Modelling and Control of a Water-Cooled Duct and Cooling System

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Abstract

In this thesis a water cooled duct and the surrounding cooling system, used for cooling hot off-gas from an electric arch furnace, is modelled for simulation. The model is then used to devise control strategies to keep the cooling water temperature at a given temperature to then be able to supply power to the district heating. The model consists of the heat transfer between the off-gas, steel of the water cooled duct and the cooling water, the transport delays of the water and a heat-exchanger used for cooling the water.

The control problem is non-trivial as there are long time-delays and non-linear behavior, combined with large load disturbances caused by variations in off-gas temperature. Different control strategies of increasing complexity and performance are presented. The results show that a simple PID-controller is not enough and different feed-forward signals, which account for how the system behaves, are also needed. Using these additions the simulated system is controlled to a satisfactory degree.
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1

Introduction

1.1 Background

Höganäs AB is the world’s largest producer of powdered metal. This powder is found in, amongst other things, electric bicycle engines, gearboxes and iron-fortified corn-flakes. ([Lindéngruppen, 2016]). In order to produce the metal powder they melt large amounts of scrap iron in an Electric Arch Furnace (EAF). This produces hot off-gas, which has to be cooled.

The duct through which the off-gas travels is water-cooled, and during the summer of 2016 this Water-cooled duct (WCD) will be replaced. The goal of this replacement is to increase melting capacity, and to use the excess heat for district heating in Halmstad. The heat is delivered via hot water resulting from the water-cooling of the off-gas in the WCD. This has a positive environmental impact compared to wasting the excess heat in cooling towers as was done before.

To be able to deliver heat to the district heating system, the water has to be of high enough temperature (125 °C), which means that the cooling water can’t be too cold. At the same time, the water is not allowed to get too hot, since if it boils it will damage the WCD. The water is pressurised, so it can reach 140 °C without boiling, but it is not allowed to reach higher temperatures. Since there are quite a lot of time delays in the systems, and the off-gas is a large load disturbance, this presents a serious control-problem.

1.2 Problem statement

The system to be controlled is illustrated in figure 1.1.
The central component is a water-cooled duct, through which the hot off-gas passes. The WCD cooling water circulates in a closed loop (the red circuit). This cooling water is in turn cooled by the water in the blue circuit, through a heat-exchanger. The water in the blue circuit is used to transfer heat to the district heating.

The water flow in the red circuit, \( \dot{m}_2 \), is constant, and not subject to control. The water flow in the blue circuit, \( \dot{m}_1 \) is however controllable. By adjusting this flow we can control how much energy is transferred from the red circuit to the blue circuit, and in this way ensure that the water in the red circuit does not get too hot. By increasing the flow in the blue circuit, more energy will be transferred between the circuits in the heat-exchanger.

There are multiple temperature sensors to aid in the control. Unfortunately the incoming off-gas temperature currently cannot be measured, since it is so hot that it would melt many sensors (it can reach a temperature of approximately 1200 °C). We can however measure the outgoing off-gas temperature (using TT1A), as well as the power supplied by the EAF. These measurements will be helpful in compensating for the off-gas variations.

The ultimate goal is to deliver a constant 8 MW to the district heating. This will be achieved by a controller that controls the water flow \( \dot{m}_1 \). To be able to deliver the heat to the district heating at an as constant rate as possible the blue circuit has a stratified hot water storage tank (an accumulator). The accumulator acts as an energy buffer, storing energy when more than 8 MW is delivered, and then delivering energy when less than 8 MW is coming from the WCD. In this way the power delivered to the district heating system can be kept constant even when the power...
from the off-gas is time-varying.

In order to effectively serve its purpose, it is important that the accumulator is stratified (see [Streckienė et al., 2011]). This, ideally, means that there are two water layers in the tank, one layer of hot water on top, and one layer of colder water below. To keep the accumulator stratified it is important to keep the temperature of the water going into it as constant as possible (at 125 °C in the system under consideration). The accumulator circuit will not be studied in this report. Therefore the goal is to keep the temperature going to the accumulator circuit as constant as possible.

1.2.1 Water-cooled duct

A simplified illustration of the WCD can be seen in figure 1.2. It consists of nine water-cooled sections. A simplified illustration of a section can be seen in figure 3.1.

Figure 1.2 Simplified illustration of WCD illustrating the general flows of off-gas and cooling-water.
The off-gas from the EAF flows straight through the sections from 1 to 9. The water flows parallel to the off-gas through sections 1 to 6. Then it turns around and flows in the opposite direction to the off-gas back through section 9 to 7.

1.2.2 Heat-exchanger

A heat-exchanger, as illustrated in figure 1.3, is a device that allows effective heat-transfer between two flows. The details of its workings will be expanded on in section 3.3.

![Heat-exchanger diagram](image)

Figure 1.3  Simplified illustration of the heat-exchanger in figure 1.1. In a real heat-exchanger the contact area between the two flows is much larger to maximize heat-transfer between them.

1.2.3 Control objective

There are multiple constraints and one objective for the control system. The water temperature out from the WCD (at TT1B) has to be less than 140°C, otherwise the whole system will shut down as a safety precaution. The outgoing off-gas temperature has to be below 600°C. The flow rate through the heat-exchanger on the accumulator side ($\dot{m}_1$) is limited to a maximum of 300 t h$^{-1}$ and a minimum of 0 t h$^{-1}$. The objective is to keep the water temperature on the accumulator side of the heat-exchanger (at TT1C) at 125°C.

The water going into the heat-exchanger from the accumulator side (at TT6C) has a temperature of around 55°C, but can vary from roughly 40°C to 65°C. This temperature depends on what happens in the accumulator circuit and the district heating. Since these are not part of the model the water temperature is a constant 55°C in the simulations, unless otherwise stated.

The goal of the thesis is to devise a control strategy which takes the temperature readings in figure 1.1 as inputs and outputs a mass flow $\dot{m}_1$ to achieve these goals.
1.3 Methodology

The first step is to create a model of the system for simulation. We start by creating a model of the WCD, which takes incoming water and off-gas temperatures and flows, and gives outgoing temperatures. This is done by splitting the WCD into sections, simulating each separately and then connecting them.

Then we model the heat-exchanger using a simple physical model. Finally the system model needs to include time delays, the biggest of which are the transport times from WCD to heat-exchanger and the time from heat-exchanger to WCD.

With a finished model of the system we proceed to conduct experiments on the simulated system, studying how the off-gas temperature and mass flows affect the system, as well as find its step response.

Finally a number of control strategies of varying complexity to control the water temperature are devised, using the information acquired in the preceding experiments.

1.4 Outline

Here we present a short outline of the contents of the thesis.

Chapter 2: Control-theory
A short description of relevant control-theory is presented for reference.

Chapter 3: Modelling
A simulation model of the water-cooled duct and surrounding cooling system is constructed.

Chapter 4: Process Behaviour
The simulated process is studied in experiments, to verify the validity of the model and to aid in the construction of control-strategies.

Chapter 5: Control
A number of control-strategies of increasing complexity are designed and tested on a number of scenarios.

Chapter 6: Conclusions
A summary of the results of the thesis and the conclusions we have drawn.
2

Control-theory

In this chapter we present a very short description of relevant concepts from control-theory.

2.1 PID-control

A PID-controller is one of the simplest commonly used control-strategies. It takes as input a measured signal \( y \) and a reference signal \( r \), and it outputs a control-signal \( u \). \( y \) is usually some measured value from the system to be controlled, and \( r \) is a value chosen externally by an operator. \( u \) is a signal which in some way affects the process, and specifically affects \( y \). The goal of the controller is to choose appropriate \( u \) to keep \( y = r \). In the case of the cooling system described in the previous chapter \( u = \dot{m}_1 \) and \( y \) is the temperature from one of the temperature sensors in figure [1.1]

A PID-controller uses the error \( e = r - y \) to decide the control-signal. From this error the control-signal is given by

\[
u(t) = K \left( e(t) + \frac{1}{T_i} \int_0^t e(\tau) \, d\tau + T_d \frac{d}{dt} e(t) \right)
\]

where \( K \) is the gain, \( T_i \) is the integral time and \( T_d \) is the derivative time. These three parameters are usually constant over time, and have to be chosen appropriately depending on the behaviour of the controlled system. One way to choose these parameters (also known as tuning the controller) is known as the AMIGO-method, which is described in section [5.1.8]

2.2 Feed-forward

Often there is some load disturbance \( l \) which affects the signal \( y \) in some undesirable way. In the case of our cooling system, the varying off-gas temperature represents a
load disturbance. The off-gas temperature will affect the water-temperature, which will make it harder for the PID-controller to maintain \( y = r \). If the controller has access to some measurement of the load-disturbance, it can use *feed-forward* to improve controller performance.

The idea is to modify (2.1) to include a feed-forward term, giving the new control-strategy

\[
  u(t) = K \left( e(t) + \frac{1}{T_i} \int_0^t e(\tau) \, d\tau + T_d \frac{d}{dt} e(t) \right) + f(l)
\]  

(2.2)

where \( f \) is some function of the measured load disturbance. Often \( f \) is simply a constant gain, so that \( f(l) = K_{ff} \cdot l \), but it doesn’t have to be.
In this chapter we develop a simulation model of the system in figure 1.1. Using the physics of heat-transfer, a set of differential equations modelling the different parts of the system are developed. The system will then be simulated in Simulink.

We start by developing a model for a single section of the water-cooled duct. Then a number of sections are combined into a model of the complete WCD. Then we create a model of the heat-exchanger. These models are then combined into a model of the complete cooling-system. Next the complete model is tested against available reference-data for verification. Finally the verification results are discussed, along with the simplifications that have been used to create it.

3.1 Single section model

3.1.1 Temperature equations

As a first step in modelling the WCD, consider one of the nine sections in figure 1.2. A simplified section of the WCD is shown in figure 3.1. We will require a model for such a section which takes incoming off-gas and water temperature as input, and gives outgoing off-gas and water temperatures as output. It will also need to keep track of the temperature of the steel in the section.

In some sections the water and the gas will flow in the same direction, and in some they will flow in opposite directions. We begin here by considering a section in which the directions are opposite (counter-flow). Then only minor modifications have to be made to model the case of parallel flow.

First consider a thin slice taken along the axial direction, i.e. the axis along which the water and off-gas are flowing. To simplify the problem, the temperature is modelled as one-dimensional, as illustrated in figure 3.2. We are interested in modelling how
3.1 Single section model

**Figure 3.1** A section of the WCD with water flowing in the opposite direction of the off-gas.

the energy contents in the three middle cubes are varying over time, by studying the energy flows.
We begin with the water, see figure 3.3. We wish to model how $T_w(x)$ varies as a function of time. The volume of the middle cube is $V = dx A_w$, where $A_w$ is the cross-sectional area of the tube at position $x$. 

Figure 3.2  Energy balances.
We start by considering the advection. Consider what happens during a short time \( dt \). Let \( \dot{V}_w \) designate the volume flow of water, i.e. the volume of water flowing through the tube per time unit. This means that during the time \( dt \) a volume \( \dot{V}_w dt \) of water with temperature \( T_w(x - dx) \) will flow into the region of consideration. This water will mix with the \( dxA_w - \dot{V}_w dt \) water of temperature \( T_w(x) \) which stays in the region. This means that the new temperature in the region will be given by

\[
T_w(x, t + dt) = \frac{T_w(x, t)[dxA_w - \dot{V}_w dt] + T_w(x - dx, t)\dot{V}_w dt}{dxA_w} \quad (3.1)
\]

We can rewrite this as

\[
\frac{T_w(x, t + dt) - T_w(x, t)}{dt} = -\frac{\dot{V}_w}{A_w} \frac{T_w(x + dx, t) - T_w(x, t)}{dx} \quad (3.2)
\]

Letting \( dx, dt \to 0 \) this becomes

\[
\frac{\partial}{\partial t} T_w(x, t) = -\frac{\dot{V}_w}{A_w} \frac{\partial}{\partial x} T_w(x, t) \quad (3.3)
\]
Chapter 3. Modelling

This is the equation for one-dimensional advection, with a velocity \( \dot{V}_w/A_w \). This makes intuitive sense, since it represents a temperature wave with velocity \( \dot{V}_w/A_w \) travelling down the pipe, where \( \dot{V}_w/A_w \) is the speed of the wave front.

However, we also need to consider the effect of the heat flowing from (or to) the steel wall. This heat transfer is given by

\[
q(x,t) = P_w \, dx \, h_w(T_s(x,t) - T_w(x,t))
\]

where \( P_w \) is the perimeter of the inside of the water channel, so that \( P_w \, dx \) is the area of contact in this slice. \( h_w \) is a convective heat transfer coefficient which depends on many factors, and we will get back to it later. This heat transfer will of course affect the water temperature. During a time \( dt \) it will add an energy \( dt \, q(x,t) \) to the water, which will give rise to a change of temperature \( dt \, q(x,t)/(dx \, A_w \, C_w) \), where \( C_w \) is the heat capacity per volume of the water. We can include this in equation (3.1) to get

\[
T_w(x,t + dt) = T_w(x,t)\left[dx \, A_w - \dot{V}_w \, dt\right] + T_w(x - dx,t)\dot{V}_w \, dt
\]

\[
+ \frac{dt \, P_w \, dx \, h_w}{dx \, A_w \, C_w} [T_s(x,t) - T_w(x,t)]
\]

which can be rewritten as

\[
\frac{T_w(x,t + dt) - T_w(x,t)}{dt} = -\frac{\dot{V}_w}{A_w} T_w(x + dx,t) - T_w(x,t) - \frac{P_w h_w}{A_w C_w} [T_s(x,t) - T_w(x,t)]
\]

so that when \( dx, dt \to 0 \) we get

\[
\frac{\partial}{\partial t} T_w(x,t) = -\frac{\dot{V}_w}{A_w} \frac{\partial}{\partial x} T_w(x,t) + \frac{P_w h_w}{A_w C_w} [T_s(x,t) - T_w(x,t)]
\]

The off-gas temperature \( T_g(x,t) \) can be modelled in the same way, which results in

\[
\frac{\partial}{\partial t} T_g(x,t) = -\frac{\dot{V}_g}{A_g} \frac{\partial}{\partial x} T_g(x,t) + \frac{P_g h_g}{A_g C_g} [T_s(x,t) - T_g(x,t)]
\]
3.1 Single section model

Note that we have shifted the sign in front of the first term on the right-hand side, because the off-gas is travelling in the negative x-direction.

Finally, we want to model the steel temperature. We start with a lumped-heat-capacity system. This assumption is reasonable if the two convective heat transfer coefficients are small enough compared to the thermal conductivity of the steel. A rule of thumb is given by

\[
\frac{h(V/A)}{k} < 0.1, \quad (3.8)
\]

where \( h \) is the convective heat transfer coefficient, \( V \) is the volume of the steel, \( A \) is the surface area of the steel and \( k \) is the thermal conductivity. ([Holman, 1992, p. 140])

The convection at both faces of the steel then gives the following equation

\[
-C_sA_s \frac{dT_s}{dx} = P_g dx h_g (T_s(x,t) - T_g(x,t)) + P_w dx h_w (T_s(x,t) - T_w(x,t)), \quad (3.9)
\]

where \( C_s \) is the volumetric heat capacity for the steel and \( A_s \) is the area of the cross section of the steel slice. The equation can be rewritten to

\[
\frac{dT_s}{dt} = -\left[ P_g h_g (T_s(x,t) - T_g(x,t)) + P_w h_w (T_s(x,t) - T_w(x,t)) \right]/(C_s A_s). \quad (3.10)
\]

Further we also want to consider the conduction between slices of the steel itself. This will be modelled as regular heat conduction,

\[
\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (3.11)
\]

Combining equation (3.10) and (3.11) we get

\[
\frac{dT_s}{dt} = -\left[ P_g h_g (T_s(x,t) - T_g(x,t)) + P_w h_w (T_s(x,t) - T_w(x,t)) \right]/(C_s A_s) + \alpha \frac{\partial^2 T}{\partial x^2} \quad (3.12)
\]

3.1.2 Heat-transfer coefficients of water and off-gas

Now we turn to the problem of calculating the convection heat-transfer coefficients \( h_w \) and \( h_g \). These coefficients will depend on quite a few things, like the geometry of the channels, the flow rate and the viscosity of the fluid or off-gas. Therefore these constants will vary over time and by section. In this section the calculations are presented without theoretical justification. For the calculations that lead to these equations see appendix B.
To calculate \( h_w \) for a given temperature \( T \):

\[
k = 0.67 \text{W m}^{-1} \text{K}^{-1}
\]
\[
d = 37.8 \text{mm}
\]
\[
A = 1.14 \cdot 10^{-3} \text{m}^2
\]
\[
\mu(T) = 2.414 \cdot 10^{-5} \times 10^{247.8/(T-140)}
\]
\[
Pr = \frac{W_w \mu(T)}{k}
\]
\[
Re_d = \frac{md}{A \mu(T)}
\]
\[
h_w = \frac{k}{d} \cdot 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}
\]

To calculate \( h_g \) for a given temperature \( T \):

\[
T_0 = 825.6 \text{K}
\]
\[
\mu_0 = 3.61 \cdot 10^{-5} \text{N s m}^{-2}
\]
\[
S = 160 \text{K}
\]
\[
C_g = 1.226 \cdot 10^3 \text{J kg}^{-1} \text{K}^{-1}
\]
\[
k = 8.32 \cdot 10^{-2} \text{W m}^{-1} \text{K}^{-1}
\]
\[
\mu = \mu_0 \left( \frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S}
\]
\[
Pr = \frac{C_g \mu}{k}
\]
\[
d = 4A_g/P_g
\]
\[
u_g = V_g/A_g
\]
\[
\rho = 1.326 \text{kg Nm}^{-1} = \frac{273.15 \text{K}}{T} \cdot 1.326 \text{kg m}^{-3}
\]
\[
Re_d = \frac{md}{A_g \mu(T)} = \frac{\rho u_g d}{\mu(T)}
\]
\[
h_{gc} = \frac{k}{d} \cdot 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}
\]
\[
\epsilon_s = \epsilon_g = 0.55
\]
\[
h_r = \frac{\sigma(T_s^2 + T_g^2)(T_s + T_g)}{1/\epsilon_s + 1/\epsilon_g - 1}
\]
\[
h_g = h_{gc} + h_r
\]

### 3.1.3 Final section equations

Now we have a final model for the section in figure 3.1 of length \( L \). We model it as three different one-dimensional and time varying temperature distributions. One for
the water \((T_w)\), one for the steel \((T_s)\) and one for the off-gas \((T_g)\). They each depend on time \(t\) and position in the axial direction (the axis along which the off-gas and water flows) \(x\).

Additionally we need boundary conditions. Assume that all temperatures are known at some initial time \(t = 0\). Also, assume that the temperature of the incoming water (from \(x = 0\)) and the incoming off-gas (from \(x = L\) for counter-flow and from \(x = 0\) for parallel flow) are known for all \(t\).

The final model is

\[
\begin{align*}
\frac{\partial}{\partial t} T_w(x,t) &= -\frac{V_w}{A_w} \frac{\partial}{\partial x} T_w(x,t) + \frac{P_w h_w}{A_w C_w} [T_s(x,t) - T_w(x,t)] \\
\frac{\partial}{\partial t} T_g(x,t) &= -\frac{V_g}{A_g} \frac{\partial}{\partial x} T_g(x,t) + \frac{P_g h_g}{A_g C_g} [T_s(x,t) - T_g(x,t)] \\
\frac{\partial}{\partial t} T_s(x,t) &= \frac{P_g h_g}{C_s A_s} [T_g(x,t) - T_s(x,t)] + \frac{P_w h_w}{C_s A_s} [T_w(x,t) - T_s(x,t)] + \alpha \frac{\partial^2}{\partial x^2} T_s(x,t)
\end{align*}
\]

for counter-flow and

\[
\begin{align*}
\frac{\partial}{\partial t} T_w(x,t) &= -\frac{V_w}{A_w} \frac{\partial}{\partial x} T_w(x,t) + \frac{P_w h_w}{A_w C_w} [T_s(x,t) - T_w(x,t)] \\
\frac{\partial}{\partial t} T_g(x,t) &= -\frac{V_g}{A_g} \frac{\partial}{\partial x} T_g(x,t) + \frac{P_g h_g}{A_g C_g} [T_s(x,t) - T_g(x,t)] \\
\frac{\partial}{\partial t} T_s(x,t) &= \frac{P_g h_g}{C_s A_s} [T_g(x,t) - T_s(x,t)] + \frac{P_w h_w}{C_s A_s} [T_w(x,t) - T_s(x,t)] + \alpha \frac{\partial^2}{\partial x^2} T_s(x,t)
\end{align*}
\]

for parallel flow.

### 3.1.4 Computer model of a section

We will use the finite difference method ([Edsberg, 2008, p. 135]) to solve the equations (3.15) numerically. The nodes are partitioned uniformly for each section of the WCD separately. Forward shift difference will be used for the time derivatives and central difference for the first order space derivatives. For the second order space
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derivatives central difference will be used.

\[
\frac{\partial}{\partial t} T(x,t) \rightarrow \frac{T(x, t + \Delta t) - T(x, t)}{\Delta t}
\]

\[
\frac{\partial}{\partial x} T(x,t) \rightarrow \frac{T(x,t) - T(x - \Delta x)}{\Delta x} \text{ for flow towards positive } x
\]

\[
\frac{\partial}{\partial x} T(x,t) \rightarrow \frac{T(x + \Delta x, t) - T(x)}{\Delta x} \text{ for flow towards negative } x
\]

\[
\frac{\partial^2}{\partial x^2} T(x,t) \rightarrow \frac{T(x + \Delta x, t) + T(x - \Delta x, t) - 2T(x,t)}{(\Delta x)^2}
\]

We discretize the time- and spatial dimensions as

\[
x_n = (n + 1/2) \cdot \Delta x, \quad n = 1, 2, \ldots, N
\]

\[
t_m = m \cdot \Delta t, \quad m = 0, 1, \ldots
\]

where \(\Delta x = L/N\). For ease of notation we let \(T_{w,m}^{n,m} = T_w(x_n, t_m)\). Discretizing (3.15) the equations to be solved are now

\[
\frac{T_{w,m}^{n+1,m} - T_{w,m}^{n,m}}{\Delta t} = -\frac{\dot{V}_w}{A_w} \frac{T_{w,m}^{n,m} - T_{w,m}^{n-1,m}}{\Delta x} + \frac{P_w h_w}{A_w C_w} [T_{s,m}^{n,m} - T_{w,m}^{n,m}]
\]

\[
\frac{T_{g,m}^{n+1,m} - T_{g,m}^{n,m}}{\Delta t} = -\frac{\dot{V}_g}{A_g} \frac{T_{g,m}^{n+1,m} - T_{g,m}^{n,m}}{\Delta x} + \frac{P_g h_g}{A_g C_g} [T_{s,m}^{n,m} - T_{g,m}^{n,m}]
\]

\[
\frac{T_{s,m}^{n+1,m} - T_{s,m}^{n,m}}{\Delta t} = \frac{P_g h_g}{C_s A_s} [T_{s,m}^{n,m} - T_{s,m}^{n,m}] + \frac{P_w h_w}{C_s A_s} [T_{w,m}^{n,m} - T_{s,m}^{n,m}]
\]

\[
+ \alpha \frac{T_{s,m}^{n+1,m} + T_{s,m}^{n-1,m} - 2T_{s,m}^{n,m}}{(\Delta x)^2}
\]

for counter-flow, with analogous changes from (3.15) to (3.16) made for parallel flow.

For \(n = 1\) or \(n = N\) we have to be careful, since \(T_{s,m}^{n,m} - 1, m\) and \(T_{s,m}^{N+1,m}\) are outside our section. This is solved by using temperatures from neighbouring sections.

We can rewrite these equations as
3.1 Single section model

\[ T^{n,m+1}_w = (1 - \Delta t \frac{P_w h_w}{A_w C_w}) T^{n,m}_w + \Delta t \frac{P_w h_w}{A_w C_w} T^{n,m}_s - \frac{\Delta t}{\Delta x} \frac{V_w}{A_w} (T^{n,m}_w - T^{n-1,m}_w) \]

\[ T^{n,m+1}_g = (1 - \Delta t \frac{P_g h_g}{A_g C_g}) T^{n,m}_g + \Delta t \frac{P_g h_g}{A_g C_g} T^{n,m}_s + \frac{\Delta t}{\Delta x} \frac{V_g}{A_g} (T^{n+1,m}_g - T^{n,m}_g) \]

\[ T^{n,m+1}_s = (1 - \Delta t \frac{P_w h_w}{C_s A_s}) T^{n,m}_w + \Delta t \frac{P_g h_g}{C_s A_s} T^{n,m}_g + \frac{\alpha \Delta t}{\Delta x^2} (T^{n+1,m}_s + T^{n-1,m}_s) \]  

(3.20)

which are straightforward to simulate, since the temperatures at time \( t_{m+1} \) depend only on those at time \( t_m \).

The boundary conditions and the ends of the simulated sections are time varying depending on the states of adjacent sections. The actual values come from the simulation of the other sections. The boundary conditions will be different in the steel compared to the water and off-gas since the heat will flow in both directions through the steel.

For initial conditions everything is assumed to be at the same temperature,

\[ T_w(x,0) = T_g(x,0) = T_s(x,0) = T_0. \]  

(3.21)

3.1.5 Section model summary

The WCD is divided into sections, each with its own geometry. Each section is then discretized with uniform nodes. The equations in (3.20) (with slight modifications in the case of parallel flow) will be solved iteratively for each time step. The boundary conditions of each section depend on the adjacent sections, except for section 1, where the incoming water and gas temperatures are given as inputs from the bigger system.

To simulate a section, the following steps will be taken:

1. Initialize all node temperatures at \( t = 0 \) \( (T^{n,0}_w) \) to some known initial values.

2. Read \( T^{0,m+1}_w, T^{N+1,m+1}_g \) from outside the model of the section. These represent the incoming temperatures of water and off-gas, either from a neighbouring section or from the outside.

3. Read \( T^{0,m+1}_s \) and \( T^{N+1,m+1}_s \) from neighbouring sections, for use in the steel conduction.
4. Update all other temperatures $T_{n,m+1}$ according to (3.20), calculating $h_w$ and $h_g$ using (3.13) and (3.14) for each spatial position.

5. Go to step 2 and repeat.

### 3.1.6 Simplifications

A number of simplifying assumptions are included in the model for a section. The geometry assumed in figure 3.1 does not correspond perfectly to the actual geometries of the sections. For example, section 5 in figure 1.2 is a 90° curve, which is not part of our model. The curve is likely to affect the heat-transfer of the off-gas. However, properly modelling this phenomena would require a much more complex 3-dimensional model of the flow and would be beyond the theoretical scope of this thesis. While the magnitude of the off-gas heat-transfer coefficient is likely not precisely correct the overall process performance should not be affected immensely by this simplification.

### 3.1.7 Simulink Implementation

An individual section of the WCD will be implemented as a Simulink S-function block. The function implements the equations in (3.20). The specific parameters for each section is set in the block and can be varied for each section.

Since simulating the WCD is the most computer intensive part of the simulation the S-function will be written in C. This gives a substantial performance increase compared to writing it in Matlab-code.

### 3.1.8 Numerical stability

The advection equations for the off-gas and the water are solved using the Forward-Time-Backward-Space method. Analysing the Courant number for the equation,

$$\sigma = \frac{a \Delta t}{\Delta x}, \quad (3.22)$$

where $a$ in this case is given by $\frac{\dot{V}}{A}$, we can determine first order convergence and stability if

$$0 < \sigma \leq 1. \quad (3.23)$$

([Edsberg, 2008, p. 175])

The heat equation is solved using the Forward-Time-Central-Space method. The condition here for stability is given by

$$\sigma = \frac{\Delta t}{(\Delta x)^2} < \frac{1}{2}. \quad (3.24)$$

([Edsberg, 2008, p. 132])
3.2 Model of complete WCD

For our equations in (3.20) the discretization has to be the same for the water, steel and off-gas. The combination gives no other restrictions than that the time step has to be sufficiently small for the given spatial resolution.

### 3.2 Model of complete WCD

The complete WCD consists of nine sections (see figure 1.2). The off-gas flows straight through the sections from 1 to 9. The water flows parallel to the off-gas through sections 1 to 6. Then it turns around and flows counter to the off-gas back through section 9 to 7. This is illustrated in figure 3.4. The parameters for each section are given in appendix A.

Between each section there is a time-delay in the water flow. This represents the time it takes for the water to exit one section and enter the next. The delay between section 6 and section 9 is longer than the others since this is where the water flow changes direction relative to the off-gas flow.
Figure 3.4  Simulink system of the WCD. The steel-connections are omitted for clarity. The transport delays between the sections are delay1 = 8.38s and delay2 = 20s.
3.3 Heat-exchanger physics

Next, consider the heat-exchanger which will be used to cool the water circulating around the WCD-circuit. Since the water temperature in the WCD-circuit will vary quite slowly compared to the dynamics of a heat-exchanger we will disregard the dynamics of the heat-exchanger and simply map inlet-temperatures to outlet-temperatures. This is expanded on in section 3.3.2. We will use what is known as the effectiveness-NTU method. ([Holman, 2009, p.540])

We consider a counter-flow heat-exchanger. There are two water flows with mass flow $\dot{m}_1$ and $\dot{m}_2$ and inlet temperatures $T_{1i}$ and $T_{2i}$. Our goal is to calculate the corresponding outlet temperatures $T_{1o}$ and $T_{2o}$. To do this we need to know the contact area $A$ (m$^2$) between the fluids, and the over-all heat transfer coefficient $U$ (W K$^{-1}$ m$^{-2}$). These are properties of the heat-exchanger. We will also need to know the specific heat capacities of the two fluids $c_1$ and $c_2$ (J K$^{-1}$ kg$^{-1}$).

To calculate the outlet temperatures from this data we first need to find which of the two fluids is the minimum fluid. This is defined as the one for which $\dot{m}c$ is the smallest. For ease of notation, lets assume that fluid 1 is the minimum fluid; if this is not the case simply swap the subscripts to get the appropriate formulas.

Now define $C_{\text{min}} = \dot{m}_1c_1$ and $C_{\text{max}} = \dot{m}_2c_2$. Then it can be shown ([Holman, 2009, p.540]) that the effectiveness $\varepsilon$ defined as

$$
\varepsilon = \frac{\Delta T(\text{minimum fluid})}{\text{Maximum temperature difference in heat exchanger}} = \frac{T_{1i} - T_{1o}}{T_{1i} - T_{2i}}
$$

satisfies the relation

$$
\varepsilon = \frac{1 - \exp[N(1 - C_{\text{min}}/C_{\text{max}})]}{1 - (C_{\text{min}}/C_{\text{max}}) \exp[N(1 - C_{\text{min}}/C_{\text{max}})]}
$$

where $N = (-UA/C_{\text{min}})$.

Now this pair of equations can be solved for $T_{1o}$ to find that

$$
T_{1o} = T_{1i} + (T_{2i} - T_{1i}) \frac{1 - \exp[N(1 - C_{\text{min}}/C_{\text{max}})]}{1 - (C_{\text{min}}/C_{\text{max}}) \exp[N(1 - C_{\text{min}}/C_{\text{max}})]}
$$

Finally, using energy balances we find that
\[ \dot{m}_1 c_1 (T_{1i} - T_{1o}) = -\dot{m}_2 c_2 (T_{2i} - T_{2o}) \]  
(3.28)

from which we get that

\[ T_{2o} = T_{2i} + \frac{\dot{m}_1 c_1}{\dot{m}_2 c_2} (T_{1i} - T_{1o}) \]  
(3.29)

### 3.3.1 Temperature-difference approximation

Next consider the specific heat-exchanger which will be used as the heat-exchanger in figure 1.1. In this case \( \dot{m}_2 = 400 \text{ t h}^{-1} \) and \( 0 \text{ t h}^{-1} \leq \dot{m}_1 \leq 300 \text{ t h}^{-1} \). This means that the fluid in the red circuit (the accumulator circuit), will always be the minimum fluid.

Thus \( C_{\text{min}} = \dot{m}_1 C_w \) and \( C_{\text{max}} = \dot{m}_2 C_w \), and so \( N = (-UA/\dot{m}_1 c_1) \) where \( c_1 = c_2 = C_w \) is the specific heat capacity of water.

Using this (3.26) can be rewritten as

\[ \varepsilon = \frac{1 - \exp \left[-\frac{UA}{C_w} (1/\dot{m}_1 - 1/\dot{m}_2)\right]}{1 - (\dot{m}_1/\dot{m}_2) \exp \left[-\frac{UA}{C_w} (1/\dot{m}_1 - 1/\dot{m}_2)\right]} \]  
(3.30)

The data for the heat-exchanger is \( A = 441.9 \text{ m}^2 \), \( U = 6192.7 \text{ W m}^{-2} \text{ K}^{-1} \). In SI-units \( \dot{m}_2 = 111.1 \text{ kg s}^{-1} \) and \( 0 \text{ kg s}^{-1} \leq \dot{m}_1 \leq 83.3 \text{ kg s}^{-1} \). Thus, all the values in (3.30) are fixed except for \( \dot{m}_1 \). We can therefore plot \( \varepsilon \) as a function of \( \dot{m}_1 \) in the allowed range. This is done in figure 3.5.
Figure 3.5 $\epsilon$ as a function of $\dot{m}_1$ for the heat-exchanger in figure 1.1.

Figure 3.5 shows that $0.96 \leq \epsilon \leq 1$. Therefore, using (3.27) we get that

$$T_{1o} = T_{1i} + (T_{2i} - T_{1i})\epsilon \approx T_{2i}$$

is an approximation that is accurate to within 4%, and that it is much more accurate than that except for when $\dot{m}_1$ is close to its maximum.

Using this approximation together with (3.29) gives that

$$T_{2o} = T_{2i} + \frac{\dot{m}_1c_1}{m_2c_2} (T_{1i} - T_{1o}) \approx T_{2i} + \frac{\dot{m}_1}{m_2} (T_{1i} - T_{2i})$$

(3.32)

This approximation will be useful later, for compensating for variations in the temperature at TT6C.

### 3.3.2 Heat-exchanger dynamics

An exact calculation of the transient response and associated time constant of a heat exchanger is quite complicated (see for instance [Gvozdenac, 2012]). In this section only a rough estimate will be presented in order to justify not including the heat-exchanger dynamics in the system model.
Chapter 3. Modelling

At any one time, 460 kg of water from each flow is contained in the heat-exchanger under consideration. The flow of water in the red circuit in figure 1.1 is fixed at 400 t h\(^{-1}\) = 111.1 kg s\(^{-1}\). This means that the transport time through the heat-exchanger is

\[
\frac{460\text{kg}}{111.1\text{kg s}^{-1}} = 4.14\text{s}
\]

Additionally, the heat-transfer between the two water-flows introduces an additional time-constant. As previously stated \(A = 441.9\text{m}^2\), \(U = 6192.7\text{W m}^{-2}\text{K}^{-1}\). The heat transfer coefficient is therefore \(h = AU = 2.74 \cdot 10^6\text{W K}^{-1}\). Each body of water has a mass of 460 kg and with a total heat capacity \(C = 460\text{kg} \cdot 4190\text{J kg}^{-1}\text{K}^{-1} = 1.93 \cdot 10^6\text{J K}^{-1}\). A rough estimate of the time-constant is

\[
\frac{C}{h} = 0.70\text{s}
\]

Combining these we get the pessimistic estimate of the total time-constant of 4.14 s + 0.70 s = 4.84 s \(\approx\) 5 s. In section 4.2 it will be shown that the time-constant of the WCD is approximately 200 s. Since a lot of simplifications are involved in the model of the WCD, it is likely that there are already errors larger than 5 s, and so the heat-exchanger dynamics will be disregarded.

3.4 Complete system model

Now we have a model of the WCD and a model of the heat-exchanger. These are now used to create a model of the complete system in figure 1.1. This system can be seen in figure 3.6.
3.4 Complete system model

Figure 3.6 Simulink model of the complete system of figure 1.1. The delays in the figure are delay_WCD_to_TT2B = 69.8 s, delay_TT3B_to_WCD = 84.4 s, delay_TT2B_to_VVX = 2.2 s and delay_VVX_to_TT3B = 5.6 s.

The WCD-block is implemented as seen in figure 3.4. The heat-exchanger-block is an implementation of (3.27) and (3.29).

As previously mentioned, the heat-exchanger is modelled as a static relation between input temperatures and flows to output temperatures. A more detailed model would have to include the dynamics of the process, the time it takes for a change in input temperature to affect the output temperature. However, since this time is so short compared to the time delays and the time constant of the WCD, the dynamics of the heat-exchanger has been disregarded.
Chapter 3. Modelling

The water transport between the WCD and the heat-exchanger are modelled as time delays. This is also a simplification, as the water pipes probably affect the water temperature slightly because of mixing of water and the steel acting as a temperature buffer. This has also been disregarded.

The reason for disregarding this is that a more realistic simulation of the water pipes would add significant model complexity with a small difference in system behaviour. Since we already have a number of sources for potential model errors (for instance, the calculations of \( h_w \) and \( h_g \)), this would likely not be worth the effort or additional computation time.

3.5 Model verification

To verify the simulation data to the real world process, the simulations are compared to real world data and manufacturer calculations.

3.5.1 Lumped-heat approximation

Our model is based on the lumped-heat approximation. This approximation is valid if \((3.8)\) is satisfied. The worst case scenario is when the heat-transfer coefficients are at their highest value. Using \((3.13)\) and \((3.14)\) gives maximum heat-transfer coefficients \( h_w = 5500 \text{ W m}^{-2} \text{ K}^{-1} \) and \( h_g = 100 \text{ W m}^{-2} \text{ K}^{-1} \) in our temperature ranges. This in turn gives \( h_w \cdot (V/A)/k \approx 0.7 \) and \( h_g \cdot (V/A)/k \approx 0.02 \).

Therefore the rule of thumb, \((3.8)\), is satisfied for the off-gas-side but not the water-side. The slowest transfer, which is the off-gas-side, will dominate. This means that the error in the approximation on the water-side is negligible, since the time delay it actually should introduce is small compared the overall delay in the system.

3.5.2 Comparison to real-world data from one section of the old WCD

There exists some run data on one section of the old WCD. This data is used to check if the transient behavior of the model matches the real world process. It is also used to study the relation between the power of the EAF and the initial off-gas temperature.

The data available are the water temperature and off-gas temperature out from one section of the WCD which is not the first. Unfortunately, we do not have data on the incoming water temperature, which seems to be varying slightly with time.

In addition to this, the cooling is not done the same way, the cooling water pipes are perpendicular to the off-gas flow and there is no parallel flow. The system is modelled with our model anyway.
3.5 Model verification

The comparison of the outgoing water temperatures of our model and the real data can be seen in figure 3.7.

3.5.3 Comparison to calculations on new WCD by manufacturer

For the new WCD, only stationary calculations by the manufacturer are available. This data is used to verify the parameters of our model so that the same approximate stationary behaviour is achieved. The calculations from the manufacturer are based on a case where the incoming water temperature is 90°C and the incoming off-gas temperature is 993.6°C. The resulting stationary temperatures coming out of each section are tabulated in table 3.1, comparing the manufacturers calculations with the results from our simulation model.

<table>
<thead>
<tr>
<th>Section</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Off-gas</td>
<td>Manufacturer</td>
<td>900.9</td>
<td>856.4</td>
<td>805.4</td>
<td>761.0</td>
<td>718.4</td>
<td>685.5</td>
<td>617.9</td>
<td>578.5</td>
</tr>
<tr>
<td></td>
<td>Our Simulations</td>
<td>918.8</td>
<td>875.3</td>
<td>822.3</td>
<td>776.6</td>
<td>733.0</td>
<td>705.5</td>
<td>634.8</td>
<td>593.4</td>
</tr>
<tr>
<td>Water</td>
<td>Manufacturer</td>
<td>95.7</td>
<td>99.3</td>
<td>102.4</td>
<td>105.1</td>
<td>107.6</td>
<td>109.5</td>
<td>117.2</td>
<td>113.3</td>
</tr>
<tr>
<td></td>
<td>Our Simulations</td>
<td>94.7</td>
<td>97.5</td>
<td>100.8</td>
<td>103.7</td>
<td>106.4</td>
<td>108.1</td>
<td>116.9</td>
<td>112.4</td>
</tr>
</tbody>
</table>

Table 3.1 Outgoing water and off-gas temperatures from each section (°C)

The water temperatures match to within 2°C. Importantly, the final outlet temperatures differ with only 0.3°C. Since part of the water flow is in the opposite direction to that of the off-gas, the final outlet water comes from section 7.
Chapter 3. Modelling

The off-gas temperatures match to within 20 °C. While this is worse than the water in absolute terms, the off-gas temperature also varies a lot more so some more error is to be expected. The outlet temperatures differ by 13.4 °C.

3.6 Discussion

3.6.1 Model verification

The model is based on the real physical equations for heat transfer systems. There are a lot of simplifications and approximations to make the calculations simple. We believe that the simplified model is justified as there is already a big uncertainty in the model parameters. As can be seen in the above sections our simulation gives results that compare relatively well to real data.

Comparing our simulation to the actual data in figure 3.7, the overall dynamics seem to match relatively well. However, the magnitude of the temperature changes is quite a bit larger in the real data than in our simulations. This can in part be explained by the fact that our model assumes a different geometry than the one that the data is taken from. This indicates that our overall model seems to work quite well, but that magnitudes of the heat transfer coefficients $h_w$ and $h_g$ are wrong. It doesn’t seem unlikely that the same will hold true in the new WCD. This means that adjustments to the model and the controllers will have to be made once real life temperature data is available from the new duct.
4

Process Behaviour

In this chapter we conduct a number of experiments on the models constructed in the previous chapter, which will aid in constructing control strategies.

First the EAF behaviour is studied. This is done by comparing the power output of the EAF and the off-gas temperature.

Next the basic behaviour of the WCD itself and then the complete system is studied to get an overview of how the system behaves and why it behaves as it does. We do this by first simulating how the water temperature changes in the WCD depending on different incoming water temperatures and off-gas temperatures. Then we connect the WCD to the cooling system and instead do simulations with different water temperatures and flows through the heat-exchanger together with different off-gas temperatures.

As we then have a basic understanding of how the system behaves we do an open-loop step response experiments on the complete system. These experiments are necessary to find appropriate controllers and how to tune them.

Lastly we study what flows through the heat-exchanger are needed to keep the water temperature constant at different off-gas temperatures. This experiment results in a non-linear function which is used for the feed-forward of the off-gas temperature.

4.1 EAF behaviour

In figure 4.1 it can be seen how the off-gas temperature is related to the power of the EAF. The off-gas temperature seems like a low-pass filtered version of the power for most parts. There are a lot of discrepancies though and the correspondence is far from perfect. This mainly occurs because the EAF process has different phases to melt the metal.
Chapter 4. Process Behaviour

Figure 4.1 Comparsion of the off-gas temperature of a specific section in the old system and the power of the EAF. The red curve is the power of the EAF and the blue curve is the outgoing off-gas temperature.

Instead of considering the power of the EAF to estimate the incoming off-gas temperature, the sensor of the outgoing off-gas temperature, TT1A, will be used. This gives a small time delay for temperature changes, but it is much more accurate.

4.2 WCD behaviour

In this experiment we test the WCD alone (without recirculating the water) by sending in water and off-gas of constant temperatures and studying the outgoing water and off-gas temperatures. In other words, we are simulating the system in figure 3.4 using constant Tw_in and Tg_in, and study how Tw_out (TT1B) and Tg_out (TT1A) vary over time. This is done with a number of different water and off-gas temperatures. Before time 0 s all steel, water and gas is at the same constant temperature as the water at time 0 s.

The results can be seen in figure 4.2. The result shows that the rise in water temperature is almost independent of the initial water temperature. It also shows how much the water temperature is expected to rise given a certain rise in off-gas temperature.

We also see that it takes approximately 200 s after the change in gas temperature to reach a new steady water temperature. This is caused by the 144 seconds it takes to transport water through the WCD (see figure 1.1) together with the time it takes to
heat up the steel pipes.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{figure42.png}
\caption{Output water temperature and off-gas temperature with different constant input off-gas temperatures and different constant input water temperatures.}
\end{figure}

\subsection{WCD and heat-exchanger with constant mass flow}

The system with the WCD connected to a heat-exchanger (the system in figure 3.6) is tested with different off-gas temperatures and different constant flows $\dot{m}_1$ through the heat-exchanger. At time 0 s all the steel and all the water has temperature 125 °C.
Outgoing water temperatures for different flows in heat exchanger, $T_{g_{in}} = 800$°C

**Figure 4.3** The result from different flows through the heat-exchanger with a stationary off-gas temperature of 800 °C

From figure 4.3 it can be seen that the water temperature oscillates. This happens because the water circulates through the heat-exchanger and WCD. In figure 4.1 it can be seen that the off-gas temperature may remain high for periods up to almost 1000 s, which is shorter than the time it takes for the water temperature to reach a stationary value.

### 4.3 Open-loop step response

To tune parameters for a controller an open loop step response is often used. In our case this means studying how the water temperature in the system is affected by a step in $\dot{m}_1$ (after a stationary state has been reached). We simulated different step responses to see the overall behavior of the system.

In figure 4.4 the multiple step responses can be seen for different flows through the heat-exchanger, with the same off-gas temperature. In order to more clearly illustrate the step response steps of magnitude 10 are applied to the process.
4.3 Open-loop step response

The process has quite a large time delay. The temperature goes down in steps. This behavior is caused by the same water circulating through the whole system, where the WCD is just one part and it takes time to heat the steel.

It is also clear that the system is non-linear; the amplitude of the step response depends on the initial water-temperature.

### 4.3.1 Open-loop step response for different off-gas temperatures

In figure 4.5 the step response for different off-gas temperatures can be seen. In this experiment the flow before the step is such that the water temperature is kept at
125 °C. This is done because this is the water temperature we ideally want the water to stay at, and so the system behaviour at this temperature is the most important.

The first step of the step response (up until around 1500 s) is almost identical for the different off-gas temperatures.

For the tuning of the controllers only the initial "step" will be considered. This can be seen in figure 4.6. This way the time constant is shorter than the delay and we have a step response from which we can tune a controller.
Three parameters are extracted from this step response. The static gain is defined as $K_p = \Delta y / \Delta \dot{m}_1$, where $y$ in this case is $TT1B$.

$L$ is the dead time of the system, the time it takes for the control signal to have an effect on the output signal. It is defined as the time where the greatest slope of the step response (the orange line in 4.6) intersects the dashed line of the initial temperature.

$T$ is the time constant of the system. It is defined as the time it takes for the output signal to reach 63\% of the way towards its final value, after the dead time.

These parameters will later be used to find suitable parameters for our PID-controllers (see section 5.1.8).
4.4 Required flows to keep temperature at 125 °C

The required flow through the heat-exchanger to keep the water at 125 °C is non-linear for different off-gas temperatures. As the off-gas temperature after the WCD is one of the least delayed measurements of the behavior of the whole system it is worth using for control. In figure 4.7 an exponential curve has been fitted to simulated flows required to keep the water temperature at 125 °C in stationarity for the complete system.

\[
\dot{m}_1 = 10.7 \text{ kg s}^{-1} \cdot \left( \exp \left[ (T_{\text{gout}} - 125 \degree \text{C}) \cdot 3.57 \cdot 10^{-3} \degree \text{C}^{-1} \right] - 1 \right)
\]

This is the function that will be used for feed-forward in our controllers.
4.5 Discussion

The system has long time delays and non-linear behavior. The long time delays, non-linear behavior and large disturbances mean that simple PID-control may not be enough.

If only the initial step is considered as in figure 4.6 a regular step response analysis can be made to tune a controller. The other steps can then be seen as predictable disturbances.

The non-linear behavior in figure 4.7 is not a big problem as it can be seen as a static non-linearity in stationarity. This means that it can be compensated for by simply using output which matches the fitted exponential curve.
Control

In this chapter we develop control strategies using the results of the previous chapter, in order to achieve the goals described in section 1.2.

A total of seven controllers are presented. They are all based on PID-control of one of the available temperature sensors. Some of them use TT1C as the controlled temperature, while others use TT1B. Some of the controllers use feed-forward, while others do not. The first four controllers compare different combinations of controlled temperature and feed-forward. The final three controllers illustrate further improvements based on specific insight into the process to be controlled.

After the controllers have been presented and explained, they are tested on a number of scenarios. Finally, the results of these tests are discussed and the effectiveness of the controllers compared.

5.1 Controllers

5.1.1 Controller 1: Single PID-control of TT1C

A very simple controller for the system is shown in figure 5.1.
5.1 Controllers

Figure 5.1  Controller 1: Simple PID-controller, using TT1C as the controlled variable.

This controller simply uses PID-control with TT1C as the controlled variable. It also includes a gain used to compensate for varying TT6C. This is described further in section 5.1.9.

While this controller might work to some extent, it makes no use of the measured off-gas temperature at TT1A. A better controller would include a feed-forward of TT1A.

5.1.2 Controller 2: Single PID-control of TT1C with offgas feed-forward

A controller with off-gas feed-forward is shown in figure 5.2.

Figure 5.2  Controller 2: PID-controller with offgas feed-forward, using TT1C as the controlled variable.

This controller compensates for the off-gas disturbance using feed-forward of...
Chapter 5. Control

TT1A. The non-linear gain is given by the exponential fit in figure 4.7 which maps off-gas temperatures to mass flows. Specifically:

\[
\dot{m}_{ff} = 10.7 \text{kg s}^{-1} \cdot (\exp[(\text{TT1A} - 125 \degree \text{C}) \cdot 3.57 \cdot 10^{-3} \text{C}^{-1}] - 1)
\]  

(5.1)

Using text-book control theory, a time delay should be added to the feed-forward, to make sure that the feed-forward cools the same water that the off-gas heats. The time delay is the time it takes for water to get from the outlet of the water-cooled duct to the inlet of the heat-exchanger, which is 72 seconds (see figure 1.1). This time-delay would improve the performance of the controller when subjected to step-loads. However, it turns out that such a time-delay actually decreases performance on ramp-disturbances. Since the real-life disturbances that our controllers will be subjected to is more ramp-like than step-like, we have decided to not include this time-delay.

While using TT1C as the controlled variable makes sense in that it represents the temperature we really want to control, it might be better to use TT1B as the controlled variable instead. The reason is that it gives information about the water temperature sooner than TT1C does, which means lower time-delays. Lower time delays means a slightly more aggressive controller can be used without inducing oscillations. Using TT1B is valid because there is almost no change in temperature through the heat-exchanger (see equation (3.31)), so TT1C is just a delayed measurement of TT1B.

5.1.3 Controller 3: Single PID-control of TT1B

For future comparison with Controller 1 we include a PID-controller without feed-forward with TT1B as the controlled variable, seen in figure 5.3.

![Figure 5.3 Controller 3: Simple PID-controller, using TT1B as the controlled variable.](image-url)
5.1.4 Controller 4: Single PID-control of TT1B with offgas feed-forward

A controller using off-gas feed-forward and using TT1B as the controlled variable is seen in figure 5.4.

![Controller Diagram]

**Figure 5.4** Controller 4: PID-controller with offgas feed-forward, using TT1B as the controlled variable.

This controller is to be compared with Controller 2. This controller should work reasonably well. However, there is one more feed-forward we can add which will improve the performance of the controller.

Looking at figure 4.5 we see that after the first "step" in the step response, there are a number of secondary steps. These steps are caused by the recirculation of water; in other words, the water that is cooled by the first step recirculates, and then passes through the heat exchanger a second time to be cooled even more. This process is repeated until a new balance is reached between the temperature increase from the off-gas and the temperature decrease in the heat-exchanger.

These secondary steps can be thought of as measurable load-disturbances, meaning that they can be counteracted. To do this we add a negative feed-forward of the control signal.

5.1.5 Controller 5: Single PID-control of TT1B with offgas feed-forward and control-signal feed-forward

This controller is shown in figure 5.5.
Now we have added a negative feed-forward of the control-signal, with a gain less than one, and a time delay. By empirical testing we have found 0.3 to be an appropriate value. The time delay should be the time between two steps in figure 4.5, which is the time it takes the water to make one complete circulation of the circuit. This time is 306 s.

Note that we need to "undo" the TT6C compensation gain before the feed-forward. Otherwise, the forward-fed control signal would be affected by the compensation gain multiple times.

5.1.6 Controller 6: Single PID-control of TT1B with offgas feed-forward and control-signal feed-forward, with P-part acting on TT2B

We mentioned the advantage of using TT1B as the controlled variable rather than TT1C. However, one disadvantage of this is that the P-part of the PID controller will instantly react to changes in TT1B by increasing or decreasing $\dot{m}_1$. However, the result of increasing $\dot{m}_1$ is a cooling of the water at the heat-exchanger, not the water at TT1B. In other words, the "wrong" water is cooled. If the controller instead uses TT2B for control, it acts on the "correct" water.
However, using TT2B would give the same disadvantage of increased delays, which will make our control worse. We want to see changes in water temperature as early as possible, so that the I- and D-parts can start reacting as quickly as possible.

One compromise is to let the P-part of the controller act on TT2B, and the I- and D-parts act on TT1B. This controller is illustrated in figure 5.6.

![Figure 5.6](image)

**Figure 5.6** Controller 6: PID-controller with offgas feed-forward and control-signal feed-forward, using TT1B as the controlled variable for ID and TT2B for P.

### 5.1.7 Controller 7: Single PID-control of TT1B with offgas feed-forward and control-signal feed-forward, with P-part acting on TT2B and off-gas temperature prediction

The big time delays are a problem when trying to control the system. Especially problematic is the off-gas disturbance, which can change in temperature quite a lot during the time it takes for water to leave the heat-exchanger and reach the WCD.

What we ideally would want is to know what the off-gas temperature will be at 90s to 234s in the future (the time it takes water at the outlet of the heat-exchanger to reach the inlet and outlet of the WCD, respectively. See figure 1.1). Unfortunately, we do not have this information. However, we can attempt to predict it using our current temperature measurements. This is what is new in Controller 7, as seen in figure 5.7.
Controller 7 attempts to predict what the outgoing off-gas temperature will be 162 s (the average of 90 s and 234 s) in the future. It does this by using the currently measured temperature, as well as its derivative. Basically, it performs a first order Taylor-expansion by approximating TT1A as

$$\text{TT1A}(t + 162s) \approx \text{TT1A}(t) + 162s \cdot \text{TT1A}'(t)$$ \hspace{1cm} (5.2)$$

In other words, it assumes that the off-gas temperature is a ramp with constant slope and approximates the future temperature accordingly. As can be seen in figure 4.1 the off-gas temperature has a ramp-like behavior most of the time, meaning that this is likely a reasonable approximation.

There are two more things to note about the controller. First off, it includes a low-pass filter before the derivative, in order to capture the overall changes in temperature while ignoring small "bumps". This filter has the added benefit of smoothing the predicted temperature, and thus the control signal, which results in a less jerky water temperature.
Secondly, it includes a limiter, with a lower limit of 0. This is done because we never want the predicted TT1A to be too low, since that could cause the controller to not cool the water enough. Since it is quite bad if the water gets too hot we have chosen to include this safety precaution.

An interesting side-note is that this prediction is equivalent to a lead-filter, if we ignore the limiter on the derivative part. To see this, note that the signal that is passed into the non-linear gain in figure 5.7 is given by

\[ TT1A \cdot \left( 1 + \frac{1}{1 + 900s} \cdot s \cdot 162 \right) = TT1A \cdot \left( \frac{1 + 1062s}{1 + 900s} \right) \]  

which is the transfer function of a lead-filter. Formulating the controller this way may simplify implementation.

5.1.8 PID Tuning

To tune the PID-controllers the *Approximate M-constrained Integral Gain Optimization* method (AMIGO) is used ([Hägglund, 2012, p. 68]). In this tuning method the process parameters, \( K_p \), \( L \) and \( T \) from a step response in figure 4.6 are used. The PID parameters are then given by:

\[
K = \frac{1}{K_p} \left( 0.2 + 0.45 \frac{T}{L} \right) = -0.296 \text{ kg s}^{-1} \text{ K}^{-1}
\]

\[
T_i = \frac{0.4L + 0.8T}{L + 0.1T} \cdot L = 122 \text{ s}
\]

\[
T_d = \frac{0.5LT}{0.3L + T} = 38.0 \text{ s}
\]

5.1.9 Compensating for varying TT6C

As seen in (3.29), the temperature at TT3B \( (T_{2o}) \) is affected by the temperature at TT6C \( (T_{1i}) \). Our controllers are tuned for the case where \( T_{1i} = 55 \degree \text{C} \). This means that the \( \dot{m}_1 \) that the controller outputs will be appropriate if TT6C = 55 \degree \text{C}. However, if the temperature at TT6C is colder (or warmer) than this, the heat-exchanger will cool the water at TT3B more (or less) than the controller expected, which will negatively affect the performance. Thus we need to compensate for this.

In order to do this we use the approximation (3.32). From this equation it can be seen that if \( T_{1i} \) is lower (or higher) than the controller expects, \( T_{2o} \) will also be lower than it expects. This is compensated for by decreasing (or increasing) \( \dot{m}_1 \). \( \dot{m}_1 \) should be adjusted in such a way to keep \( \dot{m}_1 (T_{1i} - T_{2i}) \) constant for varying \( T_{1i} \). This can be achieved by multiplying \( \dot{m}_{1c} \) from the controller by \( (55 \degree \text{C} - T_{2i})/(T_{1i} - T_{2i}) \), so that the final control signal is
\[ \dot{m}_1 = \frac{55^\circ C - TT2B}{TT6C - TT2B} \cdot \dot{m}_{1c} \]  

(5.5)

### 5.1.10 PPI-control

The *Predictive PI*-controller (PPI), is a controller suitable for processes with long time delays ([Hägglund, 2012, p. 151]). It can be derived as a special case of a Smith-predictor, with a choice of parameters which gives an especially simple control law. The control law of a PPI-controller is given by:

\[
{u(t)} = K \left( e(t) + \frac{1}{T_i} \int e(t) \, dt \right) + \frac{1}{T_i} \int \left( u(t - L) - u(t) \right) \, dt
\]

(5.6)

and the controller parameters are chosen as:

\[
K = \frac{1}{K_p} \\
T_i = T_p \\
L = L_p,
\]

(5.7)

where \(K_p, T_p\) and \(L_p\) are the parameters found from a step response as in figure 4.6. This controller was tested and discarded. The result was very oscillatory and also unstable for some test cases. The results can be seen in appendix C.

We believe that the non-satisfactory performance is caused by the secondary steps in figure 4.5 which act as disturbances that are not part of the inner model of the PPI-controller.

### 5.1.11 Implementation

The different proposed controllers are built up in Simulink and tested on the model of the WCD and heat-exchanger. The implementation looks similar to what can be seen in figures 5.1-5.7, but in Simulink notation and with proper anti-windup. The controllers cannot be tested on the real process as it is not yet available. These simulation results will act as guidelines to what kind of controller is most suitable for the real process.

### 5.1.12 Testing

Each controller will be tested in a number of ways, using the simulated process, to evaluate their performance. They will be tested on step-loads of different magnitudes, on ramp-loads of different slopes, and on the real disturbance data.
5.2 Results

In this section the different controllers are subjected to different test-scenarios, and the results are presented. Only the most interesting results will be presented here, with the rest being presented in appendix D.

5.2.1 Step-load up

In this test we let the controllers achieve stationarity at 125°C, and then subject them to a step in the off-gas of a few different magnitudes.
Chapter 5. Control

**Step-load up, Controller 1**

![Graphs showing temperature and mass flow rate responses](image)

**Figure 5.8** Controller 1 subjected to different step-loads with reference temperature 125°C.
**5.2 Results**

*Step-load up, Controller 2*

---

**Figure 5.9** Controller 2 subjected to different step-loads with reference temperature \(125^\circ\text{C}\).
Chapter 5. Control

**Step-load up, Controller 3**

![Graph showing step-load up for Controller 3 with different conditions.]

**Figure 5.10** Controller 3 subjected to different step-loads with reference temperature 125°C.
5.2 Results

**Step-load up, Controller 4**

![Graphs showing TT (°C) over time and mdot (kg/s) over time with Tg in (°C) as a constant reference temperature of 125°C.]

**Figure 5.11** Controller 4 subjected to different step-loads with reference temperature 125°C.
Chapter 5. Control

**Step-load up, Controller 5**

Figure 5.12  Controller 5 subjected to different step-loads with reference temperature 125°C.
5.2 Results

*Step-load up, Controller 6*

![Graphs showing temperature, mass flow rate, and inlet temperature over time.]

**Figure 5.13** Controller 6 subjected to different step-loads with reference temperature 125°C.
Step-load up, Controller 7

Figure 5.14  Controller 7 subjected to different step-loads with reference temperature 125°C.
5.2 Results

5.2.2 Step-load down

In this test we reverse the step from the previous test, to test how the controllers handle a downwards step-load. Only the results for controllers 3 and 7 are shown. The results for the other controllers can be found in appendix [D].
Step-load down, Controller 3

Figure 5.15 Controller 3 subjected to different downward step-loads with reference temperature 125°C.
5.2 Results

**Step-load down, Controller 7**

![Graphs showing temperature and mass flow rate over time for Controller 7 subjected to different downward step-loads with reference temperature 125°C.]

**Figure 5.16** Controller 7 subjected to different downward step-loads with reference temperature 125°C.
5.2.3  Ramp-load up

In this test we let the controllers achieve stationarity at 125°C, and then subject them to a ramp in the off-gas of a few different slopes. Only the results for controllers 3, 4, 6 and 7 are shown. The results for the other controllers can be found in appendix D.
Ramp-load up, Controller 3

Figure 5.17  Controller 3 subjected to different ramp-loads with reference temperature 125°C.
Chapter 5. Control

Ramp-load up, Controller 4

Figure 5.18 Controller 4 subjected to different ramp-loads with reference temperature 125°C.
**5.2 Results**

**Ramp-load up, Controller 6**

![Graph showing TT, mdot, and Tg_in over time.](image)

**Figure 5.19** Controller 6 subjected to different ramp-loads with reference temperature 125°C.
Chapter 5. Control

**Ramp-load up, Controller 7**

![Graphs showing control variables over time](image)

**Figure 5.20** Controller 7 subjected to different ramp-loads with reference temperature 125°C.
5.2.4 Real load

In this test we test the controllers 3, 4, 5, 6 and 7 using the real off-gas temperature data. The results for the other controllers can be found in appendix D.
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Real load, Controller 3

Figure 5.21  Controller 3 subjected to a load based on real-world data with reference temperature 125°C.
5.2 Results

**Real load, Controller 4**

Figure 5.22  Controller 4 subjected to a load based on real-world data with reference temperature 125°C.
Figure 5.23 Controller 5 subjected to a load based on real-world data with reference temperature 125°C.
5.2 Results

Real load, Controller 6

Figure 5.24 Controller 6 subjected to a load based on real-world data with reference temperature 125°C.


Real load, Controller 7

Figure 5.25  Controller 7 subjected to a load based on real-world data with reference temperature $125^\circ$C.
5.2 Results

5.2.5 Compensation for water temperature from accumulator

As discussed in section 5.1.9, we need to compensate for the water temperature coming from the accumulator. To test how well this works, we let Controller 7 run on the real disturbance, using a few different constant temperatures for this water. The setup is the same as for the experiment in figure 5.24, so ideally the water temperature would behave exactly the same as in this plot.
Chapter 5. Control

**Real load, TT6C = 30 °C**

![Graph showing temperature, mass flow rate, and power over time with annotations for TT1B, TT1C, TTref, mdot, and Power.

Figure 5.26  Controller 7 subjected to a load based on real-world data with reference temperature 125°C. TT6C = 30 °C.
5.2 Results

Real load, $TT6C = 80 \degree C$

Figure 5.27 Controller 7 subjected to a load based on real-world data with reference temperature 125\degree C. TT6C = 80 \degree C.
5.2.6 Cold-start

In these tests Controller 7 is started with a lower initial temperature (a "cold-start"), with and without a gradual ramp-up of the reference temperature.

**Cold-start without reference-ramp**

![Graph showing temperature and flow rate over time](image)

**Figure 5.28** Controller 7 subjected to a cold-start.
### Cold-start with reference-ramp

![Graph showing temperature (°C) over time (s) with various labels and curves for TT1B, TT1C, and TT_ref.]

![Graph showing mass flow rate (kg/s) over time (s) with different line types for Control signal, Feed-forward, and Extracted Power.]

![Graph showing temperature (°C) over time (s) for Tg_in and Tg_out.]

**Figure 5.29** Controller 7 subjected to a cold-start with ramp-up of reference temperature.
5.3 Discussion

5.3.1 Upwards step
The initial behavior of the step load (figures 5.9 and 5.11-5.14) is unavoidable as there is a delay before the water from the heat-exchanger reaches the WCD as can be seen in figure 1.1. In other words, it is impossible for the controller to suppress the first "bump" (which goes to slightly above 150°C) without knowing in advance that the load-disturbance is coming.

The only way to guarantee that the water temperature constraint is satisfied is to have a low enough initial water temperature. It would require a lower temperature than 125°C. Fortunately there are no direct continuous step loads in the real off-gas temperatures, as can be seen in figure 4.1. The off-gas temperature is more ramp-like.

The simple PID controllers (1 and 3) have a temperature that continues to rise, as can be seen in figures 5.8 and 5.10. The other controllers use the suggested feedforward of the off-gas temperature from sensor TT1A which completely mitigates this problem.

With Controllers 1-4 there are large slow oscillations with a period time of about 1250 s. This behavior comes from, as seen in the step response in figure 4.5, that a change in flow gives additional steps because of the recirculation of the water through the heat-exchanger. The control signal compensation suggested for controller 5, greatly reduce this problem as can be seen in figure 5.12.

The step response of Controller 6 is discussed in section 5.3.8 and that of Controller 7 is discussed in section 5.3.9.

5.3.2 Downwards step
The downward steps in figures 5.15 and 5.16 may seem to indicate poor controller performance, with the oscillations and slow convergence. However, this response is largely unavoidable. Note that the controllers with off-gas temperature feed-forward almost instantly shut off the cooling as soon as the temperature decreases. The water temperature in the WCD and heat-exchanger remains unchanged as we can only cool it further, not heat it up without hot off-gas. When the water circulates there will be oscillations depending on the state of the water before the cooling stopped. As the control signal goes to zero, there will be no flow on the accumulator side of the heat-exchanger. If there is no flow, it does not matter what the temperature of the water is.

The feed-forward makes sure that the cooling stops earlier, which means that much
of the water does not get cooled to much. The downside is that there will be a large temperature difference between the water that got cooled before the feed-forward reacted and the water that was not cooled, which results in the large oscillations as the water circulates.

5.3.3 Upwards ramp

The response to the upward ramp disturbance is more or less a milder version of the response to the step disturbance. As can be seen in figure 5.17, the performance of Controllers 3, which does not have off-gas feed-forward, is very poor. Adding off-gas feed-forward is a huge improvement, as can be seen by the performance of Controller 4 in figure 5.18.

However, there again are oscillations with a period time of about 1250 s. Like before, the control-signal feed-forward mostly gets rid of this, as can be seen in figure 5.19.

Finally, the off-gas temperature prediction of Controller 7 works best of all the controllers. This is discussed further in section 5.3.9.

5.3.4 Real disturbance

The real disturbance consists mostly of ramp-loads with different slopes. If a controller works well for the upwards ramp it should therefore work well here also. This disturbance is for the old system and may not accurately represent the disturbances of the new system. The initial off-gas temperature is a big uncertainty as there is no sensor data for it on the old system and we do not know how it will change for the new system.

The goal is to have an outgoing water temperature of 125 °C while never reaching a temperature above 140 °C. All the controllers except Controller 1 and Controller 3 (which do not have any off-gas feed-forward) manage this within some margin. The temperature does go significantly below 125 °C at some points for all controllers, but as there is almost no water flow at these temperatures it is no problem. Since the off-gas is quite cool at these times here is not much energy added to the system.

The estimated power output goes up to about 15 MW, but the mean power output is only around 5 MW. The goal is to have a mean power output of 8 MW, but as there are a lot of uncertainties in the model and the disturbance the mean value may not be very accurate.

5.3.5 Compensation for water temperature from accumulator

Comparing figure 5.25, figure 5.26 and figure 5.27 it can be seen that the compensation for variations in TT6C works very well; TT1B and TT1C are basically
unaffected. The extracted power is also affected, which is exactly the purpose of the compensation.

The only big difference is the control-signal, which has a larger magnitude the higher the temperature at TT6C. This is to be expected since the compensation is designed to increase the control-signal to compensate for lower temperature differences.

5.3.6 Controlling TT1B vs. controlling TT1C
As mentioned in section 5.1.2 using TT1B instead of TT1C as the controlled variable has the advantage that the controller notices changes in water temperature quicker, and therefore can it counteract a load disturbance faster.

This is clearest when comparing Controller 1 and Controller 3. For example, comparing the step load responses in figure 5.8 and figure 5.10 we see that Controller 3 has significantly less overshoot than Controller 1. The same can be seen in the ramp response in figure 5.17.

With off-gas feedback the difference is smaller, but can still be seen by comparing the ramp load responses of Controller 2 and Controller 4 in figure D.14 (in appendix D) and figure 5.18.

One disadvantage of using TT1B rather than TT1C can be seen by comparing the step-responses of Controller 2 compared to Controller 4 in figure 5.9 and figure 5.11. Controller 4 induces extra oscillations with a period of around 300s. The reason for this, and the solution to it, will be discussed in section 5.3.8.

5.3.7 Control-signal feedforward
As discussed in section 5.1.4 Controller 5 includes a control-signal feedforward meant to dampen the oscillations with a period of around 3500 s. Comparing the step responses of Controller 4 and Controller 5 in figure 5.11 and figure 5.12 it is clear that this works very well. The undershoot of Controller 4 is almost completely removed. The same improvement can be seen by comparing Controller 4 and Controller 5 in all the other scenarios as well.

5.3.8 P-part on TT1B vs. TT2B
As discussed in section 5.3.6 one disadvantage of Controller 4 compared to Controller 2 is the extra oscillations with a period of around 300 s it creates. This is explained in section 5.1.6 where Controller 6 is created to fix this issue. This is done by letting the P-part of the controller act on TT2B, while letting the I- and D-parts act on TT1B.
5.3 **Discussion**

The biggest difference this makes can be seen by comparing the step response of Controller 5 and Controller 6 in figure 5.12 and figure 5.13. Controller 6 dampens the high frequency oscillations noticeably more than Controller 5. However, the difference is not huge.

The difference can also be seen with the real load in figure 5.23 and figure 5.24. Again, the differences are not enormous, but in comparing TT1B at around 3500s for the two controllers Controller 6 manages to dampen the oscillation better than Controller 5.

### 5.3.9 Off-gas temperature prediction

In order to examine how well the off-gas temperature prediction works we compare Controller 6 and Controller 7.

As mentioned in section 5.1.7, the off-gas prediction is based on the assumption of a ramp-like behaviour of the disturbance. For this reason we would expect Controller 7 to perform well when this is a good approximation, and less well when it is not.

With a step-load, the ramp-like approximation is not a good one. Accordingly, it can be seen by comparing figure 5.13 and figure 5.14 that Controller 7 performs quite a bit worse than Controller 6 when subjected to a step-load. Controller 7 has quite a large undershoot, caused by the derivative of the off-gas temperature being very large for a short time. Even with the low-pass filter this causes the controller to cool the water too much.

However, with a ramp-load we would expect much better performance, since this is the case it was designed for. Comparing figure 5.19 and figure 5.20 this is indeed the case. Controller 7 performs significantly better than Controller 6. The downside compared to Controller 6 can be seen when the ramp flattens out, which causes the off-gas temperature prediction of Controller 7 to be too high, causing an undershoot in the water temperature. However, this undershoot is not very severe.

Finally, we turn to the real load case. As discussed in section 5.1.7 it has quite a ramp-like behaviour; the temperature usually rises with quite a constant slope over periods of time of around 500s to 1000s. For this reason Controller 7 should perform quite well on the real load.

Comparing figure 5.24 and figure 5.25 we see that this is indeed the case. Specifically, Controller 7 performs significantly better than Controller 6 at 500s to 1000s and at 3000s to 3500s. These are the times where the load-disturbance is most ramp-like.
On the down-side Controller 7 performs slightly worse than Controller 6 at around 1500s and at around 4000s to 4500s. However, overall the performance of Controller 7 is better than Controller 6. Especially since it keeps the maximum temperature of the water quite a bit lower, which is one of the most important aspects of the control problem.

5.3.10 Cold-start

Looking at figure 5.28 we see that when the initial temperature of the system is significantly lower than the reference temperature there is a risk of getting a large overshoot in water temperature. One way to mitigate this problem is to slowly increase the reference temperature, like in figure 5.29. This seems to be an effective solution to the problem.

5.3.11 Tuning

The simulated model has many approximations and parameter uncertainties. This means that the behavior of the real system will be different. The model will hopefully describe all of the major dynamics of the system. If this is true then only the specific tuning of the controllers’ parameters will need to be different, while keeping the overall structure the same. Of the parameters used for tuning, $L$, $T$, and $K_p$, only $L$ is known to be accurate for the real system as it only depends on the flow, geometry and distances of the cooling system, which is data that we have.

The tuning can be done by applying the AMIGO method (see section 5.1.8) which yields satisfactory results for the simulations. However, it might not be possible to find a step-response of the real system. The tuned parameters from the simulations can also be used and then manually adjusted during a real run.

Tuning the off-gas temperature feed-forward is harder as the tuning used in the simulations are based on a scenario of constant load-disturbances, which will not happen for the real process. Our suggestion is to use the non-linear gain proposed in this report and then manually tune it if it is not satisfactory.
6 Conclusions

6.1 Control performance

The simulations in this thesis provide necessary information on the approximate behaviour of the system when using the different control strategies. As can be seen in the results it will not be enough with a simple PID-controller and one of the more advanced control strategies that were proposed is needed. If one of the more advanced controllers is used, then the control objectives and constraints are satisfied. The step-load breaks the constraints for all controllers, but as discussed, this behaviour is impossible to control and it is not something that will happen in the real system.

The major dynamics of the system seems to be due to the time-delays and recirculation of the water. The recirculation dynamics are handled by a properly tuned PID, but the suggested control-signal feed-forward provides even better performance. Because of the time-delays in the system the PID is tuned to be very slow and is then very slow to react to load disturbances. The suggested off-gas temperature feed-forward compensates very well for the otherwise slow controller and also reduces the problem with the time-delays. With the suggested prediction for the off-gas temperature the feed-forward reduces the time-delay problem even further.

The simulated model of the WCD system is based on many approximations and there are many uncertainties in parameters. The simulation results will probably differ quite a lot from the real process, but all the major dynamics should be represented. For this reason we believe that the control strategies presented in this report (especially Controller 7) will be useful also for the real process. It will however likely require extra tuning of parameters to compensate for the differences between our model and the real system.

If the general system behaviour is captured in our model, and the off-gas temperatures will behave similar to the case we have tested against, Controller 7 is likely to
be the one that performs the best out of all the controllers discussed in this report. However, Controller 4 is quite a bit simpler in its implementation, and its performance is not catastrophically worse than that of Controller 7.

Since Controller 7 is based on Controller 4, with extra components added to improve performance, a reasonable strategy for controlling the real system is to start with Controller 4. Since it is relatively simple it will be easier to fine-tune its parameters in real-time to achieve acceptable performance. Once Controller 4 has been tuned to a satisfactory level, it can be expanded to Controller 5, Controller 6 and finally Controller 7. This gradual build-up of the controller minimizes the risk of unpredictable and erroneous behaviour.

6.2 Further work

6.2.1 Predicting off-gas temperature drops

One venue of possible further improvement would be to have some way to predict when the off-gas temperature will suddenly drop. If such an event could be predicted some time in advance the oscillations induced by the sudden drops could be reduced by lowering the control-signal beforehand, in anticipation of the event. The idea is similar to the predictive off-gas feed-forward of Controller 7, in that information about how the off-gas temperature will behave a few minutes into the future allows for better temperature control.

It is likely that there are signals available to the control system which would allow a robust prediction of the temperature drops some time in advance. However, because we have not had access to these signals we have not been able to test this idea further. Since the worst performance of the controllers occurs at the sudden off-gas temperature drops (because the control-signal cannot go below 0) this idea could improve the control performance by a large amount.

One difficulty in doing this is that if the controller wrongly predicts a temperature drop, it would lower the cooling which would result in a large overshoot in water-temperature. To avoid this the logic used to predict a temperature-drop has to be very robust and probably rely on more than just one signal indicating a coming drop.

6.2.2 Accumulator modelling

Another continuation of our work would be to include the accumulator circuit in the process model. This could give a more detailed evaluation of the ability of the system to deliver the required power. However, the main difference from the perspective of our controller design would be a more realistic model of the behaviour
of the water temperature at TT6C. Since this has a negligible effect on controller performance we have not prioritized this avenue of research.

### 6.2.3 Using measured EAF power

Another possible improvement would be to somehow use the available information of the EAF power. This could provide some additional information of how the off-gas temperature will behave. This has not been explored in this thesis, since there is far from a direct relation between the EAF power and the resulting off-gas temperature. However, since the EAF power data sometimes gives earlier information about how the off-gas temperature will behave (see figure 4.1), it is possible that there is some way to incorporate it to improve control.

### 6.2.4 PPI

While the PPI-controller didn’t achieve robust control of the system, it could very well be a useful tool if explored further. Some informal experiments showed that it could be able to control the system without any feed-forward, if a solution to the oscillatory behaviour is found. This is because it can use the implicit internal system model to control the system much more aggressively than a normal PID-controller.

### 6.3 Summary

The goal of this thesis was to find a control strategy which could deliver heat from a cooling system for a water-cooled duct to district heating. By modelling the system, simulating it and selecting an appropriate controller we believe to have achieved this goal.

We recommend a controller connecting to the system as in figure 6.1. The structure of the controller should be correct, but its parameters will need to be adjusted when used on the real process for proper performance.
Figure 6.1  Simplified schematic of the water-cooled duct and cooling system with Controller 7. Unused temperature sensors have been omitted.
Literature


A

Manufacturer data

<table>
<thead>
<tr>
<th>Section</th>
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<th>6</th>
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<td>No</td>
<td>Yes</td>
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</tbody>
</table>

**Table A.1**  Section specific parameters

$C_w$ | $4053 \text{ kJ m}^{-3} \text{ K}^{-1}$

$C_g$ | $1635.9 \text{ J Nm}^{-1} \text{ K}^{-1}$

$C_s$ | $3846.5 \text{ kJ m}^{-3} \text{ K}^{-1}$

$\alpha$ | $11.72 \text{ mm}^2 \text{ s}^{-1}$

**Table A.2**  Parameters used in the simulations
Calculation of $h_w$ and $h_g$

In this appendix the theory for calculating $h_w$ and $h_g$ is presented.

B.1 Heat-transfer coefficient of water

The first step to calculating $h_w$ is to determine if the water flow at a specific instant is laminar or turbulent. To this end we introduce the Reynolds number as

$$Re_d = \frac{\dot{m}d}{A\mu} \quad (B.1)$$

where $\dot{m}$ is the mass flow, $d$ is the hydraulic diameter, defined as $d = 4P/A$ where $P$ is the perimeter of the cross-section of the channel and $A$ is the area of the cross-section. Finally $\mu$ is the dynamic viscosity of the water.

In circular tubes, a rough criterion for the type of flow is that the flow is turbulent for $Re_d > 2300$, otherwise it is laminar. ([Holman, 2009, p.217]).

In our case the mass flow of water in a single tube depends on the number of tubes. The total mass flow of water is $\dot{m}_t = 400\text{t h}^{-1}$. The mass flow for a single tube then depends on how many tubes $n$ share this mass flow, so that the mass flow per tube is $\dot{m} = \dot{m}_t/n$. $n$ varies between 140 and 240 in the WCD under consideration.

Each tube has an inner diameter of 37.8 mm. For circular tubes the hydraulic diameter is the same as the diameter, so $d = 37.8\text{mm}$. The cross-sectional area of a tube is $A = \pi d^2/4 = 1.14 \cdot 10^{-3}\text{m}^2$.

The dynamic viscosity of water is temperature-dependent. An empirical formula for the temperature-dependence is given in ([Al-Shemmeri, 2012, p.17-18]) as
Appendix B. Calculation of \( h_w \) and \( h_g \)

\[
\mu(T) = 2.414 \times 10^{-5} \times 10^{247.8/(T-140)} \text{ kg m}^{-1} \text{ s}^{-1}
\]

(B.2)

where \( T \) is the water temperature in Kelvin. This formula is accurate to within 2.5\% for \( T \) between 0°C and 370°C. Our water temperatures will stay well within these limits.

Now we have everything needed to calculate Re for a given number of tubes and a given temperature. The highest number of tubes in the channel is \( n = 240 \), which will give the lowest \( \dot{m} \) which in turn gives the smallest Re. Using this \( n \) gives that

\[
\text{Re}_d = 1.19 \times 10^4 \gg 2300 \text{ for } T = 10^\circ C
\]

(B.3)

and

\[
\text{Re}_d = 1.16 \times 10^5 \gg 2300 \text{ for } T = 200^\circ C
\]

(B.4)

from which we conclude that the flow definitely is turbulent.

For turbulent flows in circular tubes the convective heat-transfer coefficient \( h_w \) is given by

\[
h_w = \text{Nu} \cdot \frac{k}{d}
\]

(B.5)

with Nu given by the semiempirical formula from ([Holman, 2009, p.259])

\[
\text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}
\]

(B.6)

The Prandtl number \( \text{Pr} \) is defined as

\[
\text{Pr} = C_w \mu / k
\]

(B.7)

where \( C_w \) is the specific heat capacity and \( k \) is the thermal conductivity. These values are also temperature dependent, but much less so than \( \mu \), so we will neglect their temperature dependence. \( T = 85^\circ C \) gives \( k = 0.67 \text{ W m}^{-1} \text{ K}^{-1} \) and \( C_w = 4.19 \text{ kJ kg}^{-1} \text{ K}^{-1} \).
Summarizing this gives a procedure for calculating $h_w$ for a given temperature $T$:

\[
\begin{align*}
  k &= 0.67 \text{W m}^{-1} \text{K}^{-1} \\
  d &= 37.8 \text{mm} \\
  A &= 1.14 \cdot 10^{-3} \text{m}^2 \\
  \mu(T) &= 2.414 \cdot 10^{-5} \times 10^{247.8/(T-140)} \\
  \text{Pr} &= W_w \mu(T)/k \\
  \text{Re}_d &= \frac{m d}{A \mu(T)} \\
  h_w &= k/d \cdot 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}
\end{align*}
\]

**B.2 Heat-transfer coefficient of off-gas**

Next we seek to calculate $h_g$. We split it in two parts, the convective heat-transfer coefficient $h_{gc}$ and the radiative heat-transfer coefficient $h_r$.

**B.2.1 Convection heat-transfer coefficient of off-gas**

We start with $h_{gc}$, the convective heat-transfer coefficient of the off-gas. The dynamic viscosity can be approximated by Sutherland’s law ([Sutherland, 1893])

\[
\mu = \mu_0 \left( \frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S}
\]  

where $\mu_0$ is the viscosity at some reference temperature $T_0$, and $S$ is a constant which depends on the type of gas. Using values of the viscosity at different temperatures calculated by the WCD-manufacturer we find that $T_0 = 825.6 \text{K}$, $\mu_0 = 3.61 \cdot 10^{-5} \text{N s m}^{-2}$ and $S = 160 \text{K}$. Also, using the manufacturer’s data we have $C_g = 1.226 \cdot 10^3 \text{J kg}^{-1} \text{K}^{-1}$ and $k = 8.32 \cdot 10^{-2} \text{W m}^{-1} \text{K}^{-1}$. Now, like before, we have

\[
\text{Pr} = C_g \mu/k
\]  

For a given normal volume flow $\dot{V}_N$, the actual volume flow $\dot{V}$ will depend on the temperature as

\[
\dot{V} = \frac{T}{273.15 \text{K}} \dot{V}_N
\]
Appendix B. Calculation of \( h_w \) and \( h_g \)

since gas density is inversely proportional to temperature for constant pressure, by the ideal gas law.

Now we can calculate Re, Nu and finally \( h_{gc} \) using the same formulas as for the water, if we are given the mass flow \( \dot{m} \) and the hydraulic diameter \( d \).

### B.2.2 Radiative heat-transfer from gas

A **black body** is a physical body which absorbs all incoming radiation. The energy \( E_b \) radiated from such a body follows the **Stephan-Boltzmann law** ([Holman, 2009, p.380])

\[
E_b = \sigma T^4
\]  
(B.12)

where \( \sigma = 5.669 \cdot 10^{-8} \text{ W/m}^2\text{K}^4 \). Most physical bodies are however not black bodies. A better approximation is that of a **gray body** which has a unitless **emissivity** \( \varepsilon < 1 \) and which radiates

\[
E_b = \sigma \varepsilon T^4
\]  
(B.13)

The radiative heat transfer between the steel pipe and the gas can be reformulated with a temperature-dependent radiative heat-transfer coefficient \( h_r \). We define \( h_r \) by \( q_r = h_r A (T_g - T_s) \), which gives that ([Holman, 2009, p.461])

\[
h_r = \frac{\sigma (T_s^2 + T_g^2) (T_s + T_g)}{1/\varepsilon_s + (A_1/A_2)(1/\varepsilon_g - 1)}
\]  
(B.14)

where \( \varepsilon_s \) and \( \varepsilon_g \) are the emissivities of the steel and the gas, and \( A_1 \) and \( A_2 \) are the radiation areas. In our situation \( A_1 \approx A_2 \), so

\[
h_r = \frac{\sigma (T_s^2 + T_g^2) (T_s + T_g)}{1/\varepsilon_s + 1/\varepsilon_g - 1}
\]  
(B.15)

From this equation we see that all we need to know about the emissivities is \( 1/\varepsilon_s + 1/\varepsilon_g \). Finding the emissivity of a gas is in general quite difficult ([Holman, 2009, p.420]) so we will instead find \( \varepsilon_s, \varepsilon_g \) which give \( h_r \) that match those calculated by the manufacturer. Doing this we find \( \varepsilon_s = \varepsilon_g = 0.55 \).


B.2.3 Total heat-transfer coefficient of gas

Summarizing the previous two sections gives a procedure for calculating \( h_g \) for a given temperature \( T \):

\[
T_0 = 825.6 \text{ K} \\
\mu_0 = 3.61 \cdot 10^{-5} \text{ N s m}^{-2} \\
S = 160 \text{ K} \\
C_g = 1.226 \cdot 10^3 \text{ J kg}^{-1} \text{ K}^{-1} \\
k = 8.32 \cdot 10^{-2} \text{ W m}^{-1} \text{ K}^{-1} \\
\mu = \mu_0 \left( \frac{T}{T_0} \right)^{3/2} \left( \frac{T_0 + S}{T + S} \right) \\
Pr = C_g \mu / k \\
d = 4A_g / P_g \\
u_g = V_g / A_g \\
\rho = 1.326 \text{ kg Nm}^{-1} = \frac{273.15 \text{ K}}{T} \cdot 1.326 \text{ kg m}^{-3} \\
Re_d = \frac{\dot{m} d}{A_g \mu(T)} = \rho u_g d / \mu(T) \\
h_{gc} = k / d \cdot 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} \\
\epsilon_s = \epsilon_g = 0.55 \\
h_r = \frac{\sigma (T_s^2 + T_g^2) (T_s + T_g)}{1 / \epsilon_s + 1 / \epsilon_g - 1} \\
h_g = h_{gc} + h_r \quad (B.16)
Figure C.1  The results from a step load using a PPI controller.
Figure C.2 The results from a step load using a PPI controller with feedforward from the off-gas temperature.
Figure C.3  The results from a step load using a PPI controller with feedforward from the off-gas temperature and control signal feedforward.
Additional Controller Results
Appendix D. Additional Controller Results

Step-load down, Controller 1

Figure D.1 Controller 1 subjected to different downward step-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

**Step-load down, Controller 2**

![Graphs showing temperature, mass flow rate, and inlet temperature over time for Controller 2.]

**Figure D.2** Controller 2 subjected to different downward step-loads with reference temperature $125^\circ C$. 

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Step-load down, Controller 4

Figure D.3 Controller 4 subjected to different downward step-loads with reference temperature $125^\circ$C.
Appendix D. Additional Controller Results

Step-load down, Controller 5

Figure D.4 Controller 5 subjected to different downward step-loads with reference temperature 125°C.
Step-load down, Controller 6

Figure D.5  Controller 6 subjected to different downward step-loads with reference temperature 125°C.
**Appendix D. Additional Controller Results**

*Ramp-load down, Controller 1*

![Graphs showing temperature and flow rate responses to ramp down loads for Controller 1 with a reference temperature of 125°C.](image)

**Figure D.6** Controller 1 subjected to different downwards ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

Ramp-load down, Controller 2

Figure D.7 Controller 2 subjected to different downwards ramp-loads with reference temperature $125^\circ$C.
Ramp-load down, Controller 3

Figure D.8  Controller 3 subjected to different downwards ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

Ramp-load down, Controller 4

Figure D.9  Controller 4 subjected to different downwards ramp-loads with reference temperature 125°C.
**Appendix D. Additional Controller Results**

**Ramp-load down, Controller 5**

- **TT (°C)**
  - TT1B
  - TT1C

- **mdot (kg/s)**

- **T_{g_{in}} (°C)**

**Figure D.10** Controller 5 subjected to different downwards ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

**Ramp-load down, Controller 6**

![Graphs showing temperature and mass flow rate over time](image)

**Figure D.11** Controller 6 subjected to different downwards ramp-loads with reference temperature $125^\circ C$. 

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Appendix D. Additional Controller Results

**Ramp-load down, Controller 7**

![Graphs of TT (°C), mdot (kg/s), and Tg in (°C) over time for Controller 7 subjected to different downwards ramp-loads with reference temperature 125°C.]

**Figure D.12** Controller 7 subjected to different downwards ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

Ramp-load up, Controller 1

Figure D.13 Controller 1 subjected to different ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

Ramp-load up, Controller 2

![Graphs showing response of TT, mdot, and Tgin for Controller 2 subjected to different ramp-loads with reference temperature 125°C.]

Figure D.14 Controller 2 subjected to different ramp-loads with reference temperature 125°C.
Appendix D. Additional Controller Results

**Ramp-load up, Controller 5**

![Graphs showing Temperature (TT), Mass Flow Rate (mdot), and Inlet Temperature (Tg_in) over time for Controller 5.]

**Figure D.15** Controller 5 subjected to different ramp-loads with reference temperature 125°C.
**Real load, Controller 1**

![Graph showing real load results for Controller 1](image)

*Figure D.16* Controller 1 subjected to a load based on real-world data with reference temperature 125°C.
Appendix D. Additional Controller Results

**Real load, Controller 2**

![Graphs showing temperature, mass flow rate, and power over time for Controller 2 subjected to a load based on real-world data with reference temperature 125°C.]

*Figure D.17*  Controller 2 subjected to a load based on real-world data with reference temperature 125°C.
Title and subtitle
Modelling and Control of a Water-Cooled Duct and Cooling System

Abstract

In this thesis a water cooled duct and the surrounding cooling system, used for cooling hot off-gas from an electric arc furnace, is modelled for simulation. The model is then used to devise control strategies to keep the cooling water temperature at a given temperature to then be able to supply power to the district heating. The model consists of the heat transfer between the off-gas, steel of the water cooled duct and the cooling water, the transport delays of the water and a heat-exchanger used for cooling the water.

The control problem is non-trivial as there are long time-delays and non-linear behavior, combined with large load disturbances caused by variations in off-gas temperature. Different control strategies of increasing complexity and performance are presented. The results show that a simple PID-controller is not enough and different feed-forward signals, which account for how the system behaves, are also needed. Using these additions the simulated system is controlled to a satisfactory degree.

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