Master's Thesis

MODELLING OF A REFRIGERATION PLANT FOR AN ESTABLISHMENT OF A LOAD FORECAST AND OPTIMISATION OF THE PLANT

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NOTE OF THANKS

First, I want to thank my partner and family for their moral support and for respecting my absence during family events while I was writing this thesis. Second, I thank my colleagues and supervisor, who provided knowledge and labouring support to make this thesis possible. Finally, thanks to my company for allowing me to attend this master's programme.

ABSTRACT

Breweries employ numerous heat-giving processes, especially during brewing and fermentation, so most breweries have industrial ammonia refrigeration plants, which work like classic refrigeration units. There, the gaseous ammonia gets compressed, condensed, and distributed to the applications where heat has to be taken out off. Here an expansion valve reduces pressure so the liquid ammonia can absorb the heat from the process fluid to be cooled, evaporating the liquid ammonia. At this point, the ammonia, carrying latent and sensible heat, is sucked back to the compressors, and the cycle begins again.

This thesis analyses the ammonia refrigeration plant of the brewery Göss, one of the largest breweries in Austria, which requires about **20% of electrical energy** and consumes nearly **one quarter** of the brewery's water usage. Currently, rising energy costs necessitate making the refrigeration plant more efficient. Hence, research should investigate the future cooling capacity load of the plant to discover how to make the cooling process operate more efficient. To discover how the system operates, this thesis elaborates on the basic theoretical foundation of refrigeration plants and their thermodynamic fundamental principle.

To create the mentioned forecast, this study examines process recordings and evaluations of the main consumers cooling demand. These selected apparatuses are the external beer cooler, the brew water cooler for cooling the hot wort, which is the beer mixture before the fermentation and the fermentation process itself in two different types of tanks. Using these data, the researcher creates abstracted load forecasts for these consumers to control the delivering pumps of liquid ammonia to the right amount of mass flow.

Subsequently, the process recordings of the fermentation tanks are divided into four different process phases. These stages need various quantities of cooling fluid, and the last stage, the main cooling and draining phase, requires the largest amount of liquid ammonia delivered. Finally, the external beer cooler and brew water cooler are investigated as recurring and evenly ongoing processes so that they are predictable in the cooling load.

In closing, the researched recordings provide an overview and abstracted view of the cooling load required by the selected consumers. Additionally, further investigation could investigate the possibility of controlling distributing pumps to deliver the sufficient amount of refrigerant medium, at the right time, in a more efficient way.

KURZFASSUNG

Brauereien besitzen viele wärmeabgebenden Prozesse, vor allem während des Brauvorganges und der Gärung, sodass die meisten Brauereien industrielle Ammoniak Kälteanlagen, gleich funktionierend wie konventionelle Kälteanlagen, im Einsatz hat. Der Vorgang dabei ist, dass das gasförmige Ammoniak verdichtet, kondensiert und flüssig zu Anwendungen mit notwendiger Kühlung verteilt wird. An der Anwendungsstelle wird das Ammoniak über ein Expansionsventil druckreduziert, sodass dieses die thermische Energie von einem anderen Prozessmedium aufnehmen kann. Durch diesen Wärmeübertragungsprozess verdampft das Kühlmedium und wird wieder über die Saugleitung in die Kompressoren eingesaugt.

Diese Arbeit analysiert die Kälteanlage der Brauerei Göss, eine der größten Brauereien Österreichs, welche über 20 % der elektrischen Energie und fast ein Viertel des Wasserverbrauchs der Brauerei benötigt. In diesen Zeiten ringen steigende Energiekosten dazu, die Kälteanlage effizienter zu machen. Daher sollte diese Untersuchung die zukünftige Kühlungskapazität und -last der Anlage ermitteln, um den Kühlungsprozess zu optimieren. Um zu verstehen, wie das System generell funktioniert, geht diese Arbeit auf das grundlegende Basiswissen von Kälteanlagen und thermodynamischen Hintergrund ein.

Um diese Prognose der Kältelast zu erstellen, werden Daten und Prozessaufnahmen der Hauptkältemittelverbraucher generiert. Diese Aggregate und Prozesse umfassen den externen Bierkühler, den Brauwasserkühler, welcher kaltes Wasser zur Kühlung der Heißwürze bereitet, und den Gärungsprozess selbst in den beiden unterschiedlichen Gärtanktypen. Mit diesen Daten werden abstrahierte Kältelastprognosen für diese Verbraucher erstellt, um die Verteilpumpen des flüssigen Ammoniaks für die richtige Menge am richtigen Ort zu steuern und zu regeln.

Die Aufnahmen der Gärtanks werden dabei in vier Prozessphasen unterteilt. Diese unterscheiden sich im Aufwand der Kältelast, wobei der letzte Schritt – das eigentliche Kühlen und Entleeren des Tanks – am meisten Ammoniak zur Kühlung benötigt. Der externe Bierkühler, sowie der Brauwasserkühler sind als wiederkehrende gleichmäßige Prozesse erkennbar, sodass diese vorhersehbar und prognostizierbar in ihrem Einsatz und der Kältelast sind.

Zusammenfassend bieten die erstellten Prognosen einen Überblick und abstrahierten Ansichtspunkt der notwendigen Kühllasten der gewählten Aggregate. Zusätzlich sollen zukünftige Untersuchungen die Verteilung von flüssigem Ammoniak untersuchen, um die Anlage damit effizienter zu gestalten.

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1 INTRODUCTION

This thesis examines the refrigeration plant of the brewery Göss to increase efficiency, therefore decrease energy consumption and minimise human interaction in the entire cooling system. This brewery is part of the Brauunion Österreich public company, a member of the Heineken group. Additionally, Göss is one of the largest breweries in Austria, with a yearly produced volume of over 1.4 million hectolitres of beer, produced in two brew lines and packaged in cans, bottles and kegs.

The main questions to be asked are:

- how can an ammonia refrigeration plant be modelled, and
- how can the needed distributed cooling medium capacity be forecasted.

The special topology of the system as a network of ammonia refrigerant must be considered, where the most parts of the plant are built up in a central location and distribute then liquid ammonia to all needed places and apparatuses, such as the following:

- glycol cooler for storage tanks,
- external beer cooler,
- brew water cooler,
- fermentation tanks, and
- others in direct cooling with any secondary fluid (direct evaporation), like ceiling room coolers.

Electrical energy consumption is comparatively high for the cooling plant than for the rest of the brewery, as displayed in fig. 1, which lists the brewery's 2021 accumulated electricity consumption. Nearly 20% of the brewery's electrical power of goes to refrigeration, whilst the cooling uses only few machines, related to other departments.

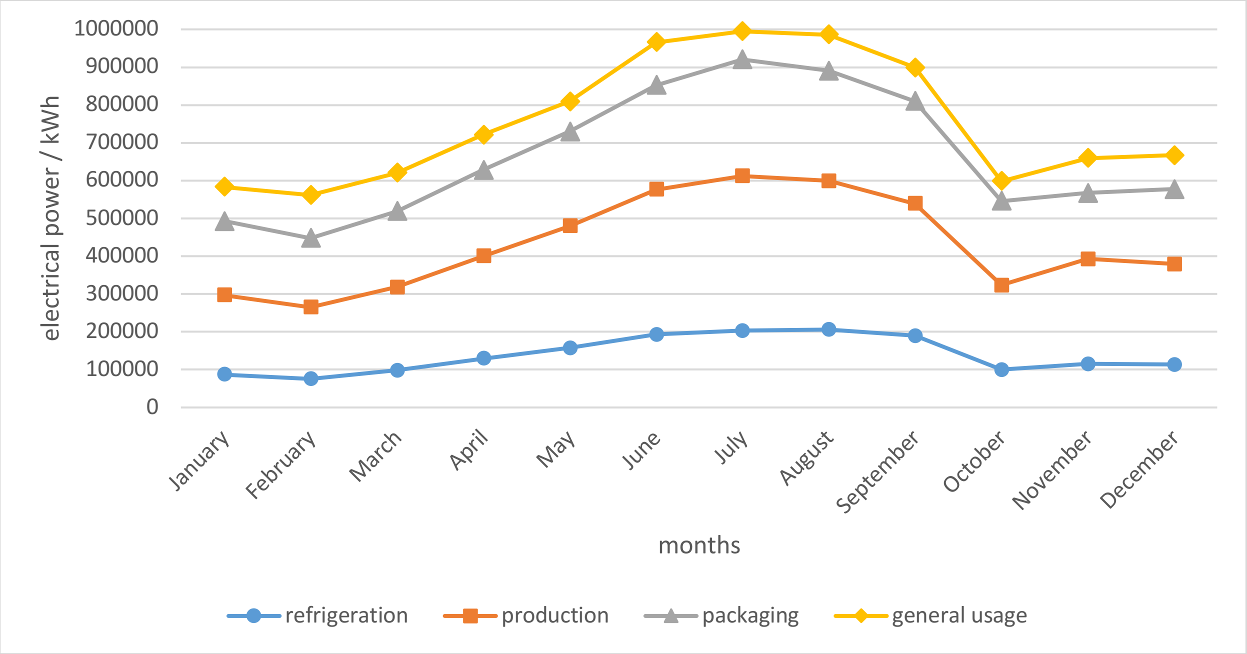

fig. 1: Accumulated energy balance, 2021 per month: electricity; source: own representation.

Introduction

Consuming nearly 40% of the factory's electric energy, production uses more electricity than any other part of the brewery to operate large motors like filtration tank filling pumps or centrifuge drive motors. Next, packaging requires over 30% of the brewery's energy to fill beer in cans, returnable bottles, and kegs.

Beyond the factory's electrical power demand, its water usage is also significant. As fig. 2 shows, the refrigeration plant uses more than one fourth of the brewery's annual water consumption. The process water is also used for cooling, namely of the shell and tube condenser. As the illustration demonstrates, this apparatus is mainly used in the summer months, and the used water gets reused in other processes, to minimise water consumption. Operating the refrigeration plant, especially the condensers with water spray for the evaporation, more efficient, would be directly visible in a lower water demand.

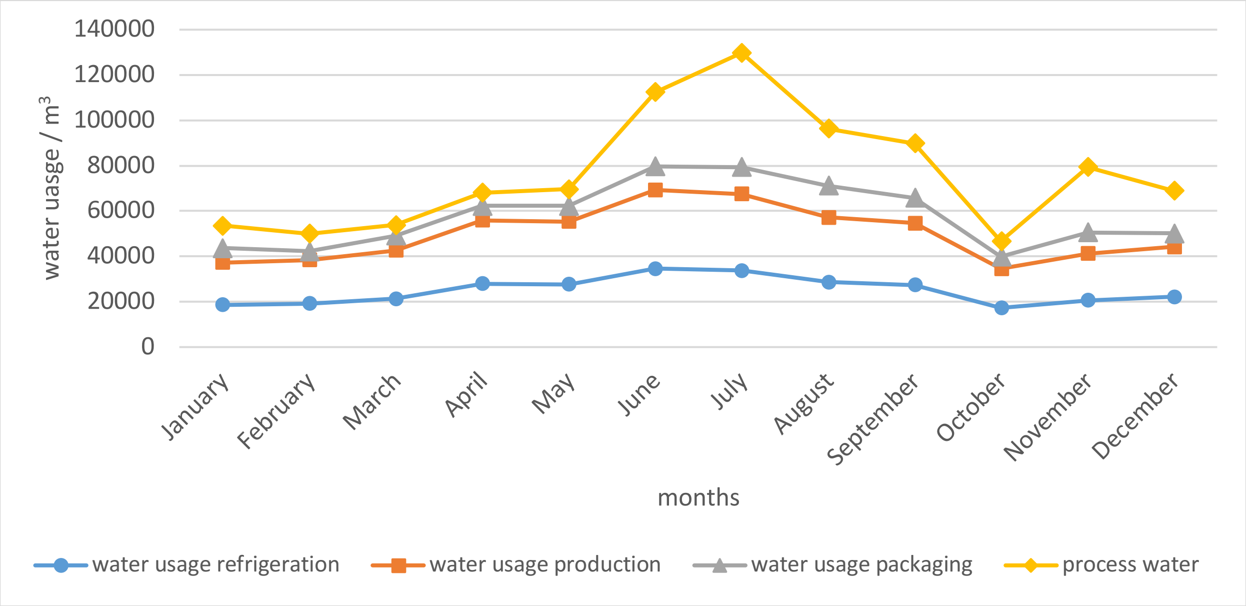

fig. 2: Accumulated energy balance of 2021 per month: water consumption; source: own representation.

The idea behind the modelling is to create a forecast of the refrigeration load using brewing, fermentation, and filling-data to maximise efficiency and produce and deliver sufficient liquid ammonia to cooling needed applications. The current standard to control the brewery's refrigeration plant employs a cascade algorithm and forced circulation pumps in liquid ammonia distribution under manual selection on process demand and season. Beyond this, the fans at the condensers use considerable electrical energy.

This model of the main large cooling consumers should be used in an internal project and as a start point of a future development to create a load forecast of all consumers and the needed amount of cooling capacity. The theoretical part consists out of the basics of thermodynamics, to describe and understand refrigeration, the main operating processes of a refrigeration plant, and a small overview of the plant in the brewery with technical information and simplified piping and instrumentation diagrams. The reference to the practical task also gives the documentation of the plant, but the main objective is the building of relations between brewing, fermentation, and cooling performance demand. Therefore, selected big consumers will be monitored, to get data with mass flows and temperatures of ammonia and process fluid itself (the heat source) over time. In addition to the main brewing and fermentation protocol a mathematical load forecast will be created to feed the plc with the needed functions, that the refrigeration plant can deliver enough coolant liquid ammonia at the right time to right spot and not too much or too less.

2 BASICS OF THERMODYNAMICS AND REFRIGERATION

This chapter describes the basics of thermodynamics and explains the necessary basics of thermodynamics in power and refrigeration units. This exploration targets energy conversion, energy transmission and the main functionality of an ammonia refrigeration plant. When discussing energy, this thesis mainly indicates heat energy, unless otherwise stated.

These general basics comprise the most important aspects of thermodynamics in this thesis, subdivided into following components:

- First law of thermodynamics
- Second law of thermodynamics
- Heat and energy transfer

Thermodynamics is basically the interface between heat, work and inner energy which describe the condition of a system. Additionally, this term describes possible changes within the system and its equilibrium conditions. Furthermore, the next chapters explain the most frequently used state variables for thermodynamics in refrigeration and heat transmission.

Beyond the main three laws of thermodynamics, the zeroth law of thermodynamics, which was afterwards investigated, explains that two systems will always achieve thermal equilibrium if there is no more heat transfer. Thus, a third system in contact with the other two systems would activate energy transfer in form of heat, until all systems reach thermal equilibrium.

In general, all technological cooling procedures perform the extraction of thermal energy from an area and are shown in a circular process. For refrigeration the physical unit of entropy is crucial, for the cooling fluid to get to maximum possible entropy, as describable by the *Gibbs-Energy* or *free enthalpy*:

$$\Delta G = \Delta H - \Delta S * T \qquad (2.1)$$

$$\frac{\Delta G}{T} = \frac{\Delta H}{T} - \Delta S$$

ΔG/J	Gibbs-Energy or free enthalpy
ΔH/J	reaction enthalpy
ΔS/J K^{-1}	entropy
T/K	temperature

2.1 First law of thermodynamics

The first law of thermodynamics is also known as the law of conservation of energy and states the following:

> "Energy cannot be produced nor destroyed, it can only be transferred, or converted from one form to another. In short, energy is conserved."[1]

A simple system, displayed in fig. 3, is used to explain this first law.

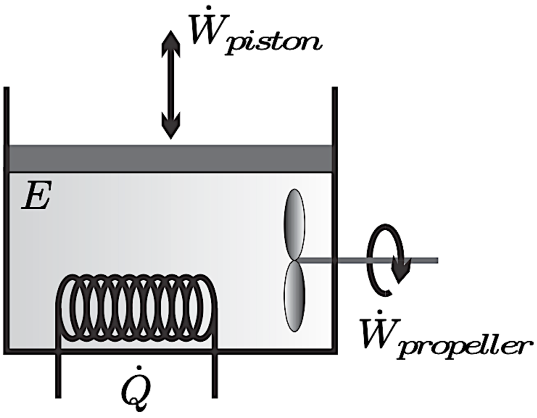

fig. 3: Closed system with energy *E* exchanging $\dot{W}$ and heat $\dot{Q}$ with its surroundings; source: Struchtrup (2014) p. 34. (unmodified).

This study formulates the first law for the simple closed system, indicating all three ways one could manipulate the system from the outside. For this system, the conservation law for energy reads as follows:

$$\frac{\mathrm{d}E}{\mathrm{d}t} = \dot{Q} - \dot{W} \qquad (2.2)$$

$$\dot{W} = \dot{W}_{piston} + \dot{W}_{propeller} \qquad (2.3)$$

E/J	total energy content of system
t/s	time
$\dot{Q}$/J s^{-1}	heat transfer rate
$\dot{W}$/J s^{-1}	total work per time = power (in sum, for propeller and for piston) [2]

Where the total energy of the system; the heat transfer rate of the system; and the total power, the work per unit time, are exchanged with the surroundings. This equation states that the change of the system's energy in time ($\mathrm{d}E/\mathrm{d}t$) is equal to the energy transferred by heat and work per unit time ($\dot{Q} - \dot{W}$). Additionally, the sign convention used shows that heat transferred into the system is positive, and work done by the system is positive. [3]

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[1] Struchtrup (2014) p. 33.

[2] Cf. Struchtrop (2014) p. 34.

[3] Cf. Struchtrop (2014) p. 34.

Following expression is used in process technology to describe the same system as in the mechanical principle, in formula 2.2 with the energy balance:

$$\frac{dH}{dt} = \dot{H}_{in} - \dot{H}_{out} + P_{in} - \dot{Q}_{out} \qquad (2.4)$$

$dH\,dt^{-1}$/J s^{-1}	energy balance of the system per time
$\dot{H}_{in}$/ J s^{-1}	to the system supplied enthalpy per time
$\dot{H}_{out}$/ J s^{-1}	from the system extracted enthalpy per time
P_{in}/W	to the system supplied power
$\dot{Q}_{out}$/ J s^{-1}	from the system extracted thermal energy per time

The expression of the total energy must be explained further, an expression that sums the kinetic, potential, and inner energy. In thermodynamics, thermal energy generally describes inner energy, but other inner energy forms are listed here:

- Chemical energy
- Energy of formation
- Nuclear energy
- Radioactive energy and
- Electrical energy.[4]

For a thermal process, as in the example above, only kinetic energy E_{kin}, potential energy E_{pot} and internal energy U are necessary for the description. Therefore, the total energy of this system is expressed as follows:

$$E = U + E_{kin} + E_{pot} \qquad (2.5)$$

E_{kin}/J	kinetic energy
E_{pot}/J	potential energy
U/J	internal energy[5]

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[4] Cf. Struchtrop (2014) p. 35.

[5] Cf. Struchtrop (2014) p. 35.

According to Bernoulli, the conservation of general mechanical energy reads:

$$const = \int \frac{\rho}{2}\, \vec{c}^2\, dV + \int p\, dV + \int \rho\, \vec{g}\, dV\, z \qquad (2.6)$$

ρ/kg m^{-3}	density
$\vec{c}$/m s^{-1}	velocity
p/Pa	pressure
V/m^3	volume
$\vec{g}$/m s^{-2}	gravitational force
z/m	elevation above reference height

Kinetic energy is the energy of mass in movement with a defined velocity. In a homogeneous physical system kinetic energy is calculated as displayed here:

$$E_{kin} = \frac{m}{2} v^2 \qquad (2.7)$$

m/kg	mass
v/m s^{-1}	velocity[6]

For inhomogeneous systems, kinetic energy is obtained by integrating the specific kinetic energy e_{kin} over all mass elements dm, expressed by density and volume:

$$e_{kin} = \frac{1}{2} v^2 \qquad (2.8)$$

$$E_{kin} = \int \rho\, e_{kin}\, \mathrm{d}V = \int \frac{\rho}{2} v^2\, \mathrm{d}V \qquad (2.9)$$

V/m^3	volume
ρ/kg m^{-3}	density
e_{kin}/J kg^{-1}	specific kinetic energy

Potential energy describes the position of a mass in a system and is given in a homogeneous system in the following equation:

$$E_{pot} = m\, g\, z \qquad (2.10)$$

g/m s^{-2}	natural gravitational acceleration
z/m	elevation above reference height[7]

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[6] Cf. Struchtrop (2014) p. 35.

[7] Cf. Struchtrop (2014) p. 35.

Z describes the elevation of a bodies centre of gravity above a reference height, and $g = 9.81$ m/s^2 is the gravitational acceleration of the Earth. Again, for inhomogeneous states, the potential energy is obtained by integrating the specific potential energy e_{pot} over all mass elements d*m:*

$$e_{pot} = g\,z$$

e_{pot}/J kg^{-1}	specific potential energy

$$E_{pot} = \int \rho\, e_{pot}\,\mathrm{d}V = \int \rho\, g\, z\,\mathrm{d}V \qquad (2.11)$$

The tiniest macroscopic elements of matter move or vibrate quickly, so every atom has microscopic kinetic energy. Additionally, every atom is a subject of interatomic forces, producing microscopic potential energy. Because these microscopic energies cannot be observed macroscopically, the internal energy *U* describes the microscopic forms of kinetic and potential energy of a macroscopic element of matter.

$$U = m\,u = \int \rho\, u\, dV \qquad (2.12)^{8}$$

u/J kg^{-1}	specific internal energy

Thermodynamic systems must combine internal energy, pressure, and volume, necessitating a value for this combination called enthalpy *H*. Total and specific enthalpy can be defined as follows:

$$H = U + p\,V \quad and \quad h = u + p\,v \qquad (2.13)$$

H/J	enthalpy of a system, formerly heat content
p/Pa	pressure
h/J kg^{-1}	specific enthalpy

Now using the enthalpy to replace internal energy to create a correlation between enthalpy, heat energy, volume, and pressure of a substance:

$$\frac{\mathrm{d}H}{\mathrm{d}t} = \dot{Q} + V\frac{\mathrm{d}p}{\mathrm{d}t} \qquad (2.14)$$

$$H_2 - H_1 = Q_{12} + \int_1^2 V\,\mathrm{d}p \qquad (2.15)$$

Finally, this equation can be employed in various calculations in thermodynamics and refrigeration.[9]

Enthalpy is basically the energetic description for mass flow in technical systems with flow.

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[8] Cf. Struchtrop (2014) p. 36.

[9] Cf. Struchtrop (2014) p. 43.

2.2 Second law of thermodynamics

The second law of thermodynamics was formulated because isolated systems tend to reach a unique, stable equilibrium state, as first described by Rudolf Clausius (1822–1888): [10]

Heat can only be transferred from hot to cold, but not vice versa.

Like the first law of thermodynamics, the second law is based on experience, so the latter has numerous other examples and definitions. Lastly, this law directs processes and restricts the transformation of heat and inner energy into work, so a value called entropy *S* is used to describe these processes, as shown below:

$$\mathrm{d}S = \frac{\mathrm{d}Q}{T} + \frac{\mathrm{d}W_{diss}}{T} \qquad (2.16)$$

$$\mathrm{d}S_q = \frac{dQ}{T} \qquad (2.17)$$

$$\mathrm{d}S_{diss} = \frac{dW_{diss}}{T} \qquad (2.18)$$

S/J K^{-1}	entropy
W_{diss}/J	work by dissipation
T/K	temperature
S_q/J K^{-1}	entropy by heat transportation
S_{diss}/J K^{-1}	generation of entropy by dissipation

Since all natural processes belong to dissipation, it applies that $\mathrm{d}S_{diss} > 0$. [11]

The second law of thermodynamics also describes the following condition changes for ideal gases:

- Isochore change of condition: volume constant
- Isobar change of condition: pressure constant
- Isothermal change of condition: temperature constant
- Isentropic change of condition: entropy constant
- Polytropic change of condition: pressure times (volume to the power of *n*) constant

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[10] Cf. Struchtrop (2014) p. 55.

[11] Cf. Cerbe, Wlhelms (2021) pp. 95-97.

Nearly all technical processes are adiabatic processes, meaning that condition shift is too swift for the heat being transferred. Therefore, the isentropic view is generally selected to calculate and describe reversible technical processes. This isentropic change of condition assumes consistent entropy, during the physicals system condition change.

In real situations the isentropic change displays also the condition, that no thermal energy is transferred in or out of the system. For example, a compression or expansion is operated in a way, that the heat “loss” is a maximum in range of dissipation.

2.3 Heat and energy transmission

Due to differences in temperature and energy, heat forms are **transferred between bodies** and **not exchanged**, as described in section 2.2. Notably, heat transfer involves the restriction of the direction, heat energy can be transferred in difference to other energy form transmission. That is, heat flows only in response to a temperature change in a system and the system boundaries, as first mentioned by Fourier, where positive heat flux needs a negative gradient.

The main heat transfer mechanisms are the following:

- Conduction,
- Convection, and
- Radiative transfer.[12]

2.3.1 Conduction

The law of Fourier states that heat conduction through a homogenous, flat wall with the wall surface *A*, a wall thickness *s* and a proportional factor λ follow an indirect linear negative temperature gradient, as shown in fig. 4. [13]

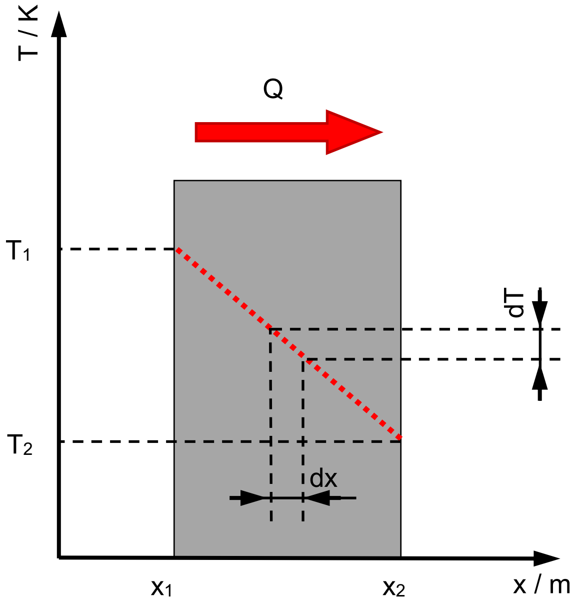

fig. 4: Stationary heat conduction through homogenous flat wall; source: Cerbe; Wilhelms (2021) p. 406. (modified).

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[12] Struchtrup (2014) pp. 39-40.

[13] Cf. Cerbe; Willhelms (2021) pp. 406-407.

The equation to describe a system like the one pictured in fig. 4, is as follows:

$$Q = -\frac{\lambda}{\Delta \mathrm{x}} A\,(\mathrm{T}_1 - T_2)t \quad with \quad \Delta x = x_1 - x_2 \qquad (2.19)$$

λ/W m^{-1} K^{-1}	heat conduction coefficient
Δx/m	length of heat conduction process
A/m^2	surface of heat conduction
x/m	position in coordinate system

With T_1 higher than T_2 and the temperature-dependent material value λ, one can calculate heat conduction. Looking at a thin slice of wall, the heat transfer would be the same as at the whole wall, and only temperature difference and thickness would change. However, since heat transfer should be positive, a negative sign is added to the factor dT/dx, demonstrating the final law of Fourier, which reads as follows:

$$Q = -\lambda\, A\, \frac{dT}{dx}\, t \qquad (2.20)$$

$$\dot{Q} = \frac{Q}{t} = -\lambda\, A\, \frac{dT}{dx} \qquad (2.21)$$[14]

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[14] Cf. Cerbe; Willhelms (2021) pp. 406-407.

2.3.2 Convection

Heat convection describes the heat transfer from fluids and gases to a solid surface. As in conduction, a proportional factor (in this case α) describes the amount of heat transfer. As can be seen in fig. 5, the microscopic profile of temperature is nonlinear.

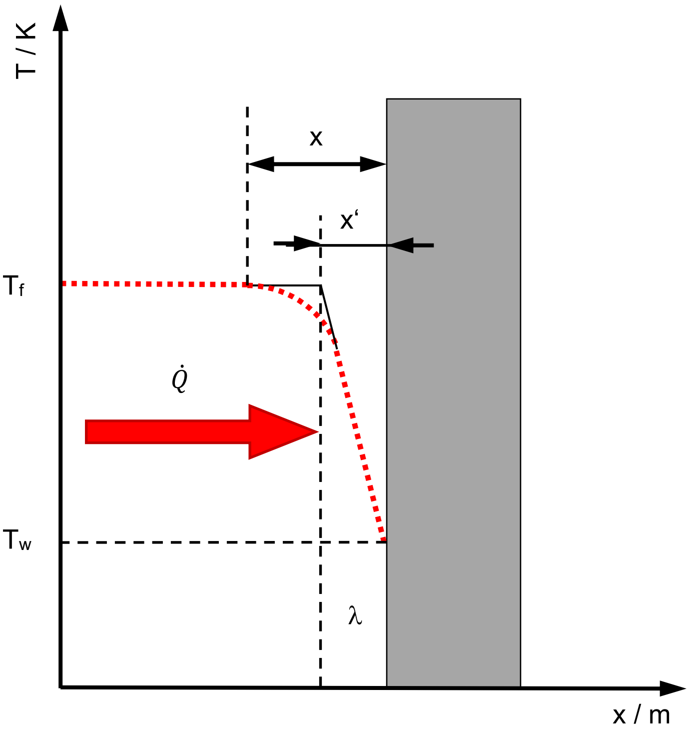

fig. 5: Heat convection between a fluid and a solid wall; source: Cerbe; Wilhelms (2021) p.415 (modified).

Therefore, Newton determined the following correlation, where T_f is the temperature of the fluid or gas, and T_w is the temperature of the wall:

$$\dot{Q} = \alpha A \left(T_f - T_w\right) \qquad (2.22)$$

α/W m^{-2} K^{-1}	heat convection coefficient
x/m	length of heat convection process (in total)
x'/m	length of the boundary layer
T_f/K	temperature of fluid
T_w/K	temperature of wall surface[15]

This coefficient α depends on numerous factors, which describe the physical characteristics, the flow condition, and the geometrical form of the heat transfer areas and is a pseudo model coefficient including

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[15] Cf. Cerbe; Willhelms (2021) p. 415.

all other parameters of influence. In a simplified point of view, if the fluid or gas is stationary against a flat wall, the heat transfer factor λ and thickness *s'* can modify the previous equation into the following equation:

$$with\ \alpha = \frac{\lambda}{s'} \tag{2.23}$$

$$\dot{Q} = \frac{\lambda}{s'}\, A\left(T_f - T_w\right) \tag{2.24}$$

The heat transfer factor α can be determined using differential equations for simple systems. Because not every system is simple, mostly practical experiments are used ascertain it empirical. Additionally, similar physical systems display correlations between certain factors, and are thus summarised to less factors to ease calculations. This factor is called the Nußelt coefficient, calculated on different functions, depending on the physical system and between the following coefficients:

- Reynold coefficient *Re*
- Péclet coefficient *Pe*
- Prandtl coefficient *Pr*
- Grashof coefficient *Gr*
- Rayleigh coefficient *Ra*[16]

For example, the mean Nußelt coefficient Nu_m for a shell and tube heat exchanger with deflector plates and transverse flow, inclusive the effects of viscosity in the boundary layer, is calculated as follows:

$$Nu_m = C\ Re^{0.6}\ Pr^{0.6}\left(\frac{\eta_f}{\eta_w}\right)^{0.14} \tag{2.25}$$

Nu/1	Nußelt coefficient
C/1	constant (0.22 for solid casing pipe and *C* = 0.25 for drilled casing pipe)
Pr/1	Prandtl coefficient (0.5 to 500)
Re/1	Reynolds coefficient (4 to 50000)
η_f/Pa s	viscosity of fluid at T_f
η_w/Pa s	viscosity of fluid at T_w[17]

With the characteristic length, the Nußelt and the heat transfer coefficient, following correlation is valid:

$$Nu = \frac{\alpha\, l}{\lambda} \tag{2.26}$$

l/m	characteristic length[18]

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[16] Cf. Cerbe; Willhelms (2021) pp. 416-417.

[17] Cf. Cerbe; Willhelms (2021) pp. 418-419.

[18] Cf. Cerbe; Willhelms (2021) p. 416.

2.3.3 Heat transfer including change of phase

Including change in phase in heat transfer, as a gas or vapour touching a wall tempered under the condensation point, it will condensate, even if the mean gas temperature is above the saturation temperature. Next, the condensate drains as a liquid and drops on the wall surface. To explain the dependence of α on various parameters, fig. 6. provides a diagram of the heat-load dependence. When a fluid is evaporated to its boiling point in a vessel using a low heating surface load due to buoyancy, the heated fluid rises to the top and evaporates mainly on the surface, which rarely develops vapour bubbles. If one increases the heating surface load, vapour bubbles will appear on the liquid's surface and rise to the top. If it has the condition of convection the fluid, the heat transfer coefficient α will rise. At the point of the critical heating surface load, the coefficient will decrease due to the development of a vapour film between the heating surface and the liquid, producing heat resistance, as shown in the figure.[19]

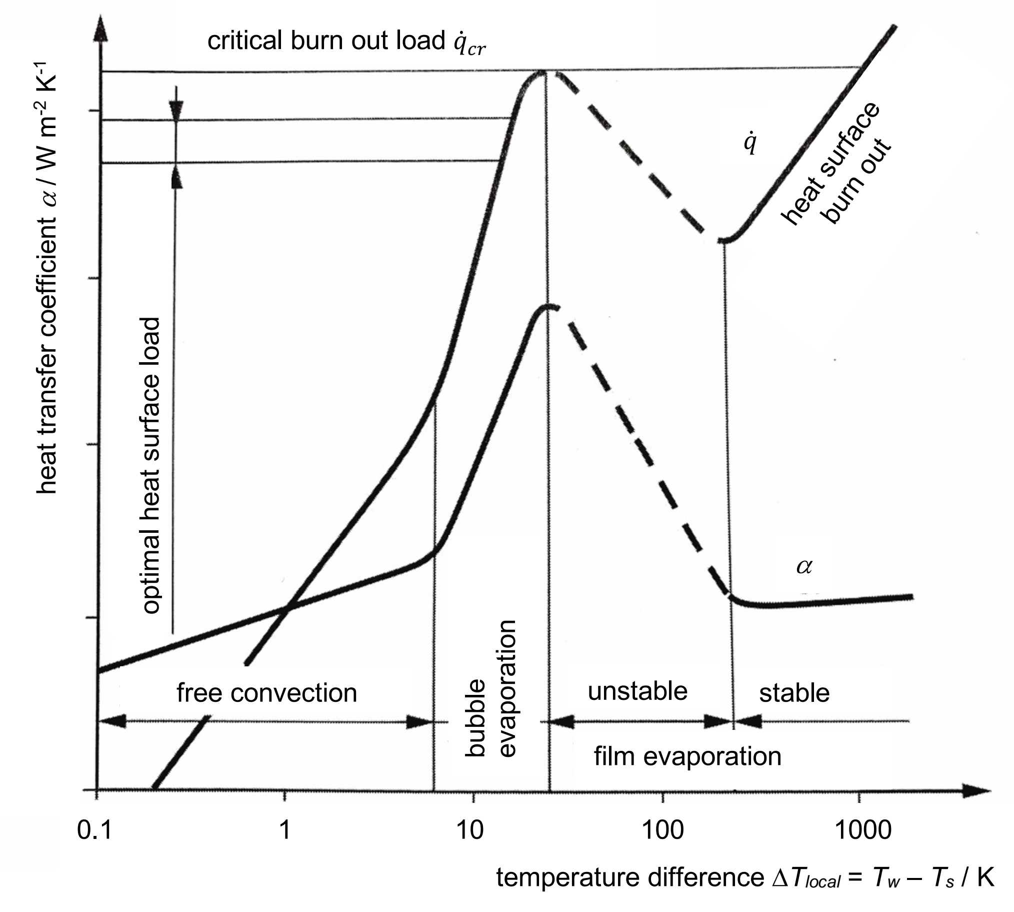

fig. 6: Heat transfer coefficient and heat plate strain of water evaporation at 0.981 bar; source: Cerbe, Wilhelms (2021) p. 430. (modified).

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[19] Cf. Cerbe; Willhelms (2021) pp. 427-430.

2.3.4 Radiative transfer

The radiation which depends on temperature of the body is called heat radiation. It increases as the temperature rises, and solid, liquid and some gaseous substances can both emit and receive this radiation, as described by Boltzmann as follows:

$$\dot{Q} = \frac{\partial Q}{\partial t} * \sigma * A * T^4 \qquad (2.27)$$

$\dot{Q}$/W	heat flow
σ/W m^2 K^4	Stefan Boltzmann number
A/m^2	surface of body
T/K	temperature of body

Beside the temperature of a body also other causes can create radiation. For example, fast moving electrons which are hitting a body cause X-rays. The movement and position of the individual photons cannot be described, so the description as a wave is used for describing such processes. In general, it is the emission of electromagnetic radiation. [20]

In fact, there is no difference between heat radiation and light radiation, except the difference between wavelengths. When radiation hits a surface, the surface can reflect, absorb, or transmit it, as displayed in fig. 7. These three types of radiation added up are equivalent to the incident radiation. Liquids and solids can receive radiation on the surface, gases in the volume.[21]

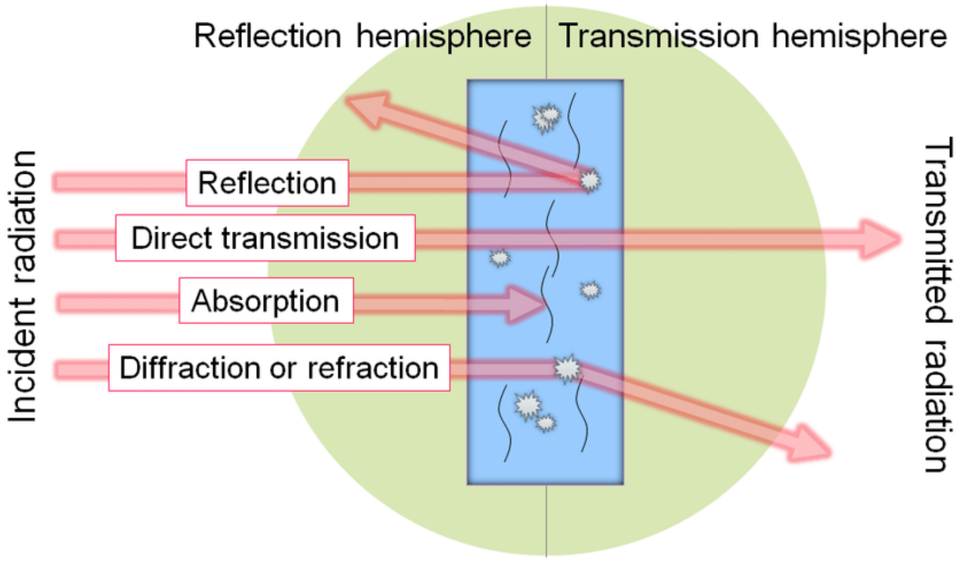

fig. 7: Heat radiation types; source: Hyll (2016) p. 19. (unmodified).

The radiative transfer is not relevant for this thesis, so this concept is not explained further in this work.

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[20] Cf. Cerbe; Wilhelms (2021) p. 431.

[21] Cf. Breidenbach (1990) pp. 79-80.

2.3.5 Heat transition

Heat transition means that the heat must be transmitted from one fluid through two or more different walls to another colder fluid following a temperature profile scheme, as pictured with the red dotted line in fig. 8. The coefficients depend on the used materials and substances the walls consist of, the type of liquids, local Reynolds numbers, local phase changes and the flow conditions. For example, walls consisting out of steel transmit heat better than walls out of concrete.

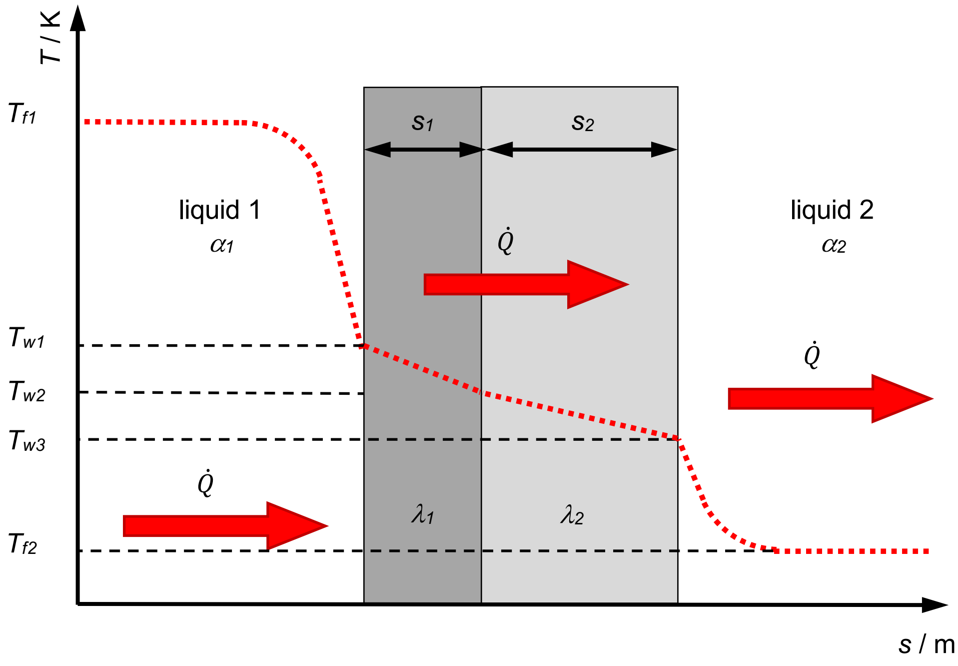

fig. 8: Heat transition, temperature profile between two liquids through two solid walls; source: Cerbe; Wilherlms (2021) p. 439. (modified).

The heat transfer between the two fluids, from the illustrated example, begins with convection between the first fluid and the first wall and then follows a conduction through the two materials, finishing with another convection between the last wall and the second fluid. To calculate now the heat transfer of such a system, one does not calculate individual operations but instead analyses the system's entire process. Therefore, the following equation describes the heat energy transfer at a flat surface with a system coefficient *k*, which describes all other coefficients required:

$$\dot{Q} = k\,A\,(T_{f1} - T_{f2}) \qquad (2.28)$$

k/W m^{-2} K^{-1}	heat transition coefficient
A/m^2	heat transition surface
T_{fx}/K	temperature of fluid x[22]

---

[22] Cf. Cerbe; Wilhelms (2021) p. 439.

To derive the system coefficient *k*, one must **add up the heat transition resistances** through the following equations:

$$R_t = \frac{T_{f1} - T_{f2}}{\dot{Q}} = \frac{1}{k\,A} \qquad (2.29)$$

$$R_t = R_{c1} + \left(\sum_{i=1}^{n} R_i\right) + R_{c2} \qquad (2.30)$$

R_t/K W^{-1}	heat transition resistance of overall system
R_{c1}/K W^{-1}	heat convection resistance between liquid 1 and wall 1
R_i/K W^{-1}	sum of heat conduction resistance of walls 1 to *n*
R_{c2}/K W^{-1}	heat convection resistance between last wall *n* and liquid 2[23]

Like electrical resistances, heat resistances are set by systems against heat transfer and calculated as follows:

$$R_t = \frac{1}{k\,A} \qquad (2.31)$$

$$R_i = \frac{s}{\lambda\,A} \qquad (2.32)$$

$$R_c = \frac{1}{\alpha\,A} \qquad (2.33)$$

Now the heat resistances and the coefficient relation can be written as shown below:

$$\frac{1}{k\,A} = \frac{1}{\alpha_1\,A} + \left(\sum_{i=1}^{n} \frac{\delta}{\lambda_i\,A}\right) + \frac{1}{\alpha_2\,A} \qquad (2.34)$$

$$k = \frac{1}{\frac{1}{\alpha_1} + \left(\sum_{i=1}^{n} \frac{\delta}{\lambda_i}\right) + \frac{1}{\alpha_2}} \qquad (2.35)$$

Notably, this equation is only valid for flat surfaces.[24] In other cases the surface *A* must be corrected or an integrative summation on the pathway of the heat flow must be elaborated.

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[23] Cf. Cerbe; Wilhelms (2021) p. 439.

[24] Cf. Cerbe; Wilhelms (2021) p. 440.

2.4 Heat exchanger

In industrial heat transmitting there are mainly three different types of flow-structures to separate and describe them:

- Parallel current flow
- Cross-current flow
- Counter-current flow

Frequently, these apparatuses are used to cool process fluids with freshwater or heat up such a medium with a heat carrier like vapour. An apparat may look difficult, but the main aspect and functionality of them is one of the three types of flow-structures, as discussed in the next chapters of this work. Additionally, all subsequent illustrations of heat exchangers, in chapters 2.4.1, 2.4.2, and 2.4.3, are pictured for shell and tube apparatuses, but the functionality in such cases is the same as that of plate heat exchangers.

In all the following temperature over distance diagrams the following expressions and variables are used:

T_{l1in}/K	temperature of liquid 1 at inlet
T_{l1out}/K	temperature of liquid 1 at outlet
T_{l2in}/K	temperature of liquid 2 at inlet
T_{l2out}/K	temperature of liquid 2 at outlet
s/m	flow path length (equivalent area of heat transmission)

Next, fig. 9 displays a small plate heat exchanger, where the plates are forced together by the two yellow pressure plates by the large screws.

fig. 9: Plate heat exchanger from the CO_2 liquefier; source: own representation.

2.4.1 Parallel current flow

As pictured in fig. 10, the heat giving and heat receiving liquid are flowing in the same direction. The process fluid of lower temperature (2) can reach maximum nearly the temperature of process fluid with higher temperature (1) at the heat exchangers outlet, as shown in fig. 11. Furthermore, this illustration demonstrates in a proportional view, the temperature difference between the two liquids is high at the inlet and low at the outlet of the heat exchanger and can move to theoretical convergence at the exchanger's outlet.

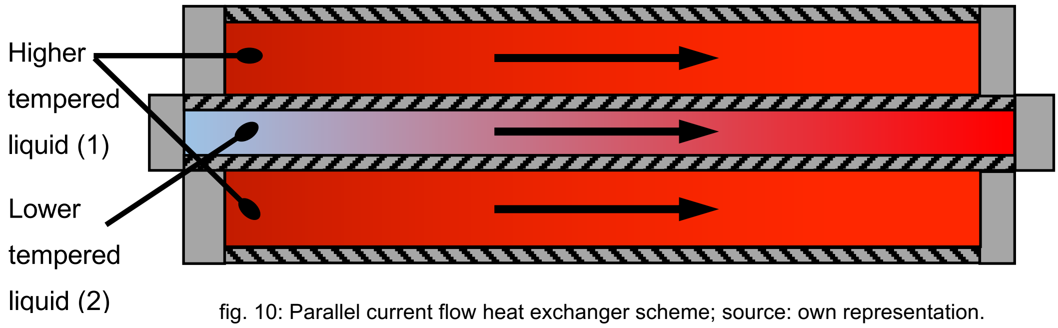

fig. 10: Parallel current flow heat exchanger scheme; source: own representation.

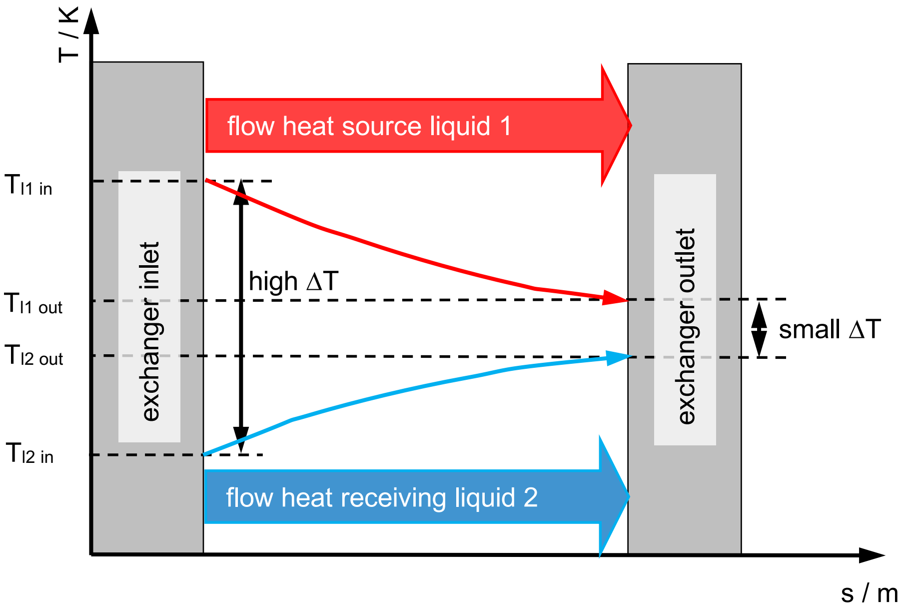

fig. 11: Parallel current flow heat exchanger temperature profile; source: own representation.

2.4.2 Counter-current flow

Unlike parallel current flow, the liquids at counter-current flow heat exchangers are directed in the opposite direction, as shown in fig. 12.

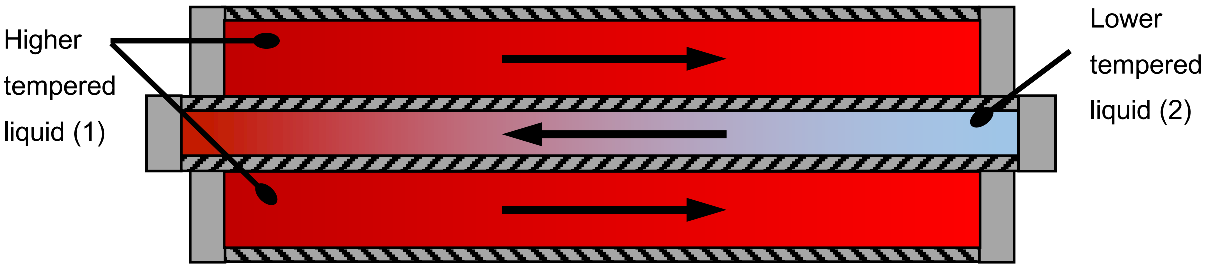

fig. 12: Counter-current flow heat exchanger scheme; source: own representation.

A significant benefit of such apparatuses is that the outlet temperature of the to be heated liquid (2) can nearly reach the temperature level of the heat source liquid (1). The temperature profile of such exchangers is displayed in fig. 13, where the large temperature difference between the liquids is on the outlet side of liquid 1, which is the inlet side of liquid 2.

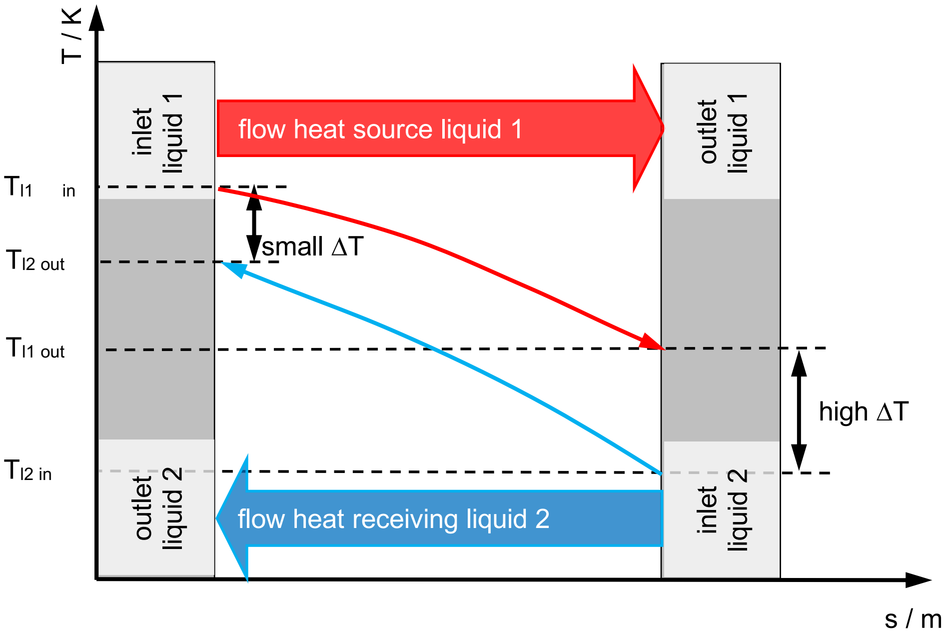

fig. 13: Counter-current flow heat exchanger temperature profile; source: own representation.

2.4.3 Cross-current flow

The cross-current flow heat exchanger is a unique exchanger form because the liquid flows are directed sideways together, as shown in fig. 14 in an orthogonal direction. This type of heat exchanger can be combined with parallel current and counter-current flows. This type of exchanger is often used for condensers, in which one medium changes its physical state from vaporous to liquid, so that at least at one side stable temperature is present.

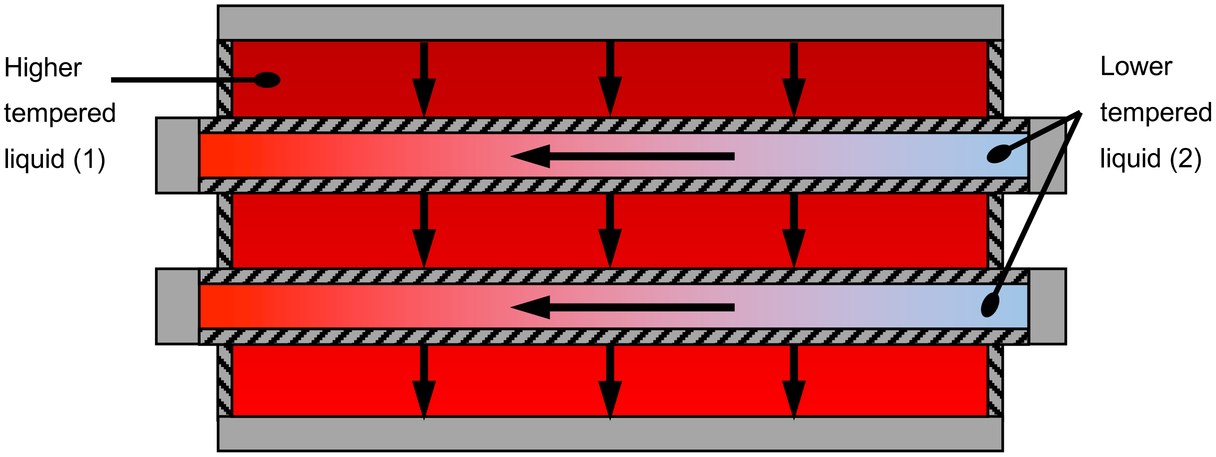

fig. 14: Cross-current flow heat exchanger scheme; source: own representation.

2.5 Functionality of refrigeration plants

The refrigeration cycle (or refrigeration process) is a closed-loop system, as revealed in fig. 15, where the cooling medium is expanded, evaporated, compressed, and condensed again. That is, the cooling medium is reused so that it cannot get lost due to costs or environmental issues. It is a follow up of so-called thermodynamic cycles.

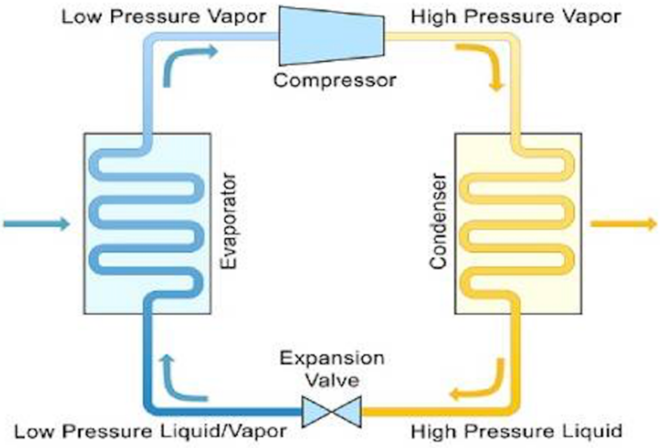

fig. 15: Refrigeration cycle; source: Super Radiator Coils (2021, unmodified).

The left side from the above figure, around the evaporator describes the low-pressure (lp), and the ride side, around the condenser, the high-pressure section (hp) of the refrigeration cycle. The following chapters describe every stage of this cycle to illustrate the refrigeration process.

2.5.1 Evaporation

The main function of refrigeration plants is to cool substances, in technical terms, to take away heat in form of thermal energy. Therefore, the first part of the cycle in this explanation is evaporation. The cooling liquid evaporates at a defined temperature and pressure, and the heat energy needed to sustain this process is provided by the medium to be cooled. According to second law of thermodynamics, described in Section 2.2, heat can be delivered only from high to low temperatures, so the refrigeration liquid must be lower tempered than the medium to be cooled.

A defined amount of refrigeration medium can absorb a defined amount of heat energy Q_0, caused by an increase of the enthalpy H, where H_1 is the enthalpy before evaporation and H_2 the enthalpy after this process.

$$Q_0 = H_2 - H_1 \qquad (2.36)$$

Q_0/J	heat energy in evaporation
H_x/J	enthalpy at time x[25]

Based on a defined mass of coolant medium m_c can be written:

$$Q_0 = m_c\,(h_2 - h_1) \qquad (2.37)$$

m_c/kg	mass of coolant medium
h_x/J kg^{-1}	enthalpy at time x

$$\dot{Q}_0 = \dot{m}_c\,(h_2 - h_1) \qquad (2.38)$$

$\dot{Q}_0$/J s^{-1}	heat energy flow
$\dot{m}_c$/kg s^{-1}	mass flow of coolant medium

$\dot{Q}_0$ expresses cooling load, and $\dot{m}_c$ represents the mass flow of the coolant medium.[26] For a needed cooling load, a certain flow of refrigerant is needed. From the density the mass flow $\dot{m}_c$ must be converted into a volume flow $\dot{V}_c$ to reach the required suction performance of the compressor. The necessary specific volume v and other parameters appear in saturation tables, including tab. 1.

$$\dot{m}_c = \frac{\dot{V}_c}{v} \qquad (2.39)$$

$$\dot{Q}_0 = \frac{\dot{V}_c}{v}\,(h_2 - h_1) \qquad (2.40)$$

$\dot{V}_c$/m^3 s^{-1}	volume flow of coolant medium
v/m^3 kg^{-1}	specific volume from saturation table[27]

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[25] Cf. Reisner (2013) p. 31.

[26] Cf. Reisner (2013) p. 31.

[27] Cf. Reisner (2013) p. 32.

T_x	p		ρ		V		h			s	
temperature	pressure		density		specific volume		specific enthalpy			specific entropy	
°C	bar		kg m^{-3}		dm^3 kg^{-1}		kJ kg^{-1}			kJ kg^{-1} K^{-1}	
	liquid *	gaseous **	liquid *	gaseous **	liquid *	gaseous **	liquid *	gaseous **	$h_{evaporate}$	liquid *	gaseous **
-40	0.97	0.85	708.95	1.026	1.4105	974.35	135.55	1108.1	972.55	0.6895	4.8866
-35	1.23	1.10	702.25	1.305	1.4240	766.38	153.62	1114.7	961.08	0.7645	4.8239
-30	1.54	1.40	695.48	1.640	1.4379	609.71	171.07	1121.1	950.03	0.8383	4.7646
-25	1.92	1.77	688.61	2.040	1.4522	490.21	188.97	1127.3	938.33	0.9110	4.7083
-20	2.36	2.21	681.64	2.513	1.4671	397.99	206.98	1133.2	926.22	0.9826	4.6549
-15	2.89	2.72	674.57	3.067	1.4824	326.05	225.09	1138.8	913.71	1.0531	4.6040
-10	3.50	3.33	667.38	3.713	1.4984	269.35	243.33	1144.2	900.87	1.1227	4.5555
-5	4.21	4.04	660.08	4.460	1.5150	224.23	261.68	1149.2	887.52	1.1914	4.5091
0	5.02	4.86	652.65	5.319	1.5322	188.00	280.16	1154	873.84	1.2593	4.4646
5	5.95	5.80	645.07	6.303	1.5502	158.65	298.79	1158	859.61	1.3263	4.4218
10	7.02	6.87	637.35	7.424	1.5690	134.70	317.57	1162.5	844.93	1.3926	4.3807
15	8.22	8.09	629.47	8.696	1.5887	114.99	336.52	1166.2	829.68	1.4583	4.3409
20	9.58	9.45	621.40	10.135	1.6093	98.66	355.65	1169.6	813.95	1.5234	4.3023
25	11.10	10.99	613.14	11.759	1.6309	85.04	374.97	1172.6	797.63	1.5879	4.2648
30	12.79	12.70	604.67	13.585	1.6538	73.61	394.51	1175.1	780.49	1.6520	4.2281

tab. 1: Extract of an ammonia saturation table; source: Reisner (2013, modified).

In refrigeration processes the ambient temperature of the evaporated ammonia is over the saturation temperature and therefore the fluid is overheated. This is caused to the fact, that the refrigerant has the urge to absorb thermal energy from the process itself and from the environment during transport between evaporation and compression.

In refrigeration technology the vapour gets overheated, which means $T > T^*_{saturation}$ at pressure p. If $T_{real} > T_x$ it is the stage of saturation. For example: if T_{real} = -5 °C at 2.3 bar T_x = -20 °C and therefore the vapour is +15 K overheated.

Note: All the above equations are based on ideal parameters with no losses and are only theoretical.

2.5.2 Compression

The compression process is the first step in the closed-loop cycle and reusage of the now evaporated refrigerant. This step compresses the saturated vapour, compared to the vapour sucked into the compressor, into a higher overheated and elevated pressure vapour, which is significant above the ambient temperature. Next, fig. 16, as example, displays the relationship between volume and pressure using the example of an air piston compressor. These piston aggregates deliver pressured air only reciprocating, every second time the piston reaches the top, in contrast screw compressors deliver a constant outlet.

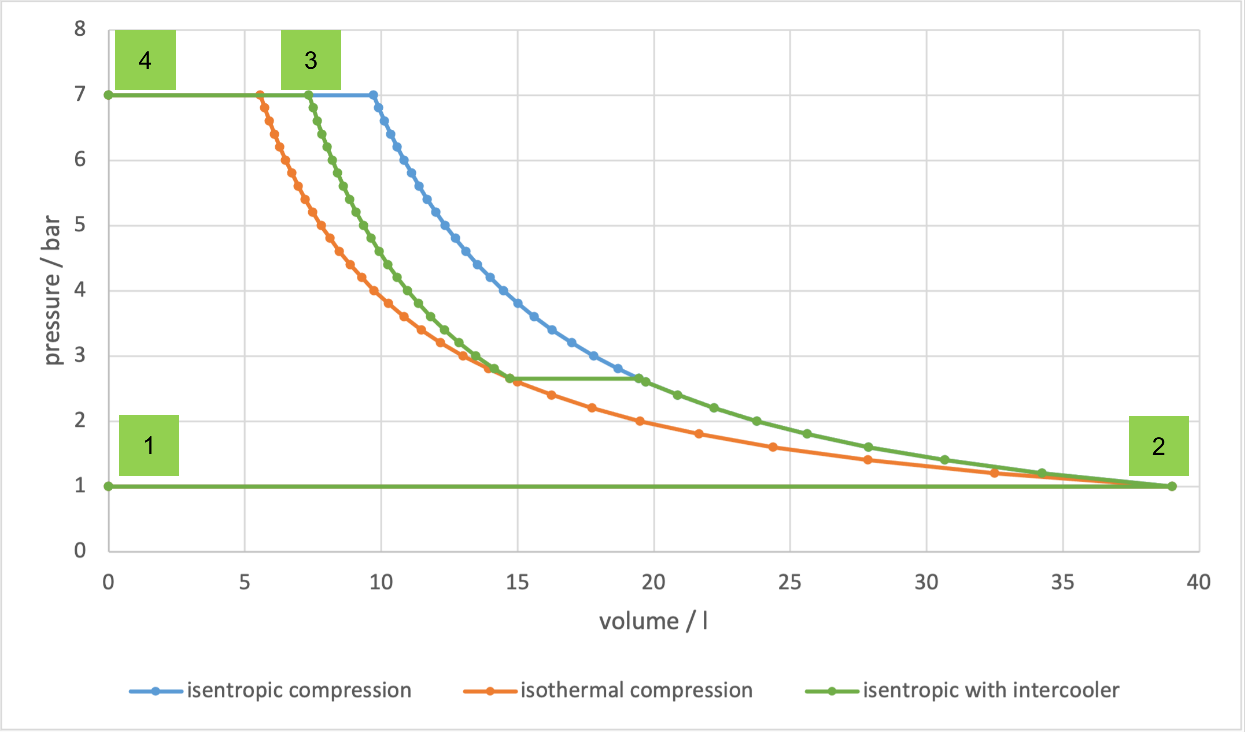

fig. 16: P-V diagram example of an air compressor; source: own representation.

Description of an idealised compression process of a fluid:

- Step 1 → 2: sucking in new fluid for compression process; inlet valve is open and outlet valve is closed. Pressure stays constant and air volume inside the compressor chamber rises, caused by piston movement (geometric intake)
- Step 2 → 3: compression of volume; inlet and outlet valve are closed. Pressure increases and volume decreases inside the compression chamber (geometric compression ratio).
- Step 3 → 4: compressed substance leaves compression chamber; inlet valve stays close and outlet valve opens. Pressure stays constant and volume of the air in piston chamber decreases to minimum to begin process again with suction step.
- Step 4 → 1: Volume stays at the minimum, and pressure decreases down to suction pressure. Cycle is repeating.

Isothermal compression is available only if all heat energy created will be removed simultaneously from the compressed fluid. Again, this is only a theoretical process and not realised in refrigeration compression processes. The isentropic compression leads to a more energy-loaded vapour, than the isothermal compression, due to the higher volume, as shown in fig. 16.

To make this overheated vapour again available for cooling, it must be condensed into a liquid and reevaporated again.

The difference between isentropic and isothermal compression:

In the very simple system, visualised in fig. 17, a simplified compression process with one piston takes place. The energy balance of this system can be described as follows:

$$\frac{dQ}{dt} = w_{in} * \dot{m} - q_{out} * \dot{m} + h_{in} * \dot{m} - h_{out} * \dot{m} \qquad (2.41)$$

Q/J	thermal energy of system
w_{in}/J	work intake
q_{out}/J	thermal energy outtake
h_{in}/J	enthalpy intake
h_{out}/J	enthalpy outtake
$\dot{m}$/kg s^{-1}	mass flow of the systems medium

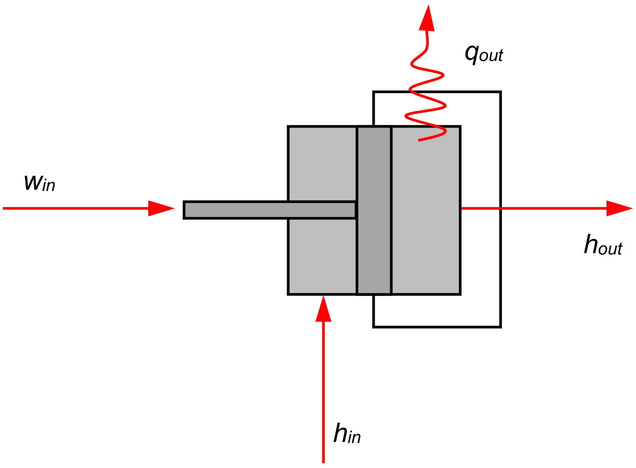

fig. 17: the difference between isentropic and isothermal compression; source: own representation.

Per definition isothermal compression needs a constant temperature in the system $dQ\ dt^{-1}$, therefore all thermal energy caused by the process must be dissipated for validity of:

$$\frac{dQ}{dt} = w_{in} * \dot{m} - q_{out} * \dot{m}$$

$$0 = h_{in} * \dot{m} - h_{out} * \dot{m}$$

$$h_{in} = h_{out}$$

On the other hand, isentropic compression has the fact, that the energy stays in the system dQ dt^{-1}. It also displays the condition that no thermal energy is transferred in or out the system.

$$0 = q_{out} * \dot{m}$$

$$\frac{dQ}{dt} = w_{in} * \dot{m} + h_{in} * \dot{m} - h_{out} * \dot{m}$$

$$0 \neq h_{in} * \dot{m} - h_{out} * \dot{m}$$

$$h_{in} < h_{out}$$

2.5.3 Condensation

At this point, the coolant medium has taken thermal energy from the cooled medium and the compression process it has passed. Condensation manages that these two energies to be given to the environment. The pressure in the system stays nearly constant, so only the temperature drops below the vapour's condensation point, which causes the overheated vapour to start to condensate again into a liquid at the condensation pressure p_c. This process occurs mainly with air and water condensers like tube bundles or plate heat exchangers.

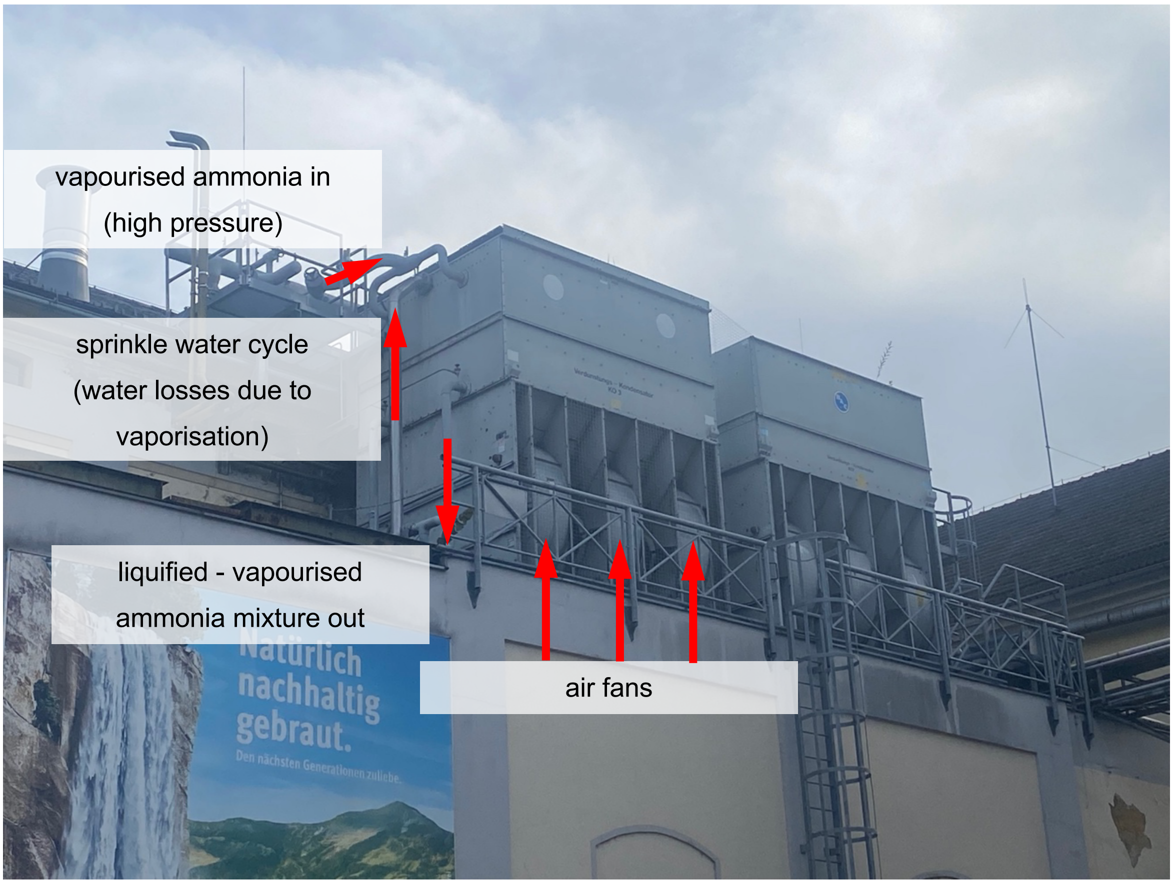

fig. 18: wet tube condensers for ammonia located on a roof; source: own representation.

2.5.4 Expansion

The coolant liquid is now unable to take thermal energy from other mediums because of its high pressure. Consequently, the liquid must be expanded to decrease the pressure, mostly through expansion nozzles. The size of the nozzle in the valve must have a defined size, which depends on the needed coolant mass flow. In industrial applications, various mass flows of the coolant medium are frequently needed, so the nozzles are under control of the temperature or pressure ratio, depending on the load.

2.5.5 Difference between dry and wet evaporation

The main difference in the operating mode between a dry and wet evaporator is, that the dry evaporator has only gaseous fluid inside the heat exchanger and the injection valve is pressure controlled, as can be seen in fig. 19 (left). With this type there is no risk of ice formation or freezing inside the secondary liquid tubes. It operates as injection evaporator, where the steam number of the ammonia is always bigger than zero inside the shell.

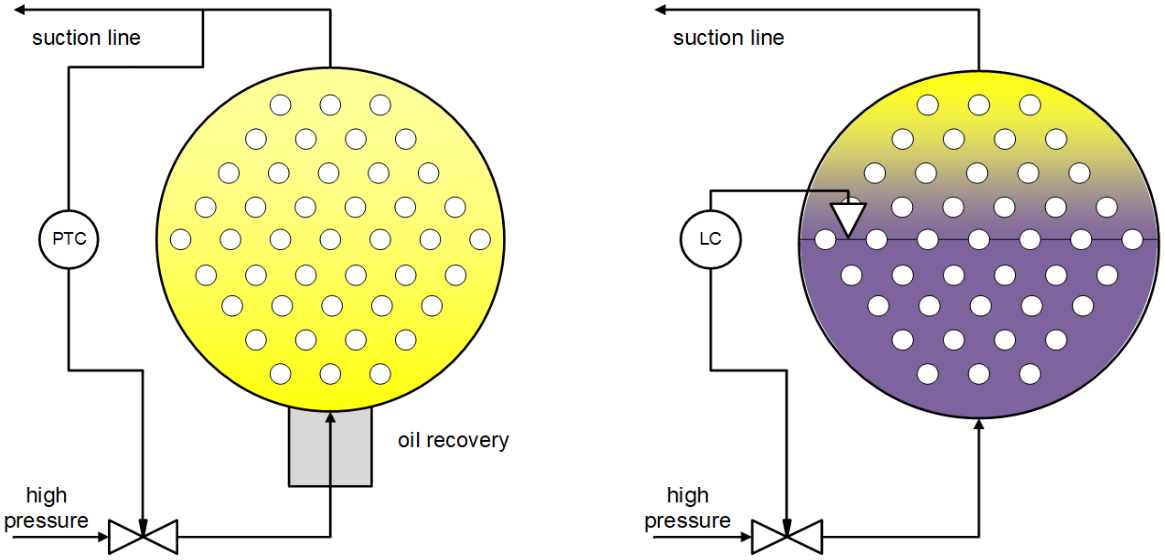

fig. 19: Dry (left) and wet evaporator (right) functionality; source: own representation.

The wet evaporator is level controlled on the level of liquid refrigerant in the heat exchanger, and the ammonia gets vaporised inside the exchangers shell, as pictured in fig. 19 (right). This type should only be used for secondary fluids with a low freezing point, like brine, artificial ice surface, or in some cases glycol, due to the fact, that in the event of fault ice formation is possible, but these evaporators are much more efficient than dry evaporators.

2.5.6 Refrigeration cycle explained with p-H diagram

The complete refrigeration cycle and enthalpy dependence can be explained using the p-H diagram, as pictured for ammonia in fig. 20.

- E → A:
    - Coolant medium acquires heat energy from medium which should be cooled down.
    - Enthalpy rises from 625 kJ kg^{-1} to1730 kJ kg^{-1}.
    - Liquid coolant medium vaporises completely.
- A → B:
    - Vapour of ammonia gets compressed.
    - Enthalpy increases from 1730 kJ kg^{-1} to 1980 kJ kg^{-1}.
    - In point B, the vapour is now overheated.
- B (C) → D:
    - Condensation of vapour back to a saturated liquid.
    - Enthalpy decreases from 1,980 to 625 kJ kg^{-1}.
    - From B to C, the overheat is removed from the vapour, part of the condensation process in the industrial view.
- D → E:
    - Adiabatic expansion, meaning no enthalpy change.
    - Temperature decreases, and the coolant liquid evaporates in part and can again extract thermal energy from other process fluids.

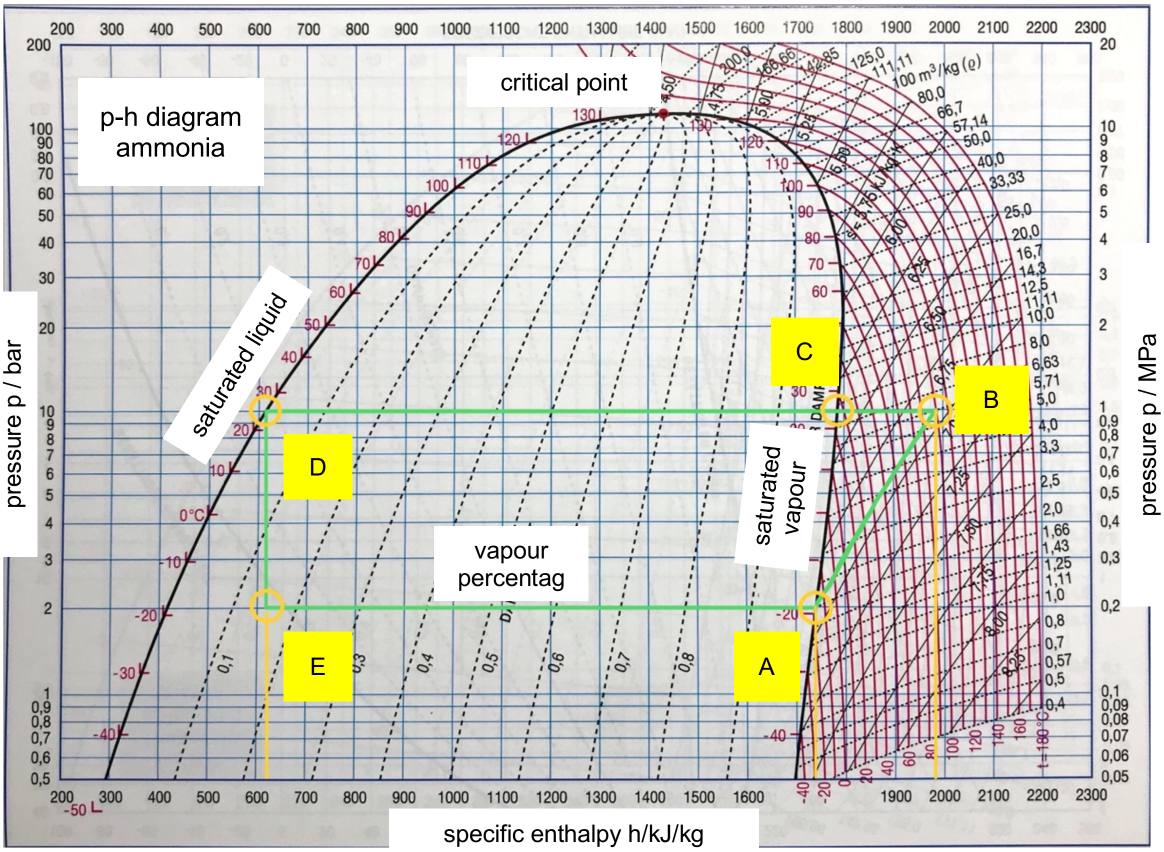

fig. 20: p-h diagram ammonia (R 717); source: Reisner (2013) p. 242 (modified).

2.5.7 Carnot-process

As the refrigerant cycle in cooling plants is a closed loop circuit, in which the refrigerant always gets through various condition changes back to its initial state, it is simplified describable with the so-called Carnot-process. This exemplification is based on thought brain experiment for ideal heat force machines, in which in a gas inside a closed container gets undergone four reversible condition changes, so called idealised reference cycle. Therefore, the initial state can be reached after every cycle, as can be seen in fig. 21. The Carnot process usually shows right turning working processes in opposite to left turning refrigeration processes. [28]

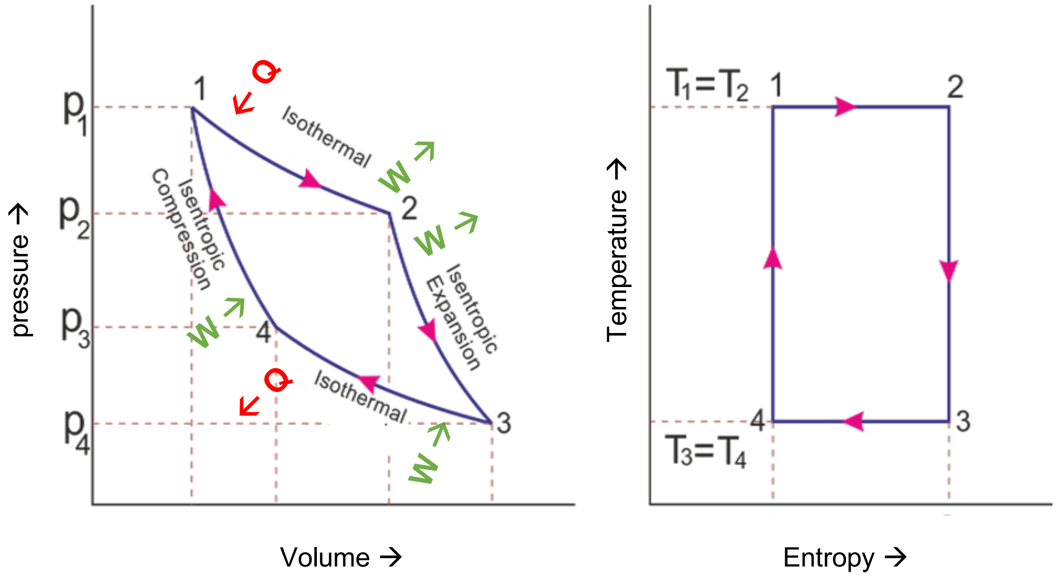

fig. 21: Carnot-process with p-V diagram (left) and T-S diagram (right); source: online source (slightly modified).

In opposite to real refrigeration plants the working medium stays gaseous in the right turning Carnot-process, caused by the inability of condensing ideal gases. The process consists of four, complete reversible, phases, whereby in every phase work is done at or to the gas. First step (1-2) isothermal thermal energy input takes place as slow as needed, so that every temperature difference between gas and outer system equals immediately and positive entropy change happens Through the gases' volume decrease is work to the outside done. Subsequent (2-3), isentropic expansion with no entropy change takes place, until pressure reaches its minimum and volume its maximum. With the lowest entropy next is isothermal compression (3-4), which means the compression is so slow, that under these circumstances every temperature difference between compression and environment gets balanced immediately. Therefore, work must be put into the gas, and thermal energy can be given to environment. Last (4-1), the gas gets isentropic compressed, which means that there is no entropy change in the gas and no temperature exchange with environment takes places, so that the gas temperature and pressure rises to initial state. [29]

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[28] Cf. Reisner (2013) p. 38.

[29] Cf. Reisner (2013) pp. 38-39.

3 REFRIGERATION PLANT DOCUMENTATION

This chapter provides an overview of the refrigeration plants structure and operating mode. The plant works as a centralised unit with compressors, condensers and a few aggregates located in the same building, which is common for industrial refrigeration plants with high capacity. The evaporators are then dislocated at different locations, where process cooling must be operated. These small refrigerant medium aggregates are fed from the first pump station, located under the gas-liquid separator 1. The other main consumers, such as fermentation tanks and large heat transmitters, receive their refrigerant from the second pump station under gas-liquid separator 2, located in the same building as the fermentation cellar. Each of the two pump stations are equipped with three identical pumps, delivering liquid ammonia to the consumers. If an aggregate like the beer cooler sends a request for cooling to the central refrigeration control, the control sets the pump on full power and is not regulated. There is, even no consumer is active, one pump delivering fluid to keep the system stabilised, so called a standby idling system.

Before this time, the storage tanks were modified to a cooling system with glycol to minimise the amount of ammonia refrigerant and safety equipment located at the storage tanks. This is caused by the fact that these types require special safety equipment and gas detecting & warning systems. In contrast to that, a glycol cooling system do not need these features. This dual-circuit system, a so-called indirect cooling cycle, with the ammonia cooling the glycol circuit, is sufficient for the storage tanks.

3.1 Simplified piping diagram

To illustrate the facilities, the researcher created a simplified piping diagram, divided into different parts for clarity. Thus, fig. 22 displays the first part, containing the compressor units, the distributers, and the condensers. Next, the second part is shown in fig. 23, including the gas-liquid separators, the collecting cylinders and cooling apparatuses. Finally, fig. 24 provides an overview of the ammonia distribution for the fermentation tanks. The tanks in this illustration are divided into three different rooms, and the tanks in the first room are no longer used as fermentation tanks because they have been modified to be pressure tanks. The second and third rooms have different methods for cooling the tanks, as explained in Section 4.1 and Chapter 5, which describe each tank's data recording and specialisation.

Each piping diagram and its aggregates display only the flow and distribution of the ammonia circuit through the following colour codes:

- Purple: ━━━ liquid ammonia after condensation
- Yellow: ━━━ overheated gaseous ammonia after compression
- Orange: ━━━ gaseous ammonia after heat absorption
- Black: ━━━ aggregates and additional information

Note: All simplified piping diagrams are visualised without any instruments and expansion valves. In fact, every aggregate has expansion valves to decrease pressure and vaporise the liquid ammonia for the cooling process.

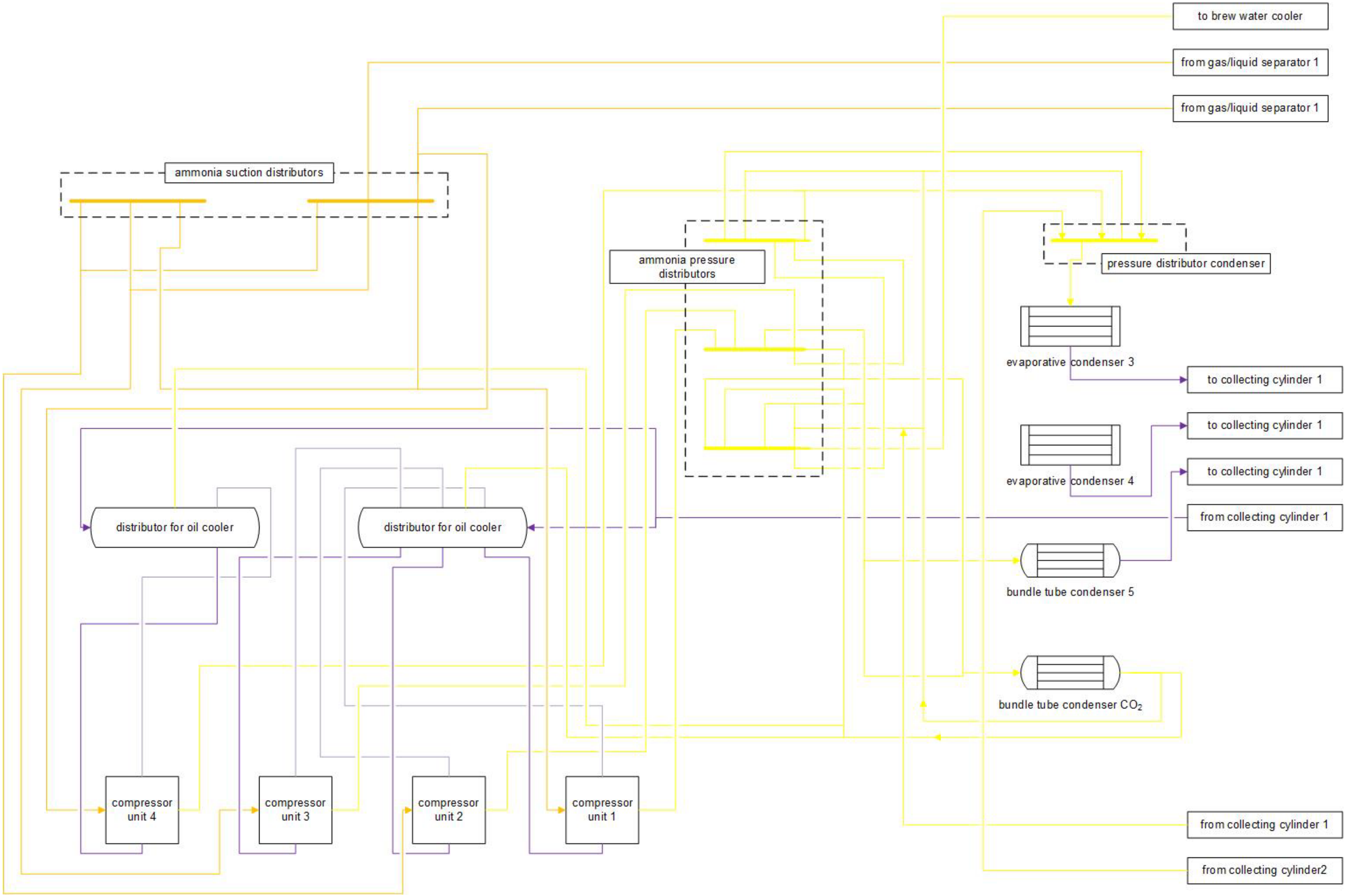

fig. 22: Simplified piping diagram Part 1: Compressors and condensers; source: own representation.

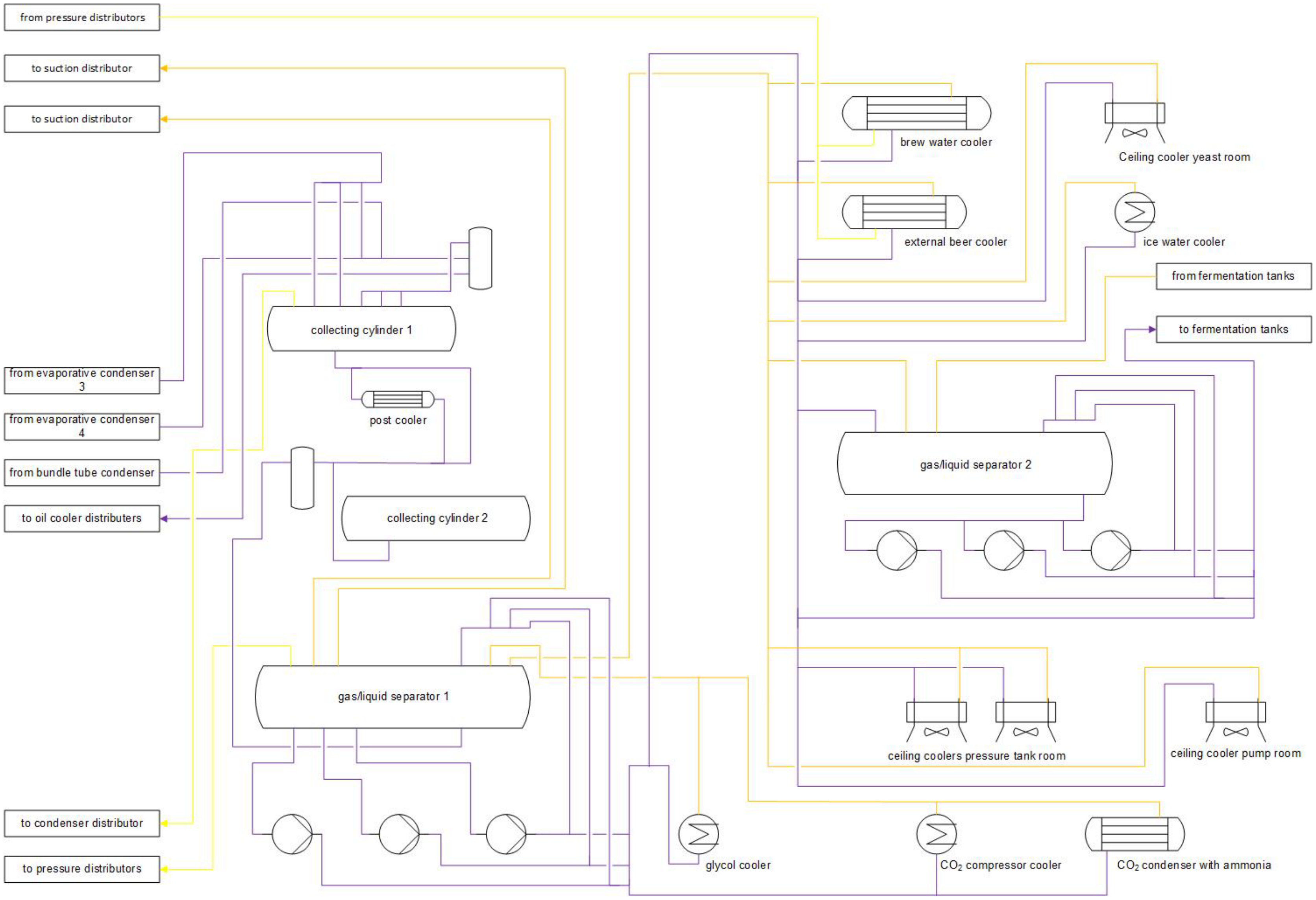

fig. 23: Simplified piping diagram Part 2: Collectors, separators, and consumers; source: own representation.

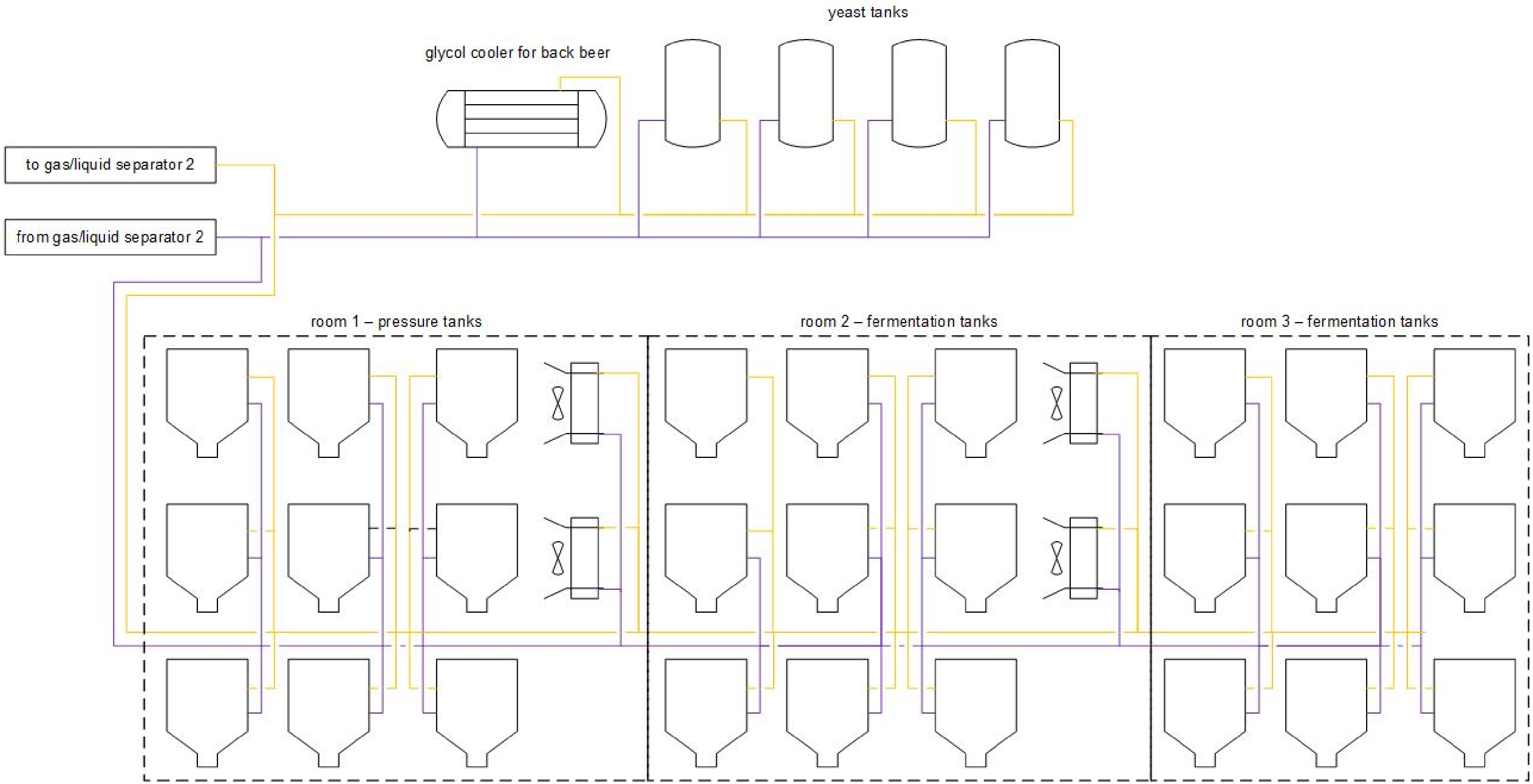

fig. 24: Simplified piping diagram Part 3: Liquid ammonia distribution of fermentation tanks scheme; source: own representation.

3.2 Machine data, functionality, and capacities

As pictured in the piping diagrams in Section 0, various consumers for cooling are installed throughout the refrigeration system. In refrigeration processes is differentiated between the high-pressure section and low-pressure section, as shown previous in fig. 15. According to the internal test book the refrigeration plant has a total mass content of 15,330 kg ammonia with a theoretical maximum cooling power of 2500 kW.

For instance, before beer and water can be cooled with liquid ammonia, the gaseous ammonia must be compressed and condensed, which is performed in the central station. To this end, the plant in the brewery Göss has four compressors, as depicted in tab. 2.

	Voltage	Electrical power	Compressor type
	V	kW	-
Compressor 1	400	327	screw vertical
Compressor 2	5000	830	screw horizontal
Compressor 3	5000	830	screw horizontal
Compressor 4	5000	830	screw horizontal

tab. 2: Ammonia compressor data comparison; source: internal documents.

At the high-pressure section of the refrigeration plant, the condensation requires two sprinkle water condensers with electrical power consumptions of 28 kW for each fan and 22 kW for each water pump and one shell and tube heat exchanger to cover the peaks of the performance needed. These sprinkle condensers have different operating modes, which are:

- Evaporative cooling with air fans,
- Variable recirculating water flow with maximum usage of 30.8 l s^{-1} of water,
- Two different fan speeds, and
- Combined evaporative cooling with air fans and water sprinkling.

To buffer and store the liquid ammonia, two collecting bottles, with a combined storage volume of 4588 l, are installed after the condensers. Due to the two different main locations of cooling applications, every location has its own gas-liquid separator. Separator 1 with a volume of 10000 l, works for the area of all the consumers between coolant compression-condensation and separator 2. However, separator 2 with 9000 l maximum capacity, feeds all aggregates in the fermentation cellar, as described in Section 0. Additionally, the pump units for distributing are mounted under each of these two separators.

As previously mentioned, the main distributing pumps under the separators each have a volume flow range from 1 to 11 m^3 h^{-1}, depending on the required delivering height, as demonstrated in the extract of the data sheet in fig. 25, where the marked line represents the installed pumps. The pumps are needed, caused by the pressure loss of the long-distance between condensers and evaporators. In small refrigeration units the condensers should always be fitted as tight as possible before the evaporator to minimise pressure losses. In big cooling plants, like the one described, it is not possible to build up the refrigeration plant in such a way. Therefore, the liquid ammonia distribution pumps hold constant pressure in the distribution manifolds that even the most dislocated expansion valve and evaporators get the right pressure.

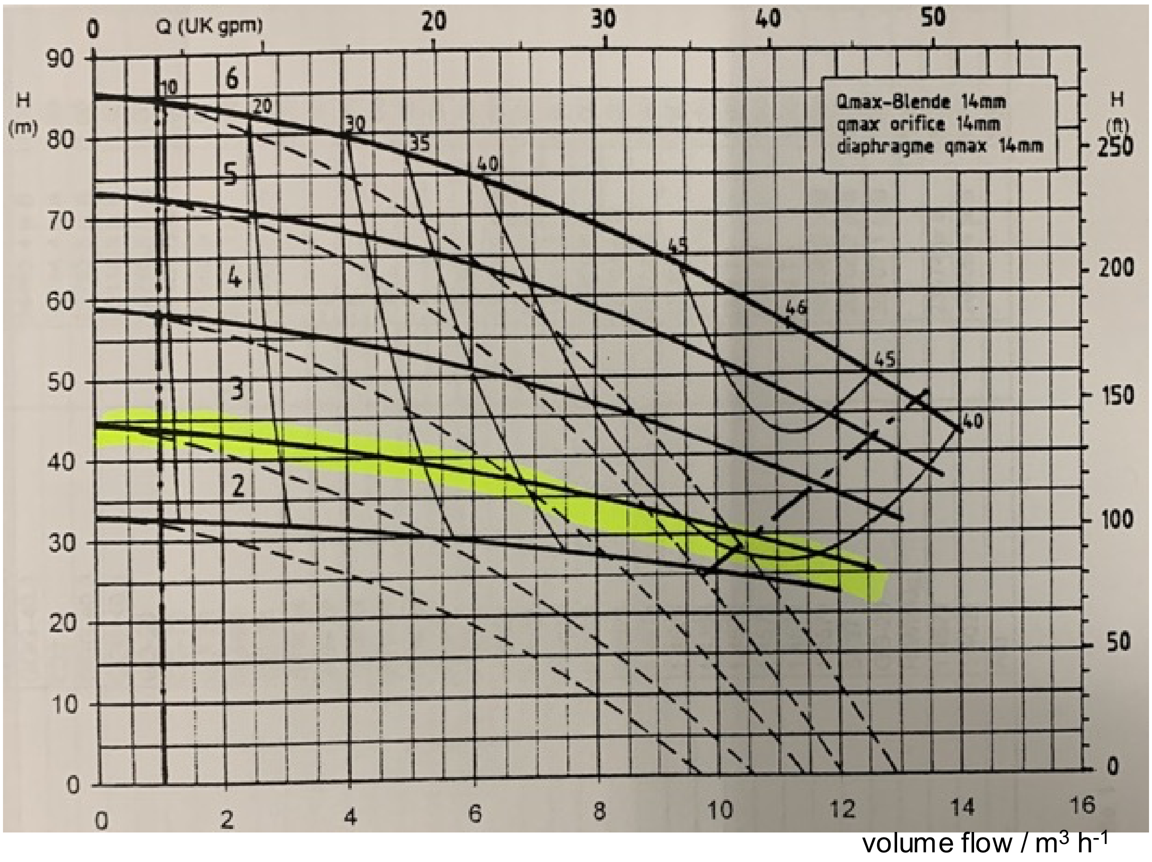

fig. 25: Data sheet liquid ammonia distributing pumps; source: Hermetic (1998).

All the next mentioned aggregates for different cooling process are on the low-pressure section of the refrigeration. The external beer cooler is used for cooling fermented and maturated beer to slightly under 0 °C before storing it in a storage tank. For this thesis investigated beer cooler is a shell and tube heat exchanger, as pictured in fig. 26, and is mounted in the fermentation cellar. This cooler was installed in 1989 and can hold up a geometric volume of 1,770 l of ammonia vapour and 740 l of beer. Additionally, beer coolers can be built up as plate heat exchanger, which better transmit heat and are more efficient but require more powerful pumps to achieve the same pressure and volume flow as the shell and tube heat exchanger. This evaporator operates in dry evaporation mode, which means that the shell is mainly dry, and no liquid is hold up inside. The big advantage of shell and tube exchangers for cooling is, that there is no caution for freezing on the secondary exchanger side, because mostly only the lower located tubes freeze, so that the upper located tubes are free for the flow of the fluid.

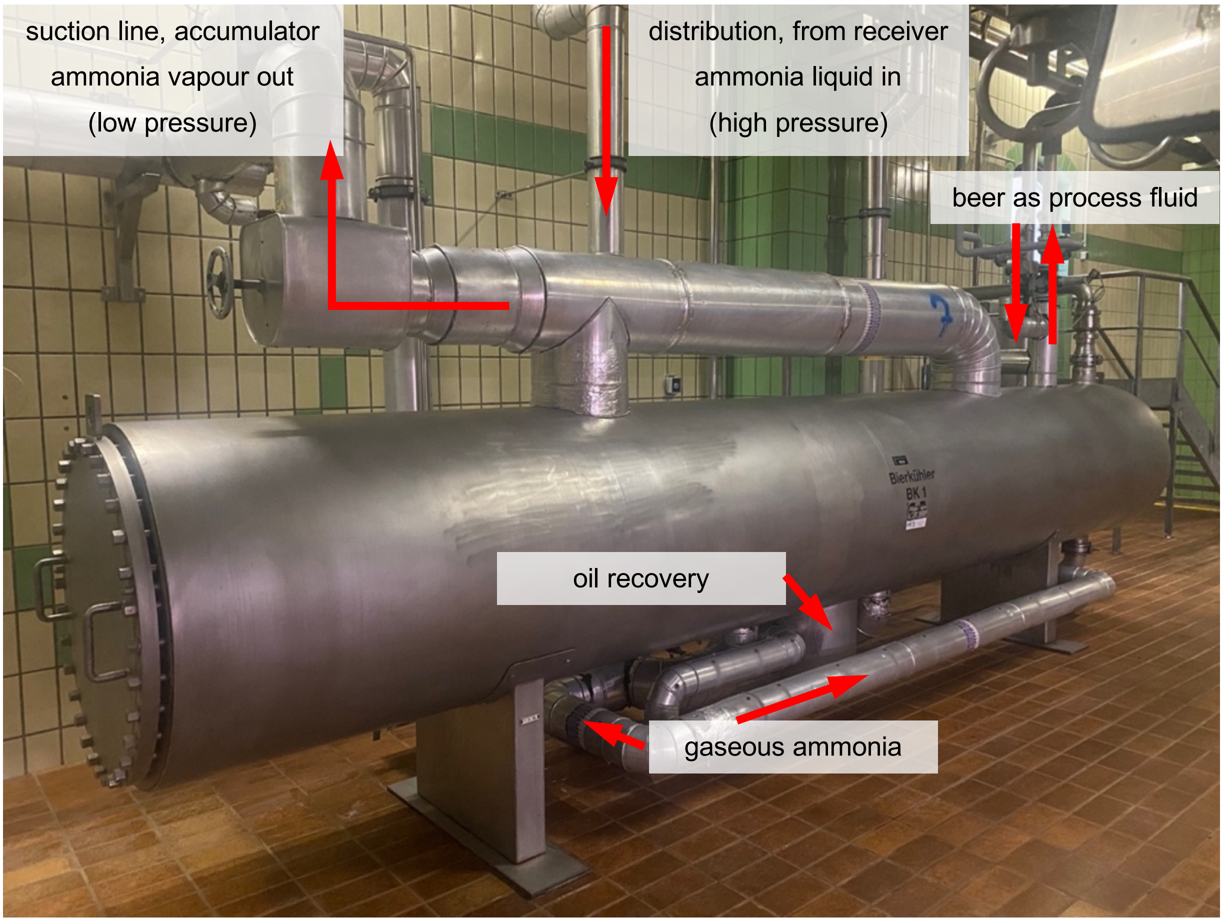

fig. 26: External beer cooler; source: own representation.

The next large cooler, which is also a shell and tube heat exchanger, is used for preparing cold brew water, reaching approximately 6 °C for the main hot wort cooler from the brew house. The hot wort is the water, malt, hop, and unfermented mixture which was cooked in the brewhouse. This fluid must be cooled before pouring it into a fermentation tank. That is, if yeast were added at this high temperature, it would die, and the amount of liquid ammonia needed to cool the wort in the fermentation tank would be excessive, so the tanks have only coat cooling plates. Next, the cold water is created by cooling normal brew water from the wells in this heat exchanger with a geometric volume of 770 l of ammonia vapour and 395 l of water inside and storing the liquid in a tank until it is needed by the hot wort cooler. This dual-circuit system protects the product from any direct contact with ammonia. The cold brew water tank serves as a buffer or storage for decoupling delivery and demand. Subsequently, the heated water is reused for heating processes throughout the brewery. A simplified process diagram is given in fig. 27.

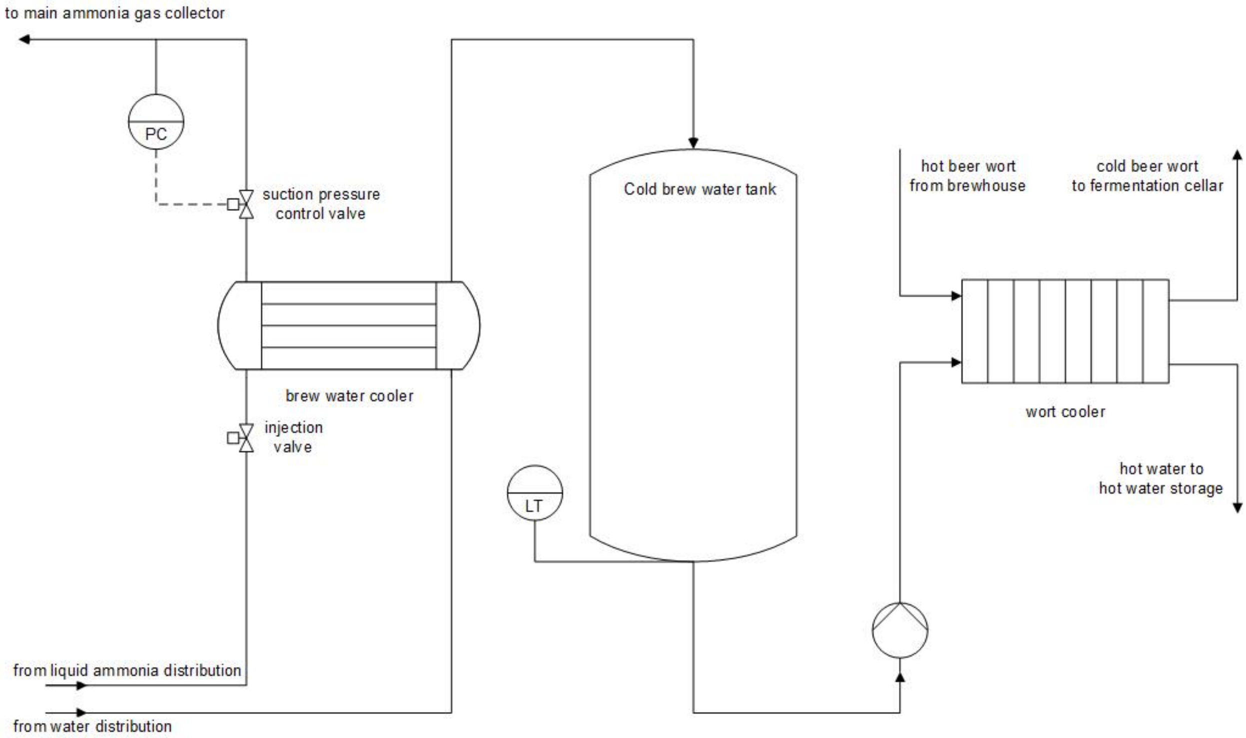

fig. 27: Simplified piping and instrumentation diagram of the brew water cooler system; source: own representation.

For cooling rooms, like the valve room for beer distributing, yeast room or fermentation tank rooms 1 and 2, ceiling air convection coolers, so called fan coils, with direct evaporation are installed, each containing a fan that blows the air through a grid containing liquid ammonia. These coolers are mostly unused in winter months due to the low temperatures outside the room. Under these circumstances cold air from outside gets blown into the room to save energy.

Some of the greatest cooling consumers are the fermentation tanks, of which 18 are installed in the brewery Göss. These are cylindroconical and fitted with cooling coats, called dimple-jackets or temp-plates, with one installed in the conical part and the others in the cylindrical one. In Göss, there are two types of tanks, with the main differences between these listed in tab. 3.[30]

Tank type	Number dimple-jackets	Total volume dimple-jackets	Tank content	Additional information
	pieces	litres	litres	-
1	3	174.0	498000	+ 2 ceiling air coolers in room
2	4	175.6	486000	+ insulation on tank surface

tab. 3: Fermentation tank differences; source: own representation

The tanks are never filled to the brim, caused to a gaseous phase of carbon dioxide above the beer, through the fermentation process formed. Thus, the tanks are built with a constructive volume reserve, for degassing and foam, of about 25 percent over the maximum amount of wort they must hold. The tank size should be approximately half of the daily wort production to keep the first beer in each tank from beginning to ferment before the tank is full. Therefor within approximately two days a single tank might be sufficient filled the latest.[31]

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[30] Cf. Kunze (2021) p. 526.

[31] Cf. Kunze (2011) pp. 510-511.

In the carbon dioxide recuperation plant, gets the through fermentation formed CO_2 recycled, to be able to be reused in the filling lines. In this plant is an additional small own version of an ammonia refrigeration plant, with 310 kg refrigerant and a cooling power of 30 kW, installed, as displayed in fig. 28. This unit comprises an ammonia compressor, a gas-liquid separator, and a plate heat exchanger to be used as a condenser. The ammonia is used for cooling and liquifying gaseous carbon dioxide into a liquid before storing it in the appropriate tanks. Notably, the condenser works with ammonia on both sides, so vapourised, overheated, compressed ammonia is liquified and cooled with ammonia from the main refrigeration plant, which is a two-step cascade cooling, for condensing pressurised CO_2.

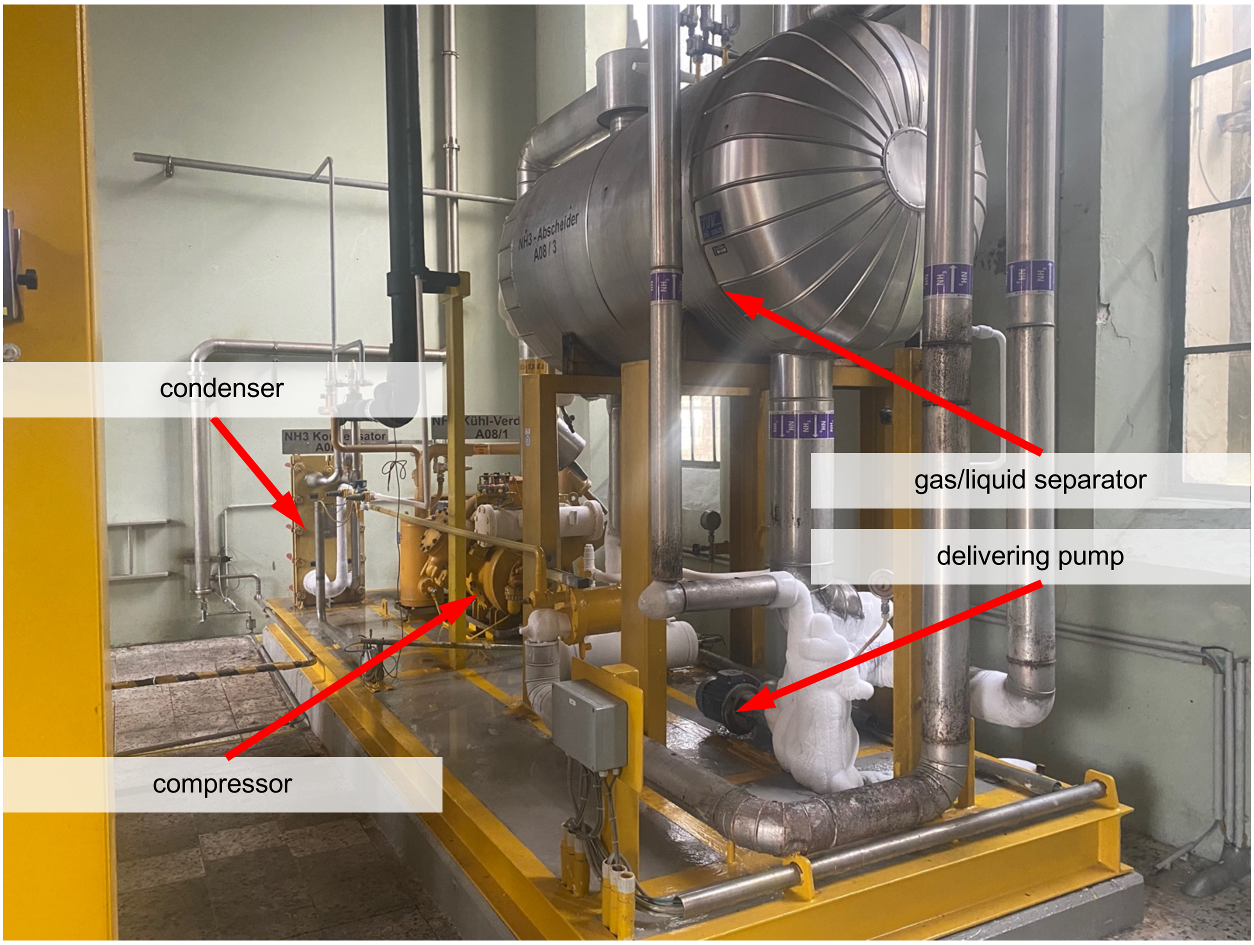

fig. 28: Integrated ammonia refrigeration unit in carbon dioxide recuperation plant; source: own representation.

Additionally, the carbon dioxide condenser has an expansion valve but no injection valve, producing a steady flow of a mixture of gaseous and liquid ammonia between the condenser and the separator, with a maximum content of 900 l geometric volume.

4 RECORDING OF TEMPERATURE AND COOLING PERFORMANCE DATA

One aim of this thesis was to record the data of the large coolant consumers. To measure the liquid flow of the ammonia, the researcher used the ultrasonic flowmeter Proline Prosonic Flow 93P from Endress & Hauser. This instrument is normally used as a replacement measuring device if inline applications fail or to check these applications' functionality. Through its varied accessories, this flowmeter can be used from pipes with diameters between 15 and 4000 mm, with maximum errors of measurement listed in tab. 4.

Pipe size range	Maximum error of measurements
mm	%
15	± 3
25 to 200	± 2
> 200	± 2

tab. 4: Flow measuring device maximum error of measurements; source: Endress & Hauser.

For add on temperature measurements on the pipes, the study employed PT100. The data of all separate mounted sensors were temporarily logged with the data logger Ecograph T RSG35 from Endress & Hauser, pictured in fig. 29. The model used has 8 independent analogue inputs and the space for six digital inputs and six digital outputs. Additionally, this work included data from the internal data logging software PlantiT from ProLeiT, such as inline temperatures, flows, pressures, tank content, and process data.

fig. 29: Endress & Hauser Ecograph T RSG35; source: Endress & Hauser.

Recording of temperature and cooling performance data

The first step before taking measurements was to check if the ultrasonic flowmeter functioned correctly by mounting and installing it in front of an electromagnetic flowmeter. on the same pipe, as depicted in fig. 30 of a simplified P&ID of the brew water processing tank. This tank is part of the process for softening water for the brewing process.

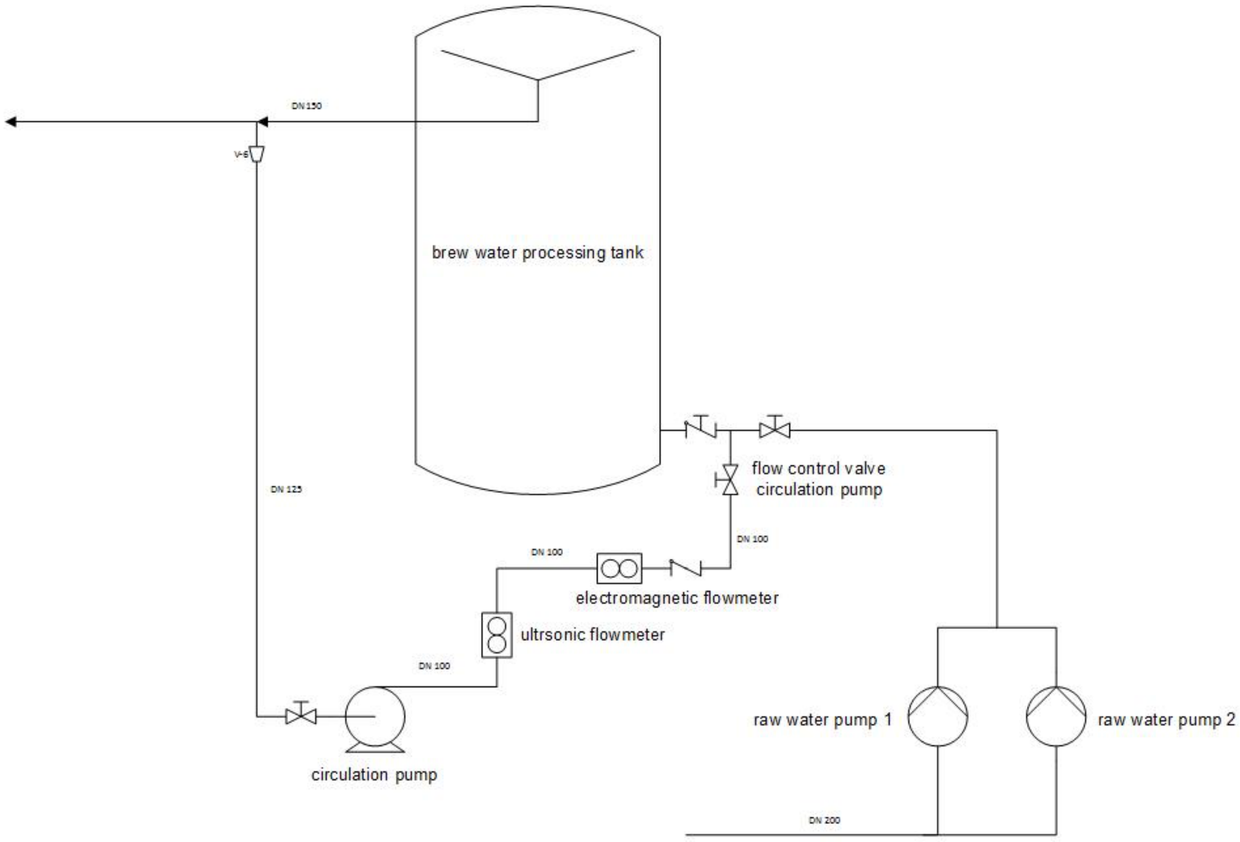

fig. 30: Simplified P&ID brew water processing tank; source: own representation.

Using the flow control valve of the circulation pump, the researcher created various settings for the water flow and recorded the values of both flowmeters, as shown in tab. 5.

Measuring number	Volume flow ultrasonic measuring device	Volume flow electromagnetic measuring device
	hl h^{-1}	hl h^{-1}
1	60.4	60.3
2	20.5	20.3
3	6.5	6.7

tab. 5: Function check of ultrasonic measuring device; source: own representation.

This table above displays that the ultrasonic measuring device is within an acceptable error range.

Recording of temperature and cooling performance data

While recording, the highest acquisition frequency rate, which can be taken out of the internal data recording system, is a time interval of 15 seconds per measuring point. On the contrary the external datalogger delivers data every second. Therefore, these two different time stamps had to be combined. This process involved calculating the average values over 15 seconds and saving them to time stamps at 15-second intervals to match the internal data logging software. Beyond this, the researcher adjusted the time stamps from both systems to match before taking measurements.

4.1 Fermentation tanks

As mentioned in Section 3.2, there are 18 fermentation tanks installed in the brewery for multiple types of beers. Generally, five brews per tank are filled for each fermentation process, as depicted in fig. 36. This entire process takes in overall about 300 or more hours to complete, from the first filling until draining. For this thesis, the two different types of tanks were measured for a whole cycle and compared, as shown next in Chapter 5.

Neither the level of beer nor the time the beer inside the tank is always the same, so there were some discrepancies in the measurements between the two tank types, which are explained later. These differences are caused mainly at working time, storage capacity, and brew volume. Working time is five days a week, if a fermentation finishes on weekend, the tank can’t be drained. The same applies to too less availability for space in storage tanks. The brew volume isn’t always constant, depending on the recipe of different types of beers and deviations in the brewing process.

As mentioned, the tanks are fitted with dimple-jackets, in which the liquid ammonia gets injected. Mounted in front of the injector valves, the ultrasonic flowmeter measures the flow rate of the liquid ammonia in the ammonia supply pipe. The PT100s were mounted on every outlet of the tank’s cooling coats to capture temperature differences, because surface PT100 are slow and inaccurate at dynamic developments of the measuring points. Next, the tank content is measured with inline electromagnetic flow meters mounted in the pipe between the brewhouse and fermentation cellar and are recorded with the internal logging software. Due to measuring tolerance and possible errors the tanks are equilibrated with level switches at different volumes of content.

To describe the filling, draining and the content a process technology approach, the so-called integrative balance, is used, where the storage and filling term are described and visualised in fig. 31. The specialty of the fermentation tanks operation is, that there are two batch processes involved. The filling which frequency depending on the brewing frequency as short time batch and the main fermentation and maturation taking over 300 hours as long-time batch process.

Recording of temperature and cooling performance data

Storage term:

The storage term tells the content inside the tank in dependence of the volume flow into and out of the storage system.

$$\frac{dV}{dt} = \dot{V}_{in} - \dot{V}_{out} \tag{4.1}$$

V/m^3	tank content (volume)
$\dot{V}_{in}$/m^3 h^{-1}	volume flow into the tank
$\dot{V}_{out}$/m^3 h^{-1}	volume flow out of the tank

Filling term:

The filling term considers exclusively the volume flow, tank level and area of content.

$$\frac{d(A * z)}{dt} = \dot{V}_{(t)in} \tag{4.2}$$

A/m^2	area of tank (circular)
z/m	level, height of level
$\dot{V}_{(t)in}$/m^3 h^{-1}	volume flow into tank at defined time

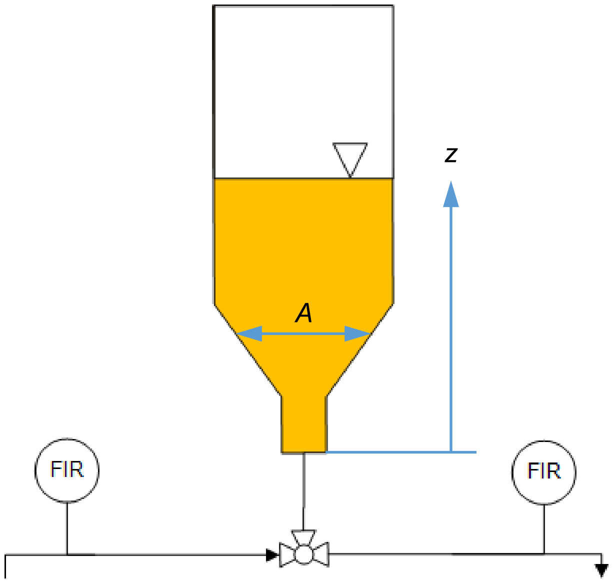

fig. 31: Integrative balance scheme of fermentation tank; source: own representation.

Recording of temperature and cooling performance data

As previously described, liquid ammonia is injected via expansion valves in the heat exchanging zones. fig. 32 shows a valve station with the injector valves of a tank with four cooling coats.

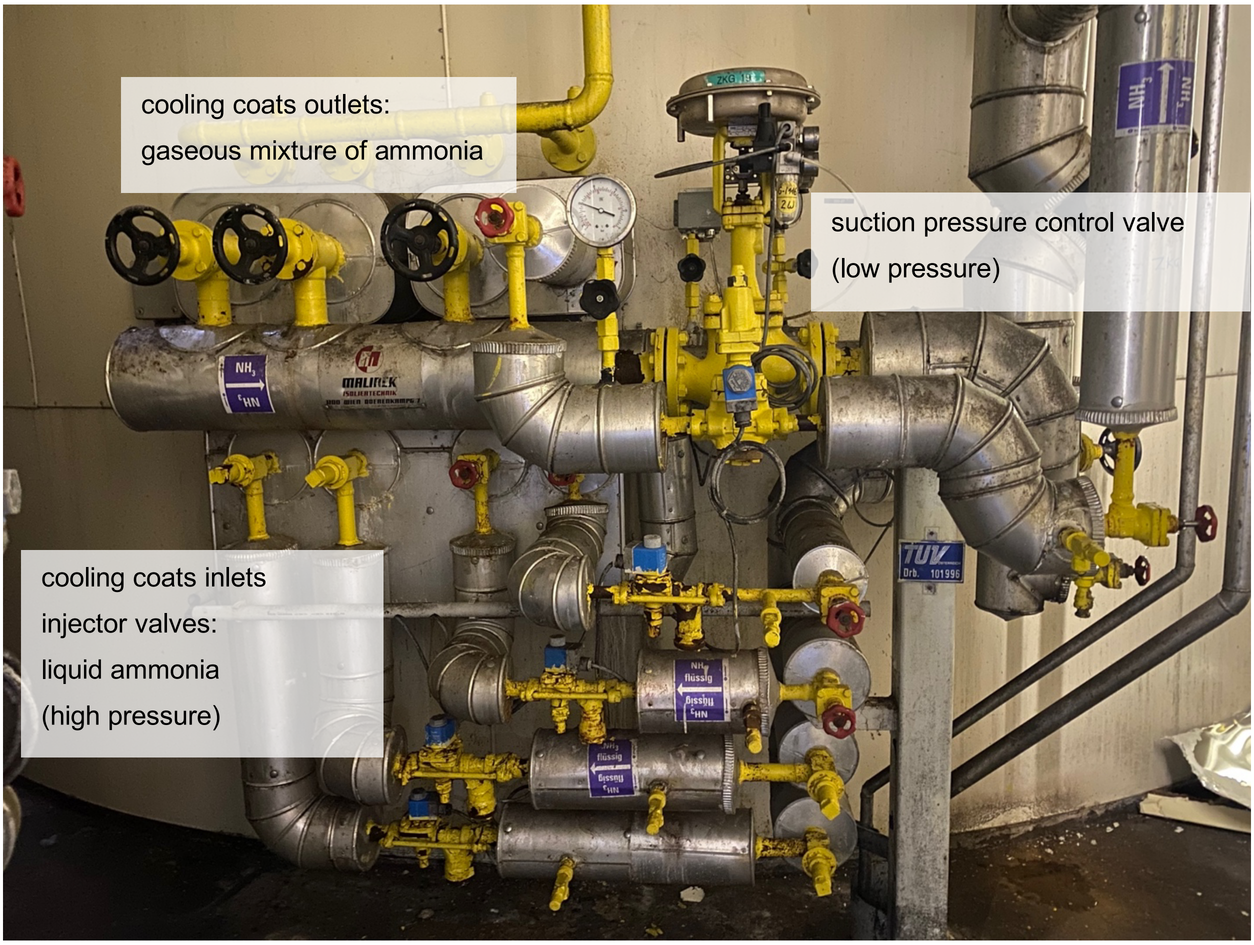

fig. 32: Instrumentation group (valve unit) fermentation tank; source: own representation.

Next, fig. 33 illustrates the control diagnosis scheme of the fermentation tank with the steam number x_{lp}, which describes the amount of vapour to total mass, as followed:

$$x_{lp} = \frac{m_{vapour}}{m_{total}} = \frac{m_{vapour}}{m_{liquid} + m_{vapour}} \qquad (4.3)$$

$x_{lp}/1$	steam number (only referred to as x in the literature)
m_{vapour}/kg	mass of gaseous ammonia
m_{liquid}/kg	mass of liquid ammonia

Recording of temperature and cooling performance data

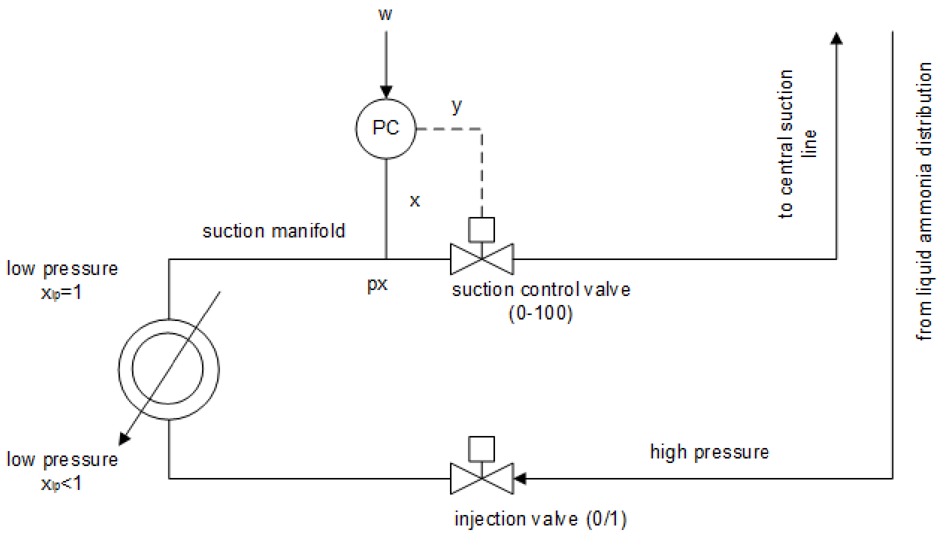

fig. 33: Control diagnosis instrumentation group fermentation tank; source: own representation.

The evaporation pressure control must manage two targets:

- Level of evaporation temperature depending on target temperature, and
- Load on the suction line, where high pressure causes less speed and vice versa.

At this point, the measured liquid flow of ammonia must be recalculated and converted into a mass flow following Formulas 4.1 and 4.2:

$$\rho = \frac{1}{v'} \qquad (4.4)$$

ρ/kg l^{-1}	density of liquid ammonia, temperature depending
v'/l kg^{-1}	specific volume of liquid ammonia

$$\dot{m} = \dot{V}\,\rho \qquad (4.5)$$

$\dot{m}$/kg h^{-1}	mass flow ammonia
$\dot{V}$/l h^{-1}	volume flow liquid ammonia

According to the density, depending on the temperature of liquid ammonia, a mean liquid temperature was used to calculate the mass flow. This dependence is nonlinear, as displayed in fig. 34.

Recording of temperature and cooling performance data

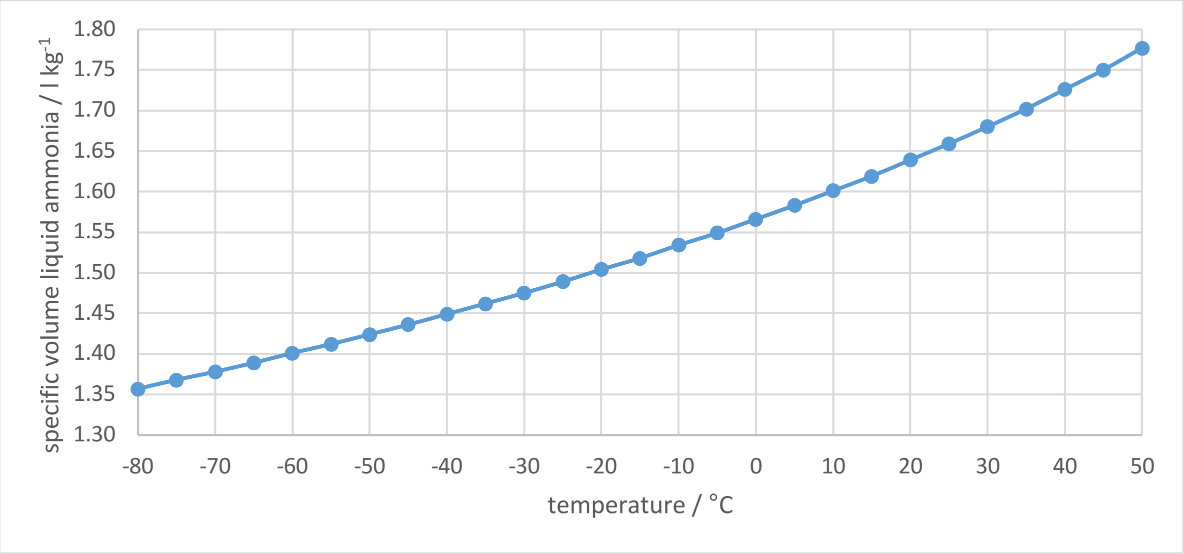

fig. 34: Specific volume dependence of liquid ammonia on temperature; source: figure own representation, data Springer (2006).

The following graph, fig. 35, displays an overview of one batch process of fermentation tank 17 with the recorded and recalculated data of liquid ammonia mass flow, temperatures, tank content and suction pressure. To improve visualisation and modelling, the researcher divided this cycle into four main phases, also displayed here:

- filling,
- fermentation,
- maturation and
- cooling and draining.

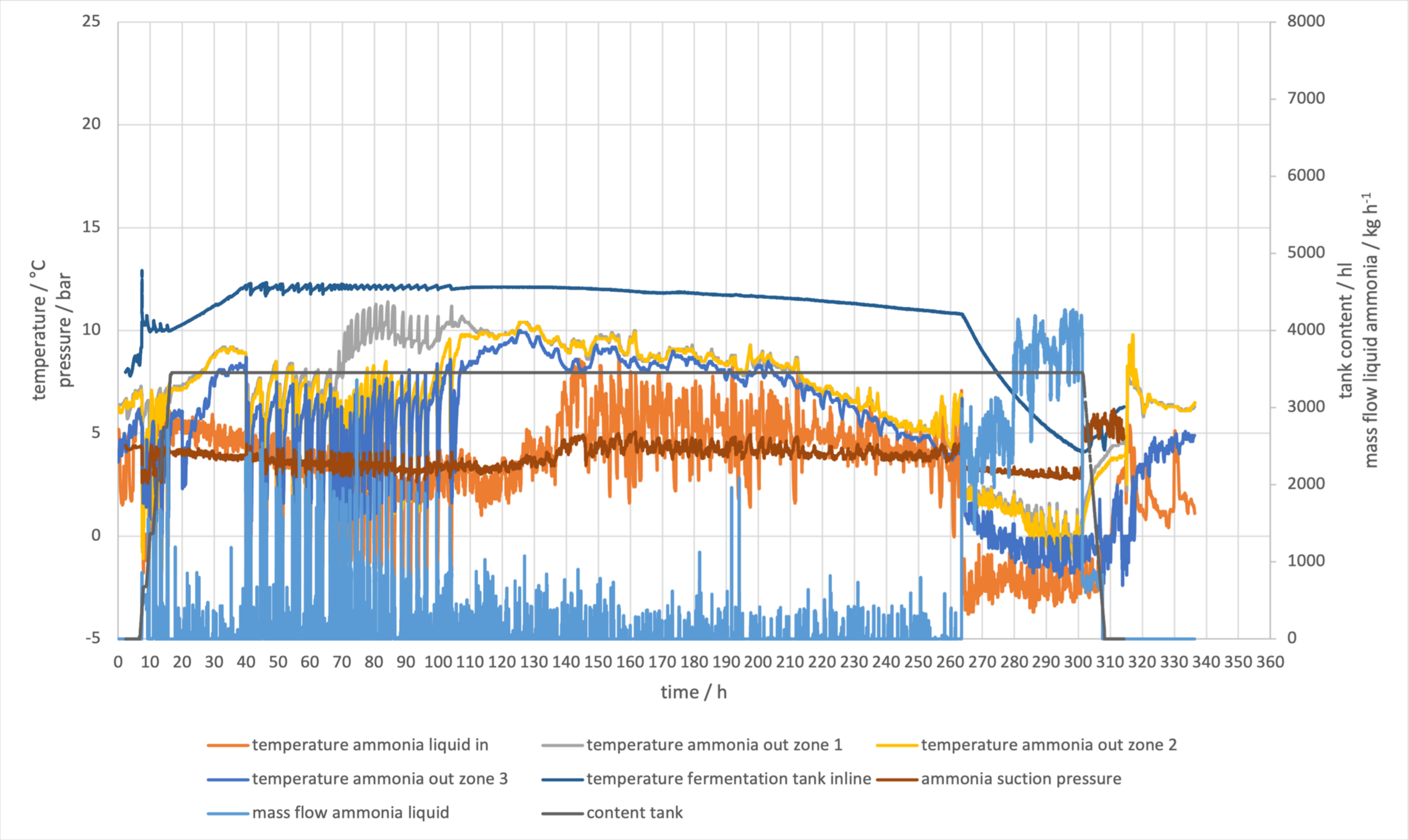

fig. 35: Fermentation tank 17 – overview (hours 0 to 360); source: own representation.

As revealed in fig. 35, the tank temperature must be regulated at a defined temperature target until all five brews are filled in one after the other. Subsequently, the amount of liquid ammonia decreases, raising the temperature inside the tank. Depending on sort of beer and yeast different temperatures are needed for fermentation. The temperature is maintained within a narrow range to keep the fermentation moving well to prevent the yeast from dying and leaving the beer half fermented. After the main fermentation, the maturation stage begins, where the beer is stored at a specific temperature to produce its unique taste and alcohol content. Approximately 260 hours after filling in the first brew, the process of cooling and draining starts. To cool the contents of the tank requires the largest amount of liquid ammonia needed at any point in the process. At a defined temperature, the beer can be transferred to a pressure tank before it is packaged in bottles, kegs or cans or loaded on a truck and delivered to another brewery for filling in non-returnable bottles, which isn't possible in Göss. Additionally, the graph shows the ammonia outlet temperatures after the evaporator and the suction pressure, which is regulated from a plc.

As mentioned above, the main phases of the fermentation process are split to optimise visualisation and explanation. Additionally, fig. 36 depicts the filling phase of the tank. The black line, which increases stepwise, shows the five brews filling the tank, and the subsequently, the temperature can rise to the fermentation level. This dependence of the temperature reflects direct to the liquid ammonia demand, is illustrated by the light blue and dark blue lines in the graph.

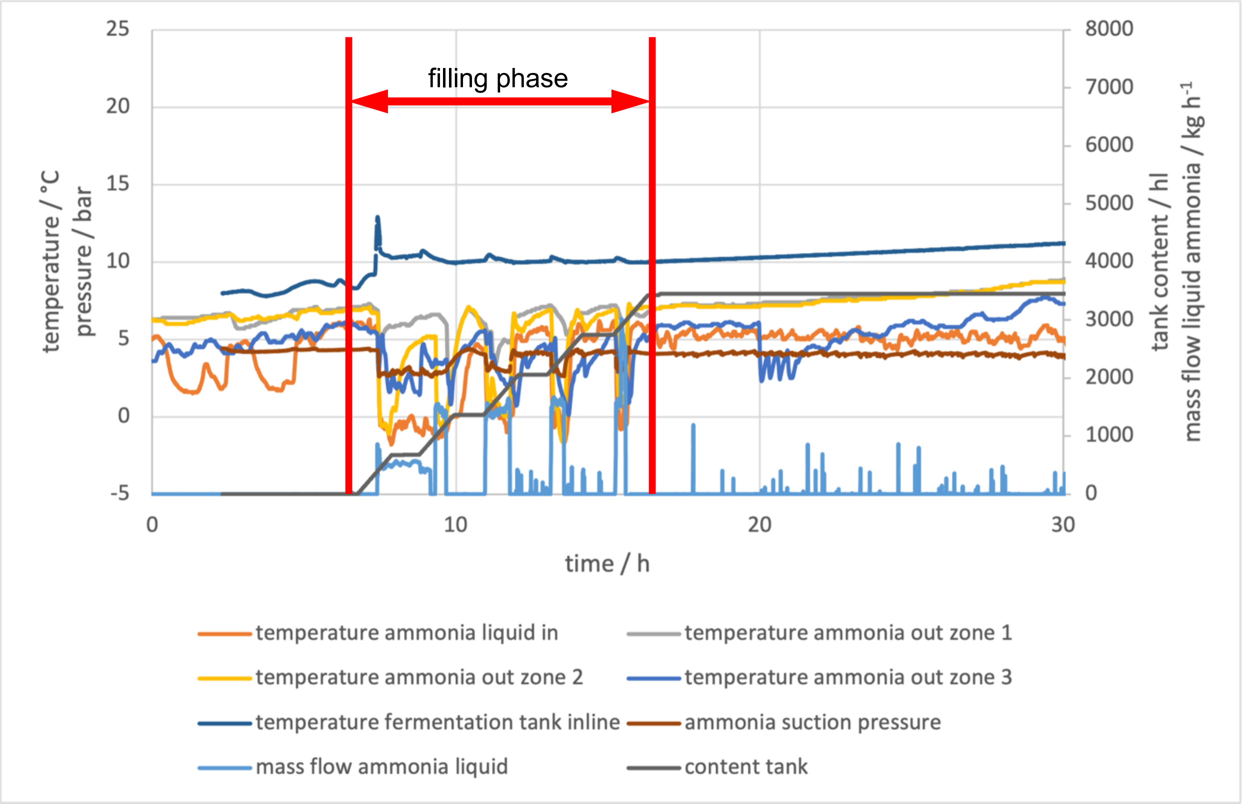

fig. 36: Fermentation tank 17 – filling (hours 0 to 30); source: own representation.

Recording of temperature and cooling performance data

Once the filling phase is finished, the fermentation process begins, as depicted for fermentation tank 17 in fig. 38. This graph shows that when the temperature in the tank reaches approximately 12 °C, the temperature is held within a narrow range to maintain fermentation. This temperature fluctuation around the fermentation point surfaces again in the required liquid ammonia mass flow. The fermentation process creates thermal energy and the ambient also transfers thermal energy into the tank, which needs the tank to be cooled as needed. It is describable by following formulas and pictured in fig. 37:

$$\frac{d(Q)}{dt} = \dot{m}_{in} * h_{in} - \dot{m}_{out} * h_{out} + \dot{Q}_{thermal} - \dot{m}_{refri.} * \Delta h + \dot{Q}_{ferm.} \tag{4.6}$$

With no process fluid filling into or draining from the tank:

$$\dot{m}_{in} * h_{in} - \dot{m}_{out} * h_{out} = 0$$

$$\frac{d(Q)}{dt} = \dot{Q}_{thermal} - \dot{m}_{refri.} * \Delta h + \dot{Q}_{ferm.}$$

As a steady state process with no temperature condition changes:

$$0 = \dot{Q}_{thermal} - \dot{m}_{refrigerant} * \Delta h + \dot{Q}_{fermentation}$$

$\dot{m}_{in}$/kg h^{-1}	mass flow process fluid into the system
$\dot{m}_{out}$/kg h^{-1}	mass flow process fluid out of the system
h_{in}/J	enthalpy of the mass into the system
h_{out}/J	enthalpy of the mass out of the system
$\dot{Q}_{thermal}$/J h^{-1}	ambient thermal energy flow into the system
$\dot{m}_{refr.}$/kg h^{-1}	mass flow refrigerant
Δh /J	enthalpy difference ammonia
$\dot{Q}_{ferm.}$/J h^{-1}	thermal energy flow from fermentation process

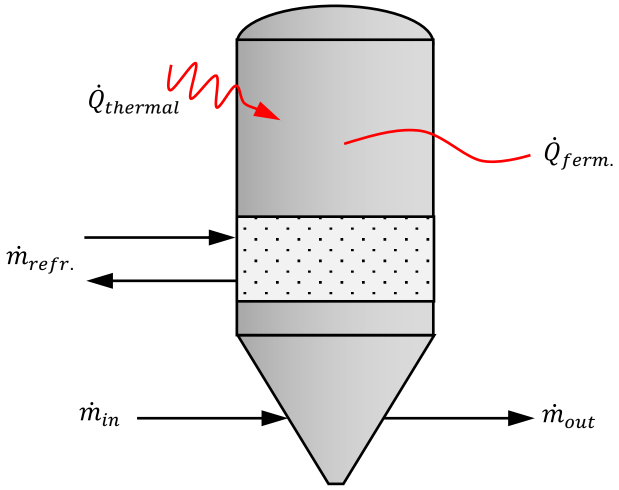

fig. 37: Thermal energy fermentation tank; source: own representation.

Recording of temperature and cooling performance data

Every time the temperature reaches its upper limit, the injection valves of all zones open to cool the tank, which is also visible in the ammonia outlet temperatures. This operates namely in the way of a two-point controller. Between a third and half of the fermentation process, the injection valve of the bottom cooling coat in the cone of the tank is not used anymore. This gets revealed in the higher temperature of the ammonia outlet zone 1. When the fermentation finishes, the time interval between the injection valve openings increases because less heat is generated in this period.

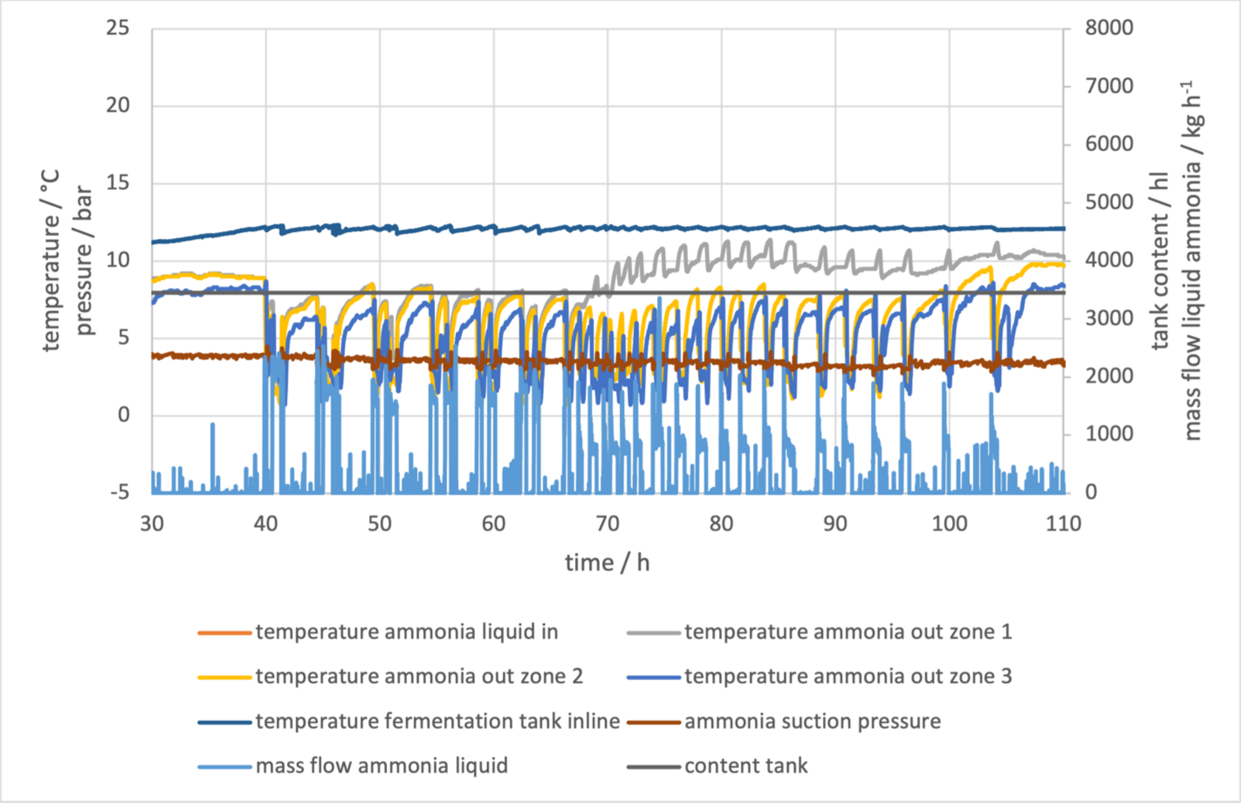

fig. 38: Fermentation tank 17 – fermentation (hours 30 to 110); source: own representation.

Recording of temperature and cooling performance data

The next phase in the beer fermentation process is the phase of maturation, displayed in fig. 39. Here only a small demand of refrigerant is needed because the main fermentation process already stopped. Refrigerant is only needed to keep the tank content on a temperature fluctuating around 12 °C as setpoint. Recognisable is, that only cooling zones 2 to 4 are in operation.

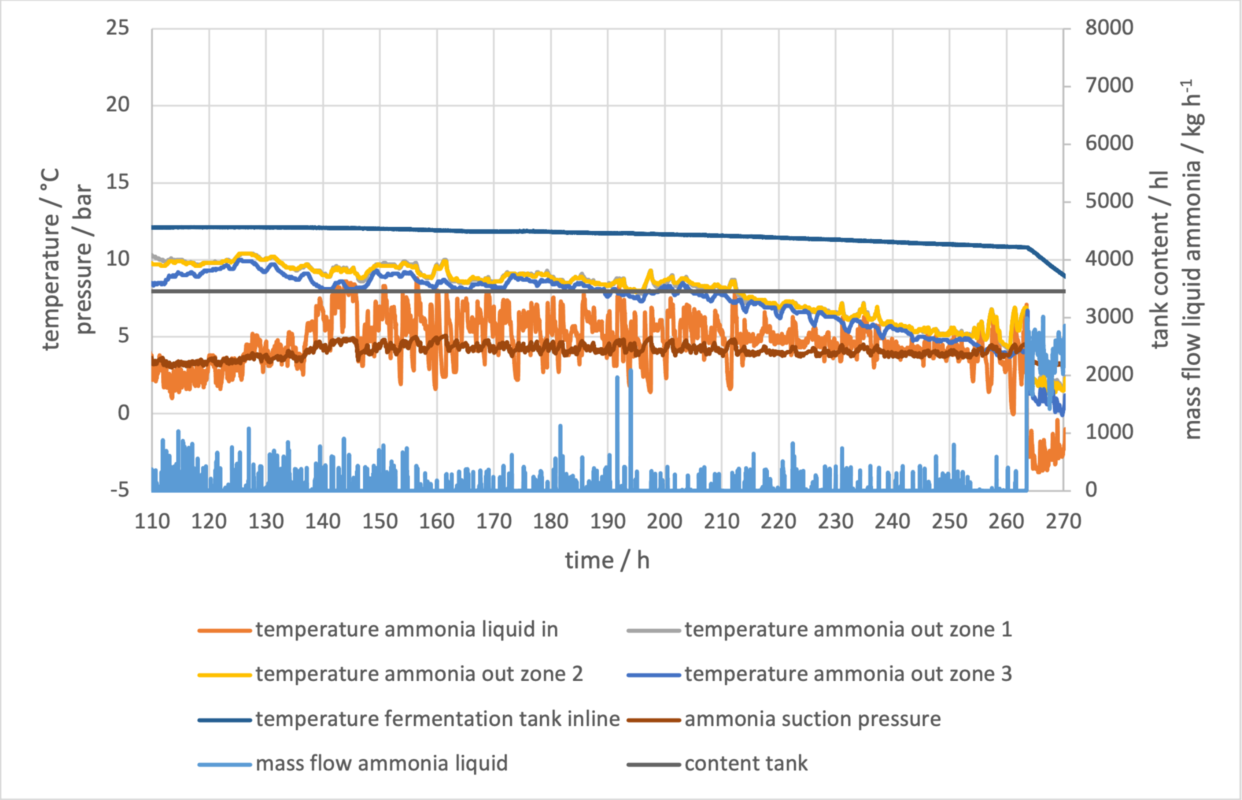

fig. 39: Fermentation tank 17 – maturation (hours 110 to 270); source: own representation.

Recording of temperature and cooling performance data

The last step is the cooling and draining phase, where the fermentation and maturation process is halted by cooling the beer to approximately 5 °C, and the fermentation is stopped biochemical. Therefore, this phase involves the greatest amount of liquid ammonia of any phase, which is used for cooling, as depicted in fig. 40, where the last hours of recordings of the beer in the tank were visualised. Once the liquid has been in the tank for around 300 hours, the tank is drained, and the beer is transferred to pressure tanks to store. As draining begins, the inceptor valves remain closed because no more cooling is necessary in this phase.

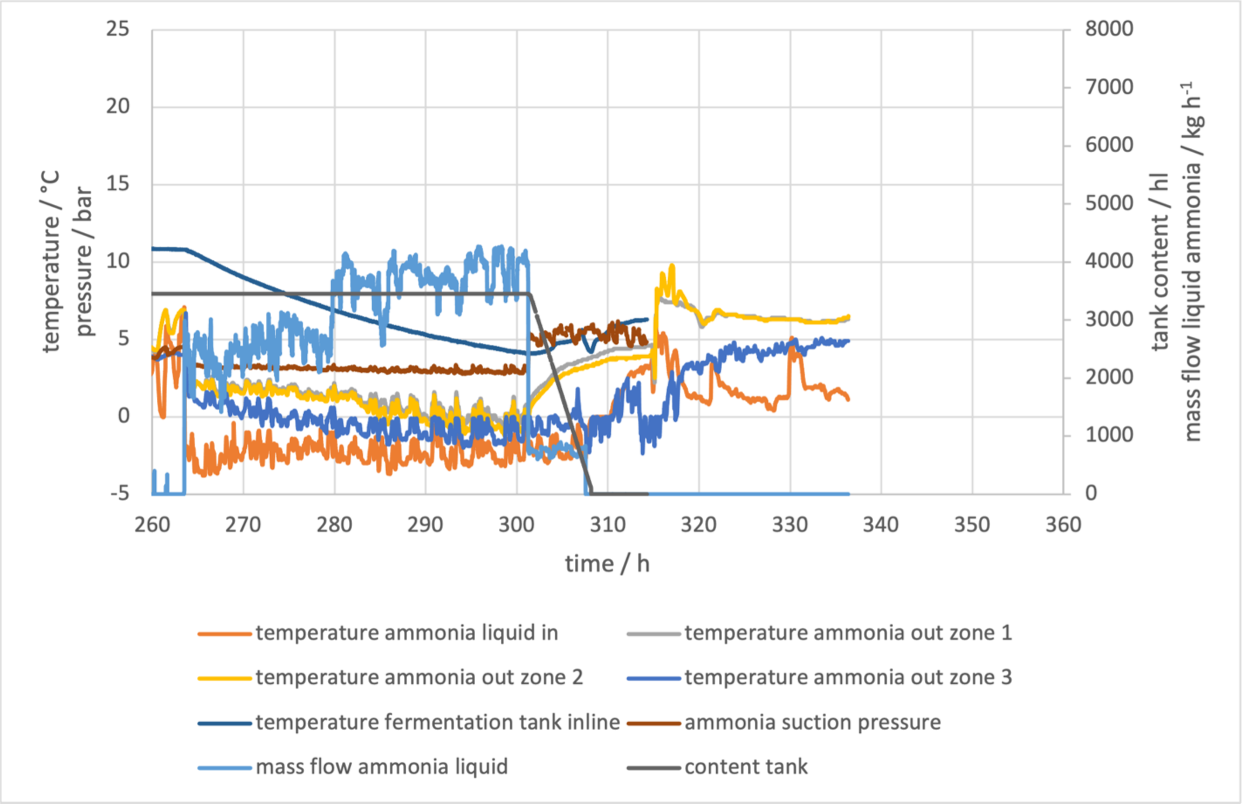

fig. 40: Fermentation tank 17 – cooling and draining (hours 260 to 360); source: own representation.

Recording of temperature and cooling performance data

With the evaporation operating at a defined temperature, caused by the pressure control, following energy balance is valid:

$$\dot{m}_{NH3} * \Delta H = \dot{Q}_{system} = \dot{Q}_{thermal} + \dot{Q}_{fermentation} - \frac{d(V * \rho * c * T)}{dt} \qquad (4.7)$$

$$\dot{Q}_{in,total} = \dot{Q}_{thermal} + \dot{Q}_{fermentation}$$

With creating a system coefficient $k_{(temp)}$:

$$\dot{m}_{NH3} * \Delta H = A * \Delta T * k_{(temp)}$$

$$A * \Delta T * k_{(temp)} = \dot{Q}_{in,total} - \frac{d(V * \rho * c * T)}{dt}$$

With following expression, a performance number *PN* can be created:

$$\dot{Q}_{in,total} = const.$$

$$\Delta T = T_{system} - T_{evaporation}$$

$$\left(T_{system} - T_{evaporation}\right) * \left(A * k_{(temp)}\right) = \dot{Q}_{in,total} - V * \rho * c \frac{d(T)}{dt}$$

$$PN = \left(A * k_{(temp)}\right) = \frac{\dot{Q}_{in,total} - V * \rho * c \frac{d(T)}{dt}}{\left(T_{system} - T_{evaporation}\right)}$$

As mentioned in this chapter's introduction, the whole process may take more time. Depending on the day of the week and time there may be a break between the time of reaching maturation stops temperature and main cooling starting time. At this point, the draining can begin delayed, as later displayed on the graphs and description of fermentation tank 19.

Next, fig. 41 outlines one cycle of fermentation tank 19. This type of tank is different than one like fermentation tank 17 in aspects such as the amount of liquid ammonia the tank consumes. As explained earlier, the process is divided into four main steps. However, a power failure occurred on one weekend during this study, so some measuring points and data are missing.

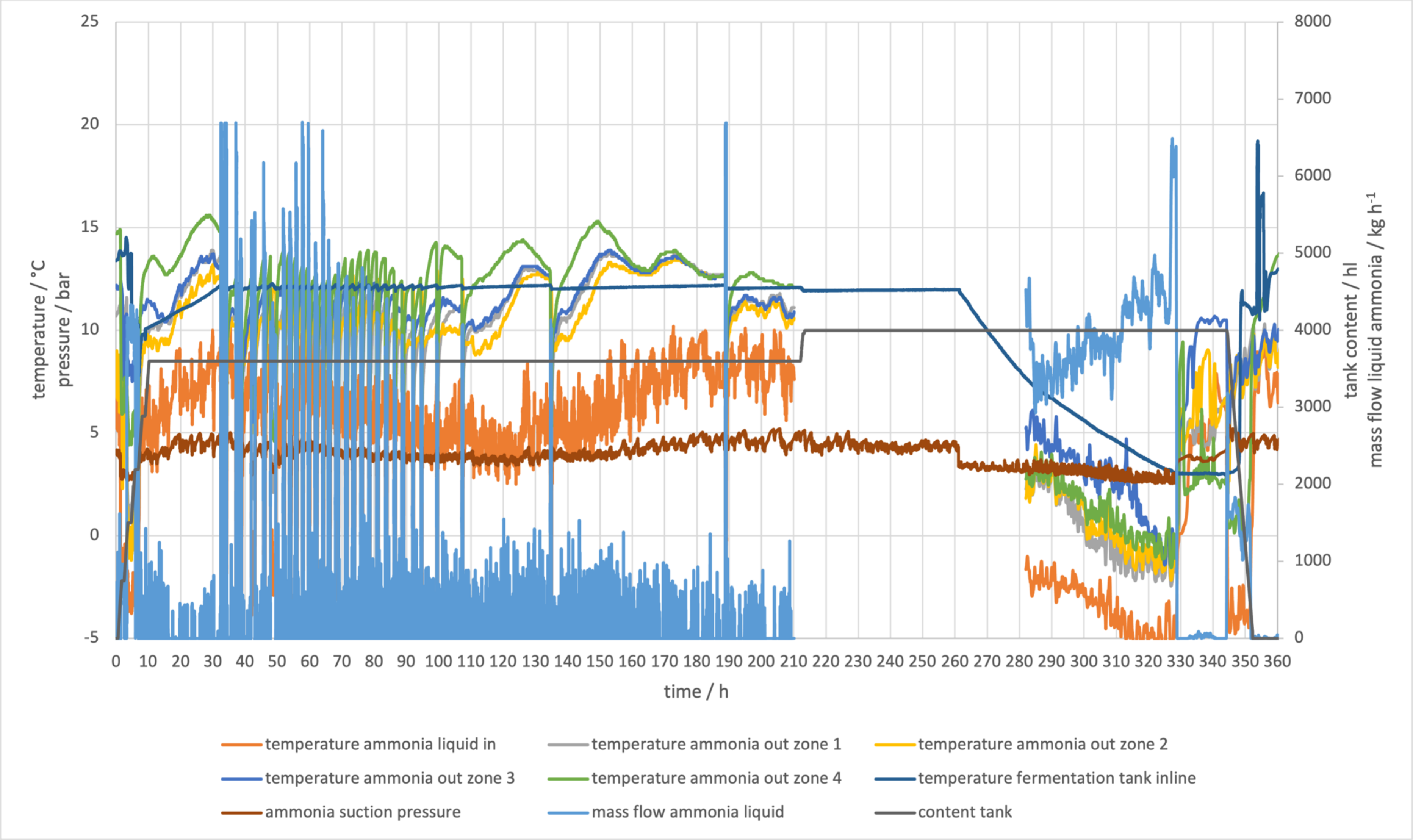

fig. 41: Fermentation tank 19 – overview (hours 0 to 360); source: own representation.

Recording of temperature and cooling performance data

Like tank 17, this tank is cooled until it is filled with the required brews, as shown in fig. 42. Subsequently, the temperature rises constantly slow to the main fermentation temperature, lowering ammonia demand after the filling.

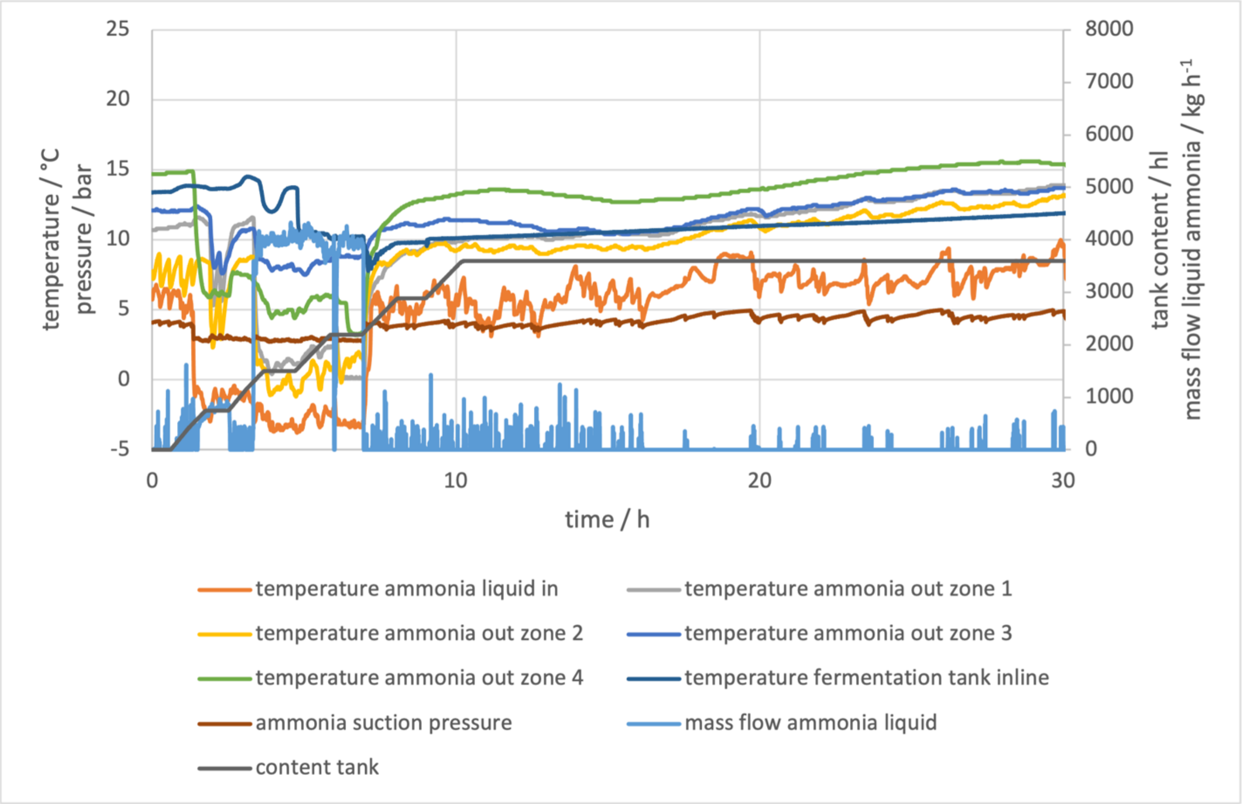

fig. 42: Fermentation tank 19 – filling (hours 0 to 30); source: own representation.

Recording of temperature and cooling performance data

Again, after the tank is filled completely and the fermentation temperature is reached, the main fermentation begins. As before the temperature range around the operating point of fermentation is very small, which produces short intervals in injection valve openings and a sufficient amount of liquid ammonia usage, as displayed in fig. 43. Additionally, the graph shows far higher but also less wider peaks in ammonia mass flow, compared to the recordings of fermentation tank 17.

Even with these differences, the analogousness in the increase of the time interval between valve opening and liquid ammonia consumption of both tanks occurs, as depicted in lower mass flow and the interval of valve openings extend until the end of fermentation.

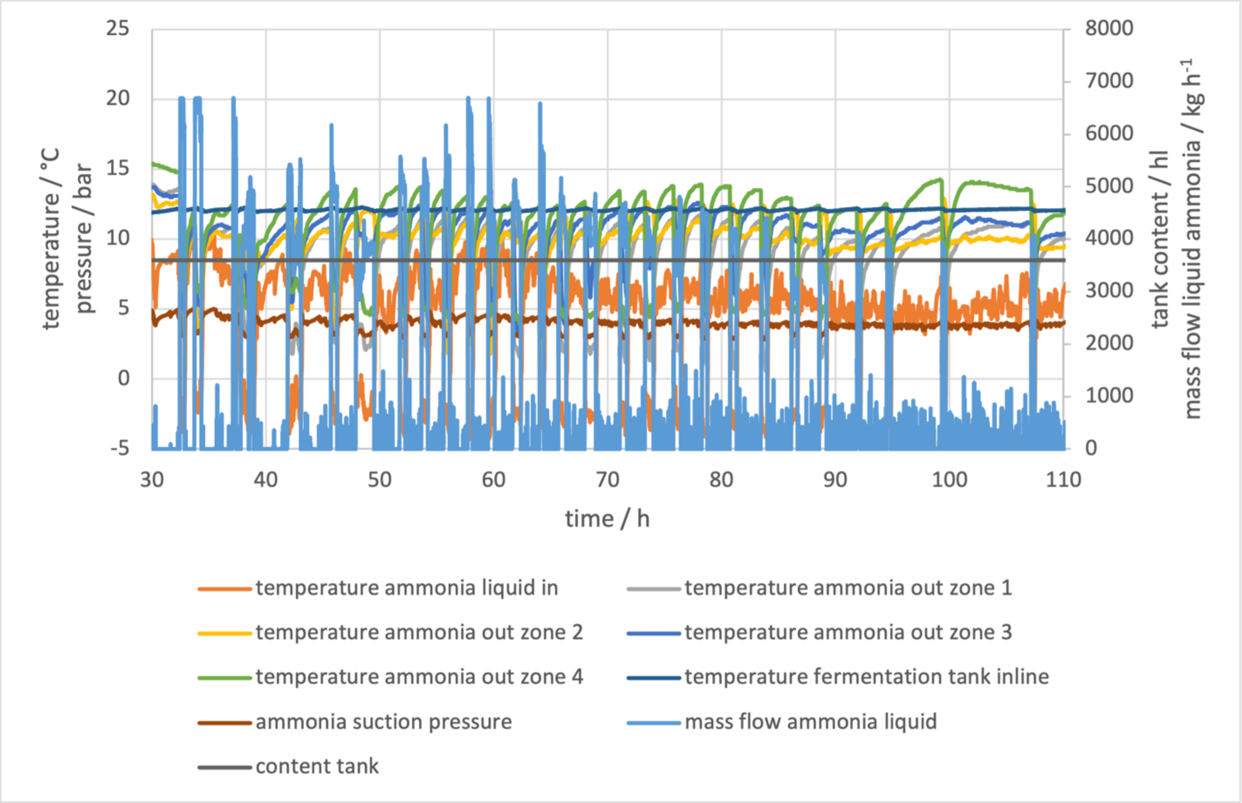

fig. 43: Fermentation tank 19 – fermentation (hours 30 to 110); source: own representation.

Recording of temperature and cooling performance data

During the maturation process, visualised in fig. 44, no significant differences are recognisable between tanks 17 and 19. There is a small but nearly constant amount of liquid ammonia usage until the point of power failure and the data loss around the 210th hour. At this point, the temperature is held at approximately 12 °C before the main cooling to halt fermentation and maturation and the draining of the tank.

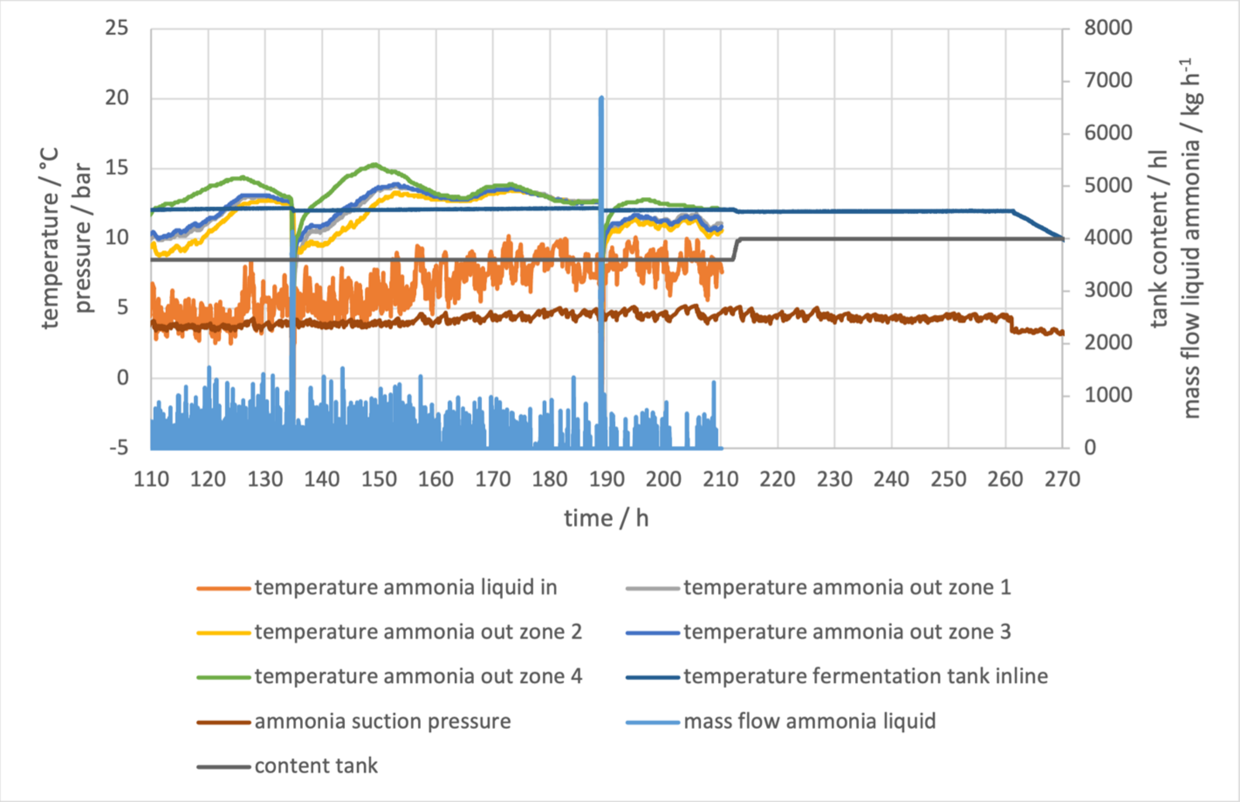

fig. 44: Fermentation tank 19 – maturation (hours 110 to 270); source: own representation.

In the 215th hour of the recording, the tank level had risen to nearly 4000 hl.

As before in tank 17, the main cooling occurs at the highest point of liquid ammonia consumption. As mentioned earlier, the tank is cooled to about 5 °C and is then maintained until the draining begins. At this recording, the time of reaching draining temperature was to a time, where the draining could not begin. Therefore, the beer inside the tank was retained for another 15 hours before pumping it into a pressure tank.

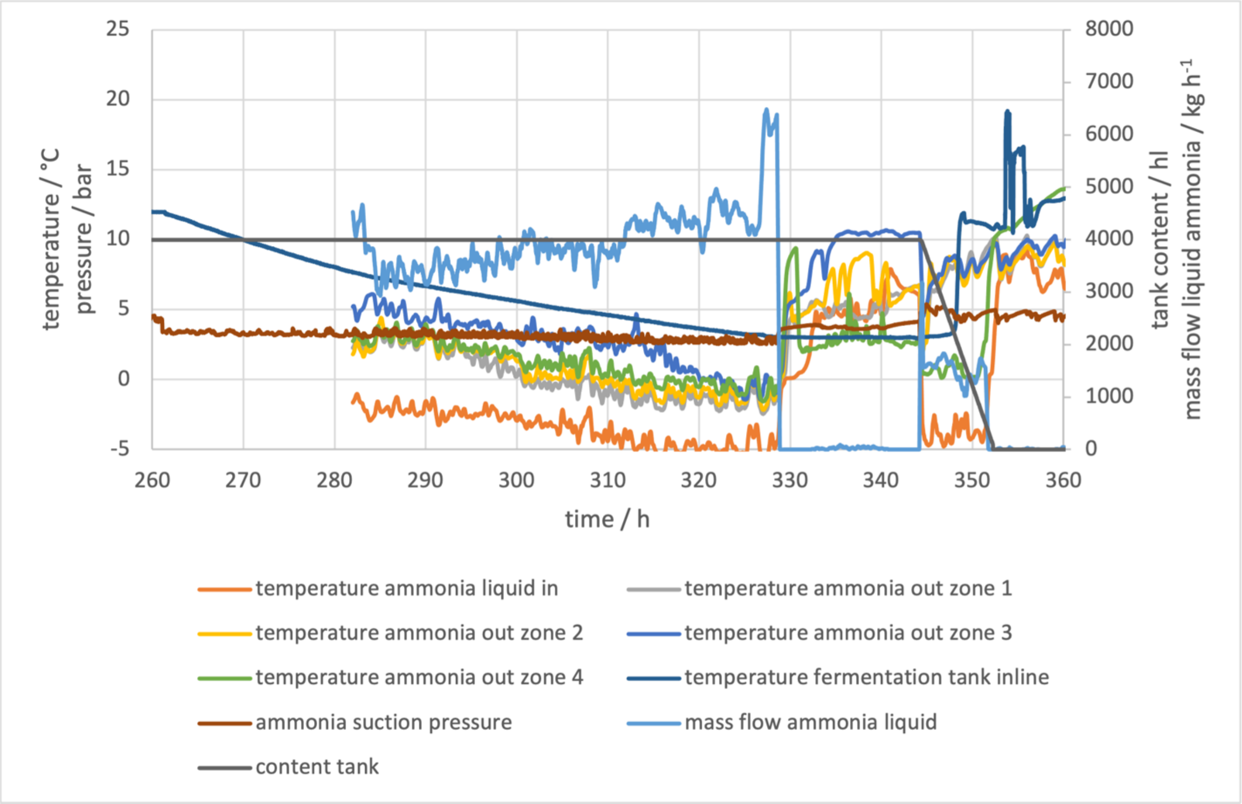

fig. 45: Fermentation tank 19 – cooling and draining (hours 260 to 360); source: own representation.

4.2 External beer cooler

The beer cooler works in a batch process, in which every main process takes about seven to eight hours, as displayed in fig. 46, an overview of 60 hours of recording. According to insulation and fittings in the liquid ammonia distribution pipe, only the mass flow of beer was recorded because there was no opportunity to mount the ultrasonic flowmeter on the liquid ammonia pipe. In this study, recording of the mass flow was accomplished by measuring the volume flow and recalculating the mass flow using the beer density, which closely resembles the density of water. From hours 26 to 29, the cooler was cleaned, visible in the graph by the high temperatures, which are even out of scale. Afterwards not used until hour 39 for a short batch process and hours 46 to 54 for a standard cooling cycle.

After the overview, data extracts again showed individual batch processes.

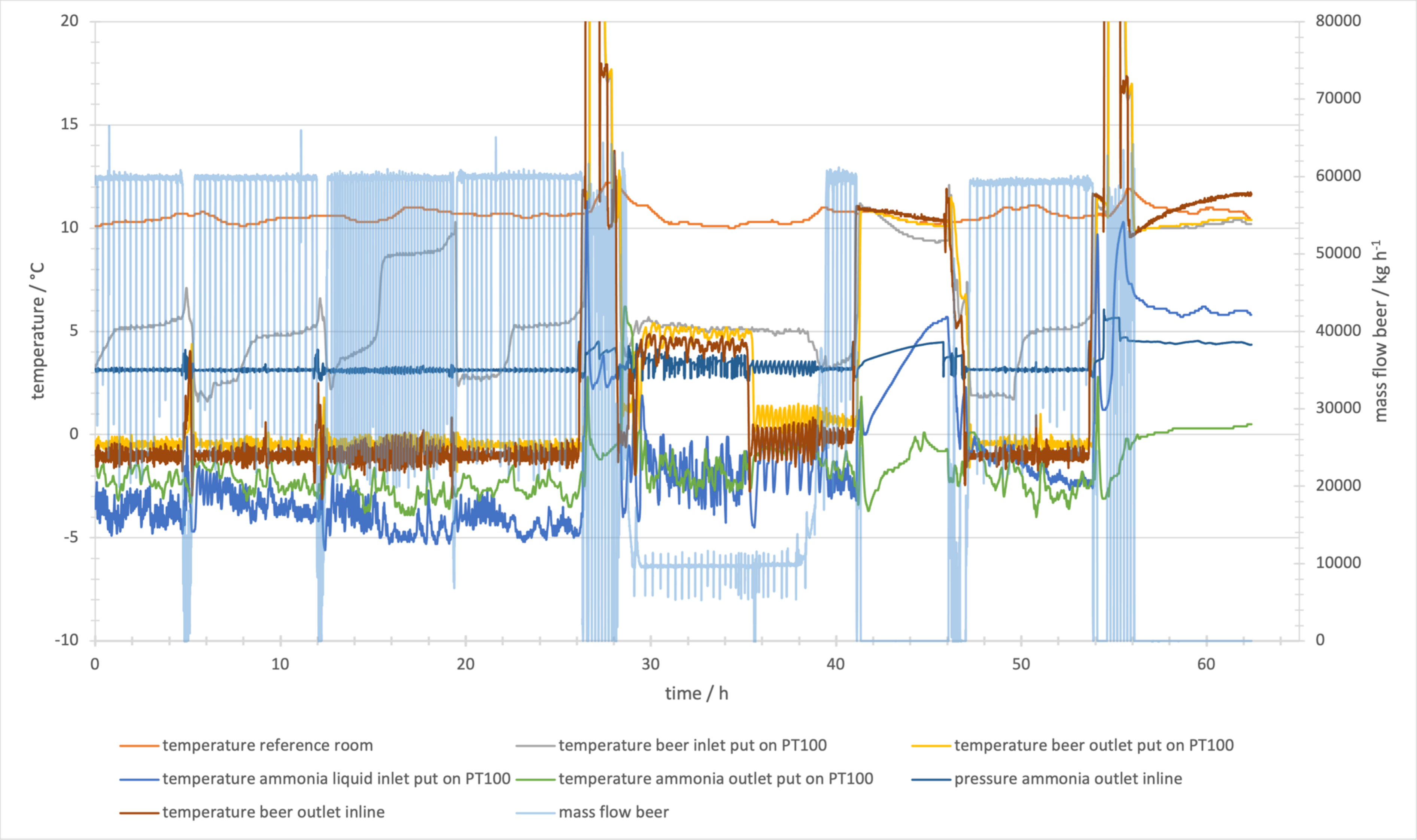

fig. 46: Beer cooler – overview; source: own representation.

Next, fig. 47 displays one batch process of the beer cooler, which demonstrates that the main goal of the cooler is to reach a defined outlet temperature. The mass flow and outlet temperature of the beer as well as also the suction pressure on the ammonia outlet are nearly constant.

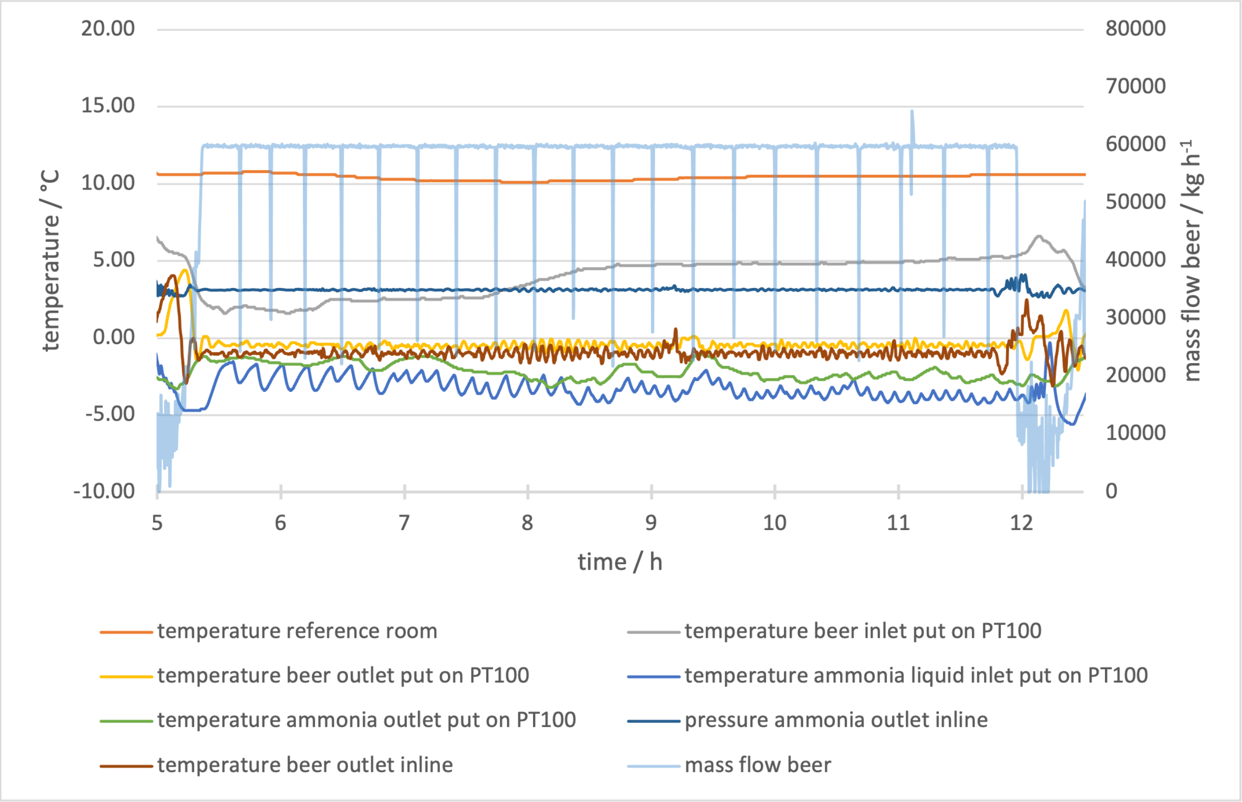

fig. 47: Beer cooler – hours 5 to 12.5; source: own representation.

The next graph, fig. 48 shows two cycles of a beer cooling process, followed by fig. 49, which displays a single procedure. Both graphs show that one main process takes about six to seven hours. The main aim of the external beer cooler is to keep the outline beer temperature constant, while the mass flow of beer is constant too. So, a certain liquid ammonia demand could be calculated using the specific data of the shell and tube cooler.

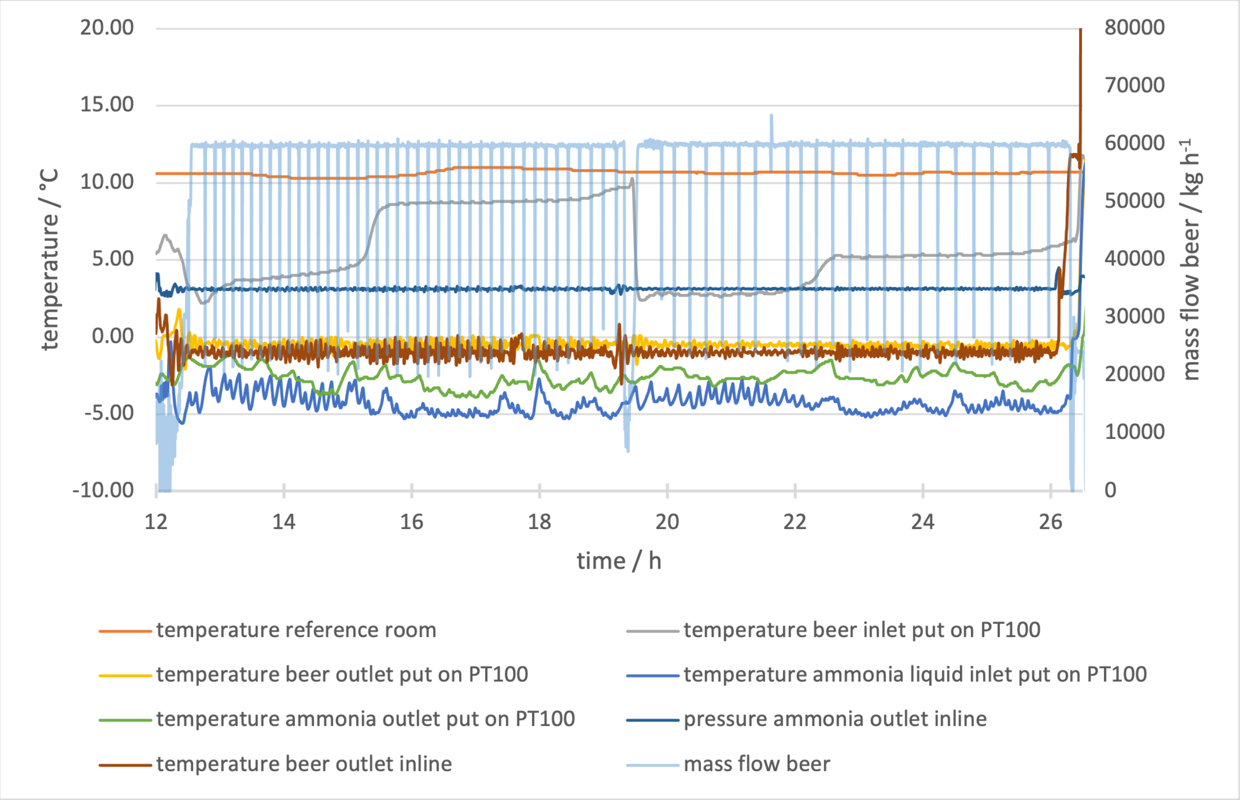

fig. 48: Beer cooler – hours 12–26.5; source: own representation.

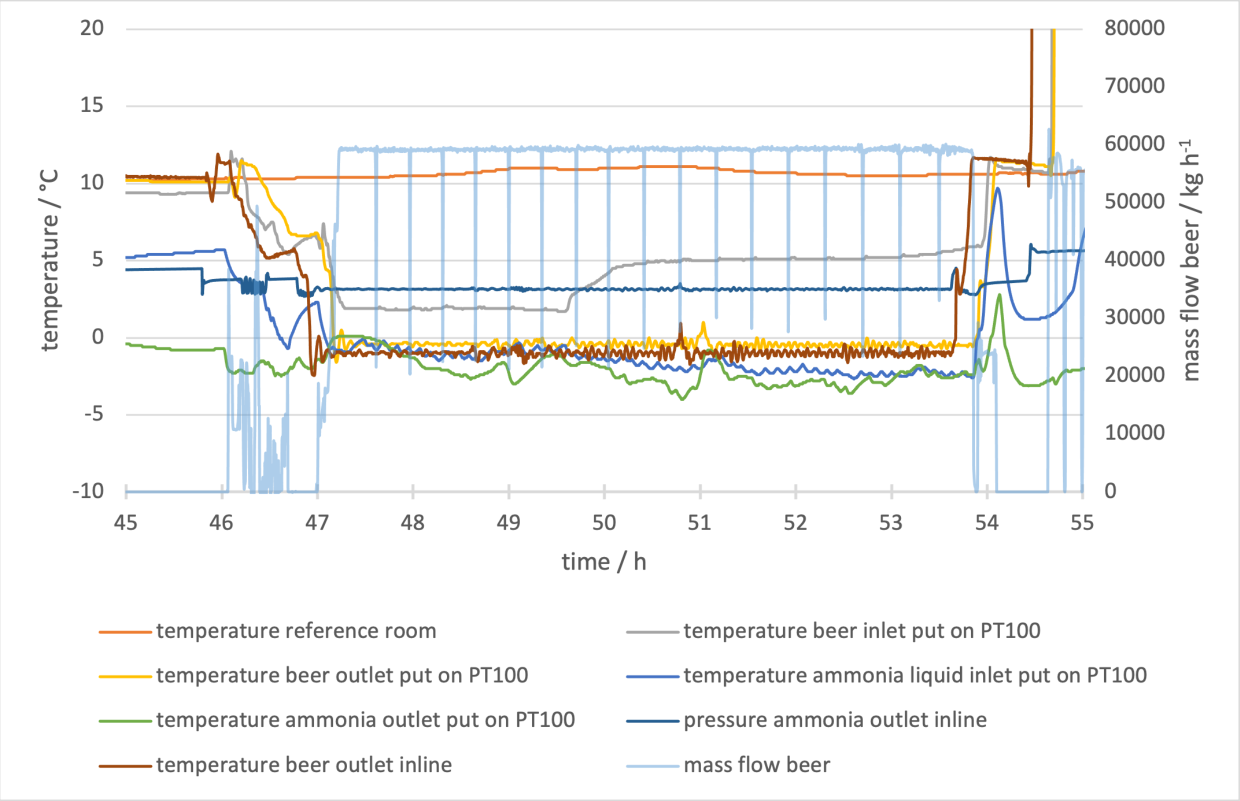

fig. 49: Beer cooler – hours 45 to 55; source: own representation.

4.3 Brew water cooler

Two measurements were conducted for the brew water cooler because only one volume flow measuring device was available and because the cooler worked every time in a comparable batched process. First, the liquid ammonia mass flow was recorded, as demonstrated in fig. 50, and then the measurements of the water mass flow were recorded, as depicted in fig. 51.

Next, the data from both measurements were edited and prepared for comparison to reveal correlations between the two mass flows, as shown in fig. 52, which displays both mass flows overlapped.

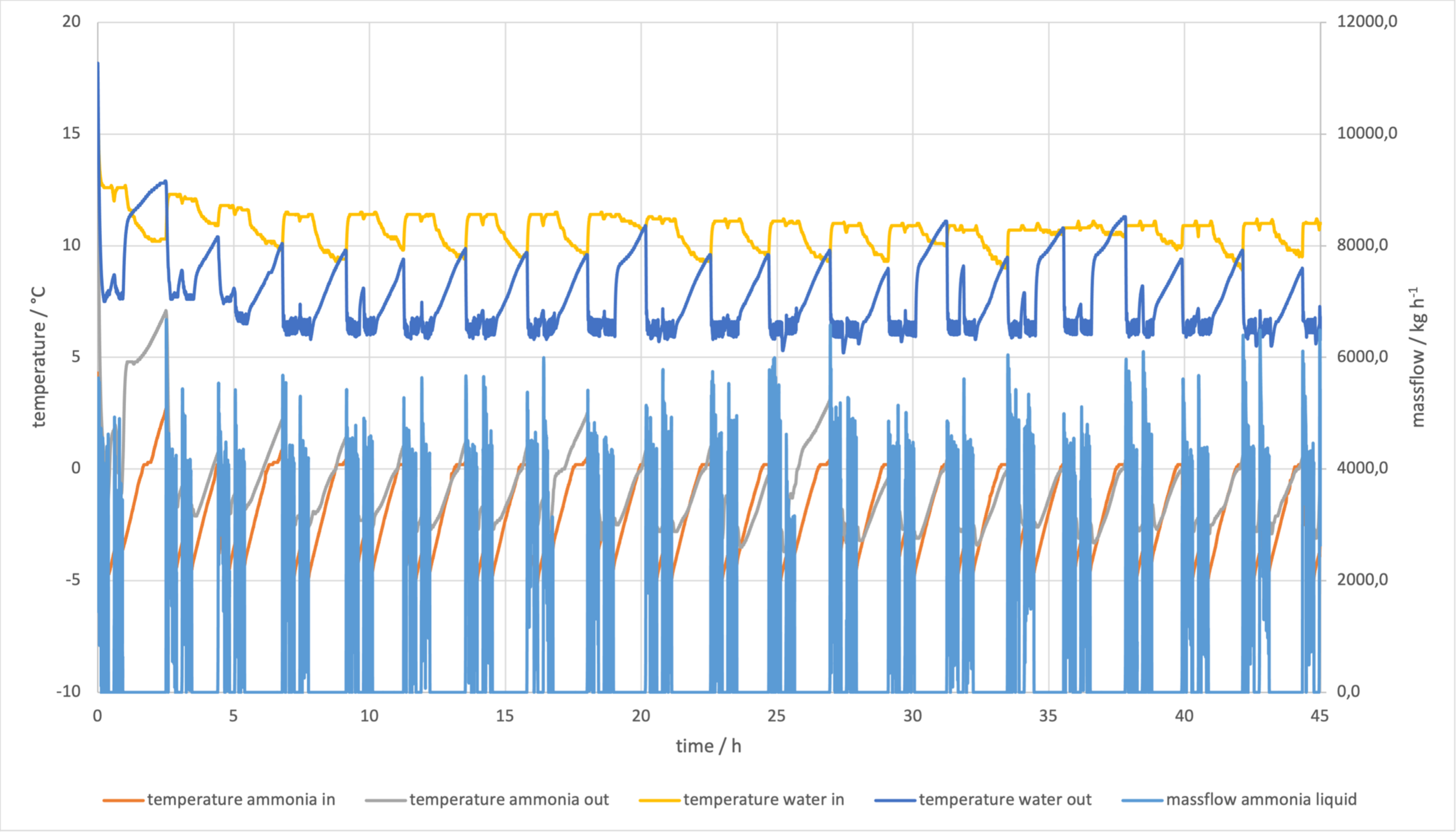

fig. 50: Brew water cooler ammonia mass flow measuring – overview; source: own representation.

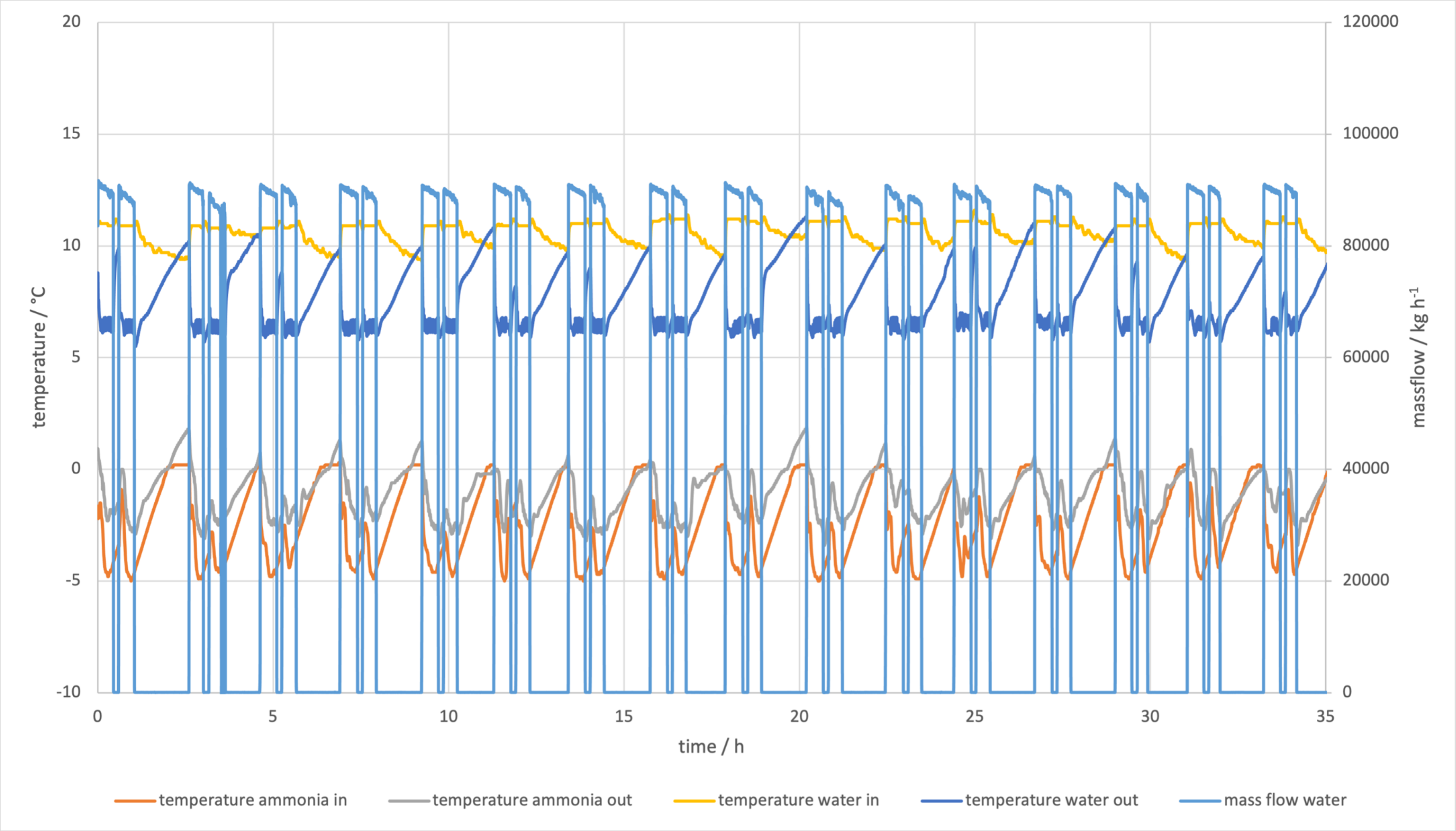

fig. 51: Brew water cooler water mass flow measuring – overview; source: own representation.

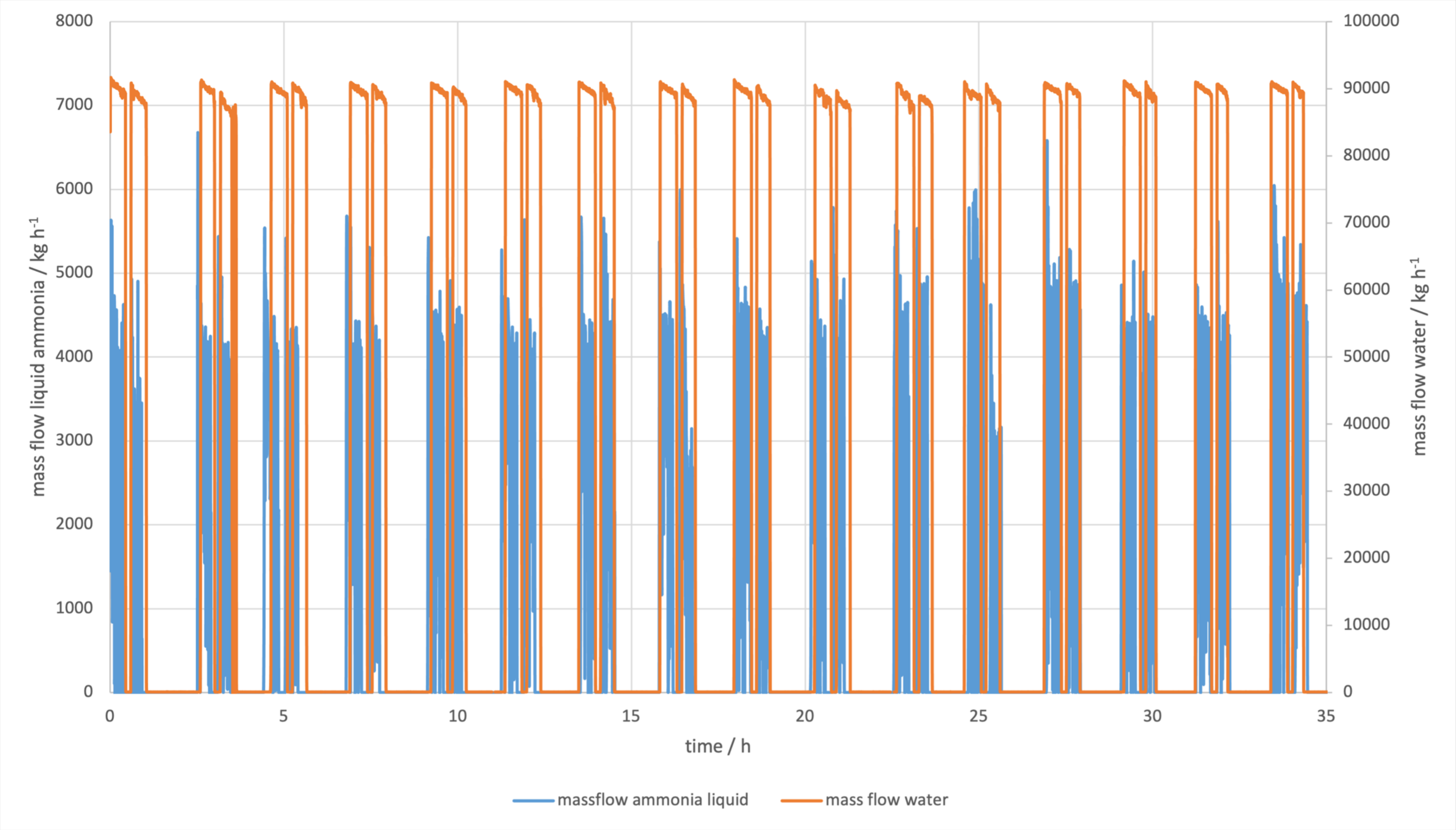

fig. 52: Brew water cooler comparison water and ammonia mass flow – overview; source: own representation.

As fig. 52 demonstrates, the mass flows of water and liquid ammonia generally follow a standardised batch process. More detailed information appears in fig. 53, which shows two of these procedures.

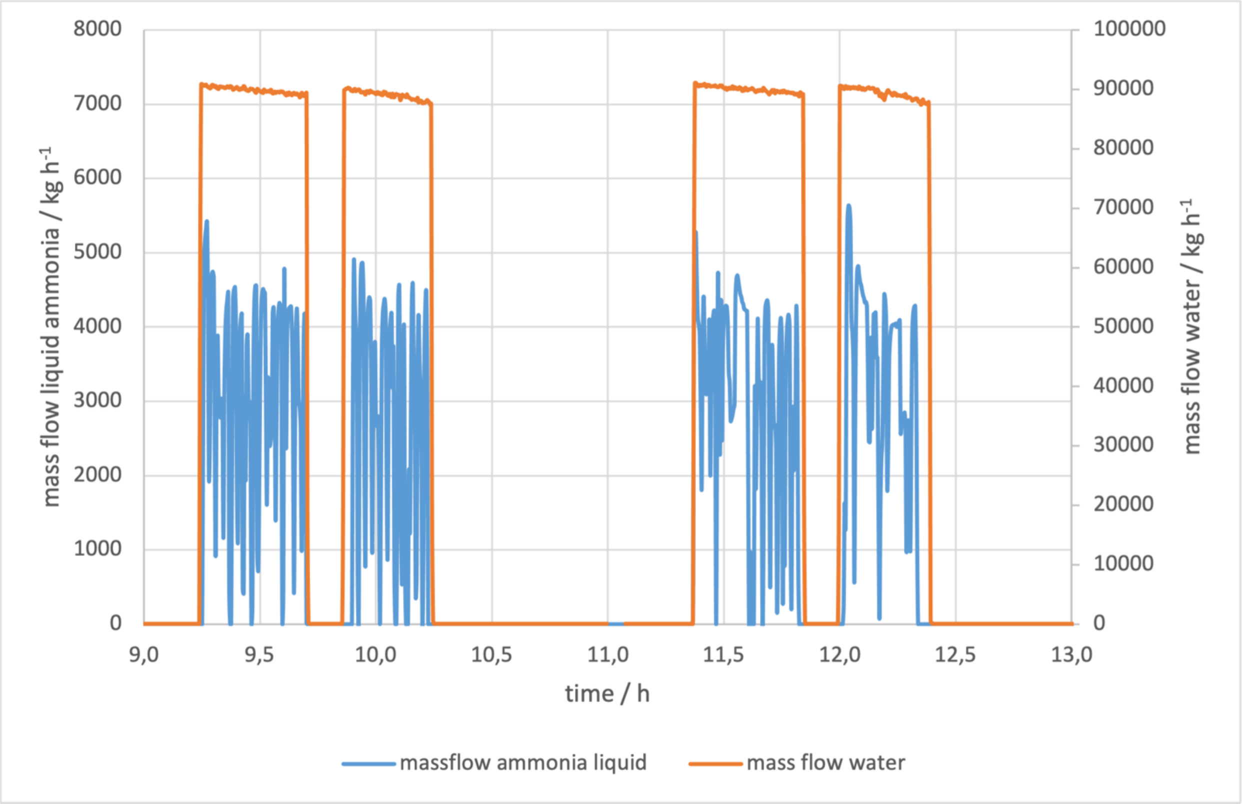

fig. 53: Brew water cooler comparison extraction of two batch processes; source: own representation.

The mass flow of liquid ammonia alternates in every recording from 4000 to 5000 kg h^{-1} by a nearly constant water mass flow of 90000 kg h^{-1} and continuous water outlet temperatures around 6 °C. In fig. 54 a control scheme of the operation is shown, where the ammonia flow first is controlled by the outlet temperature of the water to get a steady water mass flow and second controlled by suction pressure. The process itself is always divided into two parts, where both fluid flows getting stopped and restarted after a few minutes.

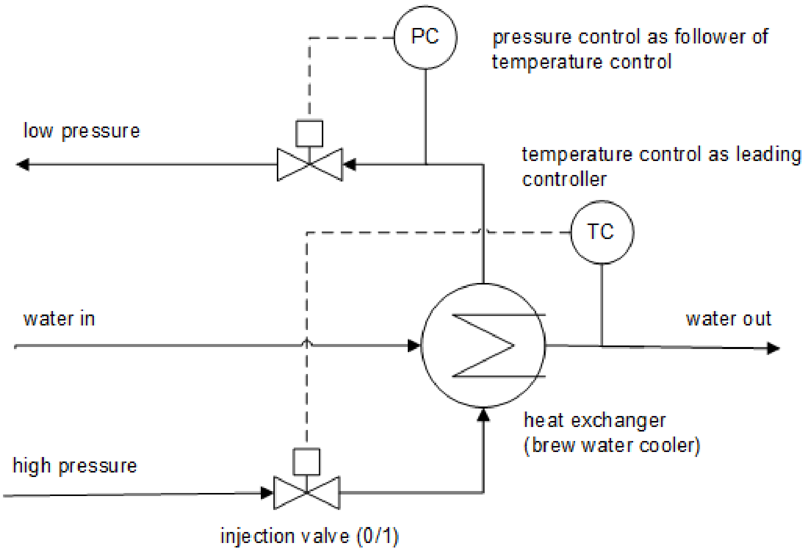

fig. 54: Brew water cooler - cascadic control scheme; source: own representation.

5 MODELLING

In this chapter, the recorded data was edited to create a useful representative forecast of the required liquid ammonia amount to optimise the plant's electrical energy usage and water consumption. The modelling itself provides an abstracted view and should be checked for feasibility, as explained in Section 5.4.

Modelling is the process of abstracting the measured values of the recorded data. The abstraction is necessary caused by at least one of the following lists:

- Data quality on which the model is based:
    - Unsteady wall thickness of pipes at measurements,
    - Injector valves opening and closing disturb flow, and
    - Unavailability to mount measurement devices.
- Complexity degree of the model:
    - Comparability and generalising of processes, and
    - Simplification of complex relationships.

5.1 Fermentation tanks

As previously explained, fermentation tanks must be used for various stages, from filling to draining. These different phases require a different amount of liquid ammonia as coolant necessary. On the other hand, the tanks are not identical; for instance, some of them have three cooling coats or are not insulated, but for that fitted with a ceiling cooler. Other tanks have four cooling coats and are insulated on the outside. Consequently, the tanks must be divided and treated in different ways. Fermentation tank 17, which is fitted with three dimple-jackets, has lower liquid ammonia mass flow rates than the flow rates at fermentation tank 19.

In the filling phase, the tanks must stabilise and hold their temperatures constant at a defined level.

$$Q = \Delta m \, \Delta h|_{refrigerant} = V_{filling} \, \rho \, c \, \Delta T_{filling} \qquad (5.1)$$

However, a detailed forecast of needed liquid ammonia amount cannot be given. While the fermentation is occurring, the needed cooling capacity increases, so the needed mass flow is again dependent on temperature. The difference between the two types of tanks is that the tank 19 type has, compared to tank 17, far higher peaks of 6,500 kg h^{-1}, decreasing steadily to 4,000 kg h^{-1} of liquid ammonia after the main fermentation. In contrast, tank 17 has steady peaks of liquid ammonia flow at a flow rate of 2,000 kg h^{-1}. Later, at the time of maturation, the mass flow rates and numbers of peaks in both tanks decrease. The signal noise of the measurement and short injector valve openings cause the seemingly steady mass flow of ammonia, but according to the temperatures of ammonia inlet, there was no mass flow.

The last step before the draining of the tank is to cool the beer. In this phase, the highest refrigerant amount of the entire process is needed. For instance, in this study, fermentation tank 17 began with a variable mass flow of about 2,500 kg h^{-1}, followed by 3,500 kg h^{-1} before the draining began. For tank 19, data is missing for the start point of the draining, but the mass flow also fluctuates from approximately 3,500–4,500 kg h^{-1} before the tank is drained.

Modelling

An abstracted but realistic trend is displayed in fig. 55, where the base amount of coolant needed, and potential peaks are visible. A summary of the above-mentioned values appears in tab. 6 and tab. 7 lists the energy balances throughout the phases.

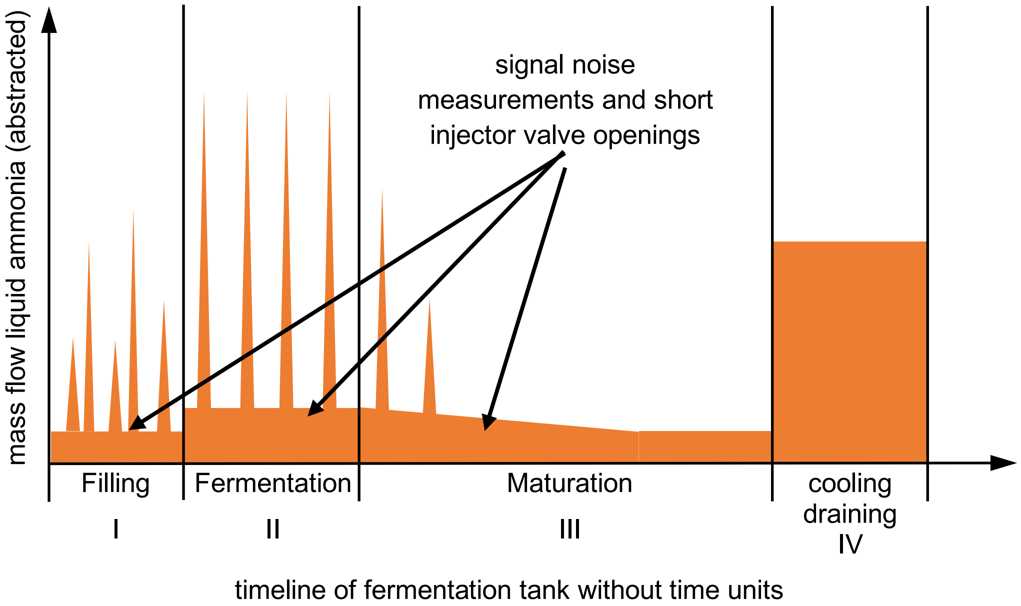

fig. 55: Abstracted trend of liquid ammonia mass flow of the fermentation tanks; source: own representation.

Process step	Tank type	Steady ammonia amount	Peaks ammonia amount	Additional information
		kg h^{-1}	kg h^{-1}	
Filling I	1	-	3000	
	2	-	4000	
Fermentation II	1	-	2000	fewer peaks
	2	-	6500	
Maturation III	1	-	2000	peaks are rare
	2	-	4000-6500	peaks are rare
Cooling / draining IV	1	2500-3500		no peaks, fluctuating mass flow
	2	3500-4500		no peaks, fluctuating mass flow

tab. 6: Summary liquid ammonia flow rates fermentation tanks; source: own representation.

Process step	Description of process and cooling
Filling I	$Q = \Delta m\, \Delta h\vert_{refrigerant} = V_{filling} * \rho * c * \Delta T_{filling}$-
Fermentation II	$\Delta m\, \Delta h\vert_{refrigerant} = V_{filling} * c,$ with c equals specific fermentation heat
Maturation III	Only steady state cooling-
Cooling / draining IV	$\Delta m\, \Delta h\vert_{refrigerant} = V_{filling} * \rho * c * \Delta T_{filling}$

tab. 7: Fermentation process phases with energy balances; source: own representation.

5.2 Beer cooler

The beer cooler has a steady mass flow of 60,000 kg h^{-1} beer in the operating mode. The usage of this apparat is not constant but generally takes about seven to eight hours per batch process, producing 420,000–480,000 kg or 4,200–4,800 hl of beer.

However, the amount of liquid ammonia in this study was not measured because of the non-ability to mount the measuring device. The beer cooler works in a batch process at mostly constant times with a constant flow of beer. Since no technical data, except the contents, is longer available, the liquid ammonia amount can only be calculated approximately, caused to losses, with the energy balance.

$$Q = \Delta m * \Delta h\vert_{refrigerant} = V_{beer} * \rho_{beer} * c_{beer} * \Delta T_{beer}$$

$$\Delta m_{refrigerant} = \frac{m_{beer} * c_{beer} * \Delta T_{beer}}{\Delta h_{refrigerant}}$$

$$\Delta m_{refrigerant} = \frac{480000\, kg * 4.19\, kJ\, kg^{-1}\, K^{-1} * 6\, K}{894.195\ \mathrm{kJ\ kg^{-1}}}$$

$$\Delta m_{refrigerant} = 13495\, kg$$

Purely on the balance sheet, considering only the vapour enthalpy and neglection of losses, 13,495 kg of ammonia would be needed for cooling 480,000 kg of beer by 6 degrees.

5.3 Brew water cooler

According to Section 4.3, the brew water cooler operates in nearly regularly cycles with nearly constant mass flows of liquid ammonia and water. Next, fig. 56 depicts an abstraction of the flows and usage time of two operating cycles.

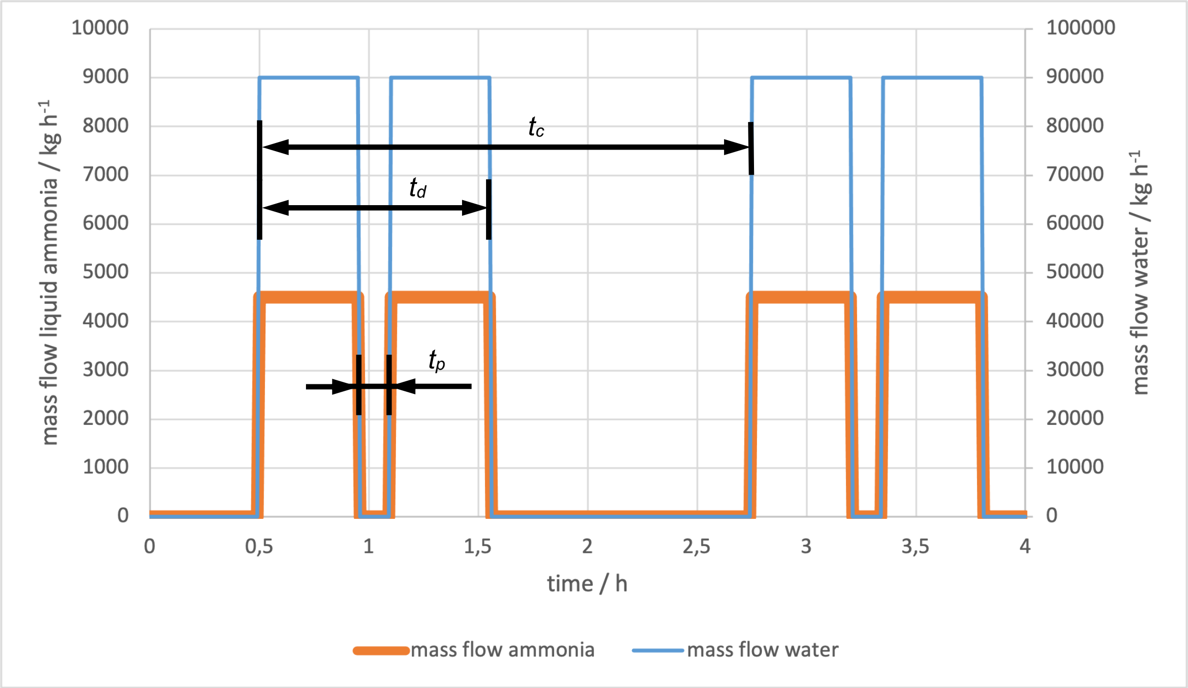

fig. 56: Abstraction beer water cooler mass flows; source: own representation.

As displayed in the illustration above, the average water flow is 90,000 kg h^{-1}, with a required average amount of 4,500 kg h^{-1} liquid ammonia to maintain the temperature of the water outlet at the desired level. Additionally, starting and stopping the heat exchanger is not necessary and not included in this abstraction because, as shown in Section 4.3, the water temperature falls rapidly to the value needed.

With the square wave shaped form of the cycle, the duty cycle can be calculated following:

$$DC_{brew\ water\ cooler} = \frac{t_d - t_p}{t_c} \qquad (5.2)$$

$$DC_{brew\ water\ cooler} = \frac{1.05\ h - 0.15\ h}{2.25\ h}$$

$$DC_{brew\ water\ cooler} = 0.400$$

DC/1	duty cycle of brew water cooler
t_d/h	duty time, operating time
t_p/h	pause time during operating time
t_c/h	cycle time from one to the next start

The duty cycle of 0.4 and the entire cycle time of 2.25 h provide an opportunity to make changes in the operating mode of the brew water cooler. To optimise the usage of liquid ammonia stock and pumping power, the post-cooler-located tank could be emptied more frequently (e.g., twice instead of only once) until it is refilled with cold water from the brew water cooler. This process is depicted in fig. 57, where the content of the tank is illustrated in blue.

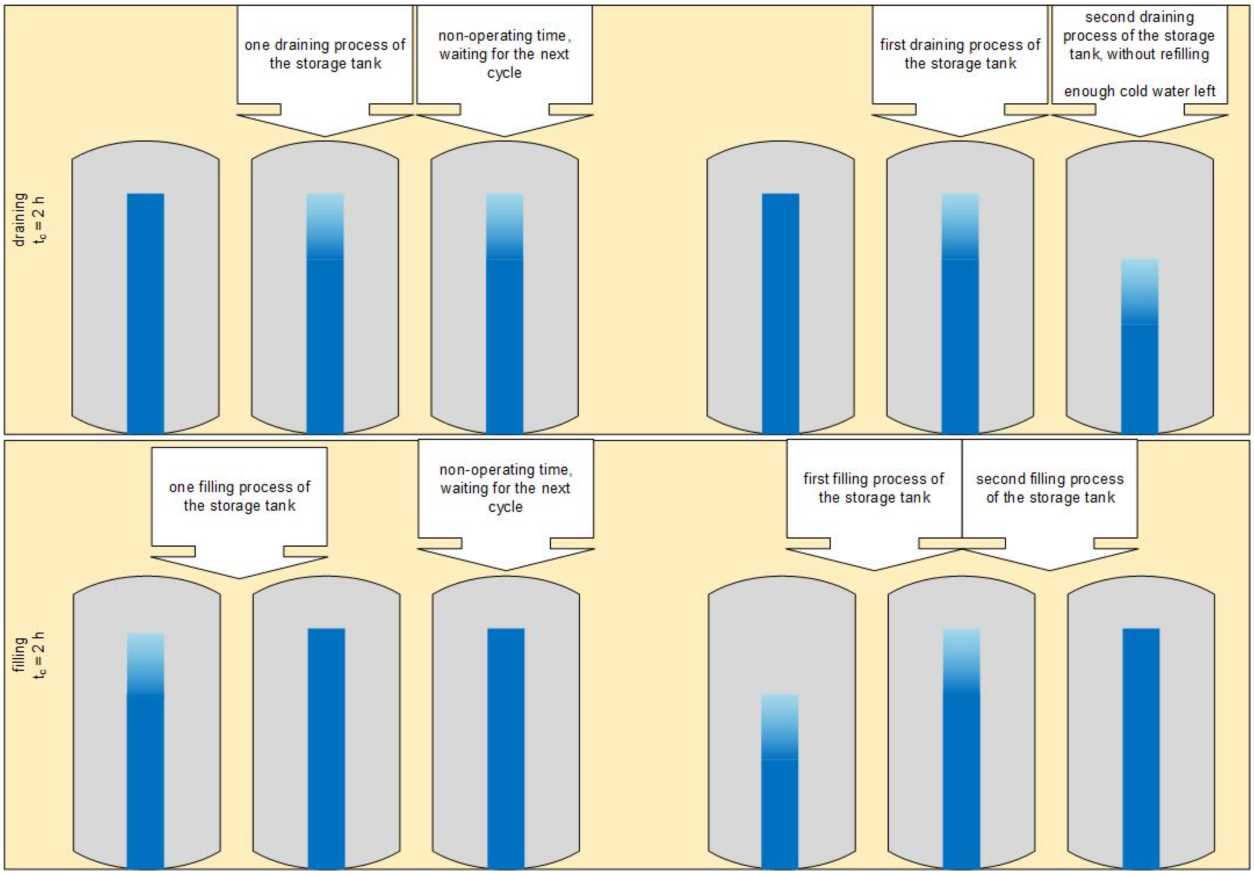

fig. 57: Post-brew water cooler storage tank – one and two processes comparison; source: own representation.

Modelling

For further analyses and optimisation, one must measure the total mass of cooled water, while operating original. This is necessary, because of the need of making operating modes with the same output at the same time. The sum of the cooled water mass is calculated as follows:

$$m_{total} = \dot{m}_{avg} * t_c * DC \qquad (5.3)$$

$$m_{total} = 90000\ kg\ h^{-1} * 2.25\ h * 0.4$$

$$m_{total} = 81000\ kg$$

m_{total}/kg	produced cooled mass of water
$\dot{m}_{avg}$/kg h^{-1}	average mass flow

Therefore, a mass of 81,000 kg of water must be cooled in the time of 2.25 h. For the idea of double filling the cooled brew water storage tank no changes in mass flow of water are present. This constant mass flow is caused by only moving the pauses and non-operating times, as fig. 58 shows. That is, double operating time of the cooler followed by a doubled non-operating time, where the liquid ammonia can be used for other applications for longer time periods. Additionally, the duty cycle would stay nearly constant if the hot wort cooler period remained constant.

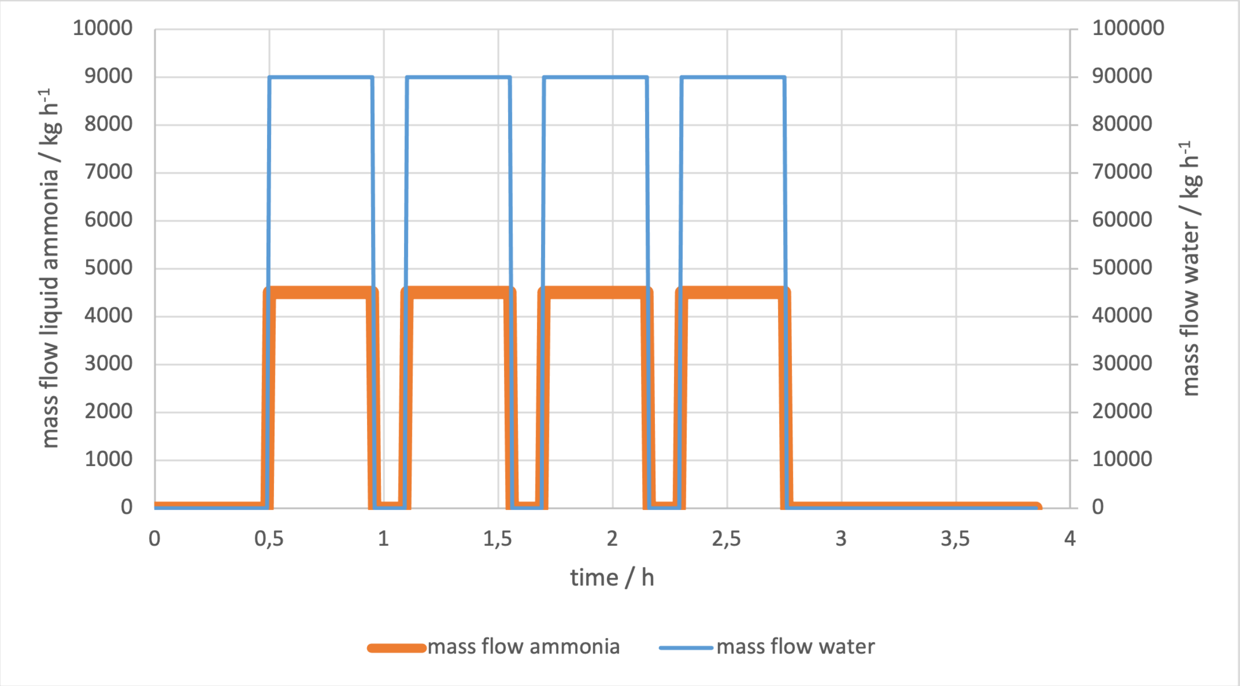

fig. 58. Abstraction brew water cooler with doubled operating times; source: own representation.

The advantage of this possible operation mode improvement is, that it would increase spare time for the usage of coolant medium for other apparatuses. To productively use this spare time, other applications with similar operation modes would have to operate alternately.

On the other hand, one could employ a nearly steadily process for the brew water cooler with only one brief pause or divide the original cycle time evenly into operating times with equal pauses. With the required total mass of water and duty cycle times of these two continuous options pictured in fig. 59 the needed mass flow of the brew water cooler can be calculated.

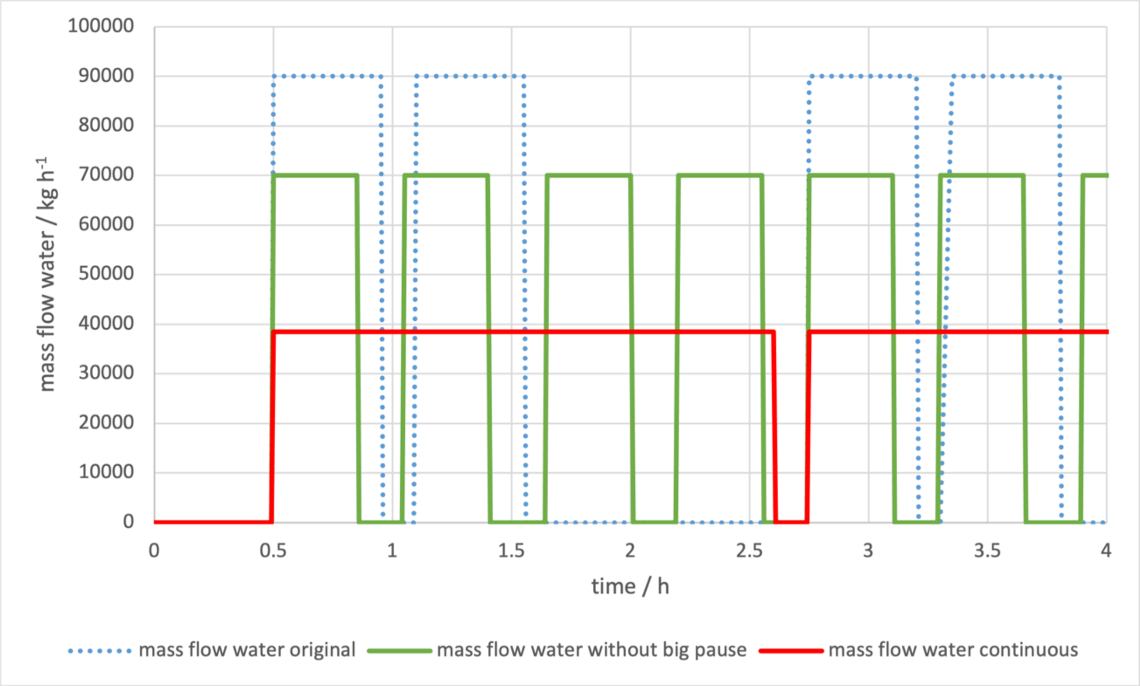

fig. 59: Possible optimisations brew water cooler; source: own representation.

To prepare enough cooled water for the hot wort cooler, one must adjust the mass flows. These values, listed in tab. 8, calculates a converted form of formula 5.2 as follows:

$$\dot{m}_{needed} = \frac{m_{total}}{t_c * DC}$$

Option	*t_p*	*Non-operating time*	*DC*	$\dot{m}_{needed}$
	h	h	1	kg h^{-1}
Original	0.15	1.1	0.400	90000
1	3*0.2 + 0.25	-	0.511	70434
2	0.15	-	0.933	38571

tab. 8: Required mass flows for different operating modes; source: own representation.

While brief pause times, an extended middle pause added, and the non-operating time being skipped completely, the needed mass flow of refrigerant decreases down to approximately 70,000 kg h^{-1}. If the brew water cooler worked in a nearly constant operation mode, the mass flow could be decreased by more than 50%, which would also minimise the need for liquid ammonia.

5.4 Feasibility check

By dividing the fermentation process into four different stages I to IV, one can more closely examine the results. The liquid ammonia mass flow increases, caused by temperature reaching a high limit, to cool the tank content. The phase with the highest consumption is the main cooling phase before the tank gets drained, which seems logical by the fact the temperature of the tank content needs to decrease several degrees. Additionally, the temperature and ammonia amount are always related, as demonstrated in fig. 60 of the maturation phase of fermentation tank 19.

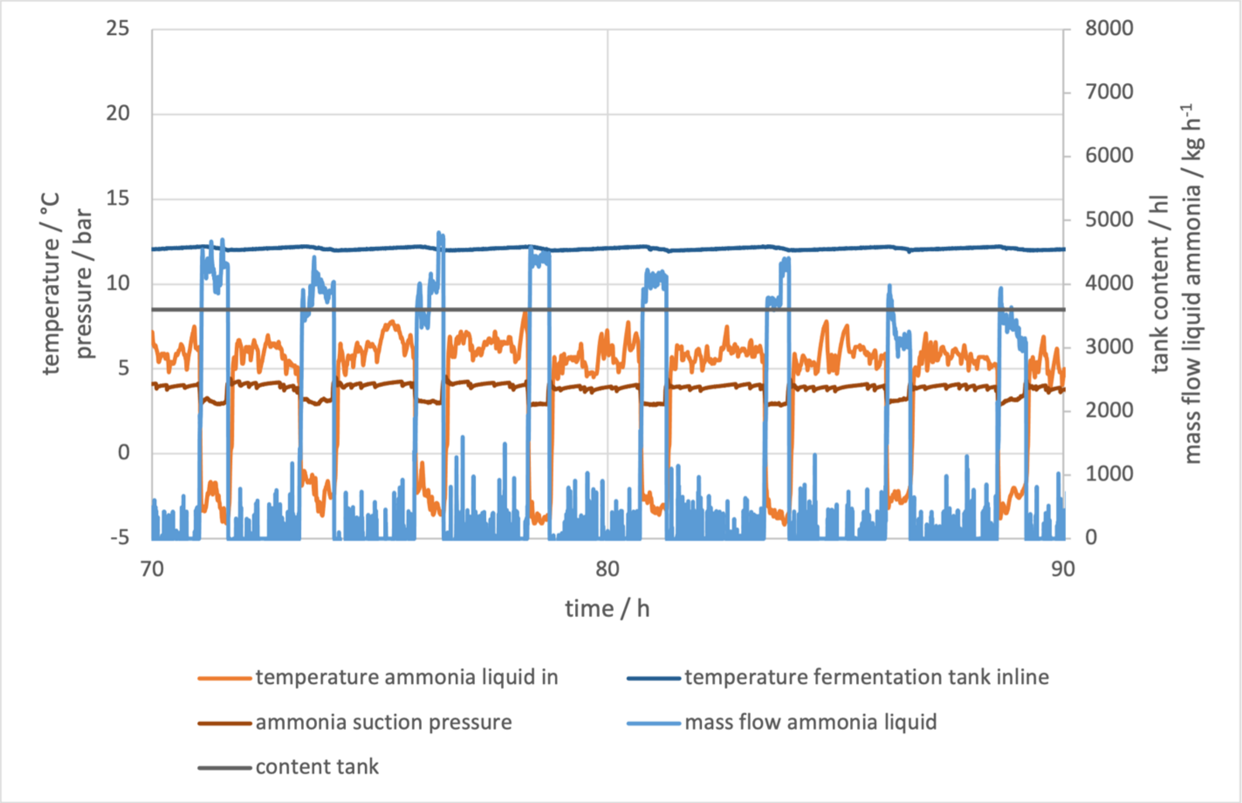

fig. 60: Excerpt maturation phase of fermentation tank 19; source: own representation.

When the tank temperature rises above a certain level, the injector valves open, and liquid ammonia is delivered into the dimple-jackets. Liquid ammonia inlet temperature decreases instantly, and after the temperature of the content reaches the low level required, the valves close and the mass flow stops instantly.

The signal noise of all measurements on the fermentation tanks is caused by incorrectly installing the ultrasonic flow meter in a down-leading liquid ammonia supply pipe. However, there is no other possibility for mounting the flow meters in front of the valve station because the main liquid ammonia distribution is just below the room roof, and supplying pipes run from top to bottom.

In this study, there was no opportunity to measure the liquid ammonia flow rates of the beer cooler, so there was no feasibility check for refrigerant for this batch process and the ammonia demand for one batch process of the beer cooler was calculated with the energy balance.

Modelling

The approach to verifying the results of the brew water cooler was to calculate the heat energy, which must be transmitted from the primary product to the refrigeration medium. To calculate needed physical data of ammonia is needed, especially the specific heat of vaporisation which can be found in tab. 1.

The brew water cooler, in the form of a shell and tube heat exchanger, cools the water by 5 °C and has mass flows of 90000 kg h^{-1} water and 4500 kg h^{-1} ammonia. Beyond this, 1 kg of water has a specific heat capacity of about 4.19 kJ kg^{-1} K^{-1} in case of application. This leads to following needed heat energy per kg to be transmitted to the refrigerant, as demonstrated here:

$$Q_{specific} = c_{water} * \Delta T \qquad (5.4)$$

$$Q_{specific} = 4.19\ kJ\ kg^{-1}\ K^{-1} * 5\ K$$

$$Q_{specific} = 20950\ kJ\ kg^{-1}$$

$Q_{specific}$/kJ kg^{-1}	specific heat energy
c_{water}/kJ kg^{-1} K^{-1}	specific heat content of water
ΔT/K	temperature difference water in and out

The measured and, with ideal conditions considered specific enthalpy differences according to tab. 1, calculated mass flows allowed the researcher to calculate mass flow ratios describing the amount of ammonia in relation to water, as follows by energy balance:

$$\left.\frac{\dot{m}_{refrigerant}}{\dot{m}_{water}}\right|_{measured} = \frac{4500\ kg\ h^{-1}}{90000\ kg\ h^{-1}}$$

$$\left.\frac{\dot{m}_{refrigerant}}{\dot{m}_{water}}\right|_{measured} = 0.05$$

With the energy balance of the system a theoretical idealised mass flow of refrigerant can be calculated as follows:

$$\dot{m} * \left.\Delta h\right|_{refrigerant} = \dot{V}_{water} * \rho_{water} * c_{water} * \Delta T_{water} \qquad (5.5)$$

$$\dot{m}_{refrigerant} = \frac{\dot{m}_{water} * c_{water} * \Delta T_{water}}{\Delta h_{refrigerant}}$$

$$\Delta h_{refrigerant} = (h'' - h') = 894.195\ kJ\ kg^{-1}$$

$$\dot{m}_{refrigerant} = \frac{90000\ kg\ h^{-1} * 4.19\ kJ\ kg^{-1}\ K^{-1} * 5\ K}{894{,}195\ \mathrm{kJ\ kg^{-1}}}$$

$$\dot{m}_{refrigerant} = 2109\ kg\ h^{-1}$$

$$\left.\frac{\dot{m}_{refrigerant}}{\dot{m}_{water}}\right|_{calculated} = \frac{c_{water} * \Delta T_{water}}{\Delta h_{refrigerant}}$$

$$\left.\frac{\dot{m}_{refrigerant}}{\dot{m}_{water}}\right|_{calculated} = \frac{4.19\ kJ\ kg^{-1}\ K^{-1} * \ 5\ K}{894{,}195\ \mathrm{kJ\ kg^{-1}}}$$

$$\left.\frac{\dot{m}_{refrigerant}}{\dot{m}_{water}}\right|_{calculated} = 0.0234$$

The calculated ideal ratio was half as large than the actual measured mass flow ratio of ammonia to water. This occurred mainly because the ideal conditions never appear, and shell and tube heat exchangers have only low efficiencies – about 40–60% of the efficiency of modern plate heat exchangers, which is greater than 90%. But as mentioned before, modern plate heat exchangers run the risk of ice formation and freezing. The latter efficiency would already double the calculated ratio of ammonia in relation to water mass flow. Another likely possibility of the big gap between measured and calculated mass flow of refrigerant to water is, that there was no complete vaporisation of the liquid ammonia, so less heat transmission occurs between the two mediums. The percentage of vaporisation was not detectable, caused to needed inline temperature and pressure measurements, which were not able to install.

The limits in evaporation:

As mentioned, there are limits in evaporation. as can be seen in fig. 61

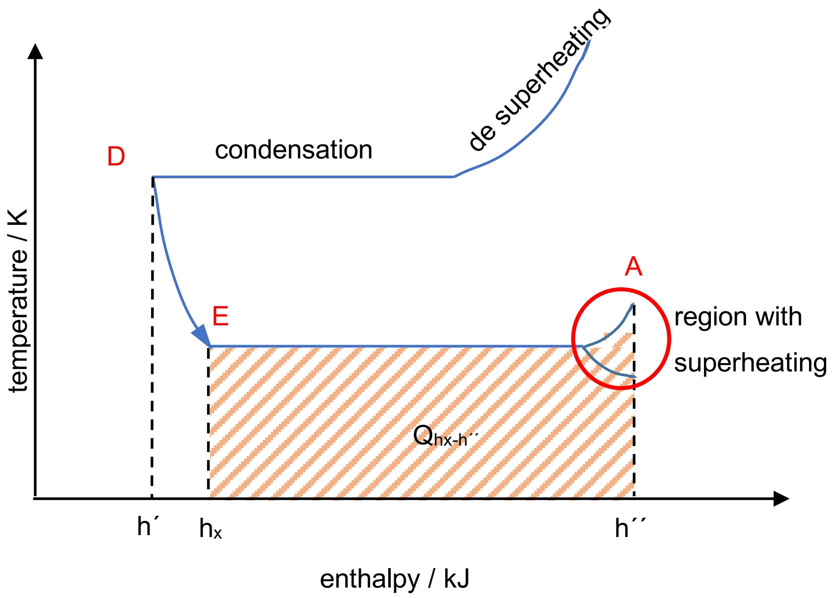

fig. 61: Simplified depiction of a T-S diagram; source: own representation.

$$h_A - h_x < (h'' - h')_{EA} \qquad with \qquad 0 \ll x \ll 0.5$$

- Step 1: D-E, pressure and temperature losses until high pressure ammonia reaches the location of the evaporator (piping between condensers and evaporator).
- Step 2: $h'_{D(local)} \equiv h_x = h_E' + x*(h'' - h')_E$.
- Step 3: E-A, area between line E-A and s-axis equals specific intake of thermal energy.

6 RESULTS AND SUMMARY

Within this work is given an abstracted view of the needed refrigeration amount for the fermentation tanks, by dividing these tanks into four different process phases. The first three phases of filling, fermentation, and maturation showed, that ammonia is only needed in peaks, to keep the tank content temperature at a defined setpoint. In the last process phase of cooling and draining a steady mass flow of refrigerant is necessary because the temperature of the beer inside the tank must be decreased by several degrees. tab. 9 shows again the investigated mass flows of liquid ammonia per type of tank and process phase.

Process step	Tank type	Steady ammonia amount	Peaks ammonia amount
		kg h^{-1}	kg h^{-1}
Filling I	1	-	3000
	2	-	4000
Fermentation II	1	-	2000
	2	-	6500
Maturation III	1	-	2000
	2	-	4000-6500
Cooling / draining IV	1	2500-3500	
	2	3500-4500	

tab. 9: Overview ammonia mass flow of fermentation tanks; source: own representation.

An ammonia consumption of the external beer cooler was not able to be determined as direct measurement. Considering the energy balance, 13,495 kg of ammonia would be needed for cooling 480,000 kg of beer by 6 degrees for one batch process. On the other hand, one important finding was, that the beer cooler worked always in batch processes with similar time periods, with beer mass flows of about 60,000 kg h^{-1}. However, there is prediction of the beer cooler load, just in fact that it must be started manually by operators. The exchangers outlet temperature, which should be slightly below zero degrees Celsius is the to be controlled value, which means ammonia gets injected more often or less, depending on this outlet temperature of the process fluid.

For the brew water cooler, which provides cold water to cool the hot wort, a ratio between water and liquid ammonia flow was investigated. This key figure indicates that for 1000 g of water are 50 g of refrigerant is necessary to extract the thermal energy out of the process fluid. This ratio was checked with a thought experiment, containing the needed heat energy to be given from the water to the ammonia. For this process nearly 21,000 kJ kg^{-1} of thermal energy had to be and could be absorbed by the measured coolant medium, even considering losses in thermal energy transfer.

In closing this study delivered a possible method of modelling refrigeration plants in breweries to forecast the needed amount of liquid ammonia for different kind of consumers. With this backbone further investigations are possible to collect enough data for a meaningful load prediction.

7 OUTLOOK

The recordings and feasibility check revealed that the required amount of liquid ammonia refrigerant for the fermentation tanks can be predicted in a range of values. Only in the main cooling stage is significant demand present, whereas the opposite is true in the other stages, where only short injector valve openings cause peakwise impulse operation in the mass flow of the refrigerant.

Therefore, future studies should involve long-term research over the next few years to collect more data on the fermentation process to implement the following opportunities and suggestions:

- Injector valve leakage monitoring via inline mass flow meters.
- Detection of low efficient tanks and pipes, due to dirt or deposits in dimple-plates, valves, and pipes,
- Optimisation in liquid ammonia distribution,
- Compressor and condensers load peak shaving, and
- Creating more exact load forecasts for each tank type.

By making process changes in the operation mode of the brew water cooler, one can fill holes in not high ammonia demand timeslots to maintain the entire refrigeration systems performance at a steady state without changing the compressors or condensers load. On the other hand, an almost constant operation mode of the brew water cooler must be checked for possible better efficiency. Hence, there is a minimisation of needed liquid ammonia to distribute, there could be the chance for operating the shell and tube condenser less time to decrease water consumption significantly.

To reduce energy consumption of the whole plant, one could change the shell and tube heat exchangers to plate heat exchangers. However, this option should be checked separately to verify whether one could conduct process engineering and ensure process stability, due to different pressure conditions and the risk of ice formation and freezing of plate heat exchangers compared to shell and tube exchangers.

Final, with collecting data over several months, even years, it is then possible to create detailed predictions of the needed cooling power for each consumer, raising the opportunity for energy savings, process stability, and predictive maintenance in ammonia distributing as also cooling apparatuses.

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GLOSSARY

english designation	german designation	description
hot wort	Heißwürze	The from the brew house finished liquid with dissolved hops and sugar converted from malt.
bottles (returnable glass bottles)	Flaschen (Mehrwegflaschen)	Multiple refill glass containers for beer and other carbonated drinks.
keg	KEG (Fass)	A small, pressurised metal barrel with a valve on one end, used to store and dispense beer and other carbonated drinks.
plc	SPS	Programmable logic control, an industrial computer ruggedized and adapted for the control of manufacturing processes.
fermentation	Gärung	The process in beer making, where sugar gets turned into alcohol.
maturation	Reifung	The process after main fermentation, where the beer gets carbonated.
draining	Entleeren	To empty a storage container, like a tank.
dimple jacket	Mantelkühlung	A machine formed sheet material for cooling, welded on a tank and carrying the refrigerant.
direct evaporation	Direktverdampfung	Direct extraction of heat without intercirculation loop.
yeast	Hefe	Unicellular fungi used for fermentation, creating alcohol out of sugar.
refrigerant	Kältemittel	Heat carrying fluid in all condition states which must convey the thermal energy by evaporators and condensers based on enthalpy.
brine	Sole	Watery solution of salts. Used in cooling applications as refrigerant.