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Energy-efficient operation of an industrial evaporative cooling

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Master thesis

Energy-efficient operation of an industrial evaporative cooling

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Thank you all for being part of this journey and helping me achieve this significant milestone.

Abstract

This thesis aims to develop a model for an industrial evaporative cooling system having a chiller and a production heat source using two cooling towers to evaluate the energy consumption of the system and find an optimal approach to minimize energy consumption without incurring investment costs. To achieve this objective, the thesis will perform a comprehensive analysis of the optimal cooling system configuration, considering various factors such as the ambient wet bulb temperature, the chiller cooling requirements and the maximum water temperature required by the production heat source.

The proposed methodology consists of developing a function for the cooling tower outlet water temperature and chiller power consumption, implemented in a Dymola numerical model. The study will investigate different control strategies and scenarios to determine the optimal approach for minimizing energy consumption.

It is essential to note that while this research focuses solely on the development and testing of the model using simulation data, real plant data validation is not included in the scope of this study. Rather, the emphasis is on demonstrating the feasibility and accuracy of the model through rigorous testing and analysis.

The expected results include the development of an efficient model and suggestions with the intention of reducing energy consumption and improving overall efficiency. The findings of this study will provide valuable insights into optimizing industrial cooling systems and may serve as a basis for future research that includes real plant data validation and practical implementation.

Kurzzusammenfassung

Ziel dieser Arbeit ist die Entwicklung eines Modells für ein industrielles Verdunstungskühlsystem mit einer Kältemaschine und einer Produktionswärmequelle unter Verwendung von zwei Kühltürmen, um den Energieverbrauch des Systems zu bewerten und einen optimalen Ansatz zur Minimierung des Energieverbrauchs zu finden, ohne dass Investitionskosten anfallen. Um dieses Ziel zu erreichen, wird in dieser Arbeit eine umfassende Analyse der optimalen Kühlsystemkonfiguration durchgeführt, wobei verschiedene Faktoren wie die Feuchtkugeltemperatur der Umgebung, der Kühlbedarf der Kältemaschine und die von der Produktionswärmequelle benötigte maximale Wassertemperatur berücksichtigt werden.

Die vorgeschlagene Methode besteht in der Entwicklung einer Funktion für die Wasseraustrittstemperatur des Kühlturms und die Leistungsaufnahme der Kältemaschine, die in ein numerisches Dymola-Modell implementiert wird. In der Studie werden verschiedene Regelungsstrategien und -szenarien untersucht, um den optimalen Ansatz zur Minimierung des Energieverbrauchs zu ermitteln.

Zu den erwarteten Ergebnissen gehören die Entwicklung eines effizienten Modells und Vorschläge zur Senkung des Energieverbrauchs und zur Verbesserung der Gesamteffizienz.

Abstract (Spanish version)

Esta tesis pretende desarrollar un modelo para un sistema de enfriamiento evaporativo industrial que utiliza dos torres de refrigeración, con el objetivo de evaluar el consumo de energía del sistema y comparar dos estrategias de control diferentes para encontrar un enfoque óptimo que minimice el consumo de energía sin incurrir en costes de inversión. Para alcanzar este objetivo, la tesis realizará un análisis exhaustivo para diferentes configuraciones del sistema de refrigeración, teniendo en cuenta diversos factores como la temperatura ambiente de bulbo húmedo, las necesidades de refrigeración del chiller y la temperatura máxima del agua fría requerida por la fuente de calor de la producción.

La metodología propuesta consiste en desarrollar una función multivariante de la temperatura del agua de salida de la torre de refrigeración y una función para el consumo de potencia del Chiller, (basado en datos empíricos), que luego se implementarán en el modelo numérico Dymola. La simulación se realizará a lo largo de un año natural completo, utilizando datos meteorológicos de la región de Friedberg, Hessen, en donde se ubica una planta industrial tomada como referencia, para obtener una buena sensibilidad en los resultados y extraer conclusiones más precisas sobre ambas estrategias de control cuando se expongan a diferentes situaciones y escenarios operativos.

Las cuestiones de investigación que se abordarán en este estudio incluyen determinar si es ventajoso operar con una sola torre de refrigeración a máxima velocidad del fan o con dos torres de refrigeración a media velocidad cuando aumentan los requisitos de refrigeración. Además, el estudio investigará si se puede ahorrar energía apagando una bomba y aumentando la velocidad del ventilador de la torre restante cuando las necesidades de refrigeración son bajas y las temperaturas ambientes de bulbo húmedo también son bajas.

Los resultados esperados de esta tesis incluyen el desarrollo de un modelo preciso y eficiente para un sistema de enfriamiento evaporativo industrial que pueda utilizarse para determinar la estrategia de control óptima, que conduzca a un consumo de energía minimizado. Además, los resultados de este estudio proporcionarán información sobre el efecto de varios escenarios operativos (configuración de la demanda de la fuente de calor y el chiller) en el consumo de energía y ofrecerán sugerencias para mejorar la instalación.

En conclusión, esta tesis presenta un enfoque innovador para optimizar el consumo energético de un sistema de refrigeración evaporativa industrial mediante el desarrollo de un modelo y una simulación exhaustivos que tienen en cuenta diversos escenarios y factores operativos. Los resultados de este estudio tienen el propósito de influir en una reducción del consumo de energía y mejorando la eficiencia global.

Contents

Latin symbols

A	m^2	surface area off the fills	
a	1/m	surface area per volume	
M_A	kg	mass air	
$\dot{M}_A$	kg/s	mass flow rate air	
M_W	kg	mass water	
$\dot{M}_W$	kg/s	mass flow rate liquid water	
P_e	W	electrical power consumption	
p	Pa	absolute pressure	
$\dot{Q}_c$	KW	cooling capacity	
T	K	Temperature	
T_W	K	Temperature of liquid water	
u	U/min	fan speed	
t	m	thickness of the fills	
V	m^3	Volume	
X	g/kg_d	Vapor mass fraction	$X = M_W/M_A$
X^*	g/kg_d	Vapor mass fraction, saturated at water temperature.	
x	m	coordinate in direction of the air flow	

Greek symbols

β	m/s	mass transfer coefficient
β_x	$kg/(m^2s)$	mass transfer coefficient
ρ	kg/m^3	density

Dimensionless numbers

NTU	Number of transfer units

Subscript indices

A	air
bD	belt drive
CT	cooling tower
C	chiller
fs	full speed
hs	half speed
d	dry
in	inlet
kin	kinetic
M	motor
out	outlet
R	removed
therm	thermal
V	water vapor

W	liquid water
Wi	water inlet
Wo	water outlet
wb	wet bulb

Abbreviations

Avg	average
CH	chiller
CT	cooling tower
HS	heat source
Max	maximum
Min	minimum
RH	relative humidity

Table references

Figures references

1 Introduction

As in any industrial process where there is a heat source, a cooling source is required that allows the process to extract a certain amount of heat, depending on its application. One of the most common cooling sources today to control the temperature of the cooling water are cooling towers. These spray the water on top of the tower, which will then fall into a fill material that will guide it and allow the exchange of heat with a turbulent flow of air produced by fans.

In this project, I will present the development of a model of an industrial evaporative cooling system for a chiller and a production heat source using two cooling towers and a basin to evaluate the power consumption of the system and compare two different control strategies. The idea is to identify the optimal control strategy, either using only one of them or both for different possible heat load scenarios, in which the energy consumption of the entire process is minimized without representing an investment cost, that is, only working with the elements available in the system.

A brief description of the complete system is necessary to understand the questions that arise about both strategies and highlight important points.

The production heat source requires a maximum cold-water temperature for the water coming from the basin. Therefore, a lower temperature does not lead to a higher efficiency.

The system also has a chiller, which is used for cooling data centre systems and building air conditioning. The chiller accepts a range of temperatures for the cold water coming from the basin. A lower temperature leads to a reduced power consumption at the compressor of the chiller.

As the water flow over the cooling towers driven by a pump is constant, in order to control the temperature of the water at the outlet of the cooling tower, it is necessary to regulate the air flow through them, therefore, the speed of the fans is the main element to control that will allow us to obtain the desired temperature.

The energy consumed by the fans of each cooling tower must also be added to the energy consumed by each pump.

Currently each cooling tower off the system operates with two fan speeds, that is, for low demand for heat extraction, it can operate with the fan at medium or half speed, when the demand increases, then the fan will operate at maximum speed.

Here is the first question, when the cooling requirement increases, is it convenient to operate with a single cooling tower, or is it better to switch a second cooling tower and work with two at half speed? These two cases will be considered as different operating strategies, and they will be simulated later.

The energy consumed by the fans is proportional to the speed cubed, so in the second case the power consumed by two fans will be lower, but it must be considered that switching to another tower would also imply activating an extra pump.

On the other hand, under low cooling requirements and low ambient wet bulb temperatures energy might be saved if one switches off one pump and lets the fan of the remaining tower run at higher speeds.

This leads to the assumption that there is an optimum for the overall consumption of electrical energy, given by the numbers of cooling towers running and the return temperature from the cooling tower(s) depending at least on:

- the ambient wet bulb temperature,
- the demand for cooling from the industrial heat source,

- the maximum cold-water temperature required by the production heat source,
- the acceptable range of temperatures for the chiller's cold-water inlet and
- the energy consumption of each cooling tower's fans and the pumps of the system.

Therefore, a detailed analysis is required to determine the optimal configuration of the cooling system based on these factors.

To answer these questions, a multivariable function of the cooling tower outlet water temperature will be developed, as well as a function for the chiller, and then implemented in a Dymola numerical model.

While this study does not involve real plant data validation, the model's performance will be assessed against a real industrial plant located in Friedberg, Hessen, Germany, by using real dimensional values and simulating a full year based on meteorological data from the region. This approach will allow for sensitivity analysis and enable more accurate conclusions about the effectiveness of both control strategies under different operational scenarios.

1.1 Motivation

The current global context of energy transition, combined with increased environmental awareness, has made the sustainable and responsible use of energy more important than ever. In response, industries are looking for ways to reduce their energy consumption and improve energy efficiency by optimizing their processes. Even sub-processes with minimal impact on the overall energy balance can offer opportunities for improvement. By analysing each process individually and identifying areas of inefficiency, significant energy savings can be achieved, resulting in both economic benefits and a reduced carbon footprint, ultimately contributing to environmental protection and promoting sustainability.

In this thesis, the study will be conducted on an evaporative cooling system, where an opportunity for improvement was identified through the implementation of an efficient strategy to control the outlet temperature of the cooling towers.

1.2 Goal

The goal of this thesis is to develop a model for an industrial evaporative cooling system using two cooling towers, to evaluate the power consumption of the system, and compare two different control strategies.

The aim is to find an optimal control strategy that minimizes the energy consumption of the entire process without having a high investment cost. To achieve this goal, a detailed analysis will be conducted to determine the optimal configuration of the cooling system based on factors such as ambient wet bulb temperature, cooling requirements from the chiller, and maximum cold-water temperature required by the production heat source.

The multivariable function of the cooling tower outlet water temperature and for the chiller power consumption will be developed and implemented in the Dymola numerical model. The simulation will be performed over the course of a year, using the meteorological data of a reference industrial plant located in Friedberg, Hessen, Germany, to obtain good sensitivity in the results and draw more accurate conclusions about both control strategies when exposed to different situations and operational scenarios.

While real plant data validation is not included within the scope of this thesis, the results and insights obtained will lay the groundwork for potential future research involving validation with actual industrial plant data. The findings of this study will contribute to enhancing energy efficiency and sustainability in industrial cooling systems.

2 State of the art

An industrial evaporative cooling system typically consists of the following components:

Evaporative Cooling Unit: The cooling tower is the main component of the system and is responsible for cooling the water used in the cooling process. It consists of a large structure that houses a series of fins or fill material that allows the water to be exposed to the air for maximum cooling.

A water pump is responsible for circulating the water from the cooling tower to the heat source, where it absorbs heat, and then back to the cooling tower for cooling.

The distribution system includes piping and valves that distribute the cooling water to the heat exchanger and then back to the cooling tower.

Fan: The fan is used to draw air through the cooling tower, which causes the water to cool down as it is exposed to the air.

Controls: The system is controlled by a variety of sensors and controllers that monitor temperature, water level, and other variables to ensure efficient operation.

2.1 Evaporative cooling towers

Cooling towers are the main elements in a cooling system, by definition from the "VDI Heat Atlas" [1]"Evaporative cooling towers are devices which are used to remove heat from various types of machinery or industrial process equipment either directly or through intermediate heat exchangers by making use of evaporative cooling" and it can theoretically cool water down to the wet-bulb temperature which is usually considerably lower than the dry-bulb temperature. Their operation is based on a principle where energy is removed from hot water in direct contact with relatively cool and dry air. So, to achieve the best mass and heat exchange ratio in cooling tower, the most important factor is the large air-water contact area and a high heat transfer coefficient. The intensive mix of cooling water and airflow together with the specific surface area of the fills achieve a high cooling performance.

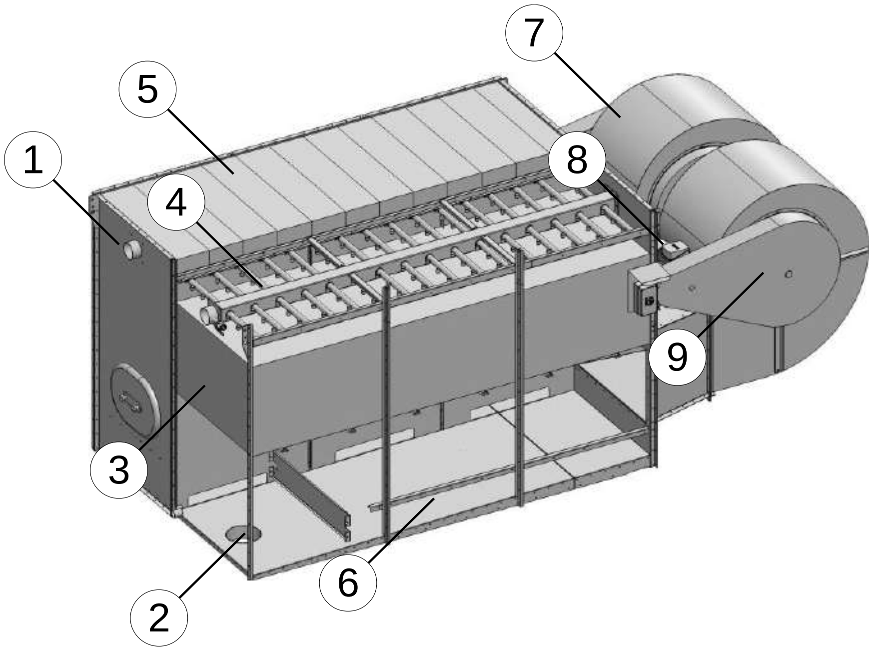
Cooling tower can be grouped into various categories as shown in Table 1.

Table 1 - Comparative table between MDCTs and NDCTs (Source: VDI Heat Atlas [1]):

	Mechanical draught cooling tower (MDCTs)	Natural draught cooling tower (NDCTs)
Open circuit	Forced draught	Counterflow
	Induced draught	Crossflow
	Counterflow	
	Crossflow	
Closed circuit	Evaporative coolers	
	Evaporative condensers	

In this facility, the technology used in cooling tower corresponds to the counter-flow technology. The technical data sheet of the cooling tower can be found in the annexes section, Table 15.

Fig. 1 shows the type of cooling tower installed in the real facility and its components.



1: Water inlet
2: Water outlet
3: Fills
4: Water distribution with spray nozzles
5: Drift eliminators
6: Basin
7: Radial fan
8: Electrical motor
9: Belt drive

Fig. 1 - Counter flow Cooling Tower (Source: Gohl-KTK, Markings by author).

Fills:

To achieve a high thermal efficiency a high surface transfer area is required. high surface densities go hand in hand with high pressure drops and they clog more easily so a compromise is often made.

The transfer coefficients used for cooling tower design are invariably determined from experimental test rigs and used in programs employing either Poppe's or Merkel's method of integration. These measurements are usually expressed in equations containing the air and water mass flow rates per unit area or their ratio.

The installation used as a reference for this project uses a cross-fluted fills in the cooling tower from the manufacturer "Enexio-water-technologies". The fill is KFP 312, the following table shows the main characteristics of the backfill.

	KPP 312	KZP 312	KFP 312	KFP 319
Contact surface [m²/m³]	232	240	238	150
Corrugation height [mm]	12	12	12	19
Channels	Cross-fluted	Cross-fluted	Cross-fluted	Cross-fluted

Fig. 2 - Features of the Fills (Source: Manufacturer's website: www.enexio-water-technologies.com, markings by author)

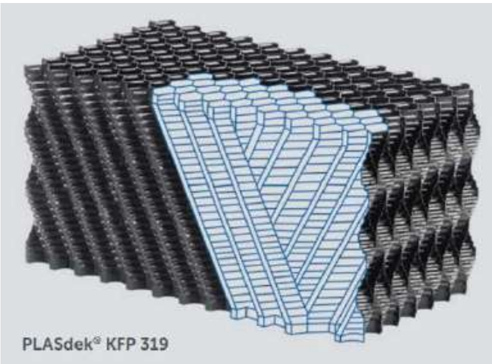


Fig. 3 - Cross-Fluted Fills (Source: Manufacturer's website: www.enexio-water-technologies.com)

Water distribution:

The water distribution uses spray nozzles. These tend to have a typical range of operation between 0.2 and 0.8 bar gauged. Below 0.2 bar the spray screen tends to break down leading to an uneven water distribution. That would allow air to bypass unwetted areas of the fills leading to bead performance. On the other hand, a pressure above 0.8 bar the pumping power becomes uneconomically high.

The pressure drops over a nozzle goes quadratic with the water volume flow rate through the nozzle and the maximum possible variation the flow rate is less than two to one.

Therefore, the water distribution system usually is designed for a fixed water volume flow rate and to have a pressure drop of around 0.4 bar.

2.2 Control System

The control system used in the plant is a cascade control system, i.e., a basic closed-loop feedback control system that monitors the water temperature at the cooling tower outlet and adjusts the fan speed and the number of units used based on the temperature information received.

Considering that each cooling tower here only has three states for the fan speed, off, half speed, or maximum speed, this control system provides limited control over the cooling process.

The temperature limits set within a certain operating range provide a reference range for the controller to compare the actual temperature value. If the temperature is within the control range, the controller will not make any changes until the next sample is taken.

If the water temperature falls below the lower limit, the controller will command a decrease in fan speed or turn it off entirely to avoid overcooling. If the temperature exceeds the upper limit, the controller will increase the RPM of the fan and, if necessary, set it to the maximum allowed speed to increase cooling. If the next sample still indicates a temperature above the limit, the control system will add a second cooling tower to the system.

To limit ware of the motor and the belt drive switching of a fan is limited to one change every 10 minutes.

There are advanced control techniques available that can optimize the cooling process and improve energy efficiency beyond basic control methods. For instance, model predictive control and adaptive control are examples of more sophisticated techniques that can achieve superior performance. Furthermore, the use of variable speed drives can provide highly precise control over the fan speed. Consideration of more than three states for fan speed control can also lead to improved control of the cooling process.

The VDF control system (Variable Frequency Drive) is currently the most widely used control system for newer installations, which consists of several components that work together to control the speed of an electric motor. This control system operates by adjusting the frequency and voltage output of the inverter based on the controller’s input. The motor speed is directly proportional to the frequency of the power supplied to it, so by adjusting the frequency, the VFD can control the speed of the motor. Each component plays a specific role in the control process.

2.3 Chiller

Air cooled and water-cooled chiller are very common equipment in HVAC (heating, ventilation, and air conditioning) systems for building. Fig. 4 shows the schematic diagram for chiller system. There are four components in vapor refrigeration cycle for air/ water cooled chiller which are compressor, condenser, metering device and the heat exchanger/evaporator. One of them that plays an important role in refrigeration system is metering device. A measuring device provides a pressure drop point. This device has the function of keeping the refrigerant in a condensed state and feeding the refrigerant to the evaporator.

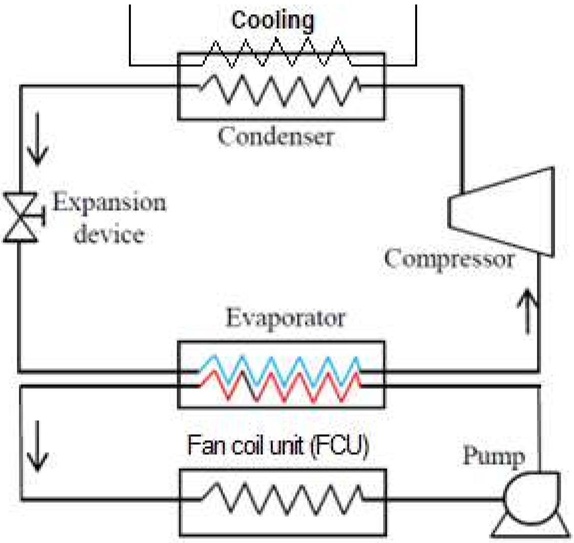


Fig. 4 - Schematic diagram of water chiller (Source: Paper: Putu Wijaya Sunu et al 2018 J. Phys.: Conf. Ser. 953 012063).

One of the key areas where chiller technology is advancing is in efficiency. Modern chillers can be up to 40% more efficient than older models. This is achieved through a combination of improvements in compressor design, refrigerant selection, and heat exchanger technology. These systems can offer significant energy savings and environmental benefits.

Variable speed drives (VSDs) can significantly improve chiller efficiency by allowing the system to adjust its output to match the cooling load of the building. Research is ongoing to develop more advanced VSD technology that can further improve chiller performance.

Many modern chiller systems incorporate heat recovery technology, which allows waste heat from the chiller to be used for other purposes, such as heating water. This further improves energy efficiency and can lead to significant cost savings.

In response to growing concerns about climate change, there is a significant amount of research focused on developing chiller systems that use low global warming potential (GWP) refrigerants. These may include natural refrigerants such as carbon dioxide or hydrofluoroolefins (HFOs).

3 Description of the real cooling plant

The industrial plant that was used as a reference for the model is a facility of Fresenius Kabi Logistik GmbH, located near the city of Friedberg, Hessen. The layout of the plant and cooling system is shown in Fig. 5 and Fig. 6.

The plant consists of 8 cooling towers like the one in Fig. 1 with seven units having three fans and one unit having two fans. The towers are arranged in four groups and supply several chillers and direct cooling for air compressors and the production facility approximately 200 m away connected by a pipe (red line in Fig. 5). A scheme of the system is given in Fig. 7.

A big basin of around 400 m^3 is located beneath the cooling towers together with a room for pumps. The water from the basin within each cooling tower is collected in the cold part of the big basin.

The total cooling capacity is around 20 MW and the nominal capacity of one unit with 3 fans is 2500 kW for the following conditions.

Table 2 - Cooling Tower Specifications:

cooling capacity	kW	2500
water volume flow rate	m3/h	238,9
water inlet temperature	°C	36,0
water outlet temperature	°C	27,0
wet bulb temperature	°C	22,0
power consumption per fan at full speed	kW	6,5
power consumption per fan at half speed	kW	1,9

The electrical motors of the fans use so called “Dahlander circuits”. Therefore, they can run on two speeds, namely approximately 1500 or 750 round per minute, without the use of a VSD.

The Fig. 5 below shows a satellite photo of the actual industrial plant. On it, the distribution line is figuratively highlighted, and the cooling towers are enclosed and shown in more detail in Fig. 6.

Fig. 5 - Overview of the plant (Source: Google Maps with markings by the author).

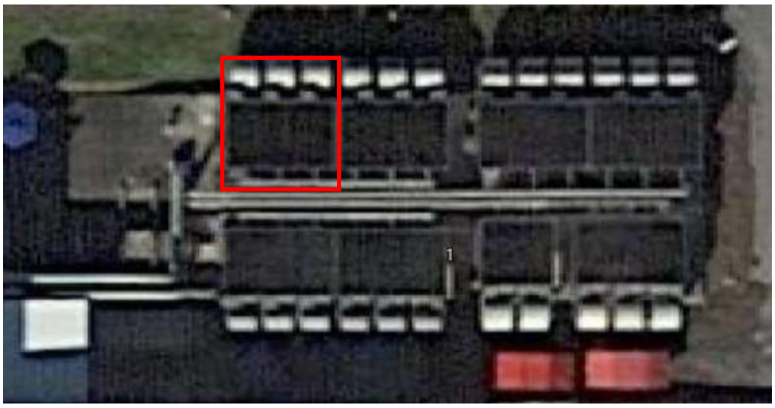


Fig. 6 - Cooling Towers of the plant (Source: Google Maps with markings by the author).

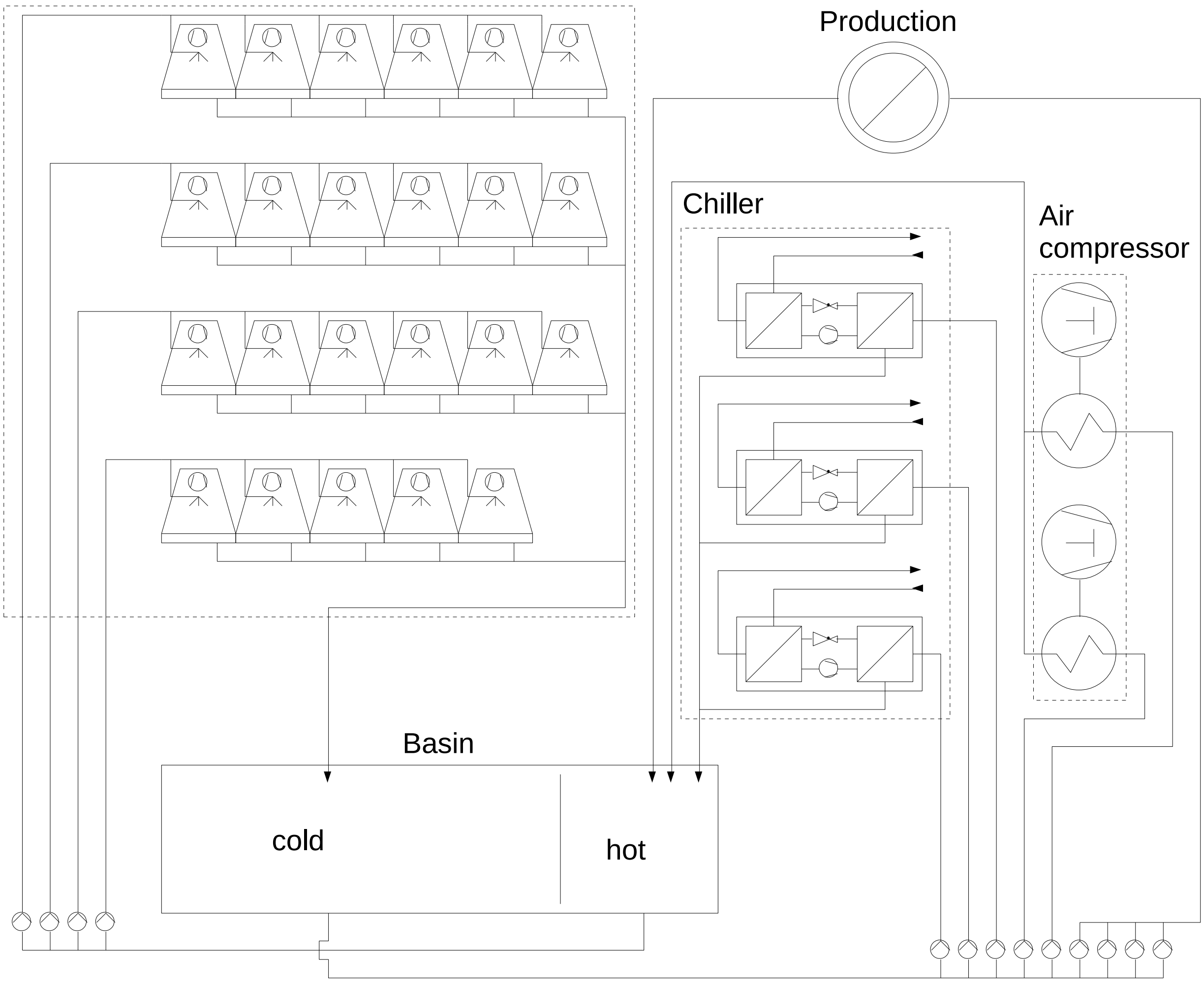


Fig. 7 - Scheme of the plant (Source: Markus Nickolay).

4 Simulation model

Dymola (Dynamic Modelling Laboratory) software was used to simulate the mode of the installation. This software tool is used for modelling and simulation of complex multi-domain systems, including mechanical, electrical, hydraulic, thermal, and control systems, it also includes a graphical modelling environment for creating and editing models, as well as a simulation engine for running simulations.

Dymola is widely used in industries such as in automotive, aerospace, robotics, process, and other applications.

The language used by Dymola is Modelica, which is an open-source equation based and object-oriented programming language, designed for modelling complex physical systems. It is designed to be flexible and easily extensible.

The models are described as collections of interconnected components, each representing a physical or mathematical entity in the system being modelled. Language provides a rich set of built-in components, including mechanical, electrical, thermal, and fluid components, as well as a variety of mathematical and control components.

4.1 Weather data

To evaluate the performance of the model against different requirements, it was necessary to study the meteorological conditions in the region and define an operational range within which the system will operate.

Under the URL https://kunden.dwd.de/obt/ the German Weather Service (Deutscher Wetterdienst, DWD) provides weather data in the form of so-called test reference years for each square kilometer of Germany. On an hourly basis the datasets represent the typical weather conditions over the course of one year. Among other values the datasets provide absolute pressure, temperature, and water vapor mass fraction.

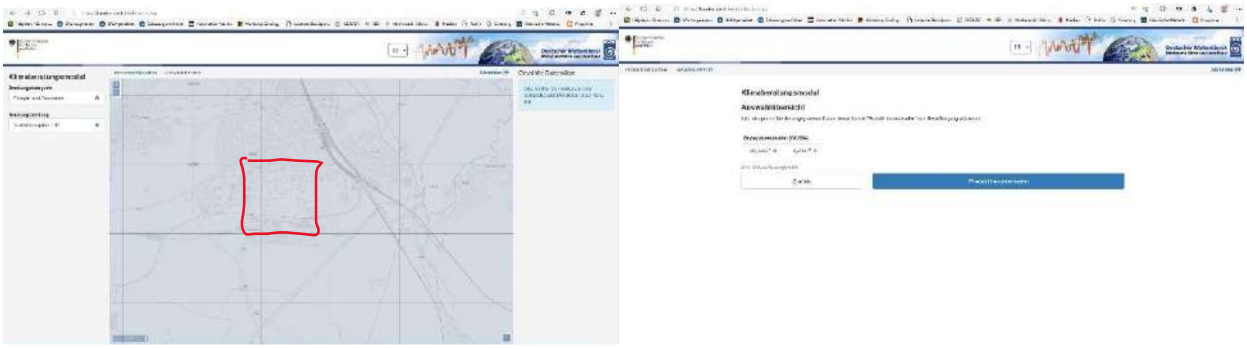


Fig. 8 - Weather data source (Source: DWD [2]) [3].

Each dataset provides 6 reference years. Three for the actual period of time based on observations in between 1995 and 2012 and three for a future period (2031 to 2060). The three datasets for each period represent either a normal year, a year with a very cold winter or a year with a very hot summer.

For this work the dataset for the actual time period and the normal year was used.

This approach allowed a comprehensive evaluation of the model's performance under different climatic conditions, which reinforced its robustness and applicability.

From these databases, the most important information was extracted and consolidated in the Table 3.

Table 3 - Weather data per year

	Average Year		Warm Year		Cold Year	
Values	**2015**	**2045**	**2015**	**2045**	**2015**	**2045**
Max T	34.3	32.6	34.7	37.8	35.1	35.1
Avg T	10.4	11.1	10.2	11.9	9.9	10.3
Min T	-10.6	-9.1	-11.7	-10.2	-15.8	-9.6
Max Twb	22.2	21.3	22.3	22.3	22.4	22.4
Avg Twb	8.3	8.8	7.7	9.1	7.5	8.0
Min Twb	-10.9	-9.3	-11.8	-10.3	-16.1	-9.9
Max p	1025	1021	1021	1022	1020	1025
Avg p	999	997	998	998	996	997
Min p	968	965	965	966	963	965
Max RH	100	100	100	100	100	100
Avg RH	79	78	76	74	76	77
Min RH	19	19	20	18	20	22

4.1.1 Implementation in Dymola

To implement the weather data into Dymola the “weather data bus” from the Modelica buildings library ([3]) was used. This is a pre-built component in Dymola that provides meteorological data for simulations. It includes models and functions that allow easy connection to different weather data sources, such as online weather services, local weather stations, or weather databases, and importing the relevant data into your model. In this case the database loaded into the model is the same as the one extracted from the German Weather Service website, which contains all the weather information for the Friedberg region. By using real-world meteorological data, simulations can provide accurate and reliable results, improving the overall quality of the model.

Fig. 9 - Weather block in Dymola

A more detailed block is shown in the annex, Fig. 45.

4.2 Set up of the industrial cooling process.

As the objective of this work was not to simulate the entire system, but rather to establish a functional model of a reduced system that could be expanded upon in future works a simplified installation was modelled using two cooling towers, a chiller, and a heat source production unit. Fig. 10 illustrates the system setup.

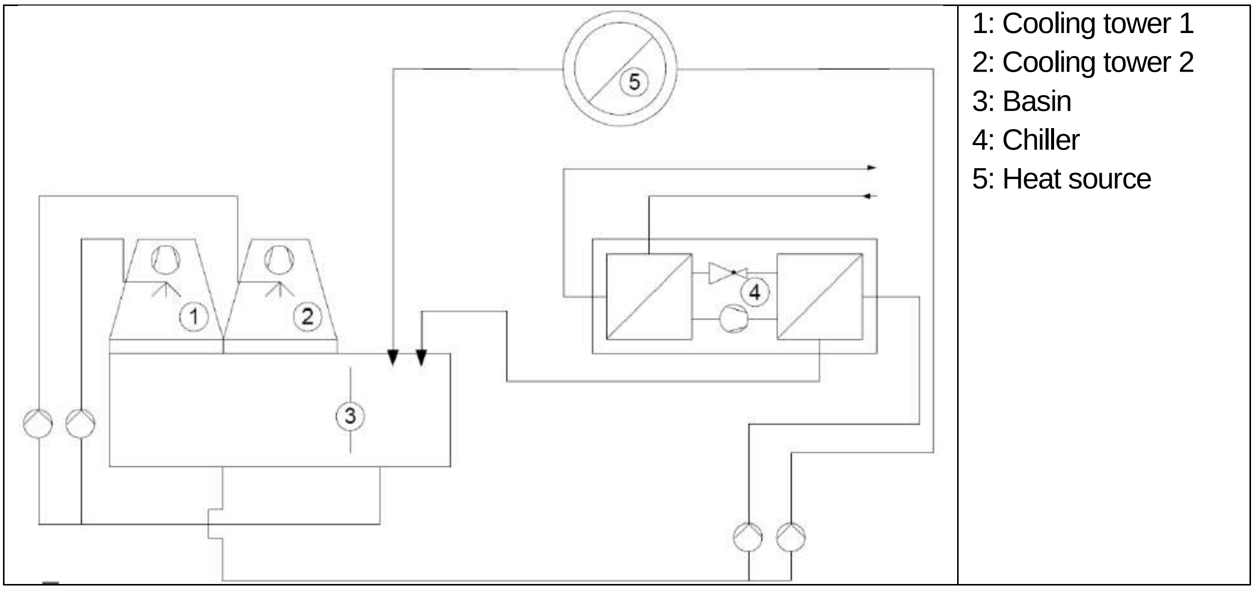


Fig. 10 - Set up Installation (Source: Setup of the model giving by Markus Nickolay).

4.2.1 Set up of the Cooling System at Dymola

The result of the complete model of the cooling system in Dymola is given in Fig. 11. This is the model of the cooling system installation that has been simulated for the two-cooling tower operating strategies (two CTs at half speed or 1 at full speed), and the 9 scenarios with different cooling requirements proposed for this project.

In the following sections, the components will be described in detail. The red lines represent the piping through which hot water from the chiller and heat source circulates, while the blue lines indicate the piping for the water that has already been cooled by the cooling towers and returned to the circuit.

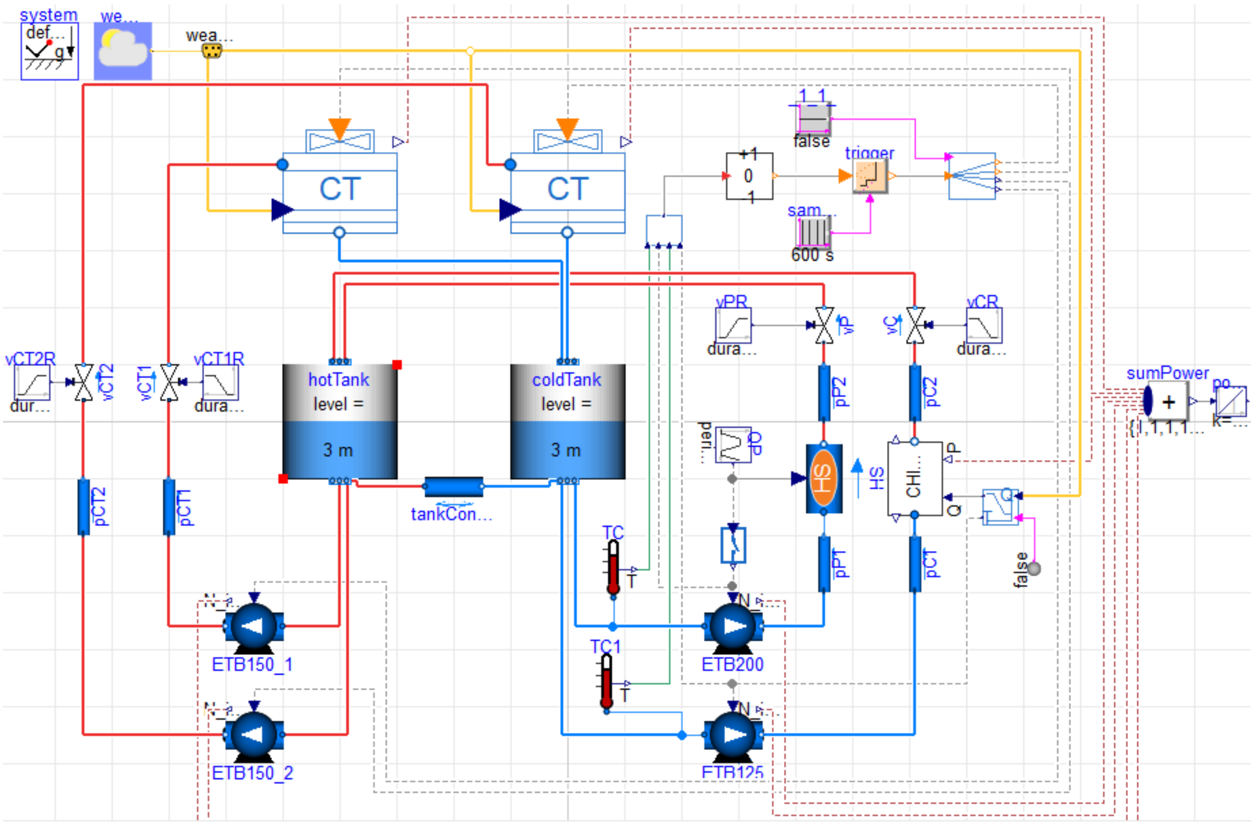


Fig. 11 - Dymola model for the cooling system.

4.3 Cooling tower

A cooling tower can be calculated according to a theory developed by Merkel ([1]). A more accurate method is given in the VDI-Wärmeatlas) ([4]).

The method is based on the calculation of the number of transfer units (NTU) from

$$NTU_{therm} = \int_{T_{W,out}}^{T_{W,in}} \frac{dX/dT_W}{X^* - X} dT_W \tag{1}$$

which must equal the kinetics of the fills given by

$$NTU_{kin} = \int_0^A \frac{\beta_X}{\dot{M}_A} dA = f\left(\frac{\dot{M}_W}{\dot{M}_A}\right) \tag{2}$$

and is for many types of fills only a function of the ratio of the water mass flow rate to the air mass flow rate.

Data for (2) can be derived from measurements or performance data from the manufacturer of the fills. In this case the data used originates from a software from the supplier of the fills. Based on the yellow marked input data in Fig. 12 the program returns the air volume flow rate need (green marking).

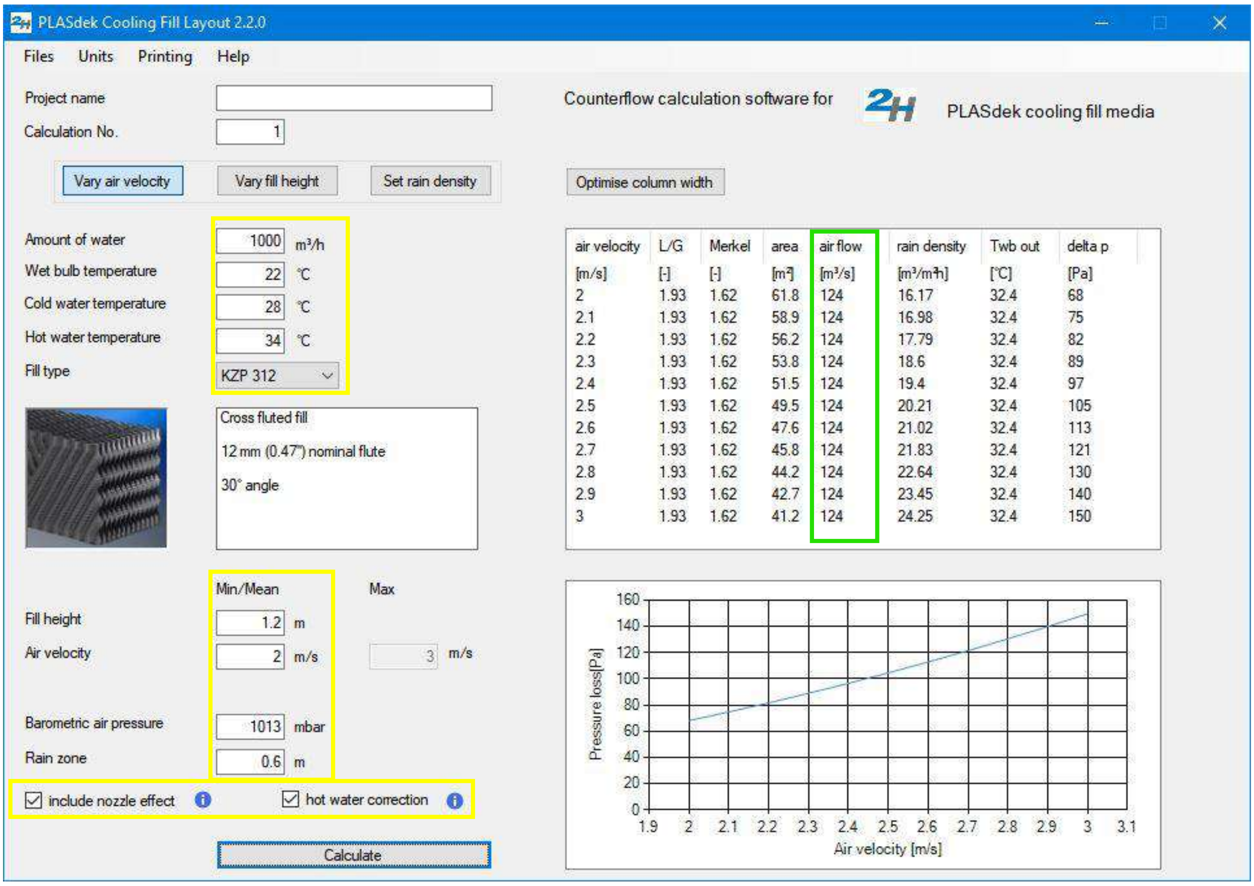


Fig. 12 - Origin of the performance of the fills (Source: ENEXIO Water Technologies GmbH).

The results given by the program were correlated an implemented in a Microsoft Excel file which also calculates the integral of (1) and outlet temperature for a given ratio of $\frac{\dot{M}_W}{\dot{M}_A}$ by M. Nickolay in former years to

$$NTU_{kin} = e^{m(x-1)+t} \tag{3}$$

with

$$t = 0.7775 \tag{4}$$

and

$$m = \begin{cases} x \leq 1: 0.5644 \\ x > 1: -0.025 \end{cases} \tag{5}$$

where

$$x = ln\left(\frac{\dot{M}_W}{\dot{M}_A}\right). \tag{6}$$

Fig. 13 shows the correlation (red line) compared to the datapoints given by the software from EN-EXIO.

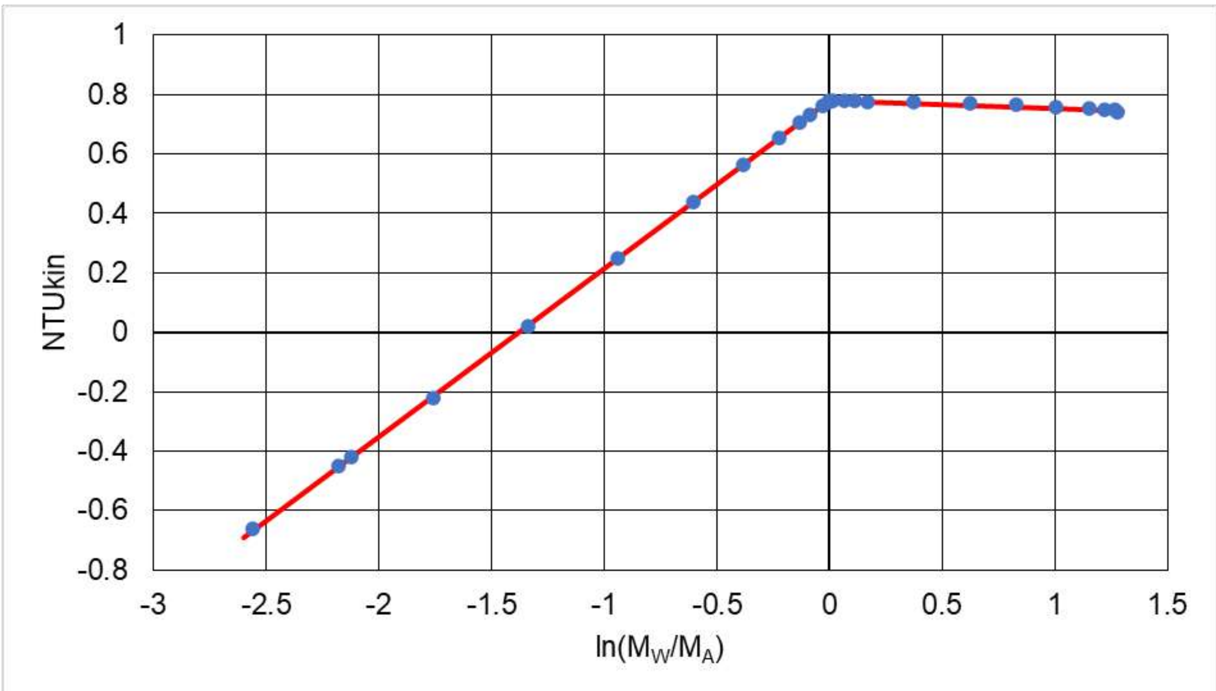


Fig. 13 - Function (3) compared to data from the program (blue dots), (Source: M. Nickolay).

4.3.1 Cooling tower function

After analyzing the weather data and defining the variables for the calculation, the next step is to discretize the data within the operating range of the system, as shown in Table 4, to calculate many cooling tower water outlet temperatures using an Excel file provided by M. Nickolay.

5808 are the calculated temperatures, which will then allow us to create a function for each operating speed (half and full), that is, 2904 temperatures for each function.

Table 4 - Cooling Tower operating range

	FanSped	p (bar)	T_{Ai} (°C)	RH (%)	T_{Wi} (°C)
Range	*half*	*0.965*	*-15*	*0*	*16*
	full	*0.995*	*-10*	*10*	*20*
		1.025	*-5*	*20*	*24*
			0	*30*	*28*
			5	*40*	*32*
			10	*50*	*36*
			15	*60*	*40*
			20	*70*	*44*
			25	*80*	
			30	*90*	
			35	*100*	
Parameter for each variable	2	3	11	11	8
Total Data	**5808**				

Having carried out all the calculations, the points obtained have been plotted. The following graph shows (in blue for full speed of the fan and orange for half speed) the different curves corresponding to different inlet water temperatures in the cooling tower.

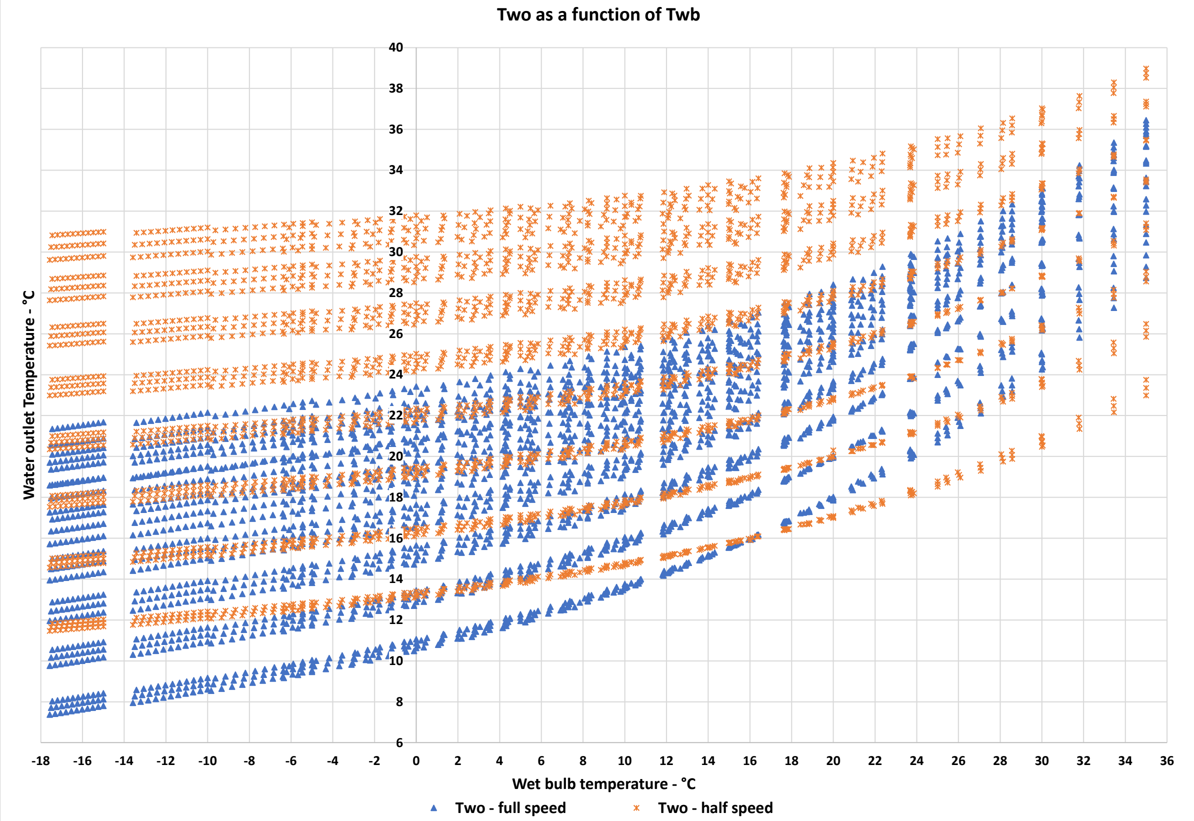


Fig. 14 - Water outlet temperature of the cooling tower for both speeds.

Taking as an example two cooling tower inlet water temperatures, for each fan speed, the trend of the outlet temperature as a function of the wet bulb temperature (T_{wb}) can be observed.

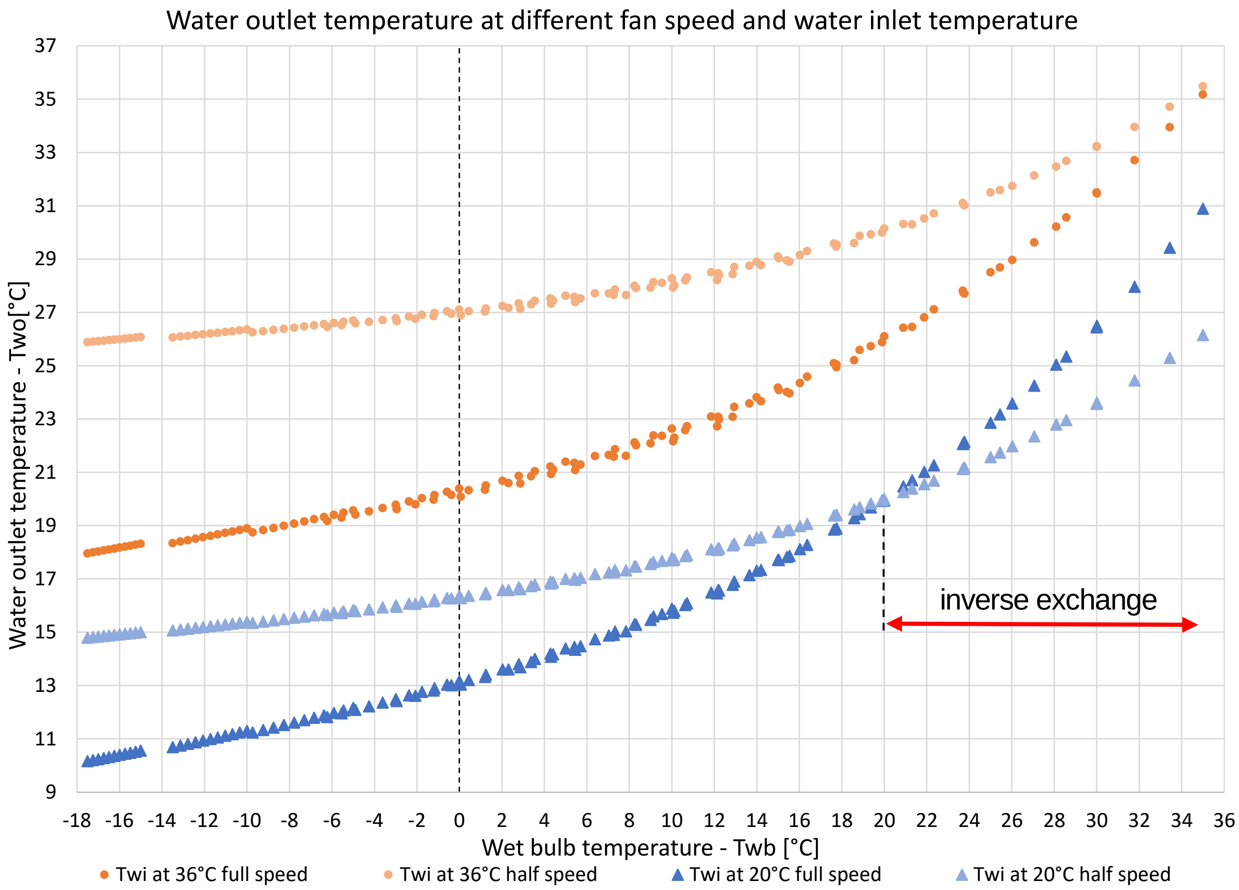


Fig. 15 - Water outlet temperature for design operating conditions, at both speeds.

From the data, these curves can be approximated as a third order polynomial function. To develop the functions for each fan speed as a function of the wet bulb temperature and the water temperature at the cooling tower inlet, the least squares method was used.

It can also be observed that the model works by exchanging heat in the opposite direction (heating the water) when the wet bulb temperature is higher than the cooling tower inlet water temperature.

4.3.1.1 Polynomial constants for each function

As a result, the following function is obtained for each speed of the cooling tower.

Water outlet temperature for full speed of the fan:

$$T_{Wo,fs} = a_{fs} + b_{fs}\, T_{Wi} + c_{fs}\, T_{Wi}{}^{2} + d_{fs}\, T_{Wi}{}^{3} \quad (7)$$

with the coefficients depending on the wet bulb temperature (trend can be observed in the annex Fig. 36)

$$a_{fs} = a_{fs,0} + a_{fs,1} T_{wb} + a_{fs,2} T_{wb}^{2} + a_{fs,3} T_{wb}^{3}, \quad (8)$$

$$b_{fs} = b_{fs,0} + b_{fs,1} T_{wb} + b_{fs,2} T_{wb}^{2} + b_{fs,3} T_{wb}^{3}, \quad (9)$$

$$c_{fs} = c_{fs,0} + c_{fs,1} T_{wb} + c_{fs,2} T_{wb}^{2} + c_{fs,3} T_{wb}^{3}, \quad (10)$$

and

$$d_{fs} = d_{fs,0} + d_{fs,1}T_{wb} + d_{fs,2}T_{wb}^2 + d_{fs,3}T_{wb}^3. \tag{11}$$

Water outlet temperature for <u>half speed</u> of the fan:

$$T_{Wo,hs} = a_{hs} + b_{hs}\,T_{Wi} + c_{hs}\,{T_{Wi}}^2 + d_{hs}\,{T_{Wi}}^3 \tag{12}$$

with the coefficients depending on the wet bulb temperature (trend can be observed in the annex Fig. 37)

$$a_{hs} = a_{hs,0} + a_{hs,1}T_{wb} + a_{hs,2}T_{wb}^2 + a_{hs,3}T_{wb}^3, \tag{13}$$

$$b_{hs} = b_{hs,0} + b_{hs,1}T_{wb} + b_{hs,2}T_{wb}^2 + b_{hs,3}T_{wb}^3, \tag{14}$$

$$c_{hs} = c_{hs,0} + c_{hs,1}T_{wb} + c_{hs,2}T_{wb}^2 + c_{hs,3}T_{wb}^3, \tag{15}$$

and

$$d_{hs} = d_{hs,0} + d_{hs,1}T_{wb} + d_{hs,2}T_{wb}^2 + d_{hs,3}T_{wb}^3. \tag{16}$$

The following table lists the constants for the final polynomial of each function.

Table 5 - Coefficients for full fan speed:

Coefficients for full fan speed	Unit	Values
$a_{fs,0}$	[°C]	0.610461084774077
$a_{fs,1}$	--	0.22951621211468
$a_{fs,2}$	[1/(°C)]	0.00429272654558364
$a_{fs,3}$	[1/(°C)^2]	0.000176125102765765
$b_{fs,0}$	--	0.686432488311475
$b_{fs,1}$	[1/(°C)]	0.00164663828388901
$b_{fs,2}$	[1/(°C)^2]	4.37755436829176E-05
$b_{fs,3}$	[1/(°C)^3]	-6.06976907321773E-06
$c_{fs,0}$	[1/(°C)]	-0.00192226891980353
$c_{fs,1}$	[1/(°C)^2]	-0.0001198549553991
$c_{fs,2}$	[1/(°C)^3]	-2.24557862293905E-06
$c_{fs,3}$	[1/(°C)^4]	1.06415463682525E-07
$d_{fs,0}$	[1/(°C)^2]	-4.82329388629608E-05
$d_{fs,1}$	[1/(°C)^3]	1.03539689577335E-06
$d_{fs,2}$	[1/(°C)^4]	0
$d_{fs,3}$	[1/(°C)^5]	0

Table 6 - Coefficients for half fan speed:

Coefficients for half fan speed	Unit	Values
$a_{hs,0}$	[°C]	-0.0653460491748675
$a_{hs,1}$	--	0.11613485515672
$a_{hs,2}$	[1/(°C)]	0.00178443835410971
$a_{hs,3}$	[1/(°C)^2]	7.76711780788867E-05
$b_{hs,0}$	--	0.870267589106905
$b_{hs,1}$	[1/(°C)]	0.00168286204663612
$b_{hs,2}$	[1/(°C)^2]	7.94198749547348E-05
$b_{hs,3}$	[1/(°C)^3]	-3.08985255147763E-06
$c_{hs,0}$	[1/(°C)]	-0.00126325972644525
$c_{hs,1}$	[1/(°C)^2]	-7.65970076732307E-05
$c_{hs,2}$	[1/(°C)^3]	-3.34562395947153E-06
$c_{hs,3}$	[1/(°C)^4]	6.78312698121485E-08
$d_{hs,0}$	[1/(°C)^2]	-0.000048
$d_{hs,1}$	[1/(°C)^3]	0.00000085
$d_{hs,2}$	[1/(°C)^4]	2.73777431041312E-08
$d_{hs,3}$	[1/(°C)^5]	-0.0000000009

The solution to each function in the range previously defined in Table 4 is plotted and it’s shown in the annexes, Fig. 39 and Fig. 40.

The accuracy of the cooling tower outlet temperature functions to be used in the Dymola model for each fan speed, presents an average and maximum error detailed in the table below.

Table 7 - Cooling tower function error.

	Average Error	Maximum Error	Average Error	Maximum Error
	%	%	K	K
Full Speed	0.9%	2.8%	0.2	0.6
Half Speed	1.1%	3.2%	0.3	1

This error is considered acceptable and can be attributed to the fact that the polynomial function used for the calculation was generated based on average values calculated at different atmospheric pressures. In order to simplify the calculation, the function does not consider the variation of pressure. However, it should be noted that this error remains within a range of 3%, which is an acceptable level of accuracy.

4.3.1.2 Power electrical consumption for the cooling tower.

Depending on the fan speed

$$P_{e,CT} = P_{max,CT}\left(\frac{u}{u_{fs}}\right)^3, \tag{17}$$

Where $P_{max,CT}$ is calculated for one unit of cooling tower, with three fans, according to the values giving in Table 2. Assuming an efficiency for the electric motor of

$$\varepsilon_{CT,M} = 0.9, \tag{18}$$

and for the belt drive of also

$$\varepsilon_{CT,bD} = 0.9, \tag{19}$$

the maximum power consumption becomes

$$P_{max,CT} = 3 \times 6.5\ \mathrm{kW}\ \times \varepsilon_{CT,M} \times \varepsilon_{CT,m} = 15.8\ \mathrm{kW}. \tag{20}$$

4.3.1.3 Element representation at Dymola.

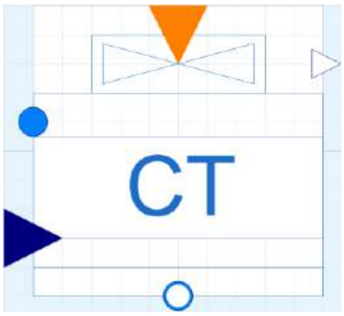


Fig. 16 - CT block in Dymola

The code programmed in the Modelica block can be seen in the annexes.

4.4 Chiller

The power consumption of a chiller providing a fixed cold-water temperature depends on the heat load and the cooling water inlet temperature into the chiller. For a fixed cooling water mass flow rate the performance data of the chiller considered here is given in Table 8.

Table 8 - Power consumption characteristics of the chiller (Source: data provided by Markus Nickolay).

Cooling water		Cooling capacity			
inlet temperature	mass flow rate	kW			
		1000	750	500	250
		Electric power consumption			
°C	kg/s	kW			
45.0	43.4	274	177	107	54
40.0	43.4	220	150	90	46
35.0	43.4	186	126	75	40
30.0	43.4	155	103	62	35
25.0	43.4	127	83	50	28
20.0	43.4	100			
Data from personal communication of M. Nickolay with DAIKIN					

4.4.1 Chiller function

The energy consumption $P_{e,C}$ of the chiller is presented graphically in Fig. 17, plotted as a function of cooling capacity or demand, using the values from Table 8 and polynomial functions derived from the data in Table 8 (dotted lines).

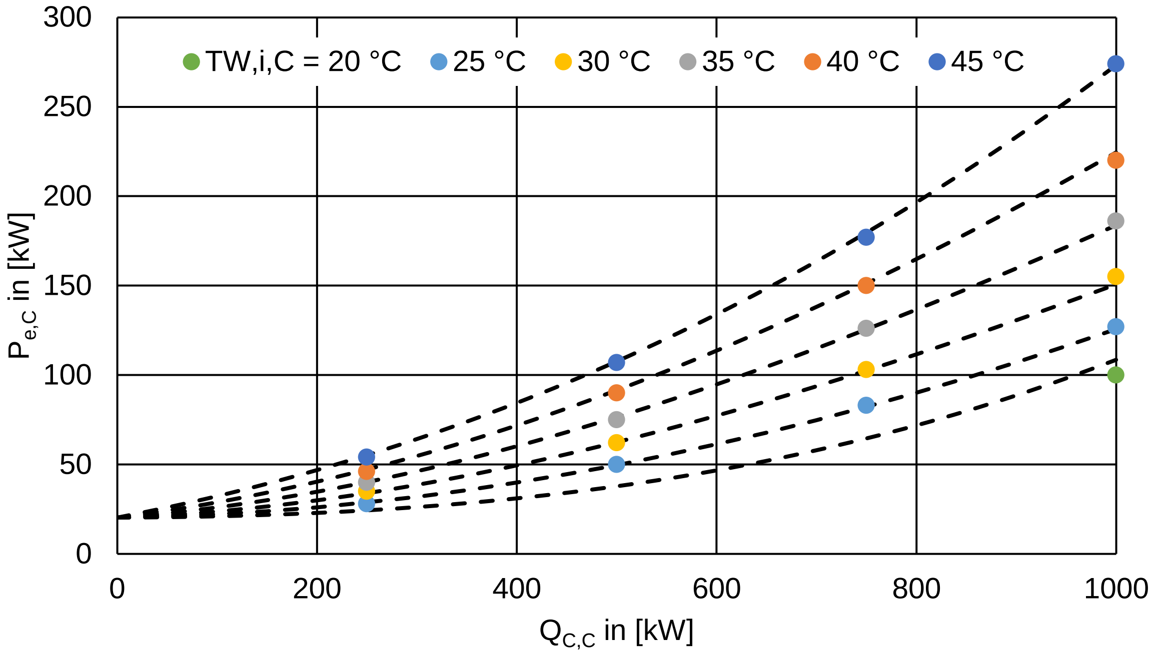


Fig. 17 - Electrical power consumption of the chiller depending on the cooling capacity $\dot{Q}_{C,C}$ for different cooling water inlet temperatures TW,i,C. Dotted lines according to (21).

It is observed that the power consumption can be approximated by the third order polynomial function (solution can be observed in the annex Fig. 41):

$$P_{e,C} = a_C + b_C \frac{\dot{Q}_{C,C}}{\dot{Q}_{C,max}} + c_C \left(\frac{\dot{Q}_{C,C}}{\dot{Q}_{C,max}}\right)^2 + d_C \left(\frac{\dot{Q}_{C,C}}{\dot{Q}_{C,max}}\right)^3 \tag{21}$$

with the coefficients depending on the cooling water inlet temperature.

$$a_C = a_{C,0} + a_{C,1} T_{W,i,C} + a_{C,2} T_{W,i,C}^2, \tag{22}$$

$$b_C = b_{C,0} + b_{C,1} T_{W,i,C} + b_{C,2} T_{W,i,C}^2, \tag{23}$$

$$c_C = c_{C,0} + c_{C,1} T_{W,i,C} + c_{C,2} T_{W,i,C}^2, \tag{24}$$

and

$$d_C = d_{C,0} + d_{C,1} T_{W,i,C} + d_{C,2} T_{W,i,C}^2. \tag{25}$$

With the coefficients given in Table 9 - Coefficients for function (22), (23), (24) and (25). equation (21) matches the data from Table 8 with an absolute error below 2.8 % except for a cooling water inlet temperature of 20°C, where the absolute relative error is 8.5 %.

Table 9 - Coefficients for function (22), (23), (24) and (25).

$\dot{Q}_{C,max}$	[kW]	1000
$a_{C,0}$	[kW]	20.3408613961802
$a_{C,1}$	[kW/°C]	0
$a_{C,2}$	[kW/(°C)^2]	0
$b_{C,0}$	[kW]	62.1867258363107
$b_{C,1}$	[kW/°C]	-6.1459680528564
$b_{C,2}$	[kW/(°C)^2]	0.158866563465633
$c_{C,0}$	[kW]	-435.474502838735
$c_{C,1}$	[kW/°C]	33.166504297054
$c_{C,2}$	[kW/(°C)^2]	-0.460894808512473
$d_{C,0}$	[kW]	471.713649870264
$d_{C,1}$	[kW/°C]	-30.690248608169
$d_{C,2}$	[kW/(°C)^2]	0.459866192731157

At the end all the electrical power consumed is changed into heat and has to be removed over the cooling towers. A smaller part is dissipated from the electrical motor of the chiller that drives the compressor to the ambient air. If we assume the efficiency of the electrical motor to be

$$\varepsilon_{C,M} = 0.9, \tag{26}$$

the heat dissipated into the cooling water becomes

$$\dot{Q}_{C,R} = \dot{Q}_{C,C} + \varepsilon_{C,M} P_{e,C}. \tag{27}$$

4.4.1.1 Element representation at Dymola.

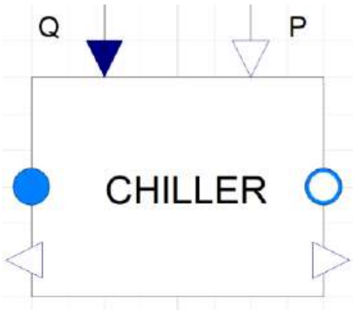


Fig. 18 - Chiller block in Dymola

The code programmed in the Modelica block can be seen in the annexes.

4.5 Smaller components

In this section some of the essential components for the operation of the system are mentioned, with their respective specifications and operating parameters, such as pipes, valves, pumps, heat source and controller.

4.5.1 Pipes

The element is taken from the Modelica standard library [5], and it is a model for a pipe flow without mass or energy storage.

“Static Pipe” is a model of a straight pipe with constant cross section and with steady-state mass, momentum, and energy balances, this is, the model does not store mass or energy. There exist two thermodynamic states, one at each fluid port. The momentum balance is formulated for the two states, considering momentum flows, friction, and gravity.

The parameters giving to the pipes were taken from the real facility and they are related to the geometry and static head, such as, length, diameter, and height.

Roughness is set as default value for smooth steel pipe and the flow reversal assumption is not allowed.

4.5.1.1 Element representation at Dymola.

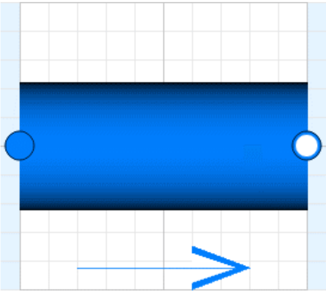


Fig. 19 - Pipe block in Dymola

4.5.2 Valves

Valve model for incompressible fluids is taken from the Modelica standard library.

The element has a parameter called “checkValve”, which can be selected as true or false. If it is false, the valve supports reverse flow, with a symmetric flow characteristic curve. Otherwise, reverse flow is stopped (check valve behavior). For this installation, “checkValve” is selected as true.

The parameters that define the operational point in the valve were set in order to ensure the design flow rate and pressure of each element of the installation,

Each valve is connected to a start ramp signal to avoid initial value problems.

4.5.2.1 Element representation at Dymola.

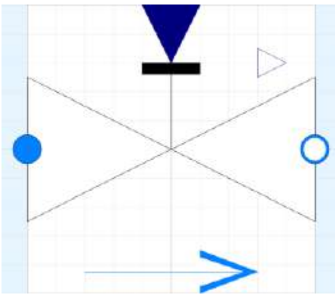


Fig. 20 - Valve block in Dymola

4.5.3 Pumps

For simulation of the pumps the model “Centrifugal pump with ideally controlled speed” from the Modelica standard library was used.

This element represents a model of a centrifugal pump with an impeller that rotates at a constant speed. The pump is assumed to be operating in an idealized manner, meaning that there are no losses due to friction, leakage, or other factors that would affect its efficiency.

The inputs provided to the model for each pump in the cooling plant are determined by their operating curves given in the annex (Fig. 42, Fig. 43 and Fig. 44).The operating curves of each pump are unique and specify the flow rate and head required for the pump to operate effectively.

4.5.3.1 Element representation at Dymola.

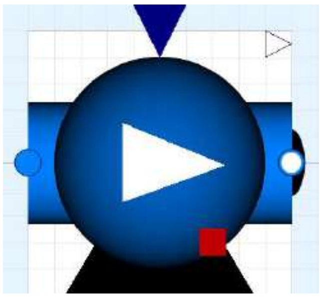


Fig. 21 - Pump block in Dymola

4.5.4 Heat Source

Basic pipe flow model without storage of mass or energy

To simulate the heat source, a pipe with heat exchange was used, connected to a trapezoidal signal generator element. This signal received by the pipe simulates the heat flow received by the fluid from the heat source.

This element can be configured to obtain either a periodic signal, which simulates the daily operation of an industrial plant with a single work shift, or a constant signal, that is, a plant that operates full time. (Fig. 25).

Both the pipe and the signal source are taken from the standard Dymola library, only that the pipe is modified by adding an input connector.

4.5.4.1 Element representation at Dymola.

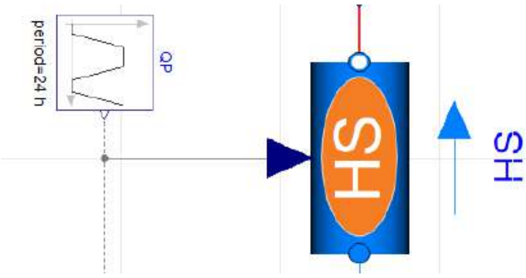


Fig. 22 - Heat source representation in Dymola

The code programmed in the Modelica block can be seen in the annexes.

4.5.5 Temperature sensors.

This component, also used from the Modelica standard library, monitors the temperature of the fluid passing its port. The sensor is ideal, that means, it does not influence the fluid.

In this installation the temperature sensor is a key element since the controller makes decisions based on these measurements.

4.5.5.1 Element representation at Dymola.

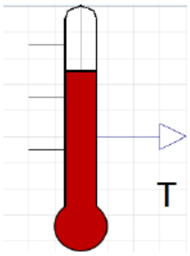


Fig. 23 - Temperature sensor block in Dymola.

4.5.6 Controller

All the elements mentioned in this section can be seen in Fig. 24.

The controller is a basic closed-loop feedback control system that monitors the water temperature at the cooling tower outlet, with a temperature sensor every 10 min, this cannot be more often because the fan motors are not allowed to switch below that time limit. This function is performed by the component called "sample", basically the output of this element is a Boolean signal "y", this is a trigger signal, the output "y" is only true at the sampling times (defined by the parameter period) and is false otherwise.

The "SpeedUpDown" component compares the water temperature sample taken from the cooling tower outlet with pre-defined temperature ranges and gives an integer as an output:

Table 10 - SpeedUpDown output

Cooling tower water outlet temperature	“SpeedUpDown” output
< Ttolow	-2
[Ttolow .. Tlow[	-1
[Tlow .. Thigh[	0
[Thigh .. Ttohigh[	+1
>= Ttohigh	+2

Every ten minutes the “trigger” element will add the signal to the sum of all previous signals up to an upper limit of 2 or a lower limit of -2.

The values that were used to define the temperature ranges were set based on the temperature required by the heat source, in this case 28°C, and are as follows:

- T to low: 27°C,
- T low: 27.5°C,
- T high: 28.5°C and
- T to high: 29°C.

This signal is then received by the “Distributer” which switches the fans and pumps of the cooling towers depending on the control strategy according to Table 11. The strategy can be selected manually before the simulation, in the "selector" element, depending on which strategy you want to evaluate.

Table 11 - Distributer switching matrix.

	Strategy 1				Strategy 2			
"SpeedUpDown" output	**Cooling Tower 1**		**Cooling Tower 2**		**Cooling Tower 1**		**Cooling Tower 2**	
	fan	**pump**	**fan**	**pump**	**fan**	**pump**	**fan**	**pump**
-2	off	off	off	off	off	off	off	off
-1	half speed	on	off	off	half speed	on	off	off
0	half speed	on	half speed	on	full speed	on	off	off
1	full speed	on	half speed	on	full speed	on	half speed	on
+2	full speed	on	full speed	on	full speed	on	full speed	on

4.5.6.1 Representation of the control model in Dymola.

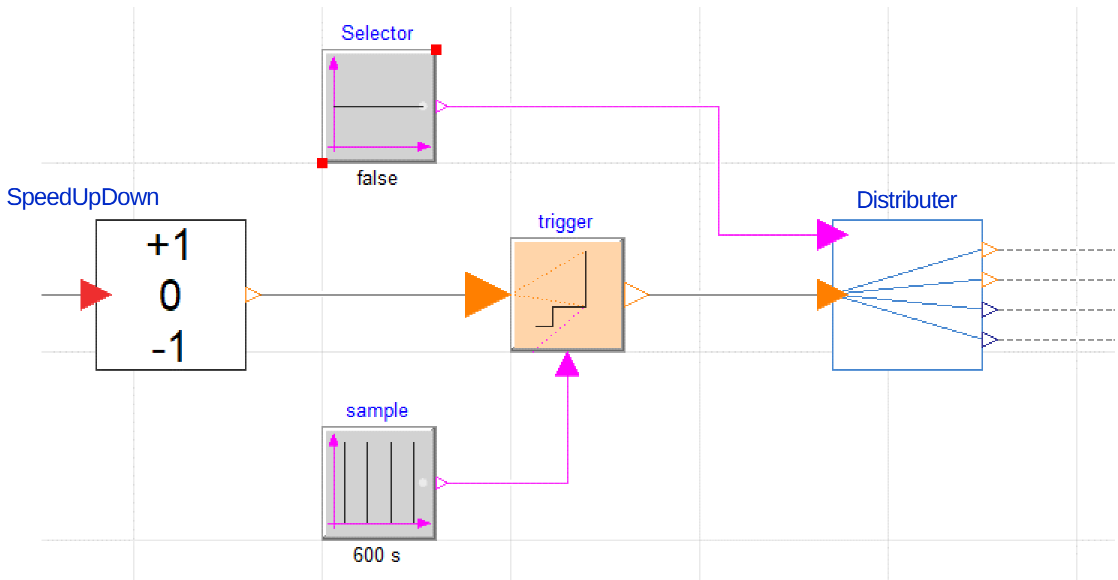


Fig. 24 - Controller representation at Dymola

The code programmed in the Modelica block can be observed in the annexes.

5 Simulations

As previously mentioned, the objective of this work is to determine which operating strategy is more beneficial from the point of view of energy consumption.

These strategies are applied when the cooling demand is such that a decision has to be made between two cooling towers operating at half speed, or only one tower operating at full speed. So, it is defined that:

- Strategy 1 refers to the case where both cooling towers operate at half fan speed and
- Strategy 2 refers to the case where only one cooling tower operates at maximum fan speed.

In addition, different operational scenarios from the point of view of cooling requirements are also presented, in order to have a broader view of the results, which allows us to reach a better conclusion.

Both strategies were simulated for all scenarios, over the course of a year, taking into account the real weather conditions of the region.

5.1 Cases

First, the different scenarios in which the model is to be simulated over the course of a year are defined:

1. Heat source and chiller full load,
2. Heat source full load, chiller off,
3. Heat source full load, chiller depending on dry bulb, no base load (Fig. 26).
4. Heat source full load, chiller depending on dry bulb, 300 kW base load (Fig. 26).
5. Heat source small load, chiller off
6. Heat source variable load, Fig. 25, chiller depending on dry bulb, 300 kW base load.
7. Heat source off, chiller full load
8. Heat source off, chiller depending on dry bulb, no base load, (Fig. 26).
9. Heat source off, chiller depending on dry bulb, 300 kW base load, (Fig. 26).

The parameters defined for the different cooling requirements of each scenario are presented in the Table 12 below.

Table 12 - Parameters for different cooling requirements.

	Production		Production Full Load				Small load	Production variable Load	Off Production and Chiller		
	Chiller		Full Load	Off	Depending on dry bulb T, no base load	Depending on dry bulb T and base load	Off	Depending on dry bulb T and base load	Full Load	Ddepending on dry bulb T, no base load	Depending on dry bulb T and base load
	Cases		1	2	3	4	5	6	7	8	9
Production Heat Source	amplitude	W	0	0	0	0	0	4.E+06	0	0	0
	rising	h	0	0	0	0	0	2	0	0	0
	width	h	0	0	0	0	0	8	0	0	0
	falling	h	0	0	0	0	0	2	0	0	0
	period	h	24	24	24	24	24	24	24	24	24
	nperiod	--	-1	-1	-1	-1	-1	-1	-1	-1	-1
	offset	W	4.E+06	4.E+06	4.E+06	4.E+06	1.E+06	0	0	0	0
	startTime	h	0	0	0	0	0	6	0	0	0
Chiller Load	OnOff	--	true	false	true	true	false	true	true	true	true
	Q0	W	1.E+06	-	0	3.E+05	-	3.E+05	1.E+06	0	3.E+05
	T0	°C	15.0	-	15.0	15.0	-	15.0	15.0	15.0	15.0
	Qmax	W	1.E+06	-	1.E+06	1.E+06	-	1.E+06	1.E+06	1.E+06	1.E+06
	Tmax	°C	35.0	-	35.0	35.0	-	35.0	35.0	35.0	35.0
	nPump	1/min	1 480	-	1 480	1 480	-	1 480	1 480	1 480	1 480
	Qmin	W	1 000	-	1 000	1 000	-	1 000	1 000	0	1 000

The "full load" as the "small load" of the heat source represents a constant demand from production, it would represent an industry that works full time, 24 hours a day, 7 days a week.

The parameters can be adjusted according to what is to be simulated, as shown in Fig. 25.

The variable load of the heat source is represented by a trapezoidal signal as follow.

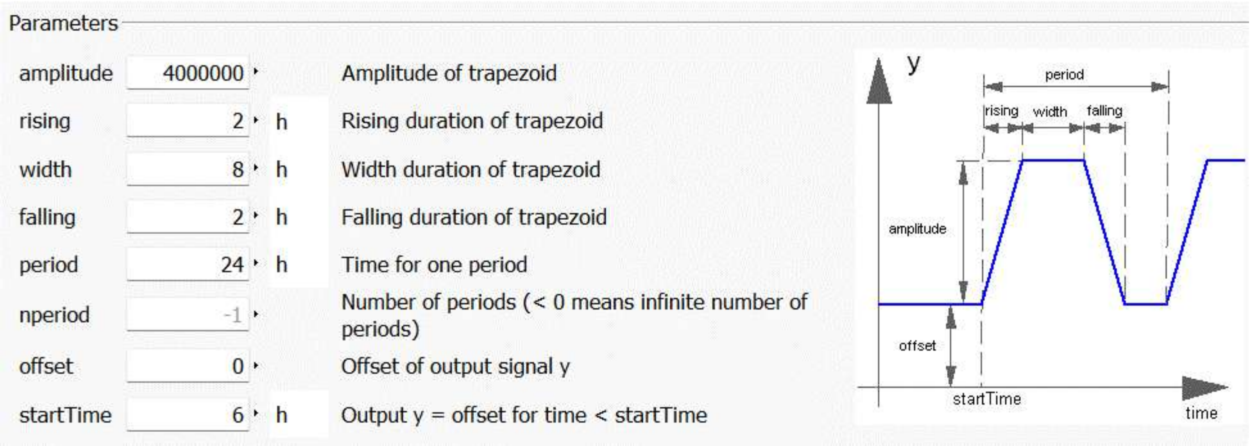


Fig. 25 -Trapezoidal signal representing the heat source load.

This trapezoidal signal would represent a variable load demanded by the production, applicable to an industry that works with a daily shift of 8hs, starting every day at 6 am.

Regarding the chiller load, four scenarios were proposed to combine with the other heat source scenarios, such as.

- Full load, i.e., Q_{max} 1000 kW.
- Load depending on the dry bulb temperature, as shown in Fig. 26, this load can be with or without a base load (Q_0= 0 kW or Q_0= 300 kW, two different scenarios), Fig. 26.
- Always off, this is, preselected or when $Q < Q_{min}$.

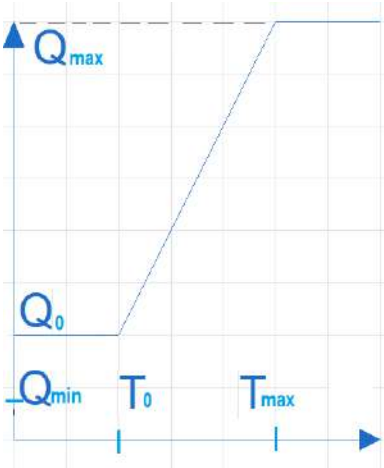


Fig. 26 - Chiller signal representing the heat load.

T_0	°C	15
T_{max}	°C	35
Q_{max}	kW	1000
Q_{min}	kW	100
Q_0	kW	300

5.2 Results

The two strategies for the nine scenarios were performed, this is a total of eighteen simulations. The collected data have been analyzed and compared to generate the results presented below. The most appropriate graph to illustrate the conclusions is the one comparing the daily energy consumption of the entire facility between the two strategies.

The values displayed in the graphs represent the difference in daily energy consumption between strategy 1 and strategy 2 Negative values indicate that strategy 1 consumed less energy than strategy 2, while positive values indicate the opposite.

It should be noted that the performance of cooling systems is highly dependent on external factors, such as weather conditions and building occupancy. Thus, the results obtained in this case study must be considered in the context of the specific conditions under which the system operates. However, the general trend observed in the results is consistent with the expected behavior of the cooling systems.

5.2.1 Heat source at full load and different conditions for the chiller

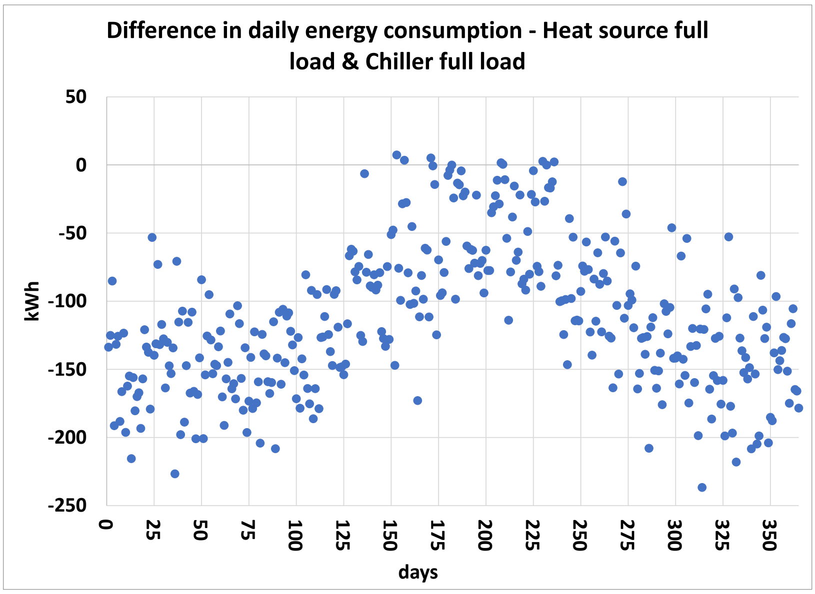


Fig. 27 - Scenario 1: Full load

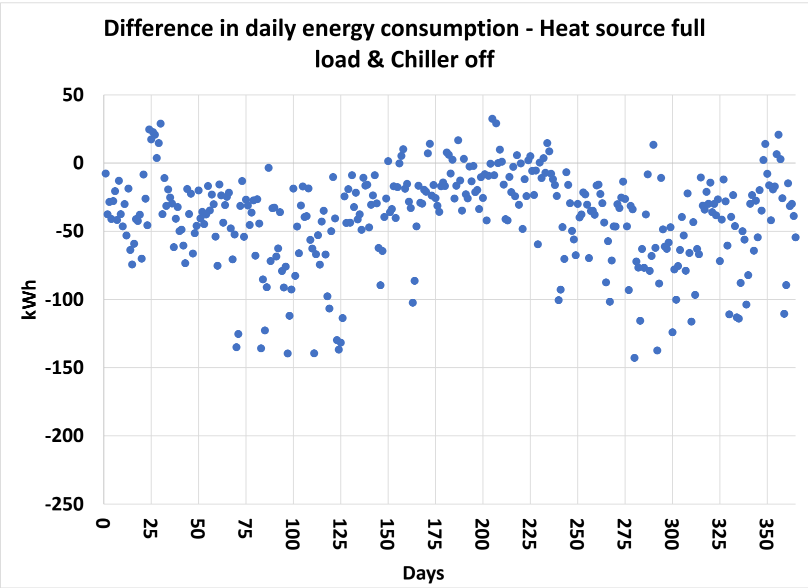


Fig. 28 - Scenario 2: HS full load, CH off

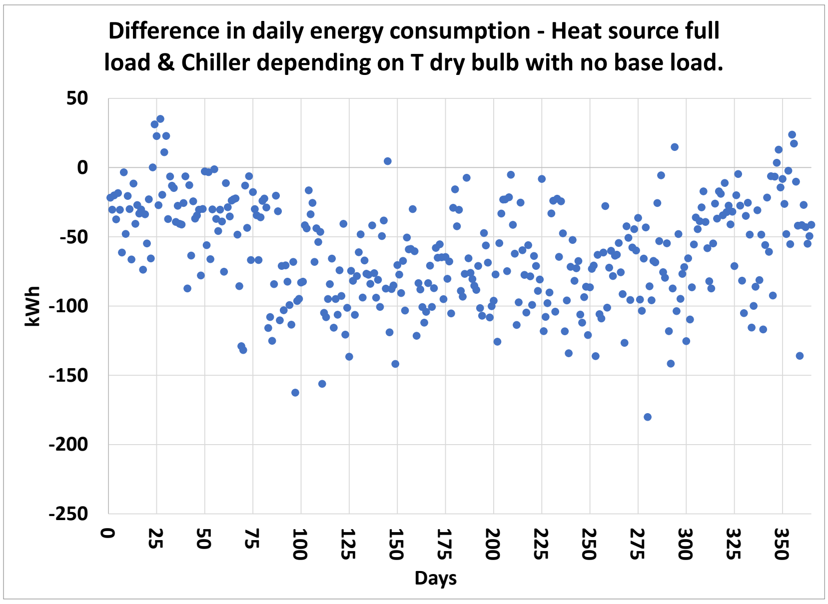


Fig. 29 - Scenario 3: HS full load, CH depending on Tdry and no base load.

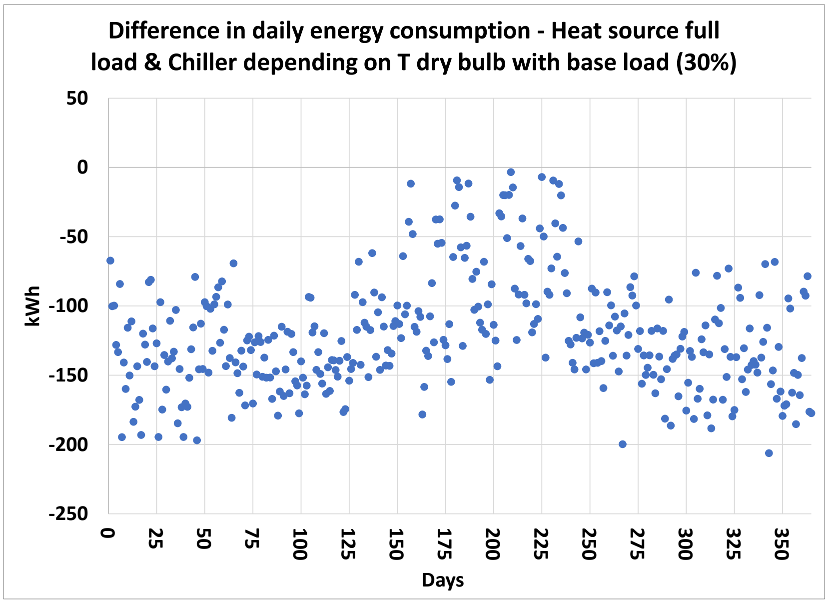


Fig. 30 - Scenario 4: HS full load, CH depending on Tdry with base load.

Although it is observed that in general the daily consumption is lower for the 4 compared cases in which the heat source operates all year round at full capacity, the greatest savings are obtained when the chiller operates at full capacity and depending on the dry bulb temperature with a base load of 30%, for strategy 1 (Fig. 27 and Fig. 30).

It should be noted that in two of the four scenarios (Fig. 27 and Fig. 30), savings were greater in the cold seasons. This is because there is less demand for cooling, so the occurrence of the system operating with the two cooling towers at medium speed is much more frequent.

In the case where the chiller operated only depending on dry bulb temperature with no base load (Fig. 29), the savings were greater in hot weather for strategy 1. The reason is that the chiller does not operate below 15 degrees, so it mostly remains off during the winter. It is observed that strategy 2 has some days in winter when the saving energy was higher than strategy 1.

In the scenario where the chiller does not operate year-round (Fig. 28), and the cooling demand of the heat source is full capacity, the greatest savings occurred in the intermediate seasons such as spring and autumn for strategy 1. This is attributed to the lower cooling demand in winter, so on very cold days, the demand can be met with just one cooling tower operating at medium speed. The opposite occurs in summer, where on very hot days, the demand for cooling is higher, so generally at least one of the two towers will operate at maximum speed. In this case, strategy 2 has some days with higher savings.

5.2.2 Small requirement for the heat source without chiller

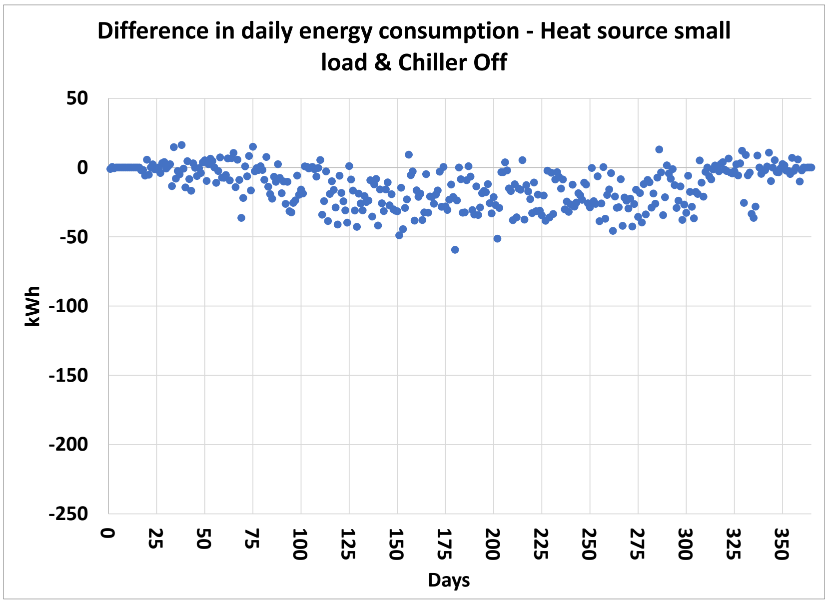


Fig. 31 - Scenario 5: HS small load, CH off

A small load refers to a small demand from the heat source, i.e., 25% of the full capacity, without a chiller. This demand is only 1 MW of production, as can be seen in Table 12.

In this case, it is observed that the opportunity for savings is also generally with the strategy where both cooling towers operate at half speed. However, it should be possible to improve the energy saving by switching between both strategies, i.e., in cold season when strategy 2 has saved more energy than strategy 1.

The greatest savings occur during the hot season when the cooling requirement is even higher. In cold seasons, the trend is not very clear, as both strategies have days with less consumption.

5.2.3 Heat Source with variable load, and chiller operating as function of wet bulb Temperature with base load of 30%.

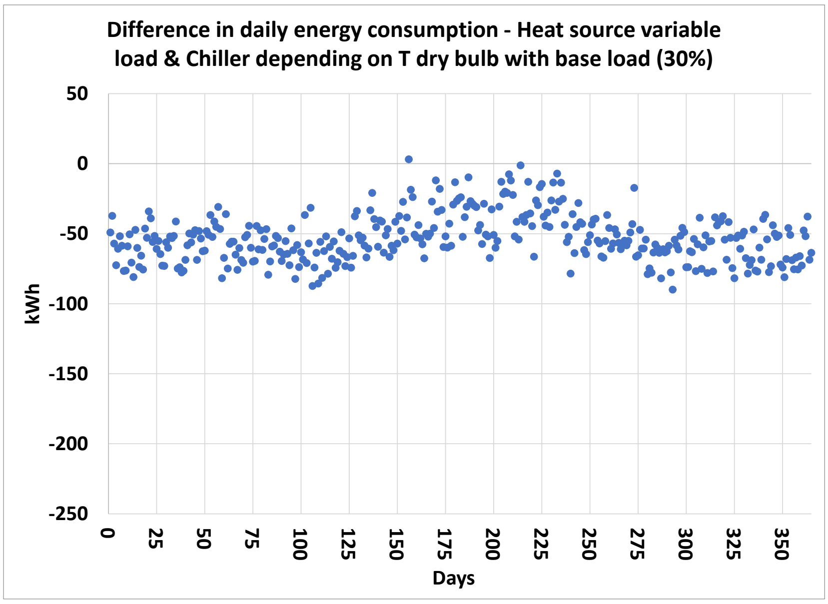


Fig. 32 - Scenario 6: HS variable load, CH depending on Tdry with base load.

In this case, one strategy has proven to be significantly more efficient than the other. The savings obtained with strategy 1 are higher, and although they are even greater during the cold seasons, they are not statistically significant. On average, the daily energy savings are estimated at 50 kWh.

5.2.4 Heat source off and chiller operating at full cooling requirements.

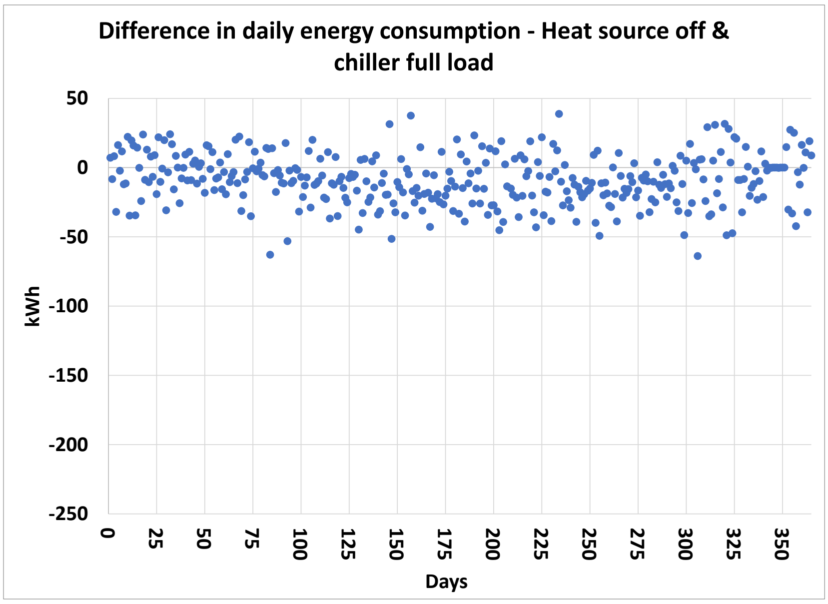


Fig. 33 - Scenario 7: HS off, CH full load.

Neither of the two strategies appears to offer a significant advantage in terms of energy savings. This can be attributed to the fact that the demand for chiller cooling is sufficiently low that one strategy is not imposed over another. However, in the summer, strategy 1 shows a slight advantage over strategy 2 on average. The days with the greatest savings for strategy 1 were observed in the intermediate seasons (spring and autumn), while for strategy 2, they were in the winter.

In this scenario it is remarkably interesting to identify the variables for which each strategy is superior to the other in terms of energy savings, and by alternating between them, the savings can be doubled.

5.2.5 Heat source off, chiller depending on T dry bulb and no base load.

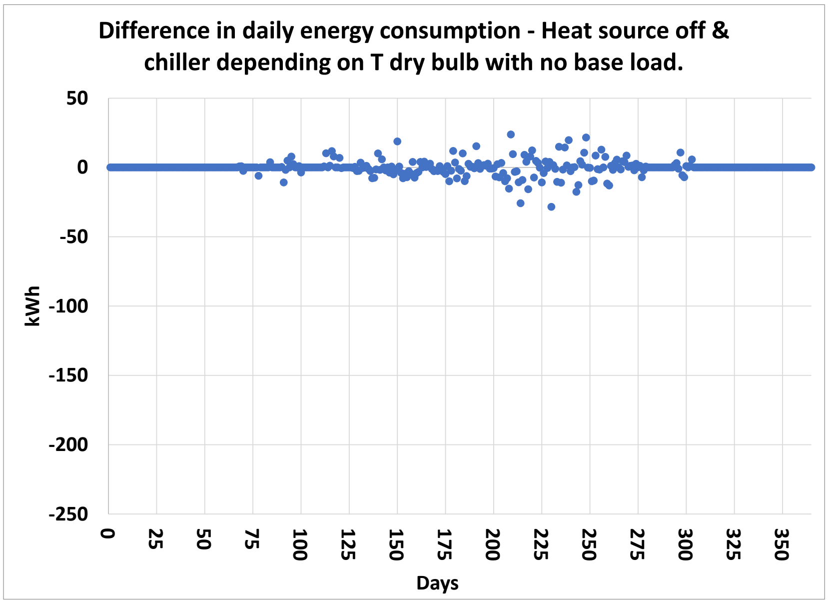


Fig. 34 - Scenario 8: HS off, CH depending on Tdry with no base load.

There are no significant differences between the two strategies in terms of their results to declare that one is better than the other. During the cold season, when the dry bulb temperature drops below 15°C, neither the chiller nor the heat source operates. As a result, this scenario is only relevant to analyse in warm season (summer and part of spring and autumn), and the best energy saving may be achieved by switching between both strategies.

5.2.6 Heat source off and chiller operating as function of T dry bulb with a base load of 30%.

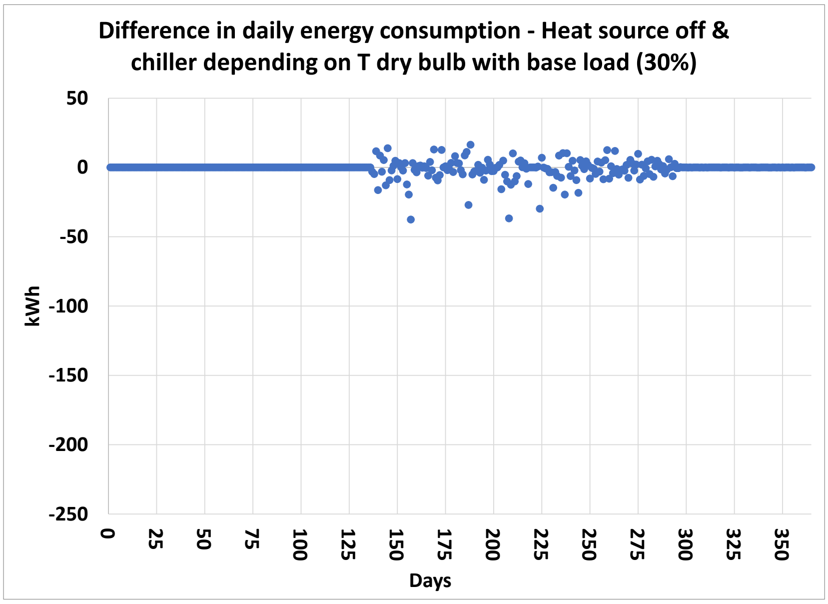


Fig. 35 - Scenario 9: HS off, CH depending on Tdry with base load.

During the winter, the cooling demand is limited to only 30% of the chiller's maximum capacity because the dry bulb temperature does not exceed 15°C. As a result, both strategies can meet the demand using only one cooling tower operating at half speed, resulting in negligible differences in energy consumption between the two strategies.

In the warmer season, when the temperature exceeds 15°C, the chiller cooling demand increases, resulting in some differences in energy consumption between the two strategies. Nevertheless, these differences are not substantial enough to determine which strategy is more efficient than the other. It should be noted, however, that the most significant savings were achieved during the operation of strategy 1, specifically on four days of the year.

5.3 Consolidating results

The Table 13 is intended to show all the consolidated results, for the different scenarios (each one could represent a different industry), comparing both strategies with the resulting energy consumption and potential savings. Here is a breakdown of the information in the table:

- Cases: This column indicates the different scenarios of combinations of the heat source (HS) and chiller (CH) cooling requirements being compared, proposed in Table 12.
- Year consumption [MWh]: This column shows the total energy consumption for each scenario, measured in megawatt-hours (MWh).
- Average daily saving [kWh]: This column shows the amount of energy that would be saved on average each day for each scenario, measured in kilowatt-hours (kWh).
- Strategy 1- savings [MWh - %]: These columns show the total amount and percentage of energy savings achieved by the first strategy being compared for each scenario over a round year.
- Highest saving [MWh - %]: These columns show the highest amount and percentage of energy savings achieved by switching strategy daily when convenient, for each scenario.
- Benefits of daily switching [%]: This column shows the percentage increase in energy savings that could be achieved by switching strategies daily when convenient, regarding to the saving of strategy 1.

Table 13 - Results of all scenarios

	Cases	Year Consumption [MWh]	Average daily saving [kWh]	Strategy 1- savings [MWh]	Highest saving [MWh]	Strategy 1- savings [%]	Highest saving [%]	Benefits of daily switching [%]
1	HS full - CH full	2142	113	41.1	41.2	2.0%	2.0%	0%
2	HS full - CH off	766	41	14.4	15.1	2.0%	2.1%	5%
3	HS full - CH Tdry	1373	62	22.2	22.6	1.6%	1.7%	2%
4	HS full - CH Tdry with base load	1248	121	44.0	44.0	3.8%	3.8%	0%
5	HS small load - CH off	562	15	4.8	5.3	0.8%	0.9%	11%
6	HS variable - CH Tdry with base load	837	53	19.3	19.4	2.5%	2.5%	0%
7	HS off - CH full load	1385	16	3.1	5.9	0.2%	0.4%	47%
8	HS off - CH Tdry	294	2	0.0	0.7	0.0%	0.2%	102%
9	HS off - CH Tdry with Base load	467	2	0.2	0.9	0.0%	0.2%	73%

The most representative energy savings that can be obtained are always with strategy 1, between 2% and 3.8% for this proposed demand, i.e., 44 MWh could be saved per year.

If the Table 13 is rearranged, based on the highest MWh saved, as shown in Table 14.

We can see that in scenarios with high cooling demand (usually where the heat source is operating at full capacity, or with variable operation added to the chiller load), strategy 1 consistently outperforms strategy 2, resulting in maximum annual savings of up to 3.8%, which translates to around 44 MWh of energy saved. However, to achieve maximum energy savings, daily switching can also provide additional benefits, resulting in an improvement of around 5% above the savings achieved by strategy 1 alone (This can be observed in column “benefits of daily switching”, Table 14).

Table 14 - Rearranged results, in terms of MWh savings.

Cases		Year Consumption	Average daily saving	Strategy 1- savings	Highest saving	Strategy 1- savings	Highest saving	Benefits of daily switching
		[MWh]	[kWh]	[MWh]	[MWh]	[%]	[%]	[%]
4	HS full - CH Tdry with base load	1248	121	44.0	**44.0**	3.8%	3.8%	**0%**
1	HS full - CH full	2142	113	41.1	**41.2**	2.0%	2.0%	**0%**
3	HS full - CH Tdry	1373	62	22.2	**22.6**	1.6%	1.7%	**2%**
6	HS variable - CH Tdry with base load	837	53	19.3	**19.4**	2.5%	2.5%	**0%**
2	HS full - CH off	766	41	14.4	**15.1**	2.0%	2.1%	**5%**
7	HS off - CH full load	1385	16	3.1	**5.9**	0.2%	0.4%	**47%**
5	HS small load - CH off	562	15	4.8	**5.3**	0.8%	0.9%	**11%**
9	HS off - CH Tdry with Base load	467	2	0.2	**0.9**	0.0%	0.2%	**73%**
8	HS off - CH Tdry	294	2	0.0	**0.7**	0.0%	0.2%	**102%**

This approach, “daily switching strategies”, could still be more relevant for some high-demand applications, such as cases 7, 8 and 9, where only the chiller operates at full capacity or the cooling demand from the heat source is small, the improvement in savings can be up to 50% (scenario 7 highlighted in Table 14), although in terms of MWh saved, it would not exceed 6 MWh per year. Nevertheless, it is more important to switch between the two strategies in these cases, because even if the energy savings are small, these could be doubled.

6 Conclusion and outlook

This thesis presents the development of models for predicting the performance of cooling towers and chillers, respectively. The cooling tower model is based on Merkel's theory with improvements by Poppe et. al. ([4]) and uses multivariable polynomial equations to estimate outlet water temperature values, while the chiller model utilizes data from the manufacturer and approximates them to a multivariable polynomial equation. Both models have been rigorously tested and have demonstrated consistent and reliable results, showing an acceptable degree of accuracy, with an error rate of less than 2.8 %.

An overall model for a plant was developed based on a real plant located in Friedberg, Hessen. The model was scaled down to two cooling towers, one chiller, and a heat source from the production process, which was tested across various scenarios with different requirements.

The energy consumption of two control strategies for cold-water temperature have been evaluated and compared with each other for nine different scenarios, each taking local weather data on an hourly basis over the course of one year into account.

The results indicate that optimizing the cooling system through different operational strategies can result in significant energy savings and reduced operation and maintenance costs in the long term. Although the model used in this study only worked with two cooling towers, the results indicate that the total energy savings could be much higher for industries with a larger number of cooling towers. The real industry used as a reference for this model has eight cooling towers.

One key finding is that there is a strong dependence between the type of scenario analyzed and the energy savings achieved, which means that savings will differ for each type of system or industry, depending on the cooling demands or requirements of the system (for example, an industrial plant that operates at full capacity, three 8-hour shifts per day, and other one that operates a single shift).

Rather than using a fixed strategy, dynamic alternation between the two strategies can result in energy savings in certain scenarios. This has been demonstrated through calculations based on daily switching.

In addition to the conclusions drawn from this thesis, there are numerous opportunities for future research and development that could further optimize the operation of cooling systems, enhance their energy efficiency, and promote sustainability.

Running the model for different cold-water temperatures could provide insights into the effect of chiller power consumption or COP on energy savings. Specifically, simulating lower cold-water temperatures than those used in this project could reveal additional opportunities for optimization.

It is important to clarify that this thesis's primary focus was on the development and rigorous testing of the models, and their performance was assessed solely through simulations using the Dymola numerical model. While real plant data validation was not included in the scope of this research, the findings serve as a crucial foundation for future work that may involve validation using actual industrial plant data.

Furthermore, future research could involve extending the model to include more cooling towers and analyzing additional scenarios to identify optimal operating parameters and conditions, as well as propose better energy-saving strategies. Additionally, verifying the model's performance with data from a real plant could improve the model's accuracy and applicability.

Moreover, the integration of smart technologies such as machine learning or artificial intelligence (AI) could enhance the efficiency of the whole system. For example, sensors and data analytics could optimize cooling operations in real-time, leading to further energy savings.

In summary, the findings of this study provide valuable insights into optimizing the operation of cooling systems and can guide decision-making in the planning and design of evaporative cooling systems for buildings and industrial facilities. With further research and development, it is possible to achieve even greater improvements in energy efficiency and sustainability.

7 Bibliography

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[8] "The Modelica Association," Modelica Association, [Online]. Available: https://modelica.org/. [Accessed 2022 9. 2.].

8 Annexes

Cooling tower datasheet:

Table 15 - Datasheet for the cooling tower installed in the real facility. (Source: Provided by Markus Nickolay)

Type			KD 3/18-32-1
Performance Data		**Unit**	**Value**
Cooling capacity		[kW]	2500
Water flow rate		[m^3/h]	238.9
Water inlet temperature		[°C]	36
Water outlet temperature		[°C]	27
Wet bulb temperature		[°C]	22
Nozzle pre-pressure required		[bar]	0.5 ± 0.15
Evaporated water rate		[m^3/h]	3.73
Technical specifications			
Fan numbers		[--]	3
Power requirements per fan		[kW]	6.5
Fan speed		U/min	230
Number of drive motors		[piece]	3
Nominal power per motor		[kW]	2.4 / 10.4
Nominal motor speed		[U/min]	720 / 1455
Nominal motor current		[A]	7 / 21.5
Protection			IP 55
Operating voltage		[V]	400
Frequency		[Hz]	50
Tolerance			± 5%
Sound Information			
Sound pressure level [(A)]		[dB]	60 ± 2
(A) - Without silencer in free field at 15 m			

Cooling tower function coefficients

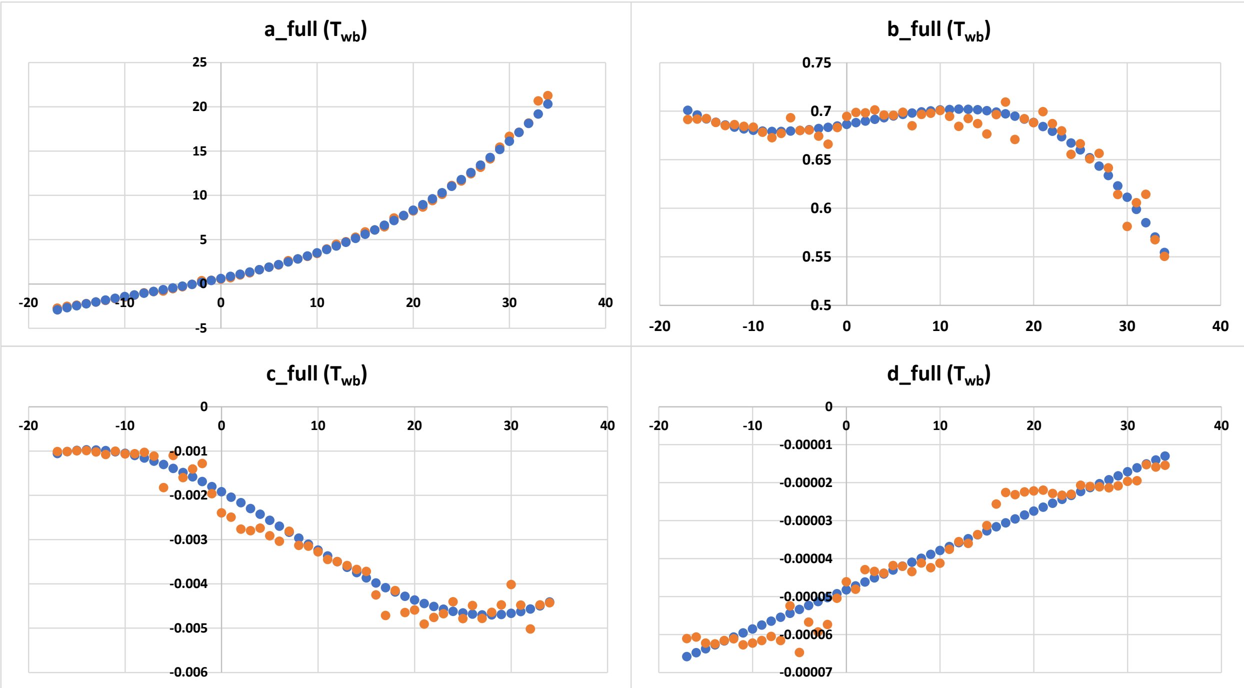

Fig. 36 - Cooling Tower coefficients depending on the cooling water inlet temperature for full fan speed.

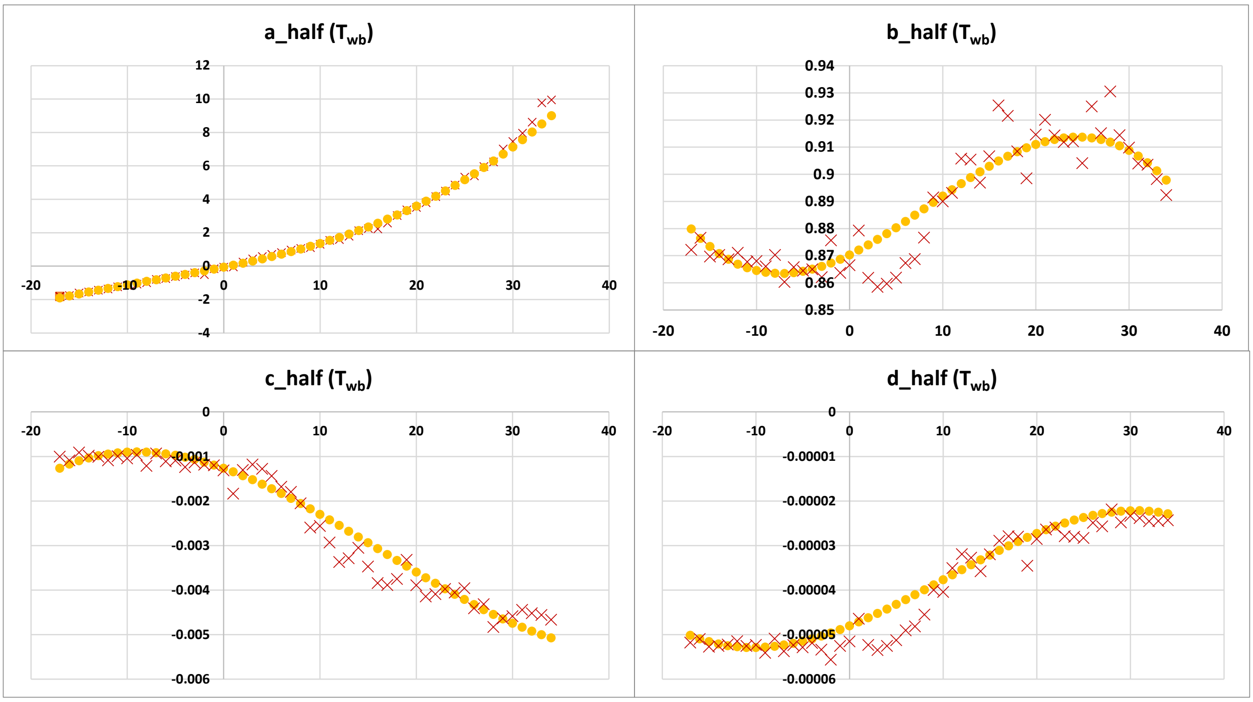

Fig. 37 - Cooling Tower coefficients depending on the cooling water inlet temperature for half fan speed.

Chiller function coefficients

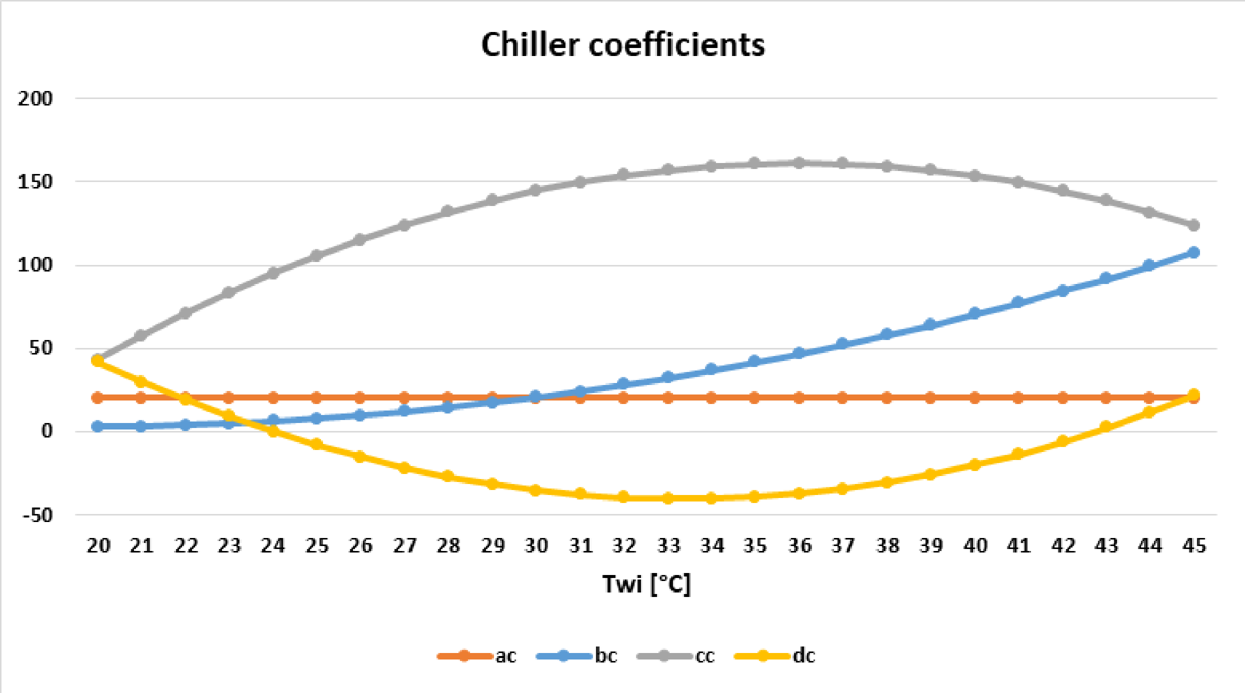

Fig. 38 - Chiller coefficients

Solution of the cooling tower function

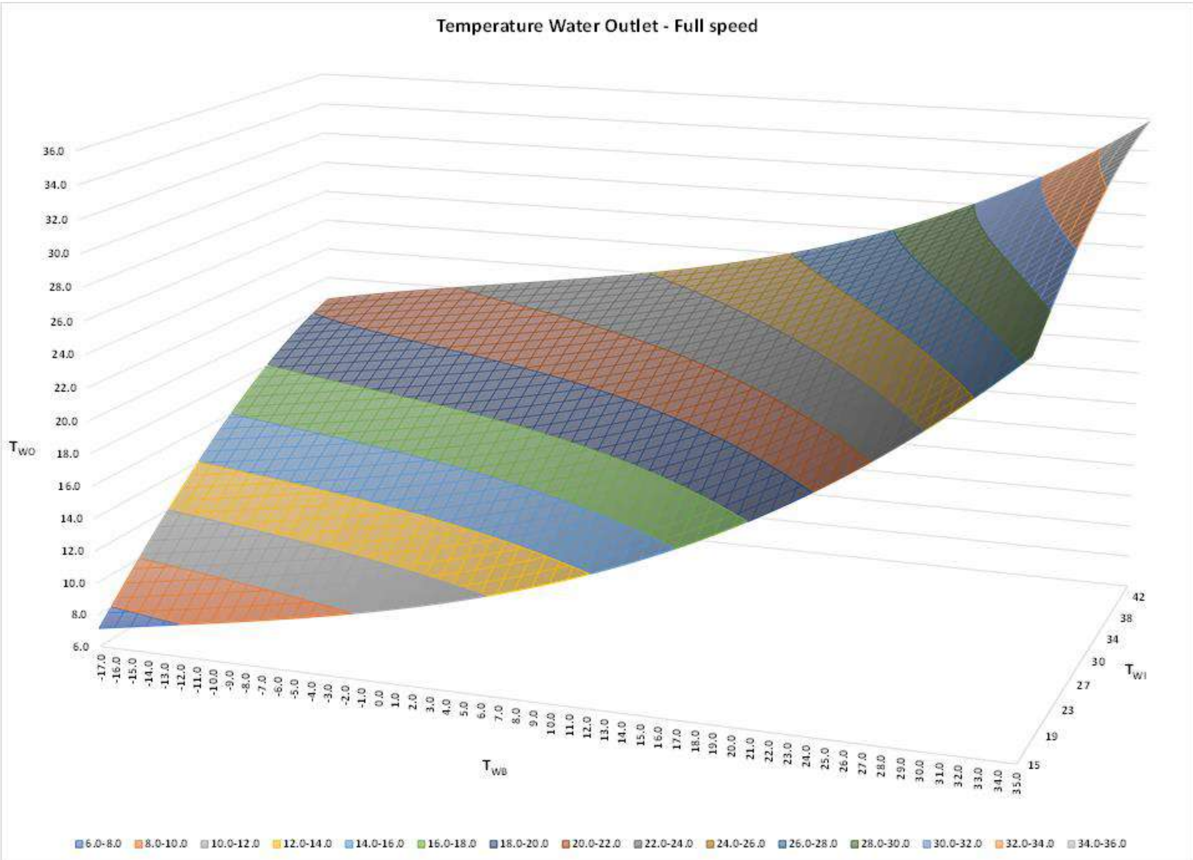

Fig. 39 - Full fan speed solution in the operational range.

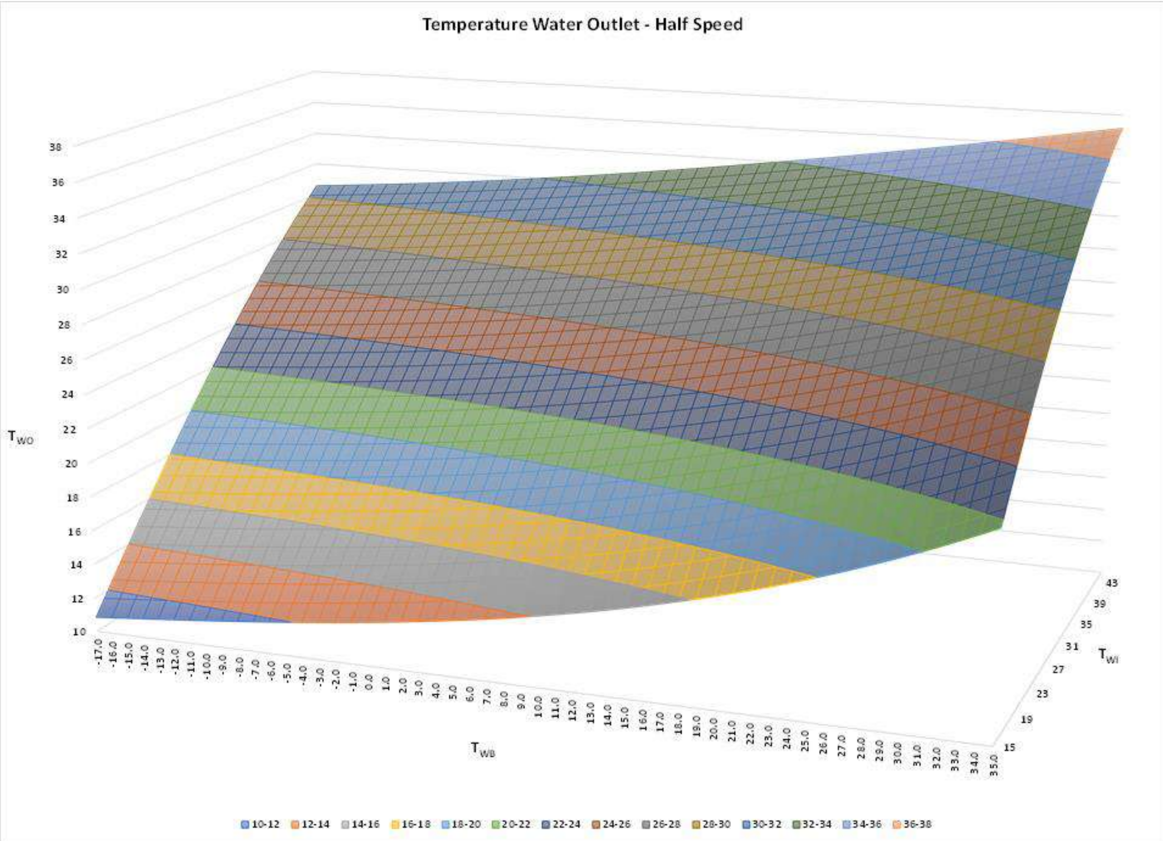

Fig. 40 - Half fan speed solution in the operational range.

Solution of the chiller function:

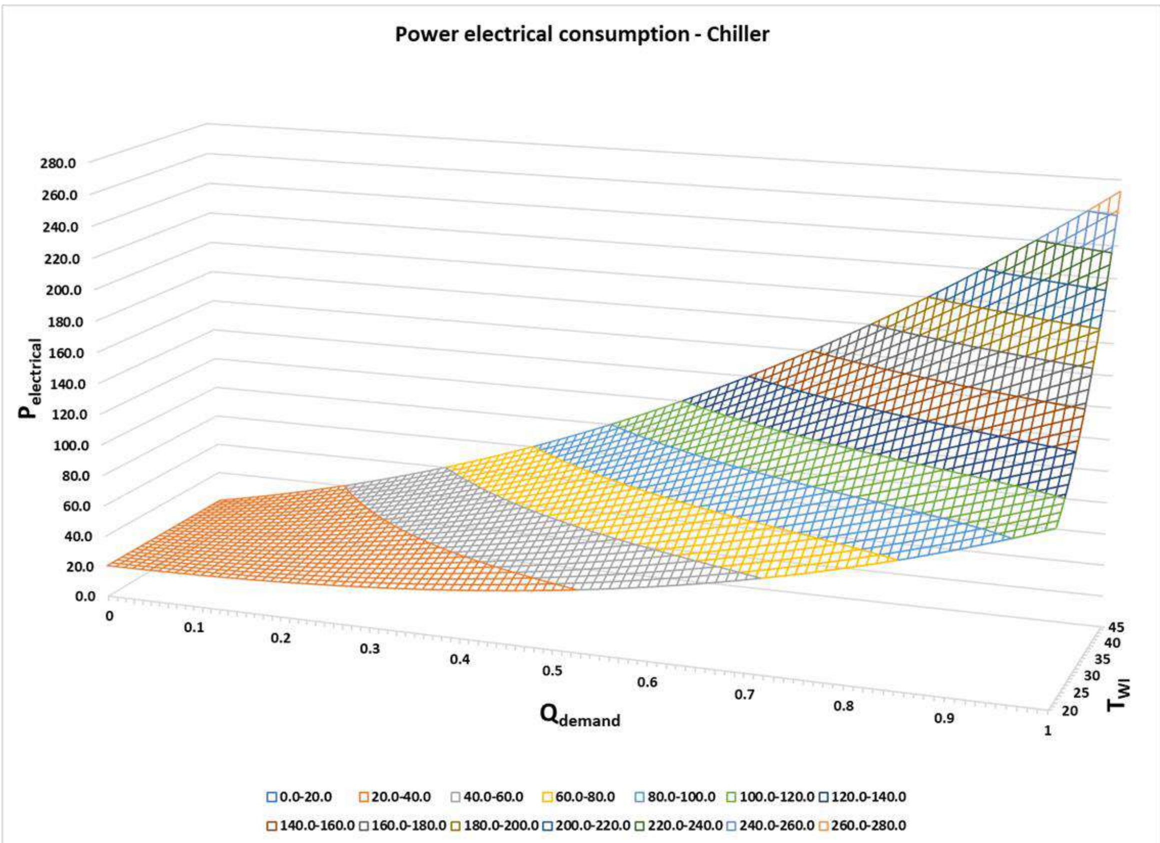

Fig. 41 - Chiller power consumption

Pumps curves:

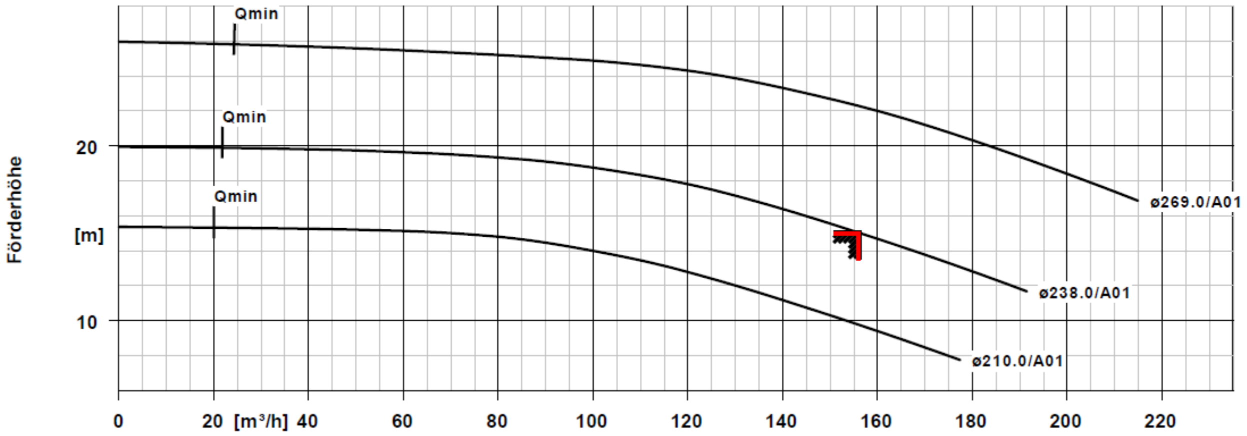

Fig. 42 - Pump curve for the chiller-pump, KSB ETN 125-100-250 GGSAA11GD301104B (Source: KSB Frankenthal).

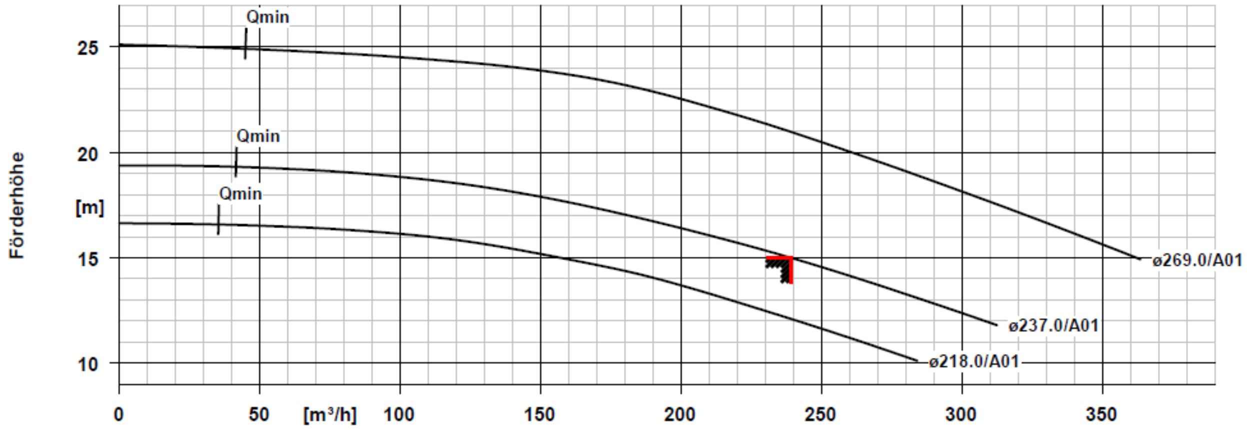

Fig. 43 - Pump curve for the cooling tower pump, ETB 150-125-250 GGSAV11D301504 (Source: KSB Frankenthal).

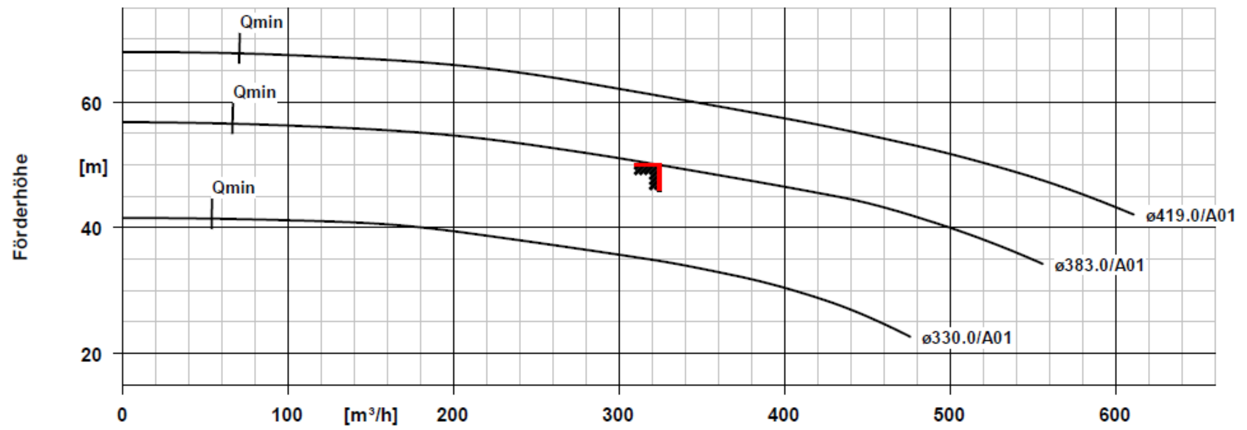

Fig. 44 - Pump curve for the production pump, ETB 200-150-400 GGSAV11D505504 (Source: KSB Frankenthal).

Weather bus model in Dymola:

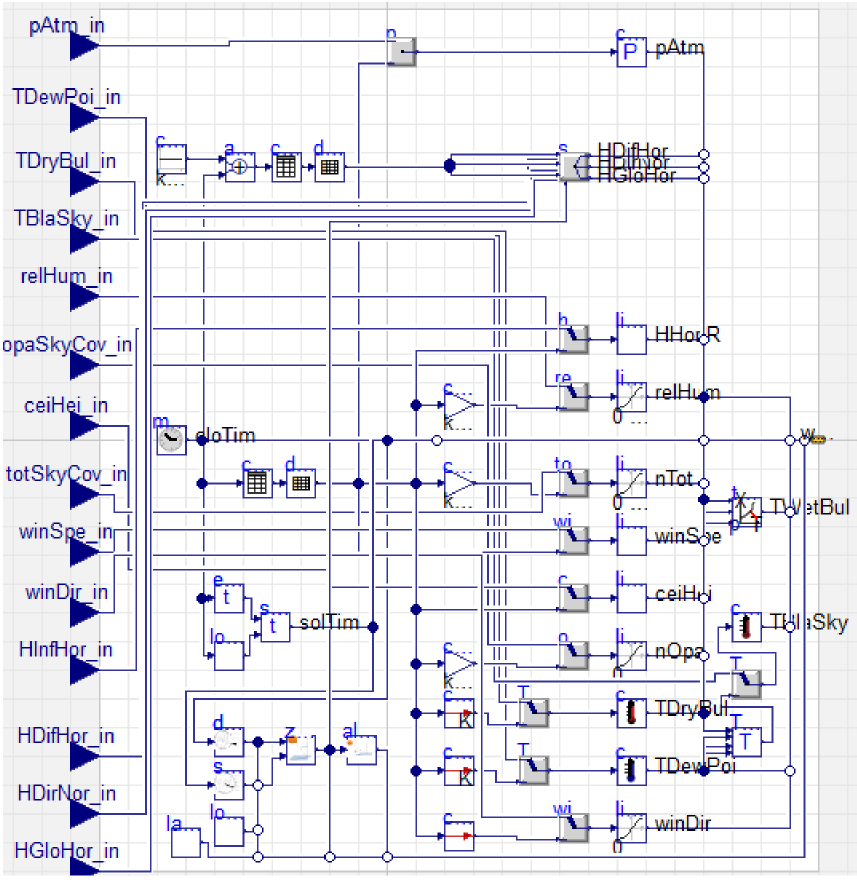

Fig. 45 - Weather bus model in Dymola

Model code of the cooling tower:

```
model CoolingTower
  import Modelica.Constants;
  outer Modelica.Fluid.System system "System wide properties";
  replaceable package Medium =
      Modelica.Media.Interfaces.PartialMedium "Medium in the component"
      annotation (choicesAllMatching = true);

    parameter Boolean allowFlowReversal=false
    "= true to allow flow reversal, false restricts to design direc-
tion (port_a -> port_b)"
    annotation(Dialog(tab="Assumptions"), Evaluate=true);

    parameter Real maxFanPower = 3*6.5/0.9/0.9 "Fan power consump-
tion in kW at full speed";

  Fluid.Interfaces.FluidPort_a          port_a(redeclare package Medium =
        Modelica.Media.Water.ConstantPropertyLiquidWater, m_flow(min=if al-
lowFlowReversal then -Constants.inf else 0))
    "Fluid connector a (positive design flow direc-
tion is from port_a to port_b)"
  Fluid.Interfaces.FluidPort_b          port_b(redeclare package Medium =
        Modelica.Media.Water.ConstantPropertyLiquidWater, m_flow(max=if al-
lowFlowReversal then +Constants.inf else 0))
    "Fluid connector b (positive design flow direc-
tion is from port_a to port_b)"
```

```
  Modelica.Blocks.Interfaces.IntegerInput FanSpeed annotation (Place-
ment(transformation(extent={{-26,90},{14,130}}),
        iconTransformation(
        extent={{-20,-20},{20,20}},
        rotation=270,
        origin={0,80})));

    Real TWi;
    Real TWo;
    Real DTW;
    Real dh;
    Real TWBi;

  Modelica.Blocks.Interfaces.RealInput TWB annotation (Placement(transfor-
mation(extent={{-120,-80},{-80,-40}}),
        iconTransformation(extent={{-120,-80},{-80,-40}})));
  Modelica.Blocks.Interfaces.RealOutput FanPower
algorithm
  if TWB > 100 then
    TWBi := TWB-273.15;
  else
    TWBi := TWB;
  end if;
  TWi := Medium.temperature(Medium.setState_phX(port_a.p, in-
Stream(port_a.h_outflow), inStream(port_a.Xi_outflow)));
  TWo := CoolingPlant20230224_0800.CoolingTowerTWO(
      TWi - 273.15,
      FanSpeed,
      TWBi) + 273.15;
  DTW := TWi-TWo;
  dh := Medium.specificHeatCapacityCp(Medium.setState_phX(port_a.p, in-
Stream(port_a.h_outflow), inStream(port_a.Xi_outflow)))*DTW;
  FanPower :=maxFanPower*(FanSpeed/2)^3;

equation

  port_a.p=port_b.p;
  0 = port_a.m_flow + port_b.m_flow;
  port_a.Xi_outflow = inStream(port_b.Xi_outflow);
  port_b.Xi_outflow = inStream(port_a.Xi_outflow);
  port_a.C_outflow = inStream(port_b.C_outflow);
  port_b.C_outflow = inStream(port_a.C_outflow);
  port_b.h_outflow = inStream(port_a.h_outflow)-dh;
  port_a.h_outflow = inStream(port_b.h_outflow);

end CoolingTower;
```

Cooling tower function:

```
function CoolingTowerTWO
  input Real TWi;
  input Integer FanSpeed;
  input Real TWB;
  output Real TWo;
// Coefficients for full fan speed
protected
 Real A0_full= 0.610461084774077;
 Real A1_full= 0.22951621211468;
 Real A2_full= 0.00429272654558364;
 Real A3_full= 0.000176125102765765;
```

```
 Real B0_full= 0.686432488311475;
 Real B1_full= 0.00164663828388901;
 Real B2_full= 4.37755436829176E-05;
 Real B3_full= -6.06976907321773E-06;
 Real C0_full= -0.00192226891980353;
 Real C1_full= -0.0001198549553991;
 Real C2_full= -2.24557862293905E-06;
 Real C3_full= 1.06415463682525E-07;
 Real D0_full= -4.82329388629608E-05;
 Real D1_full= 1.03539689577335E-06;
 Real D2_full= 0;
 Real D3_full= 0;
// Coefficients for half fan speed
 Real A0_half= -0.0653460491748675;
 Real A1_half= 0.11613485515672;
 Real A2_half= 0.00178443835410971;
 Real A3_half= 7.76711780788867E-05;
 Real B0_half= 0.870267589106905;
 Real B1_half= 0.00168286204663612;
 Real B2_half= 7.94198749547348E-05;
 Real B3_half= -3.08985255147763E-06;
 Real C0_half= -0.00126325972644525;
 Real C1_half= -7.65970076732307E-05;
 Real C2_half= -3.34562395947153E-06;
 Real C3_half= 6.78312698121485E-08;
 Real D0_half= -0.000048;
 Real D1_half= 0.00000085;
 Real D2_half= 2.73777431041312E-08;
 Real D3_half= -0.0000000009;
 Real A;
 Real B;
 Real C;
 Real D;
 Real TWBC;

algorithm

  // compute coefficients from fan speed and wet bulb temperature
  if FanSpeed == 2 then
    // Fan at full Speed
    A:=A0_full + A1_full*TWB + A2_full*TWB^2 + A3_full*TWB^3;
    B:=B0_full + B1_full*TWB + B2_full*TWB^2 + B3_full*TWB^3;
    C:=C0_full + C1_full*TWB + C2_full*TWB^2 + C3_full*TWB^3;
    D:=D0_full + D1_full*TWB + D2_full*TWB^2 + D3_full*TWB^3;
  elseif FanSpeed == 1 then
    //Fan at half Speed
    A:=A0_half + A1_half*TWB + A2_half*TWB^2 + A3_half*TWB^3;
    B:=B0_half + B1_half*TWB + B2_half*TWB^2 + B3_half*TWB^3;
    C:=C0_half + C1_half*TWB + C2_half*TWB^2 + C3_half*TWB^3;
    D:=D0_half + D1_half*TWB + D2_half*TWB^2 + D3_half*TWB^3;
  else
    //Fan is off --> no cooling
    A:=0;
    B:=1; //follows TWo = TWi
    C:=0;
    D:=0;
    end if;
// compute water outlet temperature
  TWo :=A + B*TWi + C*TWi^2 + D*TWi^3;
end CoolingTowerTWO;
```

Model code of the Chiller:

```
model Chiller
import Modelica.Constants;
outer Modelica.Fluid.System system "System wide properties";

replaceable package Medium =
      Modelica.Media.Interfaces.PartialMedium "Medium in the component"
      annotation (choicesAllMatching = true);

  parameter Boolean allowFlowReversal = system.allowFlowReversal
    "= true to allow flow reversal, false restricts to design direc-
tion (port_a -> port_b)"
    annotation(Dialog(tab="Assumptions"), Evaluate=true);
  // Initialization
  parameter Medium.AbsolutePressure p_a_start=system.p_start
    "Start value of pressure at port a"
    annotation(Dialog(tab = "Initialization"));
  parameter Medium.AbsolutePressure p_b_start=p_a_start
    "Start value of pressure at port b"
    annotation(Dialog(tab = "Initialization"));
  parameter Medium.MassFlowRate m_flow_start = system.m_flow_start
    "Start value for mass flow rate"
     annotation(Evaluate=true, Dialog(tab = "Initialization"));

  parameter Real Qmax = 1000000 "design heat load in W";
  parameter Real eta = 0.9 "efficency of the motor";
  parameter Real minFlowRate = 0.1 "kg/s";

  Real TWi;
  Real T_a "Temperature at port a";
  Real T_b " Temperature at port b";
  Real dh_port_a;
  Real dh_port_b( start=0);
  Real dh;

  Modelica.Blocks.Interfaces.RealInput Qload annotation (Placement(transfor-
mation(extent={{-18,20},{22,60}}),
        iconTransformation(
        extent={{-10,-10},{10,10}},
        rotation=270,
        origin={-40,70})));

  Modelica.Blocks.Interfaces.RealOutput power annotation (Placement(transfor-
mation(extent={{54,30},{74,50}}),
        iconTransformation(
        extent={{-10,-10},{10,10}},
        rotation=270,
        origin={40,70})));

  Fluid.Interfaces.FluidPort_a          port_a(redeclare package Medium = Me-
dium, m_flow(min=if allowFlowReversal then -
          Constants.inf else 0))
    "Fluid connector a (positive design flow direc-
tion is from port_a to port_b)"
  Fluid.Interfaces.FluidPort_b          port_b(redeclare package Medium = Me-
dium, m_flow(max=if allowFlowReversal then +
          Constants.inf else 0))
```

```
    "Fluid connector b (positive design flow direc-
tion is from port_a to port_b)"

  Modelica.Blocks.Interfaces.RealOutput TWO_output
    annotation (Placement(transformation(extent={{74,-50},{94,-30}}), icon-
Transformation(extent={{74,-50},{94,-30}})));
  Modelica.Blocks.Interfaces.RealOutput TWI_output annotation (Place-
ment(transformation(extent={{74,-50},{94,-30}}),
        iconTransformation(extent={{-74,-50},{-94,-30}})));
algorithm
  //inlet tempeerature
  T_a := Medium.temperature(Medium.setState_phX(port_a.p, in-
Stream(port_a.h_outflow), inStream(port_a.Xi_outflow)));
  T_b := Medium.temperature(Medium.setState_phX(port_b.p, in-
Stream(port_b.h_outflow), inStream(port_b.Xi_outflow)));
  if port_a.m_flow > 0 then
    TWi :=T_a;
    else
    TWi :=T_b;
    end if;

    TWI_output:=T_a-273.15;
    TWO_output:=T_b-273.15;

  // power consumption
  // change in specific enthalpie
  if (abs(port_a.m_flow)> minFlowRate) and (Qload > 0) then
    power := ChillerPowerConsumption(
        Qload,
        Qmax,
        TWi - 273.15);
    dh :=abs((Qload + eta*power)/abs(port_a.m_flow));
  else
    power := 0;
    dh :=0;
  end if;

  // change of specific enthalpie at port A
  if abs(port_a.m_flow) > minFlowRate then
    dh_port_a :=0;
    dh_port_b :=dh;
  else
    dh_port_a :=dh;
    dh_port_b :=0;
  end if;

equation
  // Mass balance
  port_a.p=port_b.p;
  0 = port_a.m_flow + port_b.m_flow;
  port_a.Xi_outflow = inStream(port_b.Xi_outflow);
  port_b.Xi_outflow = inStream(port_a.Xi_outflow);
  port_a.C_outflow = inStream(port_b.C_outflow);
  port_b.C_outflow = inStream(port_a.C_outflow);
  port_b.h_outflow = inStream(port_a.h_outflow)+dh_port_b;
  port_a.h_outflow = inStream(port_b.h_outflow)+dh_port_a;
end Chiller;
```

Chiller function:

```
function ChillerPowerConsumption
  input Real Q;
  input Real Qmax_;
  input Real T;
  output Real p;

protected
  Real A0 = 20.3408613961802;
  Real A1 = 0;
  Real A2 = 0;
  Real B0 = 62.1867258363107;
  Real B1 = -6.1459680528564;
  Real B2 = 0.158866563465633;
  Real C0 = -435.474502838735;
  Real C1 = 33.166504297054;
  Real C2 = -0.460894808512473;
  Real D0 = 471.713649870264;
  Real D1 = -30.690248608169;
  Real D2 = 0.459866192731157;
  Real q =  Q/Qmax_;
  Real A =  A0 + A1*T + A2*T^2;
  Real B =  B0 + B1*T + B2*T^2;
  Real C =  C0 + C1*T + C2*T^2;
  Real D =  D0 + D1*T + D2*T^2;

algorithm
  p:=max(0,A+B*q+C*q^2+D*q^3);
end ChillerPowerConsumption;
```

Code of the Heat Source block:

```
model StaticPipeWithHeatFlow "Basic pipe flow model without stor-
age of mass or energy"

  // extending PartialStraightPipe
  extends Modelica.Fluid.Pipes.BaseClasses.PartialStraightPipe;
  Real q;

  // Initialization
  parameter Medium.AbsolutePressure p_a_start=system.p_start
    "Start value of pressure at port a"
    annotation(Dialog(tab = "Initialization"));
  parameter Medium.AbsolutePressure p_b_start=p_a_start
    "Start value of pressure at port b"
    annotation(Dialog(tab = "Initialization"));
  parameter Medium.MassFlowRate m_flow_start = system.m_flow_start
    "Start value for mass flow rate"
     annotation(Evaluate=true, Dialog(tab = "Initialization"));

  FlowModel flowModel(
          redeclare final package Medium = Medium,
          final n=2,
          states={Medium.setState_phX(port_a.p, inStream(port_a.h_out-
flow), inStream(port_a.Xi_outflow)),
                 Medium.setState_phX(port_b.p, inStream(port_b.h_out-
flow), inStream(port_b.Xi_outflow))},
          vs={port_a.m_flow/Medium.den-
sity(flowModel.states[1])/flowModel.crossAreas[1],
```

```
               -port_b.m_flow/Medium.den-
sity(flowModel.states[2])/flowModel.crossAreas[2]}/nParallel,
          final momentumDynamics=Fluid.Types.Dynamics.SteadyState,
          final allowFlowReversal=allowFlowReversal,
          final p_a_start=p_a_start,
          final p_b_start=p_b_start,
          final m_flow_start=m_flow_start,
          final nParallel=nParallel,
          final pathLengths={length},
          final crossAreas={crossArea, crossArea},
          final dimensions={4*crossArea/perimeter, 4*crossArea/perimeter},
          final roughnesses={roughness, roughness},
          final dheights={height_ab},
          final g=system.g) "Flow model"
     annotation (Placement(transformation(extent={{-38,-18},{38,18}})));
  Modelica.Blocks.Interfaces.RealInput Qin annotation (Placement(transfor-
mation(extent={{-40,40},{0,80}}),
        iconTransformation(
        extent={{-20,-20},{20,20}},
        rotation=270,
        origin={0,62})));
equation

  // Mass balance
  port_a.m_flow = flowModel.m_flows[1];
  if abs(port_a.m_flow) > 0.1 then
    q = Qin/port_a.m_flow;
  else
    q = 0;
  end if;
  0 = port_a.m_flow + port_b.m_flow;
  port_a.Xi_outflow = inStream(port_b.Xi_outflow);
  port_b.Xi_outflow = inStream(port_a.Xi_outflow);
  port_a.C_outflow = inStream(port_b.C_outflow);
  port_b.C_outflow = inStream(port_a.C_outflow);

  // Energy balance, considering change of potential energy
  // Wb_flow = v*A*dpdx + v*F_fric
  //         = m_flow/d/A * (A*dpdx + A*pressureLoss.dp_fg - F_grav)
  //         = m_flow/d/A * (-A*g*height_ab*d)
  //         = -m_flow*g*height_ab
  port_b.h_outflow =inStream(port_a.h_outflow) - system.g*height_ab + q;
  port_a.h_outflow =inStream(port_b.h_outflow) + system.g*height_ab - q;
  annotation (defaultComponentName="pipe",
Documentation(info="<html>
<p>Model of a straight pipe with constant cross section and with steady-
state mass, momentum and energy bal-
ances, i.e., the model does not store mass or energy.
There exist two thermodynamic states, one at each fluid port. The momen-
tum balance is formulated for the two states, taking into account
momentum flows, friction and gravity. The same result can be obtained by us-
ing <a href=\"modelica://Modelica.Fluid.Pipes.DynamicPipe\">Dynam-
icPipe</a> with
steady-state dynamic settings. The intended use is to provide simple connec-
tions of vessels or other devices with storage, as it is done in:
</p>
<ul>
<li><a href=\"modelica://Modelica.Fluid.Examples.Tanks.EmptyTanks\">Exam-
ples.Tanks.EmptyTanks</a></li>
<li><a href=\"modelica://Modelica.Fluid.Examples.InverseParameteriza-
tion\">Examples.InverseParameterization</a></li>
```

```
</ul>
<h4>Numerical Issues</h4>
<p>
With the stream connectors the thermodynamic states on the ports are gener-
ally defined by models with storage or by sources placed upstream and down-
stream of the static pipe.
Other non storage components in the flow path may yield to state transfor-
mation. Note that this generally leads to nonlinear equation systems if mul-
tiple static pipes,
or other flow models without storage, are directly connected.
</p>
</html>"),
end StaticPipeWithHeatFlow;
```

Controller code:

“FanSpeedUpDown” block:

```
block FanSpeedUpDown "y=+1: Deamand for mor air, y=-1: Demand for less air"
  TemperatureInput T annotation (Placement(transformation(extent={{-120,-
20},{-80,20}}),iconTransformation(extent={{-120,
            -20},{-80,20}})));
  Modelica.Blocks.Interfaces.IntegerOutput s annotation (Placement(transfor-
mation(extent={{100,-10},{120,10}}),
        iconTransformation(extent={{100,-10},{120,10}})));

  parameter SI.Temperature Ttolow = Tlow-1;
  parameter SI.Temperature Tlow= 273.15+28;
  parameter SI.Temperature Thigh = 273.15+29;
  parameter SI.Temperature Ttohigh = Thigh+1;
equation

  if T < Ttolow then
    s = -2;
  else
    if (T>=Ttolow) and
                          (T<Tlow) then
      s = -1;
    else
      if (T>=Tlow) and (T<Thigh) then
        s = 0;
      else
        if (T>=Thigh) and (T<Ttohigh) then
          s = 1;
        else
          s = 2;
        end if;
      end if;
    end if;
  end if;

  //s=if (T>Thigh) then 1 elseif (T<Tlow) then -1 else 0;
end FanSpeedUpDown;
```

"Trigger" block:

```
block TriggeredAddLimited
  "Add input to previous value of output, if rising edge of trig-
ger port. Limit Results."
  extends Modelica.Blocks.Interfaces.PartialIntegerSISO;

  parameter Boolean use_reset = false "= true, if reset port enabled"
        annotation(Evaluate=true, HideResult=true, choices(checkBox=true));
  parameter Boolean use_set = false
    "= true, if set port enabled and used as default value when reset"
        annotation(Dialog(enable=use_reset), Evaluate=true, HideRe-
sult=true, choices(checkBox=true));
  parameter Integer y_start = 0
    "Initial and reset value of y if set port is not used";
  parameter Integer maxY = 2
  "Limit y to maxY";
  parameter Integer minY= -2
  "Limit y to minY";

  Modelica.Blocks.Interfaces.BooleanInput trigger annotation (Placement(
        transformation(
        extent={{-20,-20},{20,20}},
        rotation=90,
        origin={-60,-120}), iconTransformation(
        extent={{-20,-20},{20,20}},
        rotation=90,
        origin={0,-120})));
  Modelica.Blocks.Interfaces.BooleanInput reset if use_reset annota-
tion (Placement(
        transformation(
        extent={{-20,-20},{20,20}},
        rotation=90,
        origin={60,-120})));
  Modelica.Blocks.Interfaces.IntegerInput set if use_set annotation (Place-
ment(transformation(
        extent={{-20,-20},{20,20}},
        rotation=270,
        origin={60,120}), iconTransformation(
        extent={{-20,-20},{20,20}},
        rotation=270,
        origin={28,98})));
protected
  Modelica.Blocks.Interfaces.BooleanOutput local_reset annotation(HideRe-
sult=true);
  Modelica.Blocks.Interfaces.IntegerOutput local_set;
initial equation
  pre(y) = y_start;
equation
  if use_reset then
     connect(reset, local_reset);
       if use_set then
         connect(set, local_set);
       else
         local_set = y_start;
       end if;
  else
     local_reset = false;
     local_set = 0;
  end if;
```

```
  when {trigger, local_reset} then
     y = if local_reset then local_set else min(max(pre(y) + u,minY),maxY);
  end when;

  annotation
<p>
Add input to previous value of output, if rising edge of trigger port
</p>

<p>
This block has one Integer input \"u\", one Boolean input \"trigger\",
an optional Boolean input \"reset\", an optional Integer input \"set\", and
an Integer output \"y\".
The optional inputs can be activated with the \"use_reset\" and
\"use_set\" flags, respectively.
</p>

<p>
The input \"u\" is added to the previous value of the
output \"y\" if the \"trigger\" port has a rising edge. At the start of the
simulation \"y = y_start\".
</p>

<p>
If the \"reset\" port is enabled, then the output \"y\" is reset to \"set\"
or to \"y_start\" (if the \"set\" port is not enabled), whenever the \"re-
set\"
port has a rising edge.
</p>

<p>
The usage is demonstrated, e.g., in example
<a href=\"modelica://Modelica.Blocks.Examples.IntegerNetwork1\">Model-
ica.Blocks.Examples.IntegerNetwork1</a>.
</p>

</html>"));
end TriggeredAddLimited;
```

“Distributer” block:

```
block Distributer
  Modelica.Blocks.Interfaces.IntegerInput s "FanSpeedDemand" annota-
tion (Placement(transformation(extent={{-120,-20},{-80,
            20}}), iconTransformation(extent={{-120,-20},{-80,20}})));
  Modelica.Blocks.Interfaces.IntegerOutput F1 "Speed for Fan 1"
    annotation (Placement(transformation(extent={{100,10},{120,30}}), icon-
Transformation(extent={{100,10},{120,30}})));
  Modelica.Blocks.Interfaces.IntegerOutput F2 "Speed for Fan 2" annota-
tion (Placement(transformation(extent={{100,70},{120,
            90}}), iconTransformation(extent={{100,50},{120,70}})));
  Modelica.Blocks.Interfaces.RealOutput P1 "Demand for Pump 1"
    annotation (Placement(transformation(extent={{100,-70},{120,-50}}),
                                                                       icon-
Transformation(extent={{100,-70},{120,-50}})));
  Modelica.Blocks.Interfaces.RealOutput P2 "Demand for Pump 2" annota-
tion (Placement(transformation(extent={{100,-18},
            {120,2}}), iconTransformation(extent={{100,-30},{120,-10}})));

parameter Integer o = 2;
```

```
Integer S;

  Modelica.Blocks.Interfaces.BooleanInput modus annotation (Placement(
        transformation(extent={{-120,60},{-80,100}}), iconTransformation(ex-
tent=
           {{-120,60},{-80,100}})));
algorithm


    S:=s+o;

  if S == 0 then
    F1 :=0;
    F2 :=0;
  elseif S == 1 then
    F1:= 1;
    F2:= 0;
  elseif S == 2 then
    if modus then
      F1:=1;
      F2:=1;
    else
      F1:=2;
      F2:=0;
    end if;
  elseif S == 3 then
    F1:=2;
    F2:=1;
  else
    F1:=2;
    F2:=2;
  end if;

  if F1 >0 then
    P1:=1472;
  else
    P1:=0;
  end if;

  if F2 >0 then
    P2:=1472;
  else
    P2:=0;
  end if;

end Distributer;
```