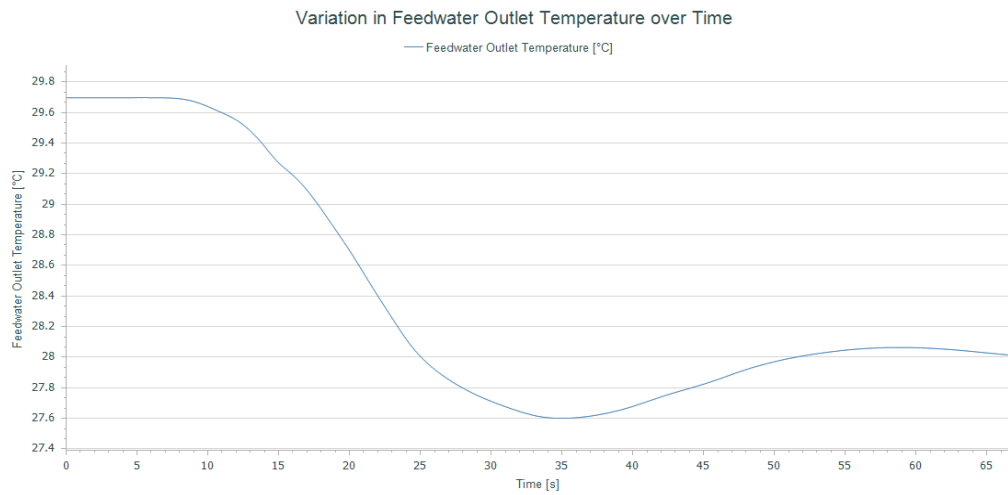
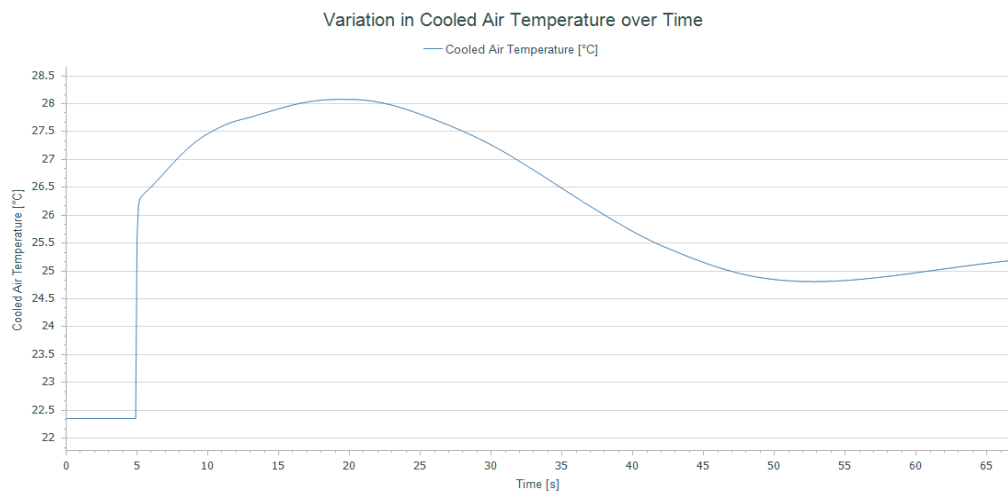


Variation in Evaporator Heat Transfer over Time

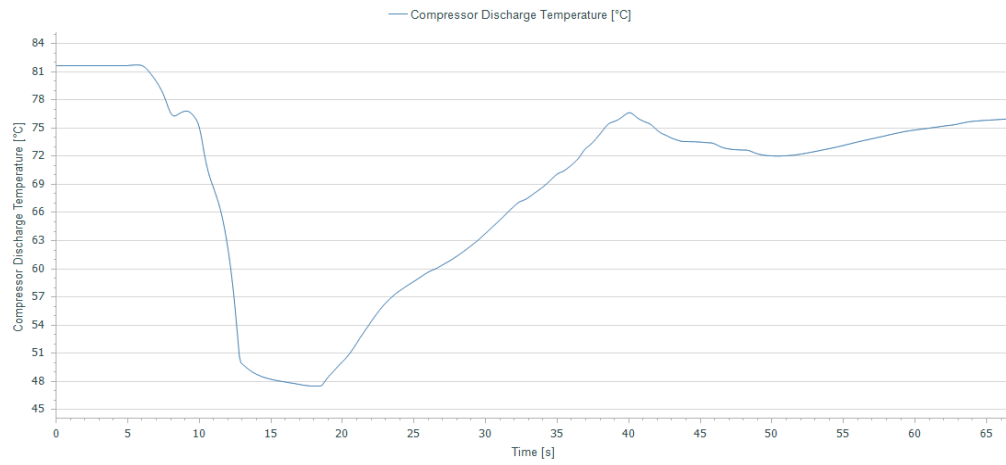


Variation in Condenser Heat Transfer over Time

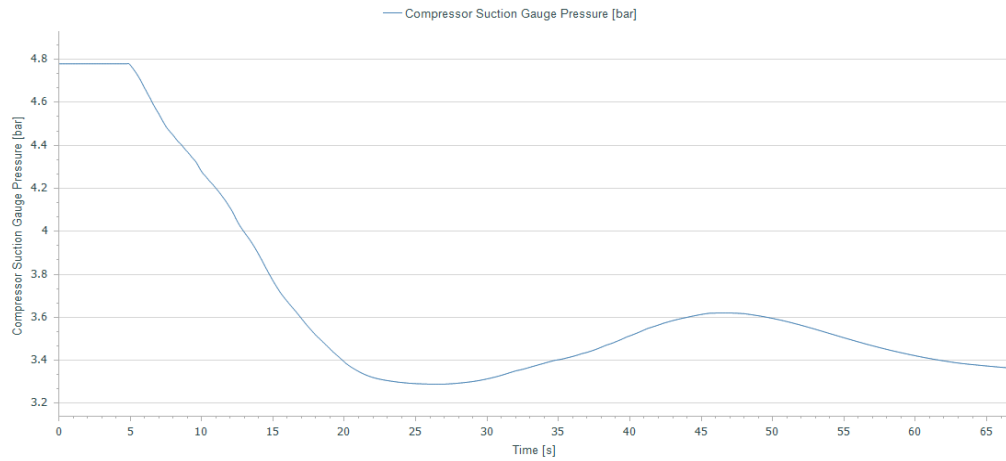




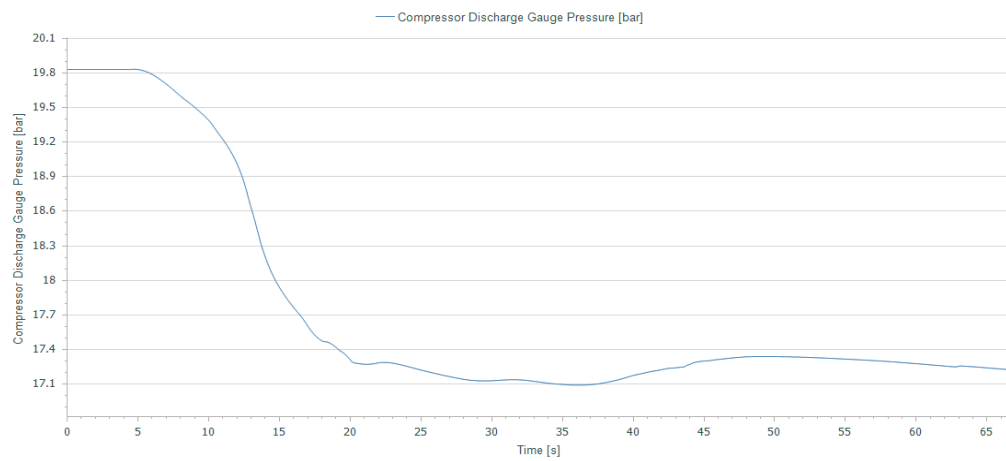
Variation in Compressor Discharge Temperature over Time

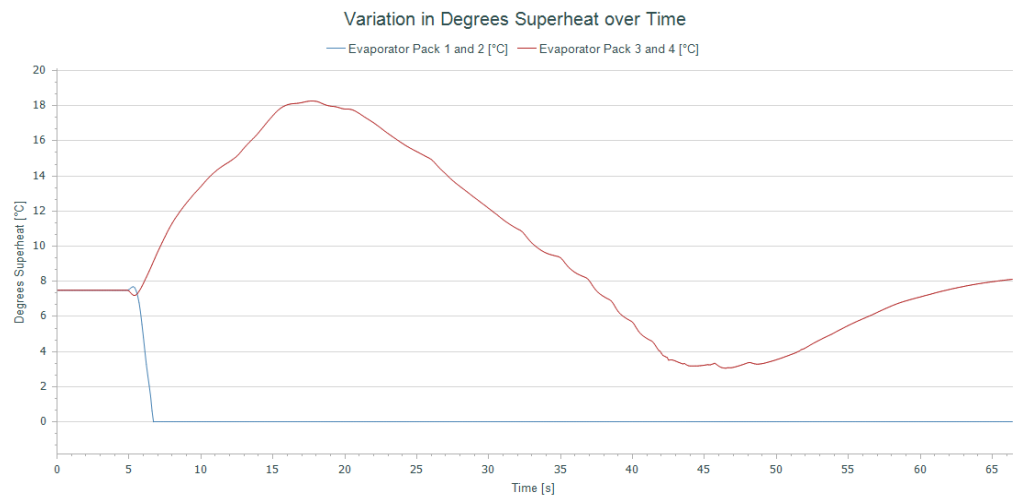
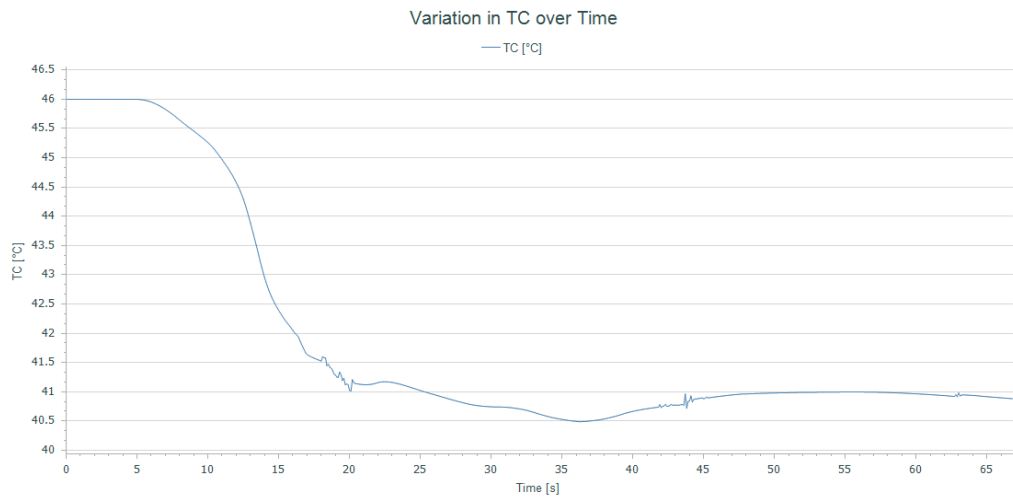
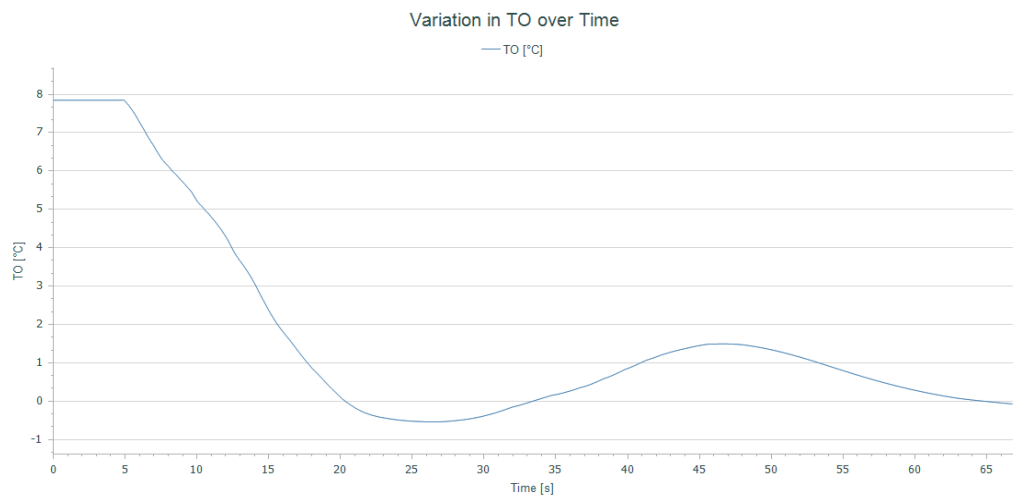


Variation in Compressor Suction Gauge Pressure over Time

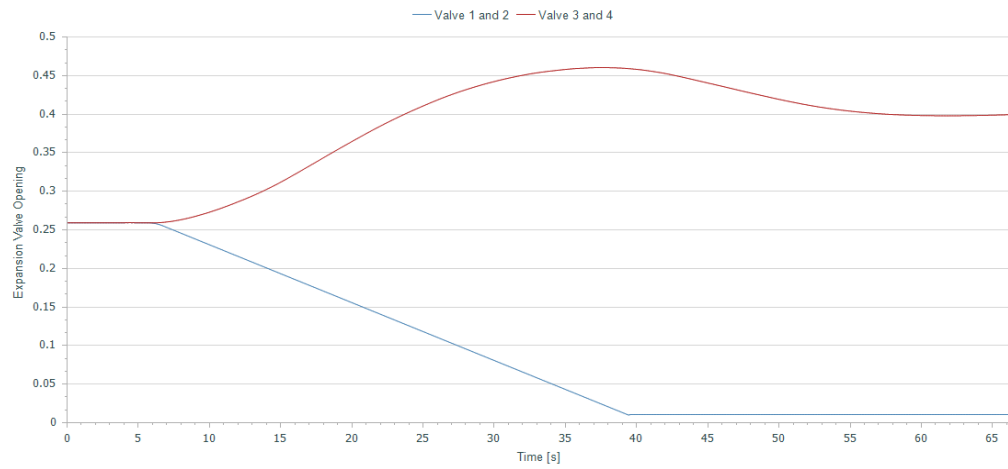


Variation in Compressor Discharge Gauge Pressure over Time

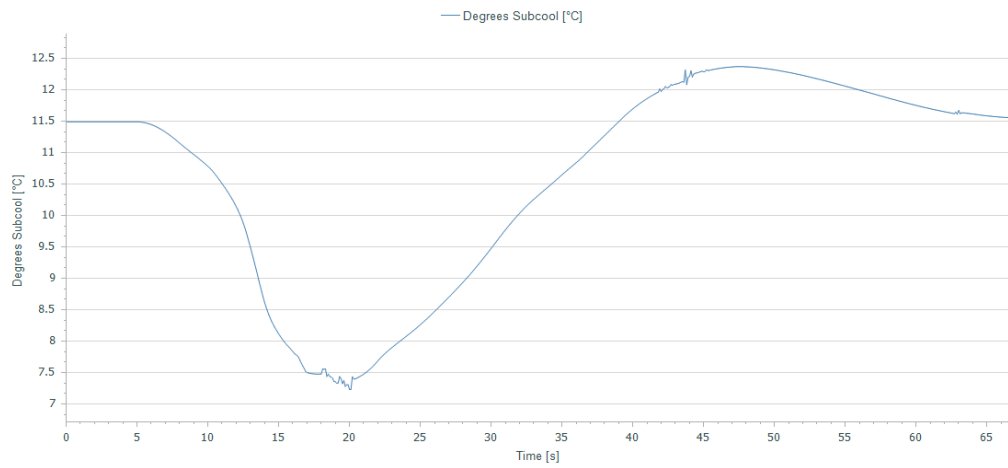




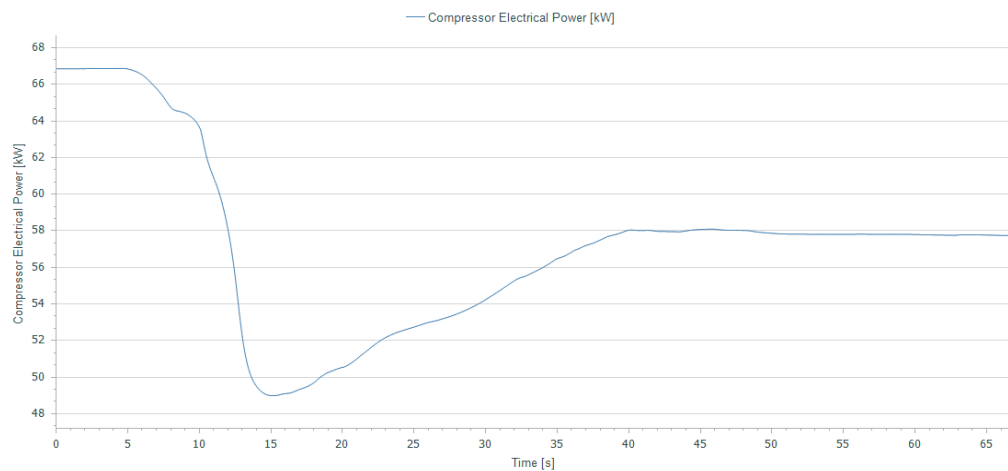
Variation in Expansion Valve Opening over Time



Variation in Degrees Subcool over Time



Variation in Compressor Electrical Power over Time



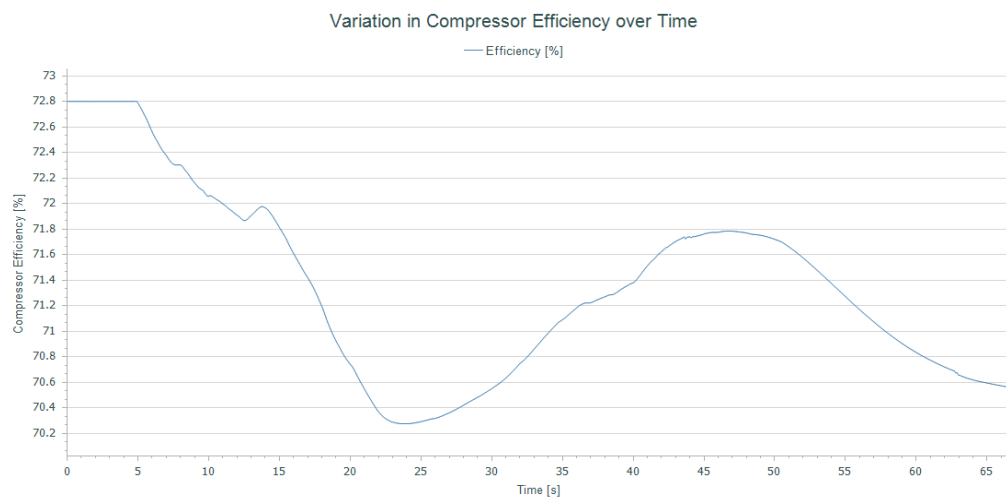


Figure 78: Case 5