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Improvement of measurement efficiency at the engine test bench

Improvement of the control strategy of the air conditioning unit

Master's Thesis in Automotive Engineering

THOMAS NOWACZYK

MASTER'S THESIS 2011: 2011-61

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Göteborg, October 2011

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Cover:
RaceCon, Dynamic Air Conditioning System, FEV

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Abstract

The improvement of measurement efficiency in industry is getting more and more important. Faster measurement procedures lead to more experiments in less time and so reduce the costs. This master thesis is focusing on the improvement of the conditioning unit on one of the test bench equipped with a small turbocharged diesel engine.

In order to test engine according to manufacturer specifications, the air going in the cylinder has to be at a certain temperature depending on the load and the speed of the engine. The intercooler, (exchanger air/air), being not present on a test bench, the intake air down-stream the turbocharger is cooled down by an air-water heat exchanger. This water is either warmed up by a heater or cooled down by a water-water heat exchanger. The temperature of the water in the cooling/heating system is controlled in order to get the desired intake air temperature. The cooling power is controlled by opening and closing a valve in a cooling circuit and the heating power is depending of the amount electricity consumed by the heater. Both of the cooling and heating units are controlled by one PI governor in a global control strategy.

The cold hydraulic circuit has to be modified in order get a more precise flow in the water-water exchanger to be able to control the intake air temperature. The cooling power is the crucial part having more than 20 kW of cooling power regarding the 6 kW of heating power. Once this step achieved, experiments will be done in order to modify and optimize the control strategy to get the best performances of the system.

Once this has been optimized, this control strategy will be adapted to other test benches with different engine size in order to get better and faster measurements.

This report will be written in English

Key words: Engine, diesel, test bench, control, air conditioning, air temperature

Content

1. Introduction and motivation for improving and developing the AirCon system

1.1. Introduction of the system

The AirCon system is an air conditioning system used in the engine test benches at FEV GmbH. It controls the air intake temperature down-stream turbocharger. The intake air being compressed, can reach high temperature down-stream the compressor at high load (until 140 °C). The goal of the system is to control this temperature in order the fit the engine requirements.

1.1.1. Introduction of the physical system
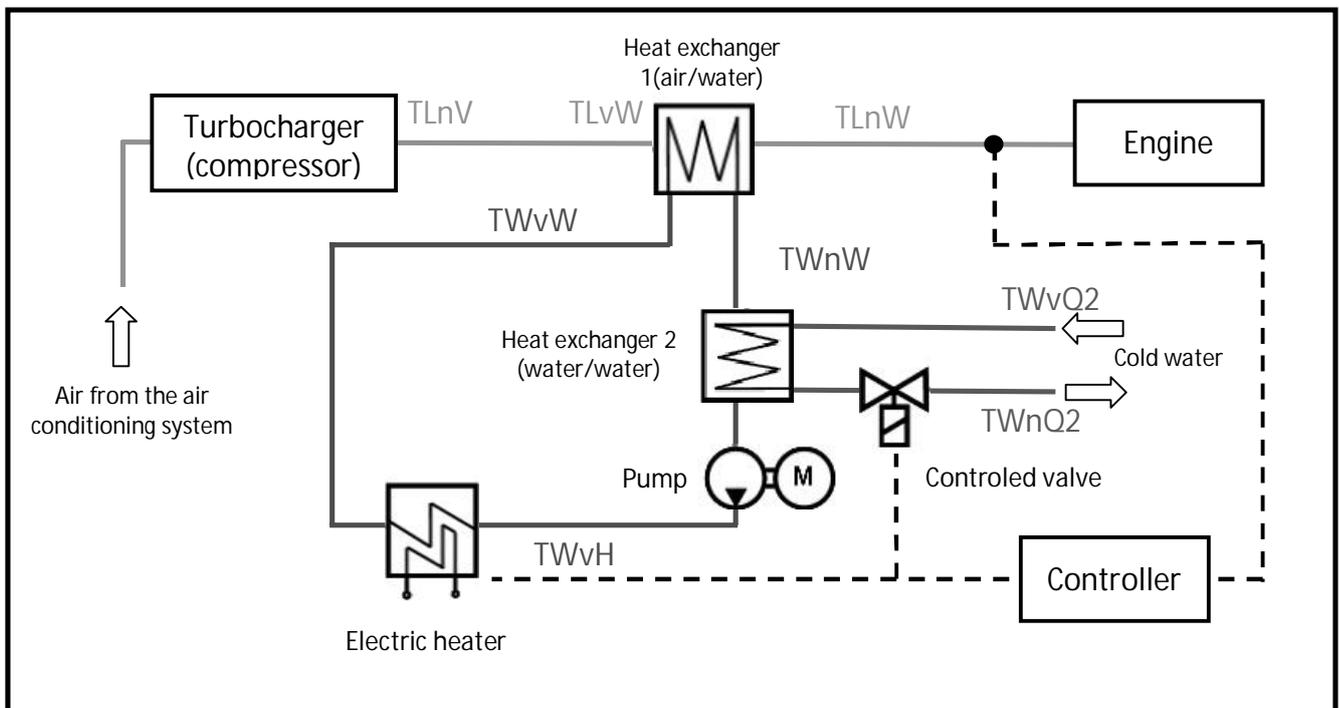
Figure 1: AirCon scheme

The system is made of two countercurrent heat exchangers, one electric heater, a water pump for the auxiliary water circuit, a controlled valve to set the cold water flow, temperature and flow sensors and the computer which controls the actuators. The auxiliary circuit is filled up with a mixture of water and glycol (70%/30%). The pump is working continually at constant mass flow. As shown on the scheme (dotted line) the controller is acting on the heater power and on the valve controlling the cold water mass flow (cooling power). The cold water comes from a cooling tower. It delivers cold water to all the test bench cells at a temperature that can vary between 4°C and 8°C depending on the outside temperature. The maximum cold water mass flow for the system is around 35 l/min.

In the following chapter all the experimentations and simulation have been made regarding a test bench equipped with a 1.3 liter turbocharged diesel engine.
The hardware parts specifications used in the AirCon can be found in Appendix 1.

The temperature labels in the system will respect the following rules:
T, for temperature, L for Luft (air), W for Water, n for nach (down-stream), v for vor (up-stream), V for Verdichter (compressor) and W for Wärmetauscher (heat exchanger).
For example, TLnV will be Temperature air down-stream compressor (Temperatur Luft nach Verdischter), TLnW will be temperature air down-stream heat exchanger (Temperatur Luft nach Wärmetaucher).
In the model the label can be followed by soll for set point or ist for simulated/measured values.
A precise scheme of the AirCon system with labels and a table with the translation of the labels are available in Figure 2 and in the list of symbols and abbreviations.

1.2. Background

On an engine test bench, the intake air temperature must be controlled to a defined value, normal using conditions, set by the manufacturer of the engine in order to fit realistic values that could be measured in a vehicle. In the case of this study, the intake air temperature has to be cooled down down-stream the compressor. Due to compression the air temperature reaches high level which cannot be tolerated.
This master's thesis will focus on heat transfers and control strategies.

1.2.1. Energy balance

Heat exchange:
The air conditioning system of the test bench is a thermodynamic system working with fluids exchanging heat with each other trough heat exchangers.
In this case, the equation of conservation of energy says that the energy lost by one fluid in the heat exchanger is gained by the other fluid.

Fluid 1:
$cp_1, \dot{m}_1, T_{1in}$

Fluid 2:
$cp_2, \dot{m}_2, T_{2in}$

Heat exchanger

Fluid 2:
$cp_2, \dot{m}_2, T_{2out}$
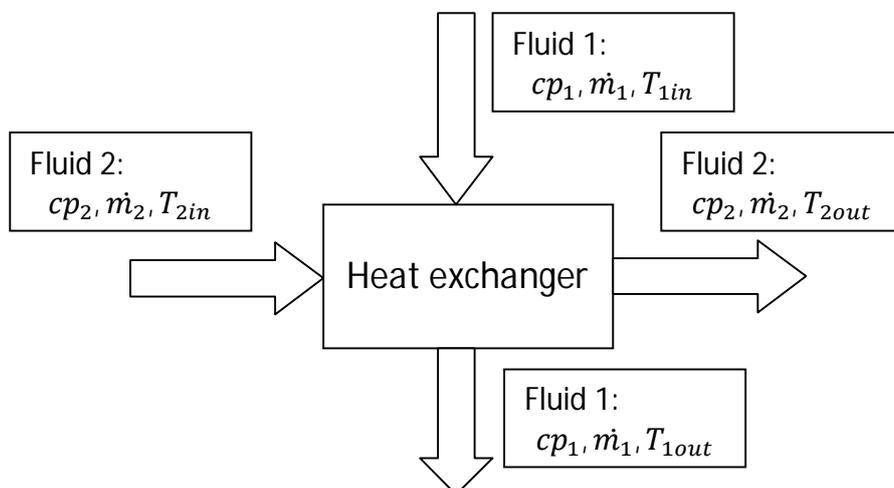
Fluid 1:
$cp_1, \dot{m}_1, T_{1out}$

The following formula is obtained, if heat losses are neglected:

$$\dot{m}_2 cp_2 (T_{2in} - T_{2out}) = \dot{m}_1 cp_1 (T_{1out} - T_{1in}) \qquad \textbf{\textit{(1.1)}}$$

With $\dot{m}_1$ and $\dot{m}_2$ the fluid mass flows in kg/s, Cp_1 and Cp_2 the heat capacity in J/K, T_{11} and T_{21} the fluid input temperatures in Kelvin and T_{21} and T_{22} the fluid outputs temperatures in Kelvin. In the report temperatures will be set in Celsius degrees, °C, since with are always working with temperature differences.

Convection:
Convection is a heat transfer resulting from the movement of a fluid. There are two different kinds of convection, natural convection when the fluid moves due to temperature difference with no external source and forced convection when the fluid movement results from an external source such as a fan.
The heat transfer due to convection is given by the following formula:

$$Q = hA(T - T_{air}) \qquad \textbf{\textit{(1.2)}}$$

With h the convective heat transfer coefficient in W/m²K, A the exchange area in m², T_{air} the temperature of the ambient air in Kelvin, and T the mean temperature of the surface in Kelvin.
In the following, these parameters, convection heat coefficient and the exchange area, will not be calculated exactly. This will be calibrated using experimental results.

1.2.2. Control, governors definitions and techniques

Laplace transform:
The Laplace transform is used in control to simplify formulas and equations. The Laplace variable will be referred as s. The signals in the Laplace domain will use capital letters, for example an input signal $u(t)$ will be written $U(s)$ in the Laplace domain.
An integration in the Laplace domain is equivalent of divided by s and a derivation of multiplying by s.

$$\mathcal{L}\left(\int u(t)dt\right) = \frac{U(s)}{s}$$

And

$$\mathcal{L}\left(\frac{du(t)}{dt}\right) = sU(s)$$

The transfer function, usually named $H(s)$, is used to describe the system. It is the ratio of the system output, y, divided by the system input, u in the Laplace domain.

$$H(s) = \frac{Y(s)}{U(s)}$$

Governors:

An engine test bench is an automated system. The human intervention during test is very limited almost nonexistent. Therefore all the systems around the engine and the engine itself are controlled.

Controlling a system can be resumed in controlling one or several values within this system. The system is using a closed loop control strategy. This means that these controlled values are compared to reference values or set points, and an error, e, is calculated. Then the governor uses this error to correct the system.

Different types of controllers can be used depending of the needs and of the level of complexity of the controlling systems:

P governor (proportional control)

The output of the P governor is directly proportional to the error:

$P_{out}(t) = K_p \cdot e(t)$ which is equivalent to $P_{out}(s) = K_p \cdot E(s)$

The transfer function is:

$$H_p(s) = \frac{P_{out}(s)}{E(s)} = K_p$$

K_p is called the proportional gain. An increase of K_p increase the velocity of the response but also leads to bigger over/undershoot and can lead to instability.

I governor (integral control)

The I governor integrates the error:

$I_{out}(t) = \frac{1}{T_i} \int e(t) dt$ which is equivalent to $I_{out}(s) = \frac{E(s)}{T_i \cdot s}$

The transfer function is:

$$Hi(s) = \frac{I_{out}(s)}{E(s)} = \frac{1}{T_i \cdot s}$$

T_i is the integration time constant. A small T_i, large I part, makes the system faster but leads to bigger over/undershoot and can also lead to instability. The big advantage of the I part is that in stationary condition the error goes to zero.

The stability of a system depends a lot of the couple of the parameters (K_p, T_i).

D governor (derivative control)

The D governor derivate the errors:

$D_{out}(t) = Td \frac{de(t)}{dt}$ which is equivalent to $D_{out}(s) = Td \cdot s \cdot E(s)$

The transfer function is:

$$H_d(s) = \frac{I_{out}(s)}{E(s)} = Td \cdot s$$

Td is the derivative time constant. A larger Td tends to make the system a bit slower but reduces the over/undershoot. The derivative term is really sensitive to noise and if Td is too large, it can lead to instability.

T1, T2 parts (first, second order filter).

These filters can be added to a P or D governor, it is then named PT1 or DT1.

The transfer function of a T1 is:

$$H_f(s) = \frac{1}{(1 + \tau \cdot s)}$$

where τ is the time constant of the filter.

The governor of a system can be made of different governors. For example a common governor is a PID, which is made adding a P, a I and a D governor together.
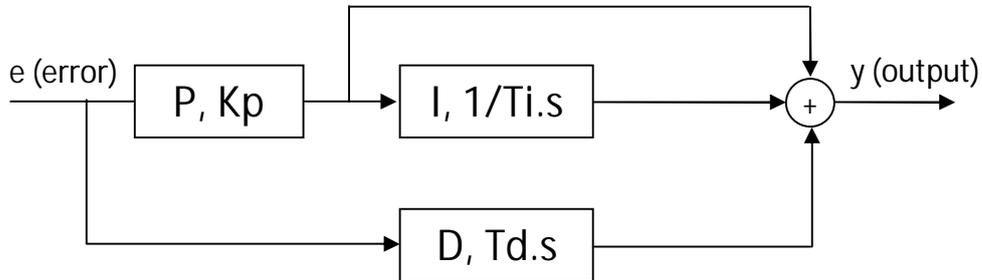
This PID architecture will be the one used in the simulation software.

To tune the governor, which means to adjust the gains and the time constants, several methods are available.

The manual method consists in tuning "manually" the parameters. This is not a mathematical method and it requires experience to get good results.

The Ziegler-Nichols method which consists in making the system oscillating only using the proportional gain and them the PID parameters can be calculating using the critical gain and the oscillation period.

Then software can be used to do automated PID tuning. Matlab/Simulink® has for example a toolbox called Control Toolbox for PID tuning.

Tuning a PID or any other controller depends on the requirements, if overshoots are allowed for example. In any case it will be done doing compromise, for example making the system slower to make it more stable.

<u>Initialization method:</u>
Governors, especially the integral part governors, can be initialized under certain condition to specified values. It is for example common to initialize the integral part to 0 with a change of set point to avoid under or overshoot.

<u>Anti Reset Windup Method (ARW Method):</u>
This method consists in freezing and/or limiting the I part of the governor in function of the governor output under certain conditions. Then in the case of a PID governor, just the P and D parts work while the I part is frozen.
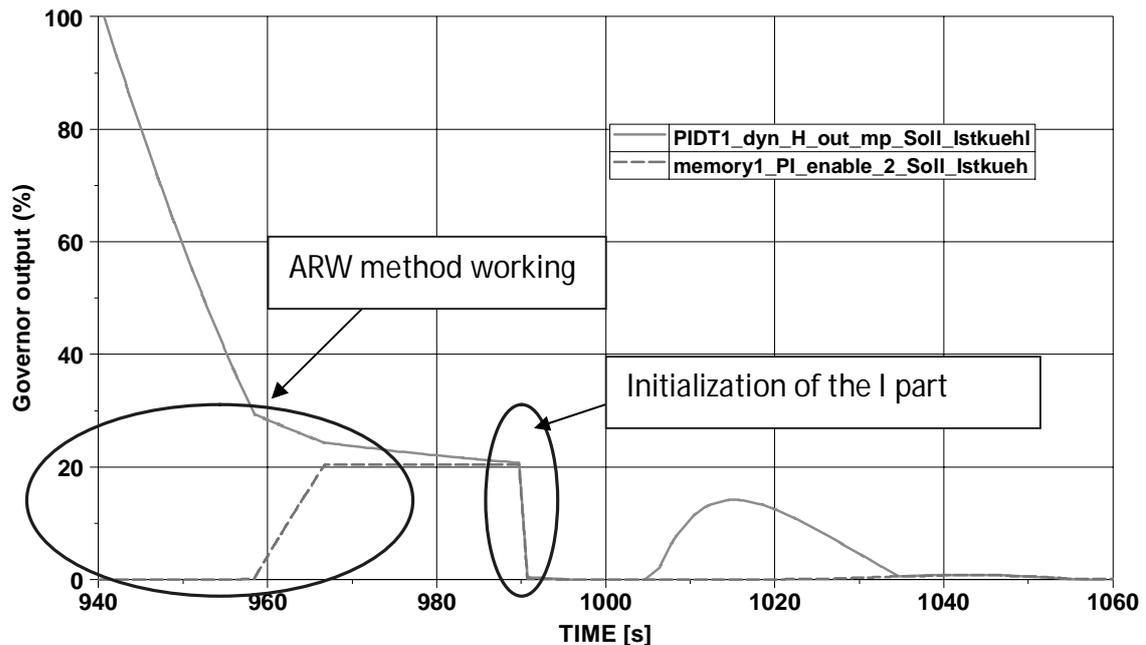
Figure 2: I part initialization and Anti Reset Windup Method

On the previous figure, the ARW method can be seen working. While the PID output (in green) is bigger than 30% the I part (in orange) is frozen. When the PID output becomes less than 30% the I part starts the work until a limit value. The I part is here limited to 20%. Then at around 990 seconds, the I part is initialized to zero. This leads to more stable response, using the P part to approach the set point and then use the I part to reach it.
The ARW method is really useful for systems with big dead times.

1.3. Introduction of the simulation model

The simulations have been made with the software ASCET® from ETAS®. The model is based on thermodynamic calculations and on assumptions made regarding previous experiments done at the test bench. The aim of this model is not to have a perfect match of the real system but to help in the development of the governing strategy.

In the simulation model, the temperature down-stream compressor, $TLnV$, the air intake temperature set point, $TLnWsoll$, and the air mass flow, MPUS, are depending on the engine speed and engine load. These variables will be calculated using 3D maps. These maps can be found in Appendix 2, 3 and 4.
Then to simulate the evolution of these variables, $TLnV$ and $MPUS$, first orders filters are used. The time constants differ depending on the variable and on the engine speed and engine load. Higher engine load and or speed will induce for example a higher air mass flow and so the heat transfers and the temperature changes will be faster.
Theses time constants have been calibrated, using measurements done on the test bench, to fit the best the reality.

The air temperature up-stream heat exchanger 1 needs to be simulated since there are losses between the compressor and the inlet of the heat exchanger. These losses are due to convection. To calculate $TLvW$ the two following equations are used:

$$P_{losses} = hS\big((TLnV - x) - T_{air}\big)$$

$\big(TLnV - x\big)$ is an estimation of the mean surface temperature of the pipe.
And

$$P_{losses} = \dot{m}_{l\bullet}\, cp_{air}\big(TLnV - TLvW\big)$$

Combining these two equations:

$$TLvW = TLnV - \frac{hS}{\dot{m}_{l\bullet}\, cp_{air}}\big((TLnV - x) - T_{air}\big) \qquad\qquad \textbf{(1.3)}$$

The values hS, x, and T_{air} are calibrated using an experiment. The results of the calculations are shown is the following graph for $hS = \textbf{2.8}\,W/K$, $x = \textbf{16°}C$ and $T_{air} = \textbf{20°}C$.
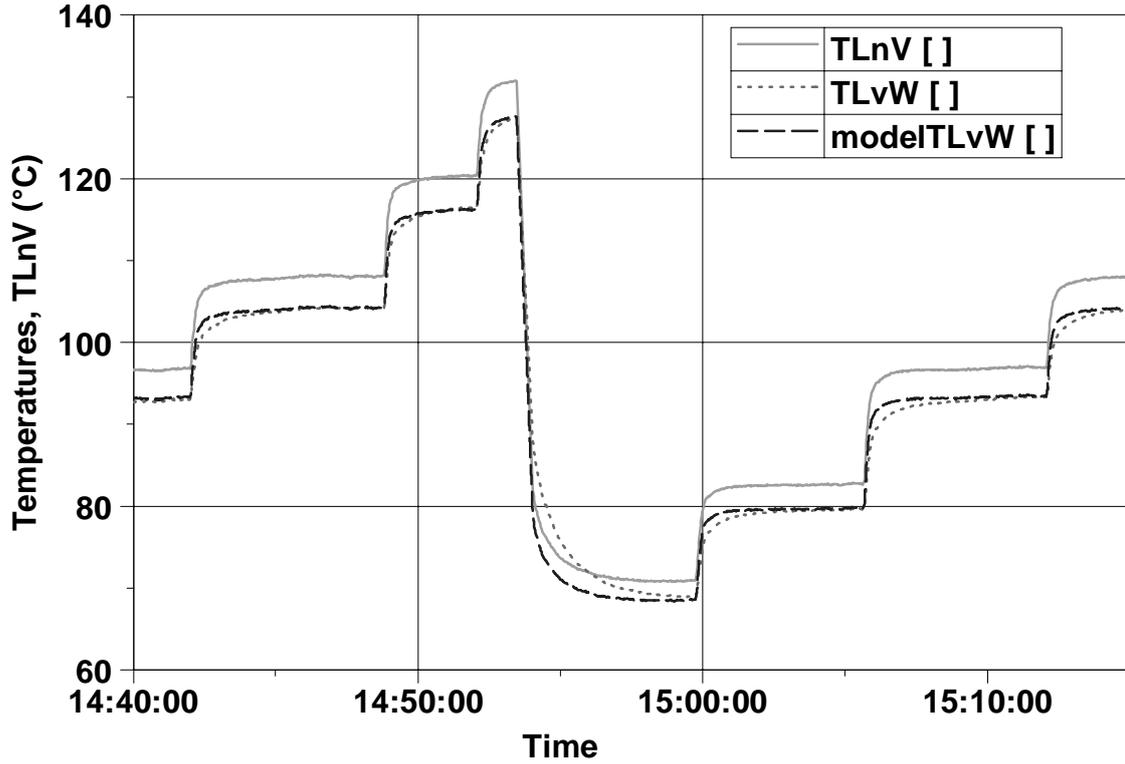
Figure 3: TLnV, TLvW and model of TLvW

The modeled $TLvW$ fits with the real temperature in stationary conditions. To make it fits better, a first order filter can be added in order to make it slower.

The equations governing the AirCon system are the following:

$$\dot{m}_l \cdot cp_{air}(TLvW - TLnW) = \dot{m}_{wg} \cdot cp_{wg}(TWnW - TWvW) \qquad (1.4)$$
$$\dot{m}_{wg} \cdot cp_{wg}(TWnW - TWvH) = \dot{m}_{wk} \cdot cp_{wk}(TWnQ2k - TWvQ2k) \qquad (1.5)$$
$$P_{heat} = m_{wg} \cdot cp_{wg}(TWvW - TWvH) \qquad (1.6)$$

Since this system has too many unknown, two temperatures are known, $TLvW$ and $TWvQ2k$, and not enough equations, assumptions and other ways to calculate temperatures have to be found.

First the mass flow of the mixture water-glycol is not known, but the mass of liquid in the auxiliary circuit has been measured, $m_{wg} = 9.05\ kg$. Then variation of the liquid temperature can be calculated using the equation of conservation of energy.

$$\dot{m}_l \cdot cp_{air}(TLvW - TLnW) + P_{heat} - \dot{m}_{wk} \cdot cp_{wk}(TWnQ2k - TWvQ2k)$$
$$= m_{wg} \cdot cp_{wg} \frac{dTWvW}{dt}$$

$$(1.7)$$

This can be simplified to:

$$\Delta TWvW = \frac{\Delta t}{m_{wg} \cdot cp_{wg}} \left(\dot{m}_l \cdot cp_{air}(TLvW - TLnW) + P_{heat} \right.$$
$$\left. - \dot{m}_{wk} \cdot cp_{wk}(TWnQ2k - TWvQ2k) \right)$$

With Δt the simulation time step, in this case it is 0.01 second. In the model all the temperatures in the auxiliary circuit are initialize to 20°C, room temperature in the test bench, and the input temperature of the cold water, $TWvQ2k$, is set to 5°C. The water temperature down-stream heat exchanger will be calculated using the heat exchange at the heat exchanger 1.
The power exchanged is:

$$P_1 = \dot{m}_l \cdot cp_{air}(TLvW - TLnW) \tag{1.8}$$

The heat exchange at each calculation step is:

$$Q_1 = \Delta t \cdot \dot{m}_l \cdot cp_{air}(TLvW - TLnW)$$

So:

$$m_{wg} \cdot cp_{wg}(TWnW - TWvW) = \Delta t \cdot \dot{m}_l \cdot cp_{air}(TLvW - TLnW)$$

$$TWnW = \frac{\Delta t}{m_{wg} \cdot cp_{wg}} \dot{m}_l \cdot cp_{air}(TLvW - TLnW) + TWvW \tag{1.9}$$

The temperature of the intake air, $TLnW$, is calculating using the water temperature down-stream heat exchanger 1, $TWnW$.

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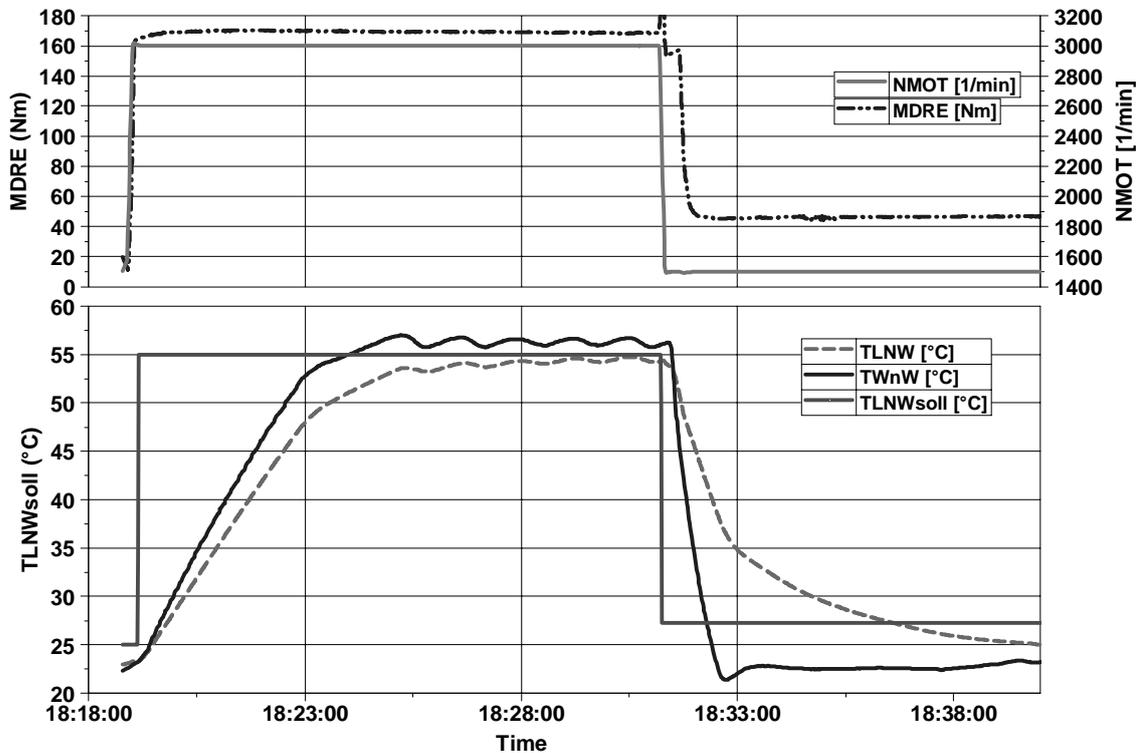
Figure 4: Intake air temperature behavior

9

On the graph, the intake air temperature, $TLnW$, has a behavior close to the water temperature down-stream heat exchanger, $TWnW$. In the model, $TLnW$ will be calculate using $TWnW$ and a first order filter. The time constant of the filter is calibrated using experiments to fit the best with reality.

The transfer function is:

$$\frac{TLnW(s)}{TWnW(s)} = \frac{1}{(1 + \tau.s)}$$

This assumption gives a very good estimation of the temperature $TLnW$ in transient conditions. Due to the transfer function, in steady state conditions both of the temperature, $TLnW$ and $TWnW$ will be equal.

In the previous equations, the heating power, P_{heat}, and the cold water mass flow, $\dot{m}_{wk}$ can be controlled. These values are directly proportional to the controller output from 0 to 100%. The heating power will vary from 0 to 6kW and the cold water mass flow from 0 and 35 l/min.

1.4. Motivation for improvement and development

Before being able to measure of an operating point, a set of measured values, temperatures, pressures... have to be stable. The intake air temperature was at the beginning of the project, the value with the longest stabilization time. The temperature $TLnW$ is considered stable when the interval set point +/-1°C is reached and when the standard deviation is less than 0.333. The standard deviation criterion is important for measured values that can vary very quickly like pressures. The main criterion is the temperature interval.

Reducing the stabilization time, makes the engine tests shorter and therefore cheaper. One hour of experiments on an engine test bench cost around 400 euro, so a lot of money can be saved by optimizing the stabilization times.

1.5. Content of the master thesis

The report will be divided in three main parts. Firstly, the current AirCon system will be described. The development of a new control strategy and simulations will then be explained and finally a comparison between different sets of experiment will illustrate the improvement of the system.

Secondly, a more advanced control strategy will be developed and simulated to be used in a near future with a new software which will be in charge of managing the future engine test benches.

And finally a complete new AirCon system will be discussed.

2. Improvement and development of the governing strategy of the AirCon system used with ADAPT®

2.1. Introduction and investigation of the ADAPT® system and of the previous control strategy

ADAPT® is a computer system handling in and output signals, measurement and actuators channels. It takes care of all the temperature, flow and pressure sensors and controls the valve opening to regulate the cold water flow and the electric heater.
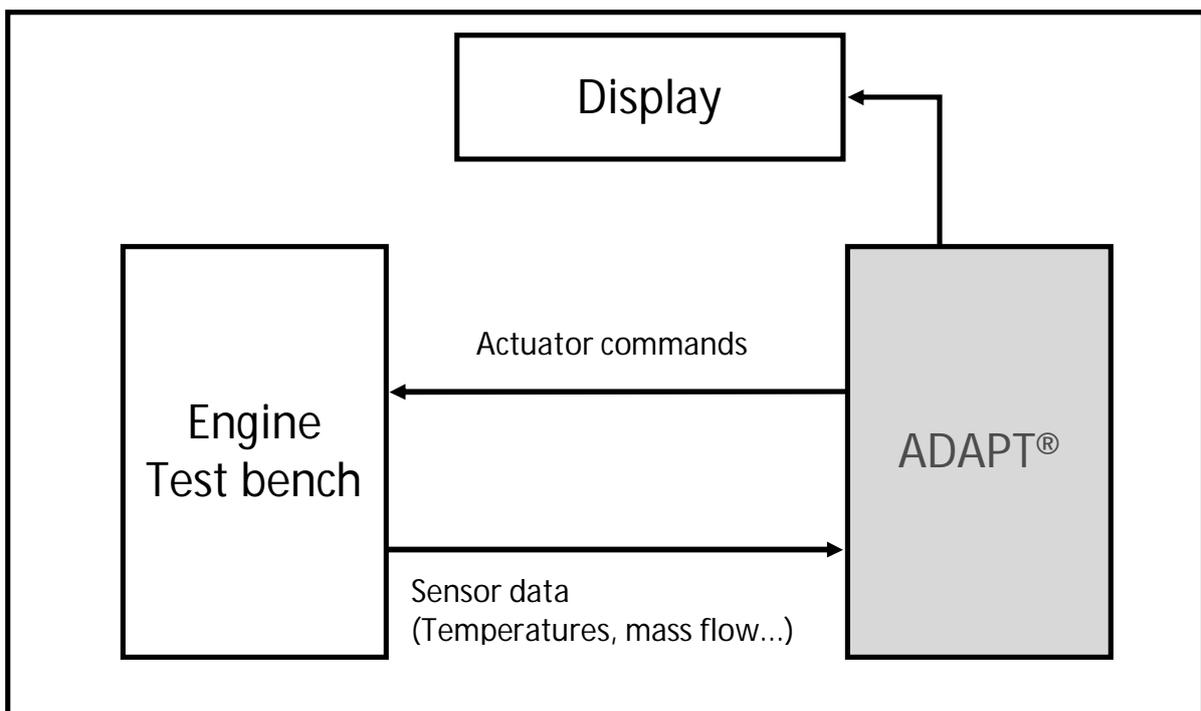
Figure 5: ADAPT® system description

ADAPT® displays the information through a window. This display window can be found in Appendix 5.

In the ADAPT® system only one PI governor is available and it is only able to do simple calculation (addition, multiplication) and basic loops (if endif, while...).

More advanced control strategy like the I part initialization or the ARW method, (describe page 6) cannot be implemented. Only a limitation of the I part can be implemented. Parameter like gains and time constants are fixed, they cannot be read from 3D maps for example.

Concerning the actuators, the heater and the valve, their behaviors have been identified with experiments.

2.1.1. Heating system
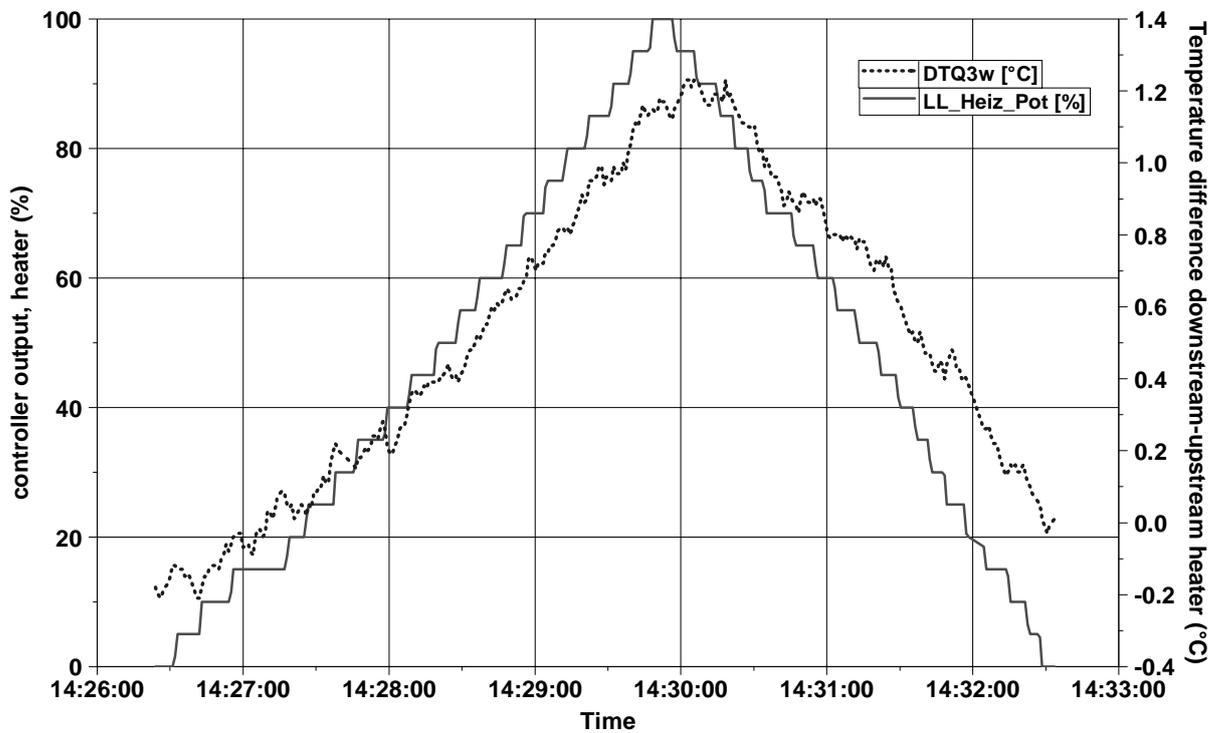
Figure 6: Heater signal and temperature difference in and out heater

On this graph, the response of the heater is nearly linear. The temperature difference being directly proportional the power, $P_{heat} = \dot{m}_{wg}.cp_{wg}(TWvW - TWvH)$ and the water mass flow being constant, it can be assumed that the power is directly proportional to the controller output and fully transferred to the fluid. In the AirCon system used for the test, the heater has a maximum power of 6 kW.
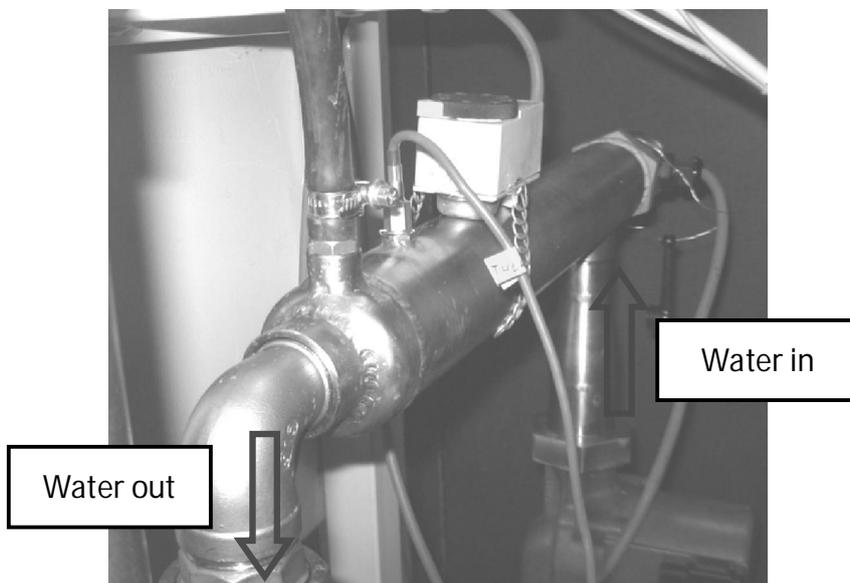
Figure 7: Electric heater

2.1.2. Cooling system
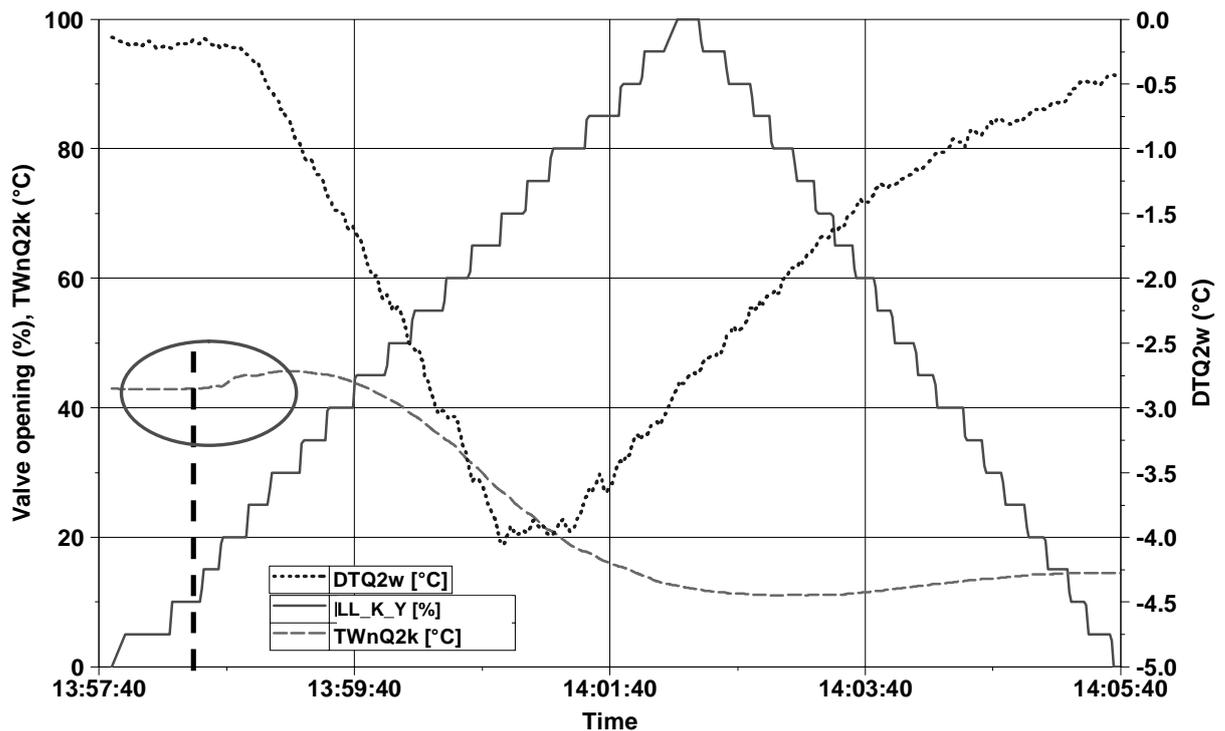
Figure 8: Cooling system identification

For the cooling system, it is less linear. The effect of the valve opening is not visible until it reaches 10% (the valve opening and the controller output can be assumed equal since the valve is programmed with a linear response, see Appendix 6). In the valve documentation, it is recommended to work above 10% opening.

It has been estimated that with the valve fully opened, the cooling power can reach up to 80 kW. In stationary conditions, the cooling power needed is much smaller than these 80 kW.

For example the cooling power needed at 150Nm, 3000 rpm is:

$$\dot{m}_l \cdot cp_{air}(TLnV - TLnW) = \frac{210}{3600} \cdot 1005 \cdot (110 - 50) = 3.5\ kW$$

This operating point would require a really small valve opening, around **3.5/80 ≈ 5%**. This has to be avoided in order to have a stable system.

A way the increase the valve opening keeping the cold water mass flow constant is to use a flow limiter up-stream the valve.
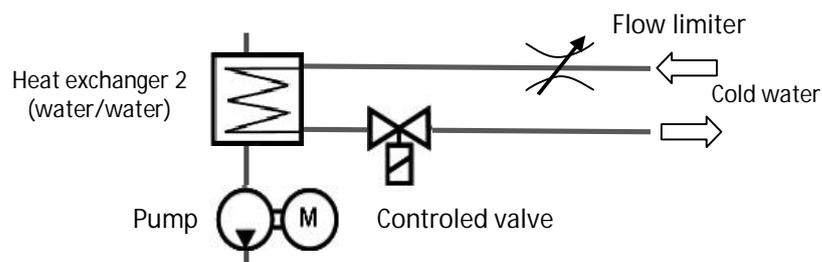
Figure 9: system with mass flow limiter

A compromise has to be found between shifting to a higher operation point for the valve, (larger opening), to make the system more stable and easier to control, and to have a reduced cold water mass flow which induced a slower system in the phases where all the cooling power is needed.

To find this compromise, a series of test have been done. Each test starts from a stationary point, 3000rpm and 170 Nm with an intake temperature, $TLnW$, equal to 55°C, and then the valve is fully open until $TLnW$ reaches 30°C.
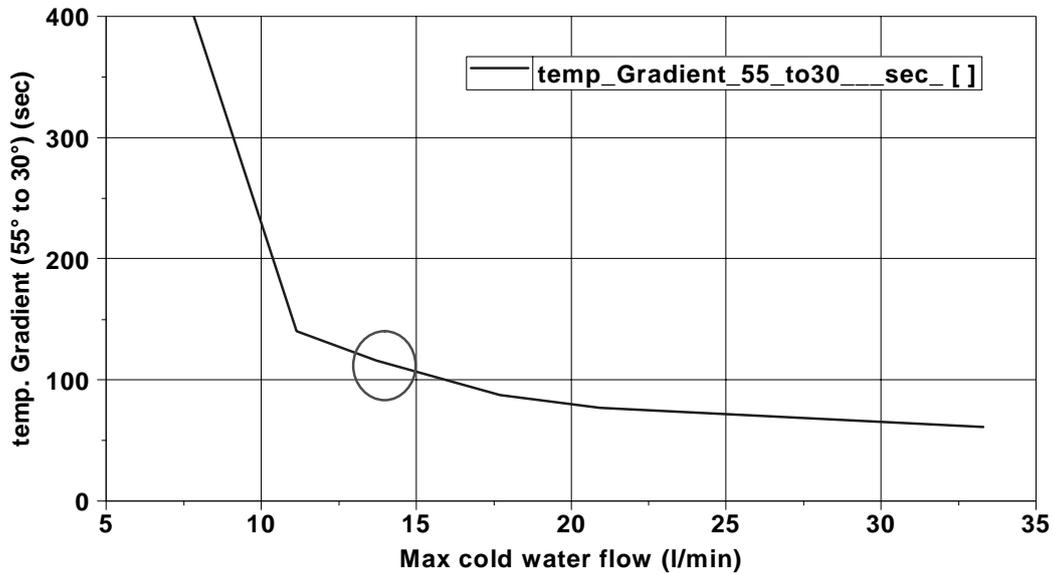
Figure 10: Temperature gradient vs. water mass flow

The temperature gradient from 55°C to 30°C has an exponential behavior. The cold water mass flow is reduced to a value of 13 l/min. This reduction of the mass flow will allow the valve to work at higher operating point.
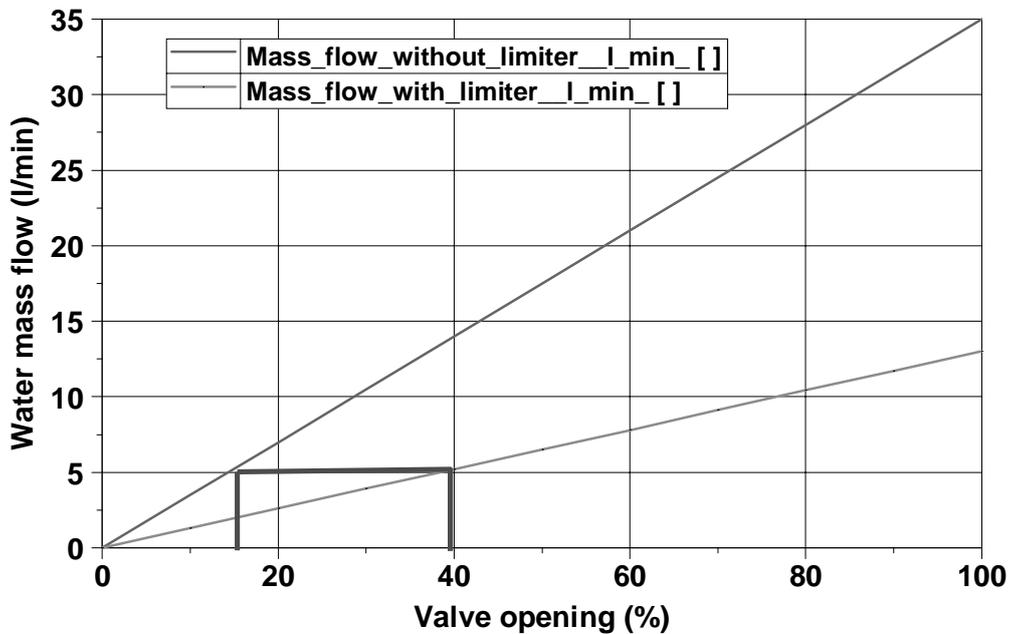
Figure 11: valve opening vs. water mass flow

With the mass flow limiter, the same cold water flow is reached with a larger opening of the valve which makes the system more stable and easier to control. For example, of the graph, 5 l/min is reach with an opening of 39% instead of 17%.
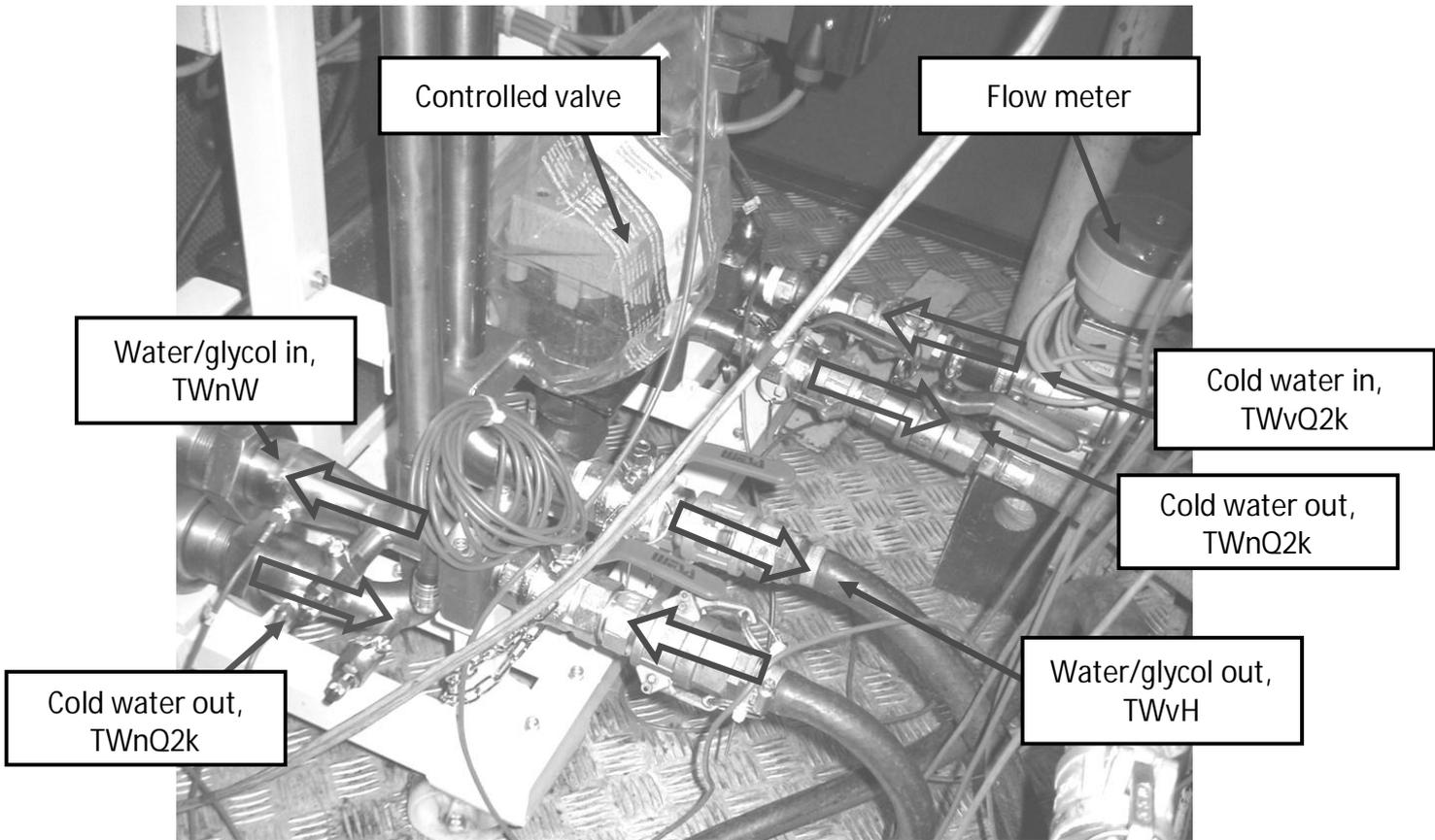
Figure 12: Heat exchanger 2 (water/water)
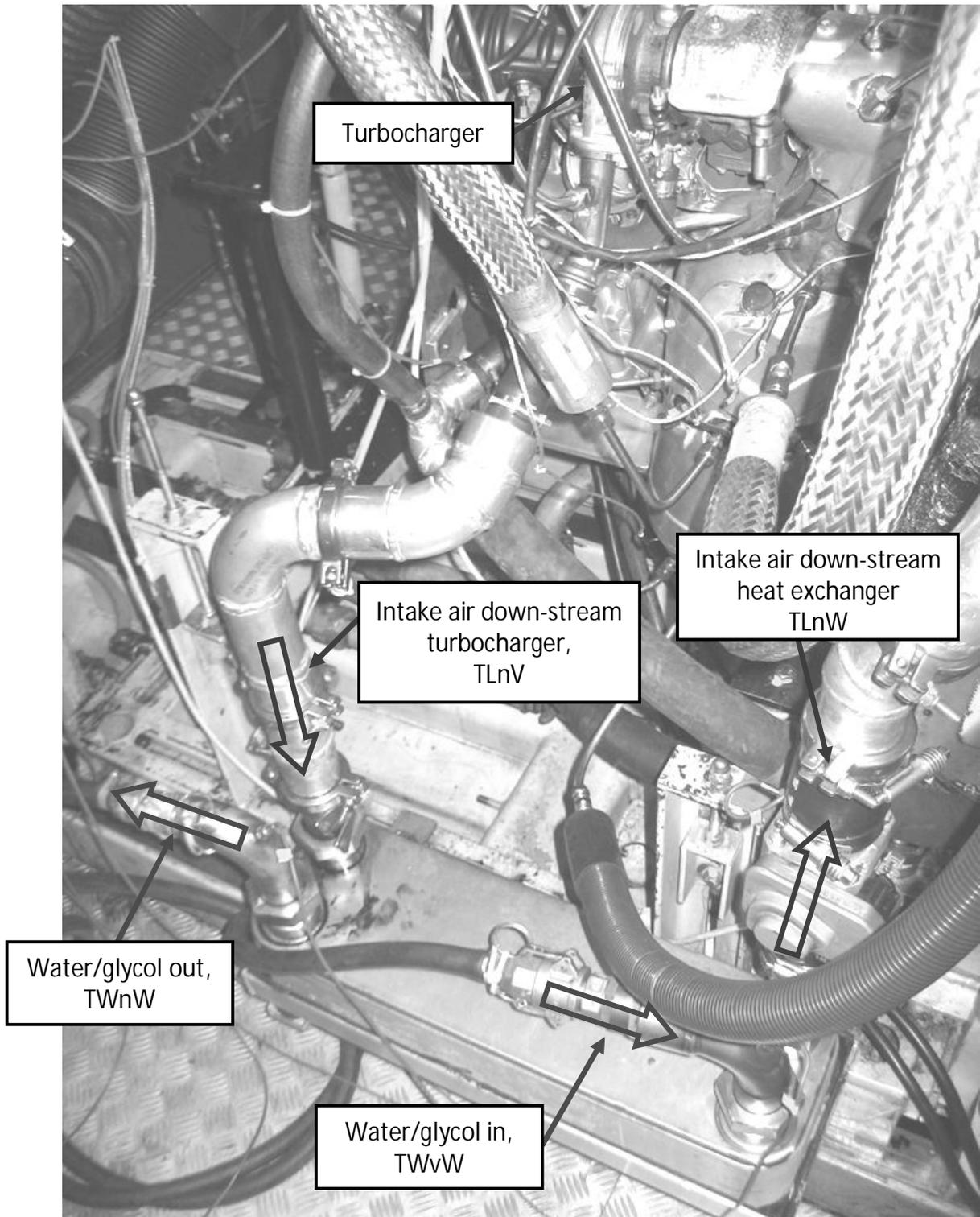
Figure 13: Heat exchanger 1 (air/water)

2.1.3. System dead times
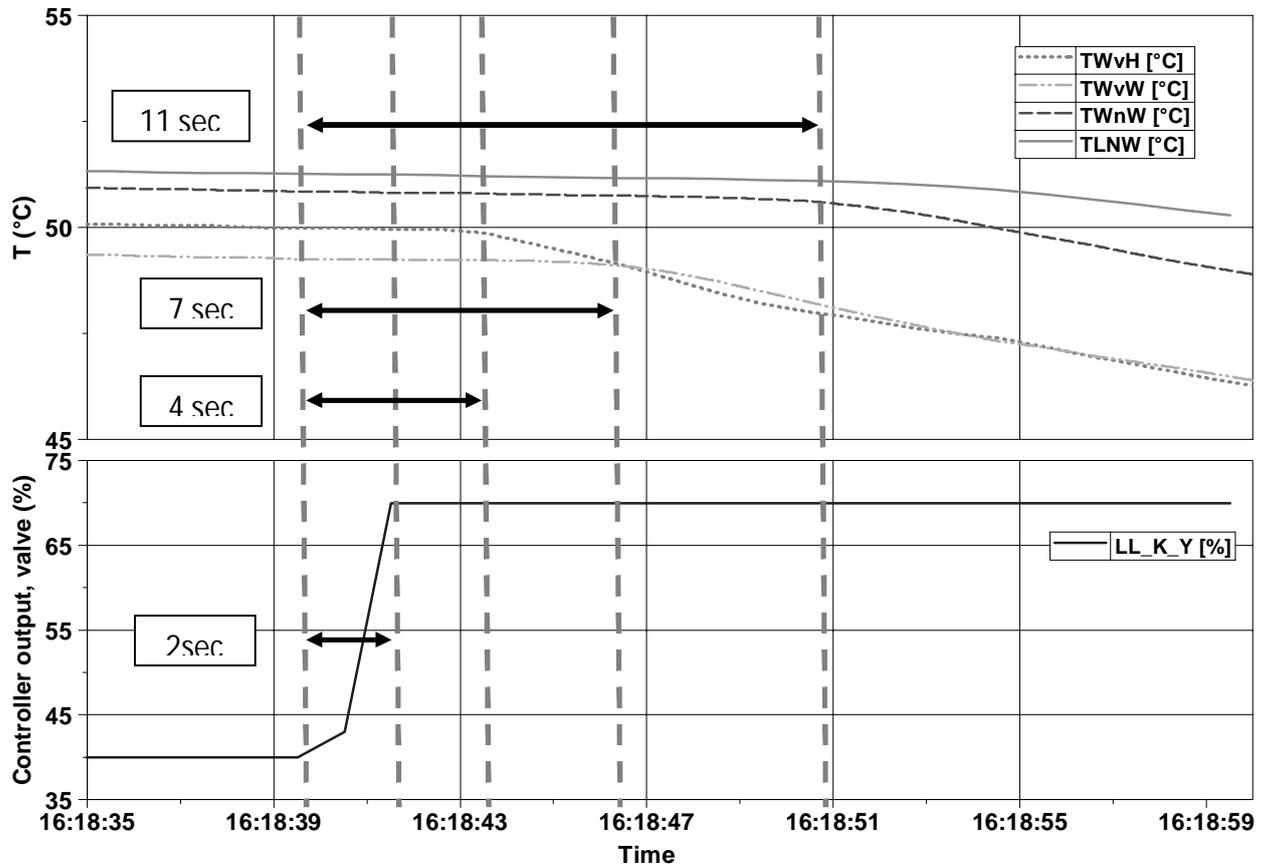
Figure 14: System dead times, Valve opening step 40 to 70%

Table 1: System dead times

label	Valve	TWvH	TWvW	TWnW	TLnW
Dead time (s)	2	4	7	11	11

The system dead times are independent of the experiment conditions.
Long dead times make the system hard to control since the effect of the controller is seen around 11 seconds after on $TLnW$. Because of the dead times the I part has to be limited. If the I part is not limited it will increase (or decrease) until the set point is reached. In this case the I part would reach a high value during the dead time.

2.1.4. Original control strategy

The original control strategy done with ADAPT® is using and simple PID. The governor works in this case with the air temperature deviation: $TLnW - TLnWsoll$.

For each change of engine operating point the set point follows a ramp from its original value to the desired value.
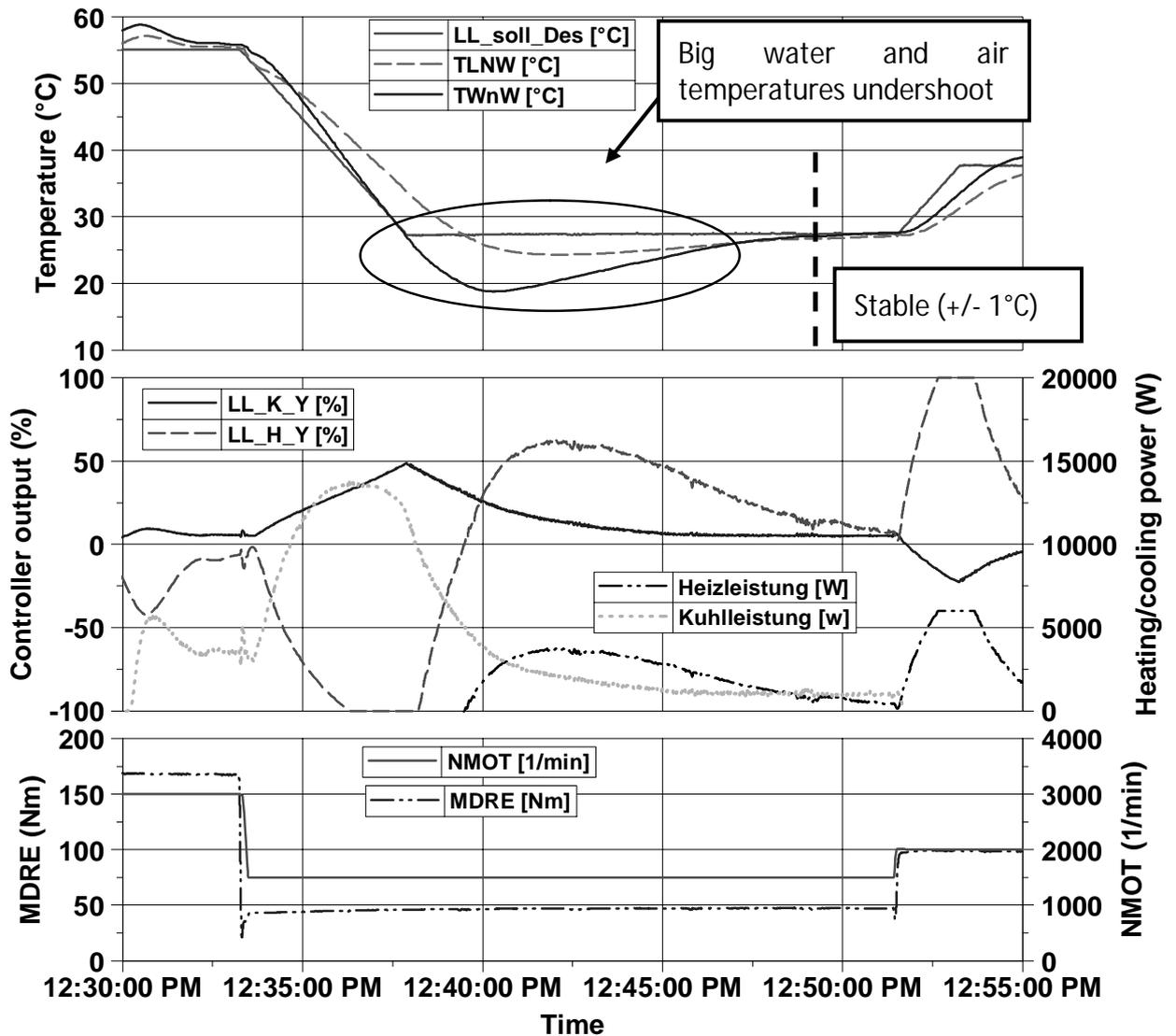
Figure 15: Experiment results with the original governor

The ramp, created to avoid big over/undershoots, is not sufficient. For the step down (around 12:40), the intake air temperature, in magenta on the first graph, $TLnW$, does a big undershoot.

On the second graph, the heater command, in red, and the valve opening, in blue, are both bigger than 0% which means that the system is heating and cooling in the same time in stationary conditions.

Looking at the power exchanged at the heat exchanger 1, see equation1.8, $TLnV$ being always bigger than $TLnW$, the system only need to support cooling power in stationary conditions. The fact of heating and cooling in the same represent a big waste of energy and should absolutely be avoided.

The parameters of the PID have been "optimized" but this is not sufficient to get the system to work properly. Bigger changes in the control strategy are required.

Table 2: PID parameters, original strategy

	Proportional gain, Kp	Integral time constant, Ti (s)	Derivative time constant, Td (s)
Original	4	0.06	0
Optimized	4	0.1	2.5
Optimized with better safety margin	2	0.1	2.5

2.2. Development and simulation of the new strategy

The aim of this new control strategy is to improve the response time and stability of the system and correct the issues of the old strategy.

2.2.1. Development of the new strategy

First a new control strategy has to be created regarding the possibilities of ADAPT®.
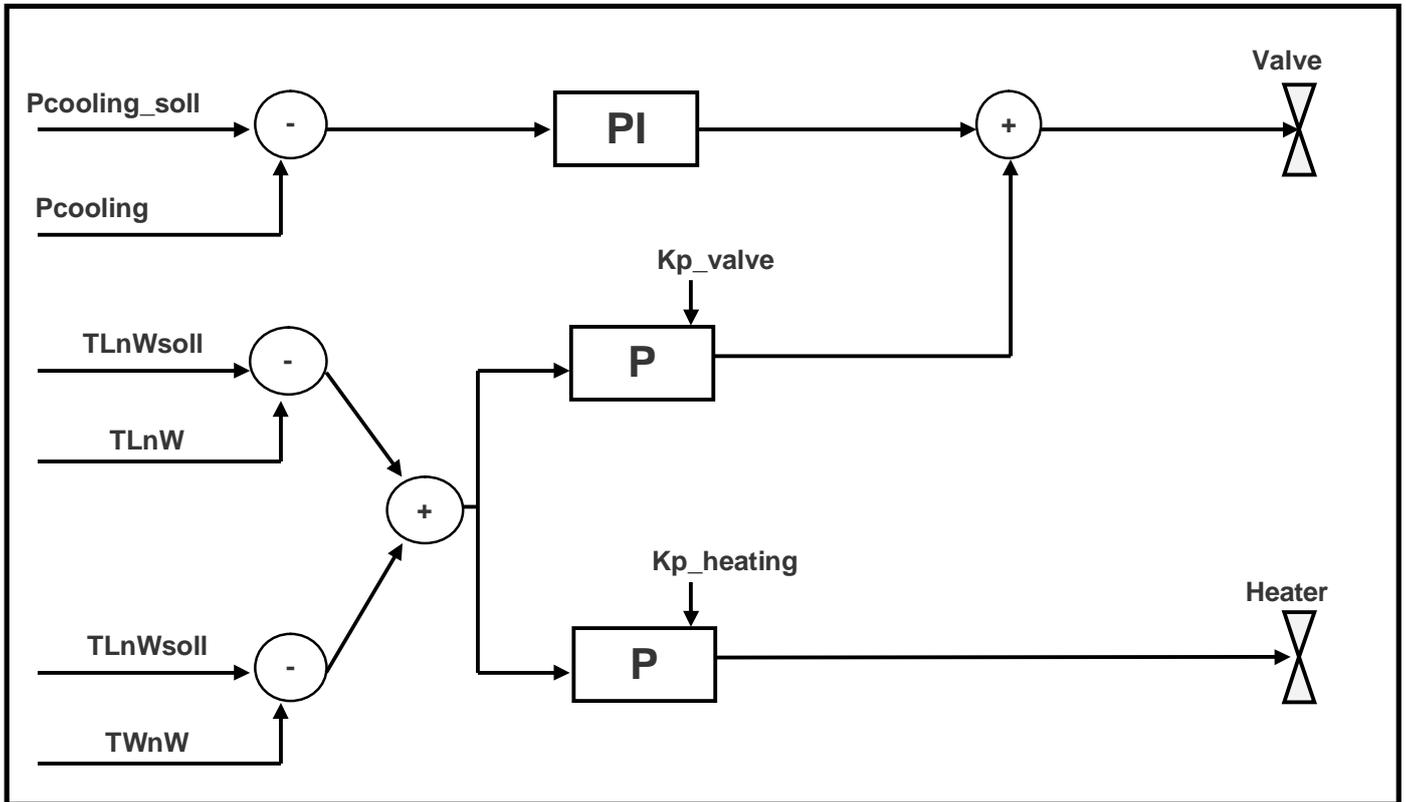
Figure 16: new ADAPT® control strategy

In this strategy the PI controller works with the cooling power deviation. The cooling power set point is calculated with the temperature set point:

$$P_{cooling\_soll} = \dot{m}_l . cp_{air}(TLnVsoll - TlnWsoll) \qquad (2.1)$$

And the actual power is calculating using the actual values of the temperature:

$$P_{cooling} = \dot{m}_l . cp_{air}(TLnV - TlnW) \qquad (2.2)$$

The P governors, called dynamics offsets, are used for the heater and in addition to the PI governor for the valve.

These P governors work with a different deviation. They use a coupled deviation, comparing the air and water temperature, $TLnW$ and $TWnW$, to the intake air temperature set point, $TLnWsoll$.

Couple deviation:

$$(TLnWsoll - TlnW) + (TWnW - TlnWsoll)$$

This coupled deviation is used to make the temperature $TWnW$ over/undershoot the air temperature set point to make the system work faster.
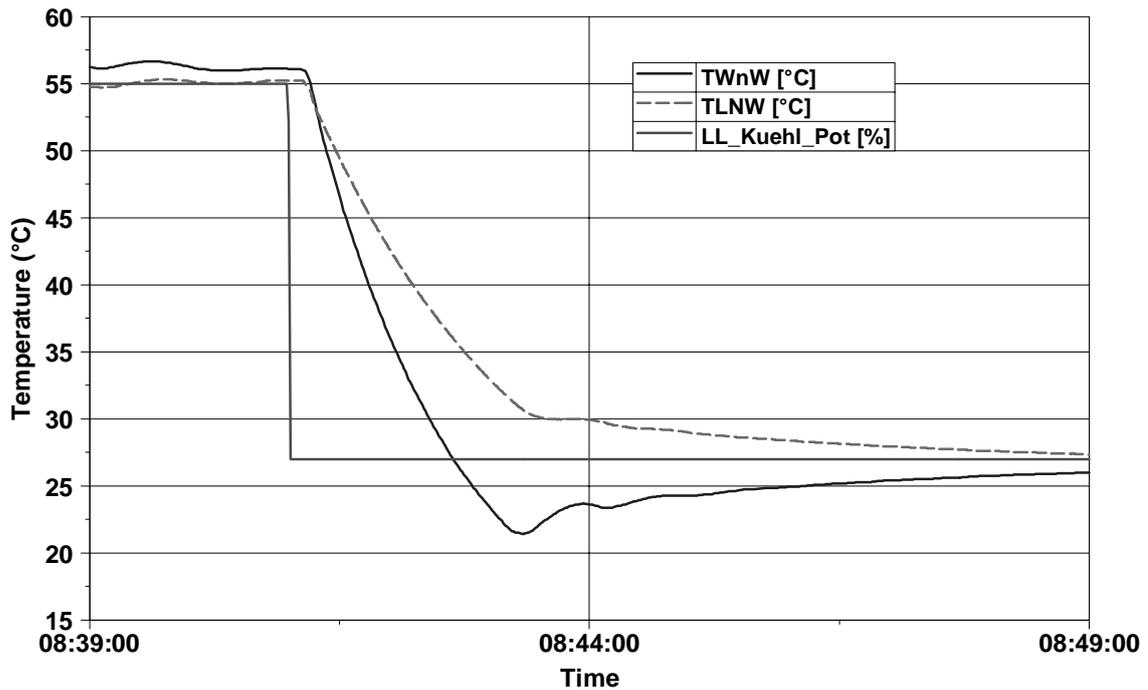
Figure 17: Step down, TLnW new strategy

On this graph that the water temperature $TWnW$ undershoot, in blue, helps the air temperature TLnW, in magenta, to reach the set point $TLnW soll$, in red, faster. The difference between $TWnW$ and $TLnW$ is important to make the system fast. On this graph, the two different phases when the P governor works, fast approach, and when the PI governor works, slower approach can be seen. The P governor gains are reduced when the $TLnW$ gets close to the set point to avoid oscillations and over/undershoot of the air temperature.

2.2.2. Simulation and precalibration of the system with ASCET®

The simulation model is using the equations describe in chapter 1.2.2 to calculate the temperatures and the control strategy is describe on figure 15.
For the controller, PI governor is used with fixed proportional gain and integral time constant, and the I part is limited with fixed values. The two P governors are used with variable proportional gain, one for the controlled valve and one for the heater. In this case the gain depends on the power deviation which allows to reduce the influence of this governor when temperature gets close to the set point to let the PI governors work alone.

The PI governor for the valve uses the following values:

Table 3: Pi parameters

	Values
K_p	0.05
T_i (s)	7
I part max values (%)	20
I part min value (%)	0

The proportional governors use the following values:

Table 4: Proportional gain for the heater

Deviation	-4	0	1	4
Gain value	0	0	3	10

The heater is not active when the deviation is higher than 0. This is made to avoid heating and cooling simultaneously.

Table 5: Proportional gain for the valve

Deviation	-2	0	2
Gain value	-50	0	-50

The gain with zero deviation is null so the PI governor can work by its own.
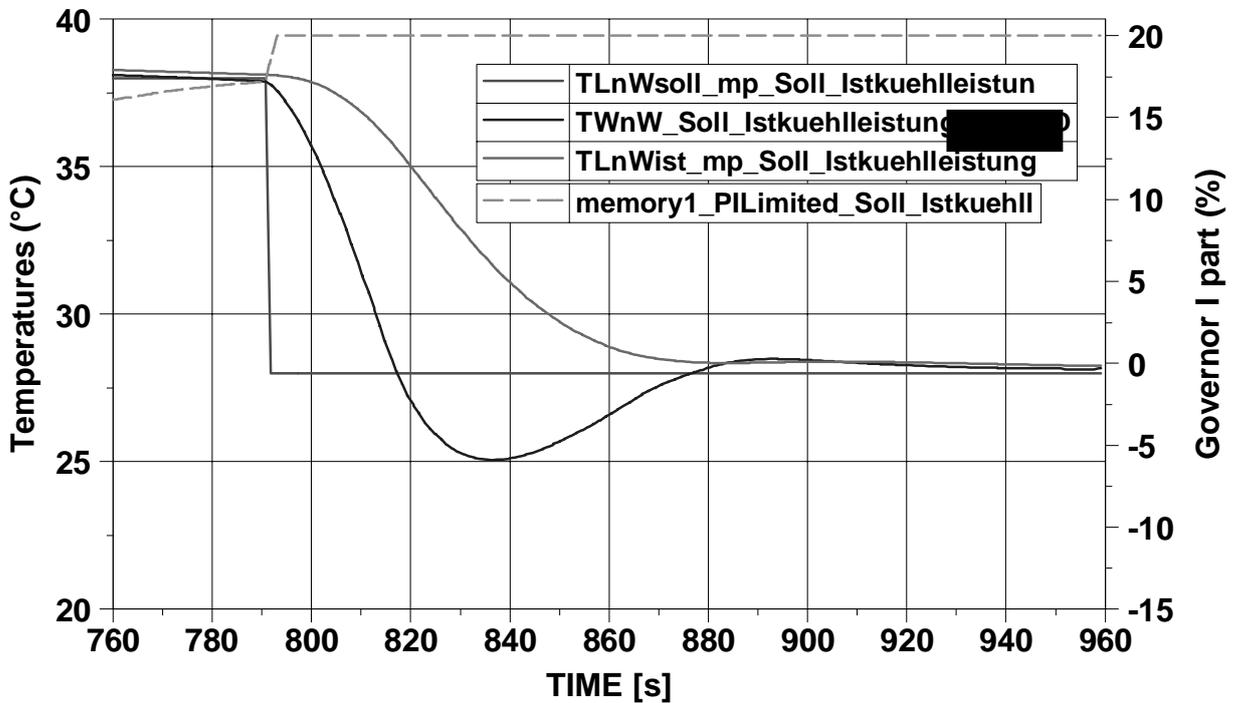
Figure 18: First ASCET® model, simulation results step down

The stationary error is due to the I part (dotted green). On the graph, the I part reaches the limit of 20%. With not enough I part the governor cannot reduce the error.

The high limit is useful to avoid undershoot, if the I part increases too much then the controller output gets too high and there is too much cooling energy which will induce an undershoot. In this case the ARW method is useful.
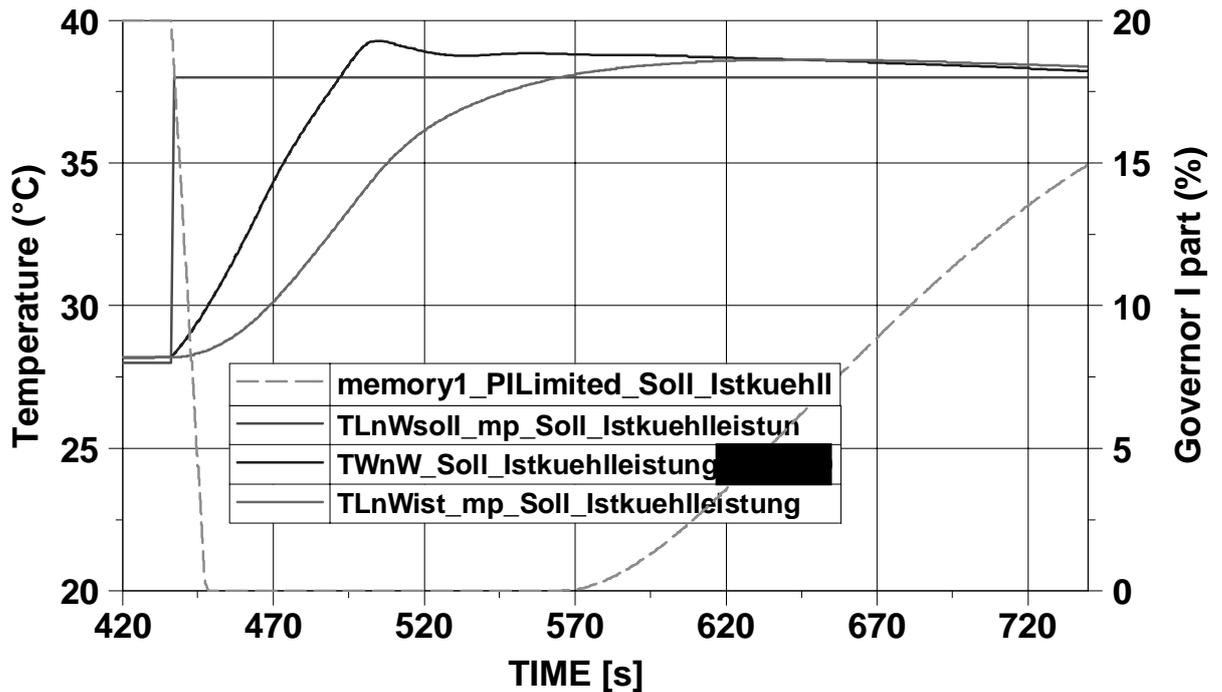
Figure 19: First ASCET® model, simulation results step up

The overshoot is also due to the I part. When the set point changes, the I part drop to its low limit. When the air temperature overshoots, the I part starts to work. The set point is then reached because of the effect to the I part.

The I part limitation is useful to reduce the time of the overshoot. Without the limitation it would have dropped and it would have taken a longer time before it get positive again.

These results show that the system is now much more stable and does not need a ramp for the set point anymore. This model shows that this new strategy would be much more efficient than the old one. Overshoot is problem when a step up is done but this overshoot is less than 1°C so the temperature can be considered stable. With the step down, the system has a stationary error, with parameters adjusted to make the error smaller than 1°C. Tests with the real system must be done in order to validate the control strategy and to set correctly all the parameters.

2.3. Experiments and evolution of the new strategy

2.3.1. First set of experiments

The set of experiment has been made with the control strategy described in chapter 2.2.1 and 2.2.2. These investigations have been made in "manual mode". The engine speed and torque and the intake air set point are set manually and the system controls automatically the heater and the valve in order to reach the temperature set point.

The investigations are done with the following PI parameters:

$$K_p = 0.1$$

$$K_i = \frac{1}{T_i} = 0.005 \ s^{-1} \ (T_i = 200 \ s)$$

During these investigations, the following operating points will be used.

Table 6: Operating points for measurements

Point number	Engine speed (rpm)	Engine torque (N.m) (Pedal position (%))	Air intake temperature set point (°C)
1	3000	170 (43)	55
2	1500	45 (21)	27
3	2000	100 (32)	38
4	800	"0" (idle) (0)	20
5	1400	100 (30)	38
6	2500	155 (40)	50
7	3500	200 (80)	55
8	1000	37 (15)	25

The tests will consist in doing steps from a point to another and wait until the intake air temperature, $TLnW$, is stable.
Each step is measured in a different measurement point, MP.

Table 7: Measurement points

Measurement Point	Initial operating point	Final operating point
First series of test done with the new control strategy		
552 MP1	1	2
MP2	2	3
MP3	3	4
MP4	4	5
MP5	5	6
MP6	6	7
MP7	7	6
MP8	6	8
Second series of test done with the old control strategy		
MP10	1	2
MP11	2	3
MP12	3	4
MP13	4	5
MP14	5	6
MP15	6	7
MP15 (done in the same MP)	7	6
MP16	6	8

With these two sets of tests, the two strategies can be compared.
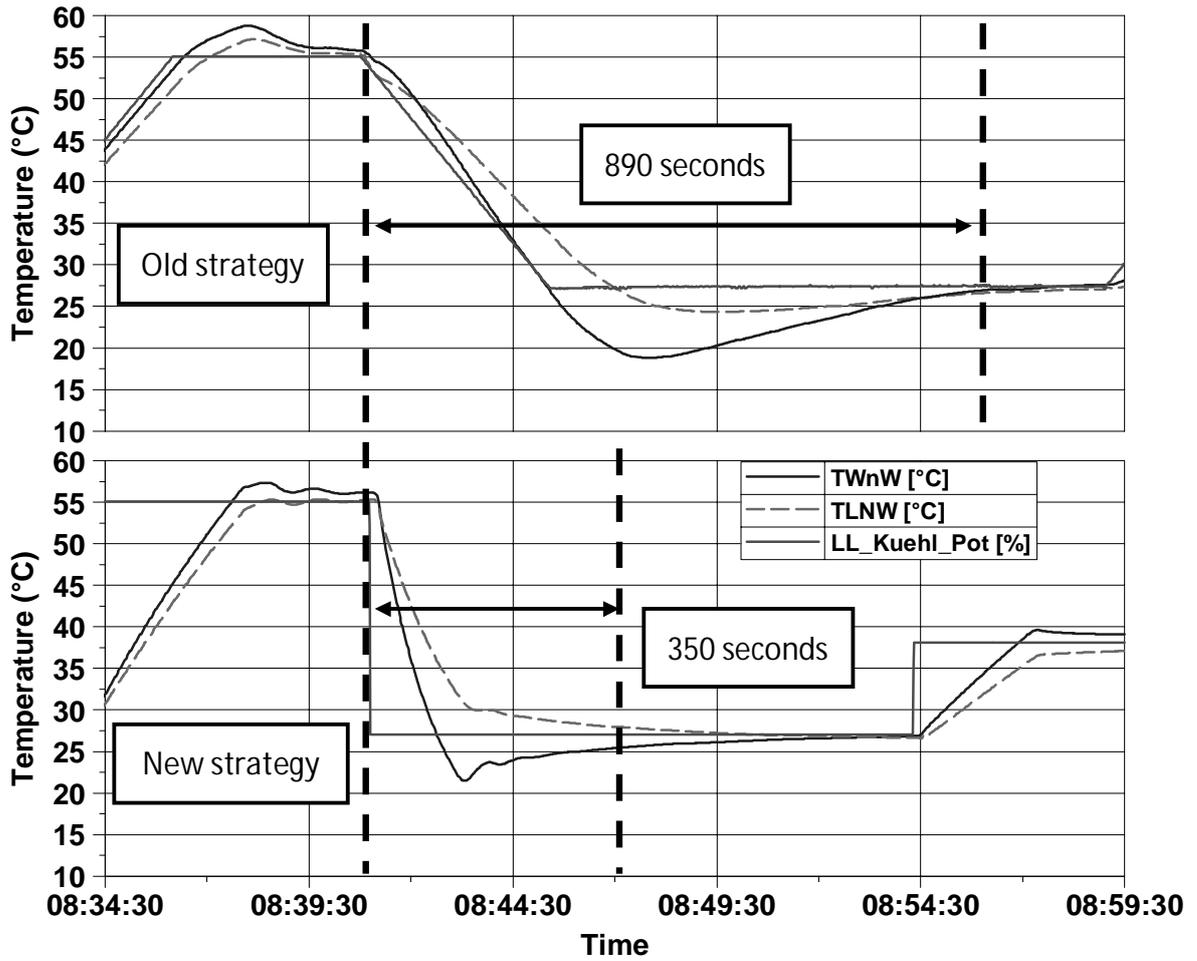
Figure 20: Comparison old/new strategy, step down (MP10-MP1)

This first graph shows that the new strategy avoids a big undershoot of the intake air temperature, $TLnW$. The system is much faster, 350 seconds instead of 890 seconds.
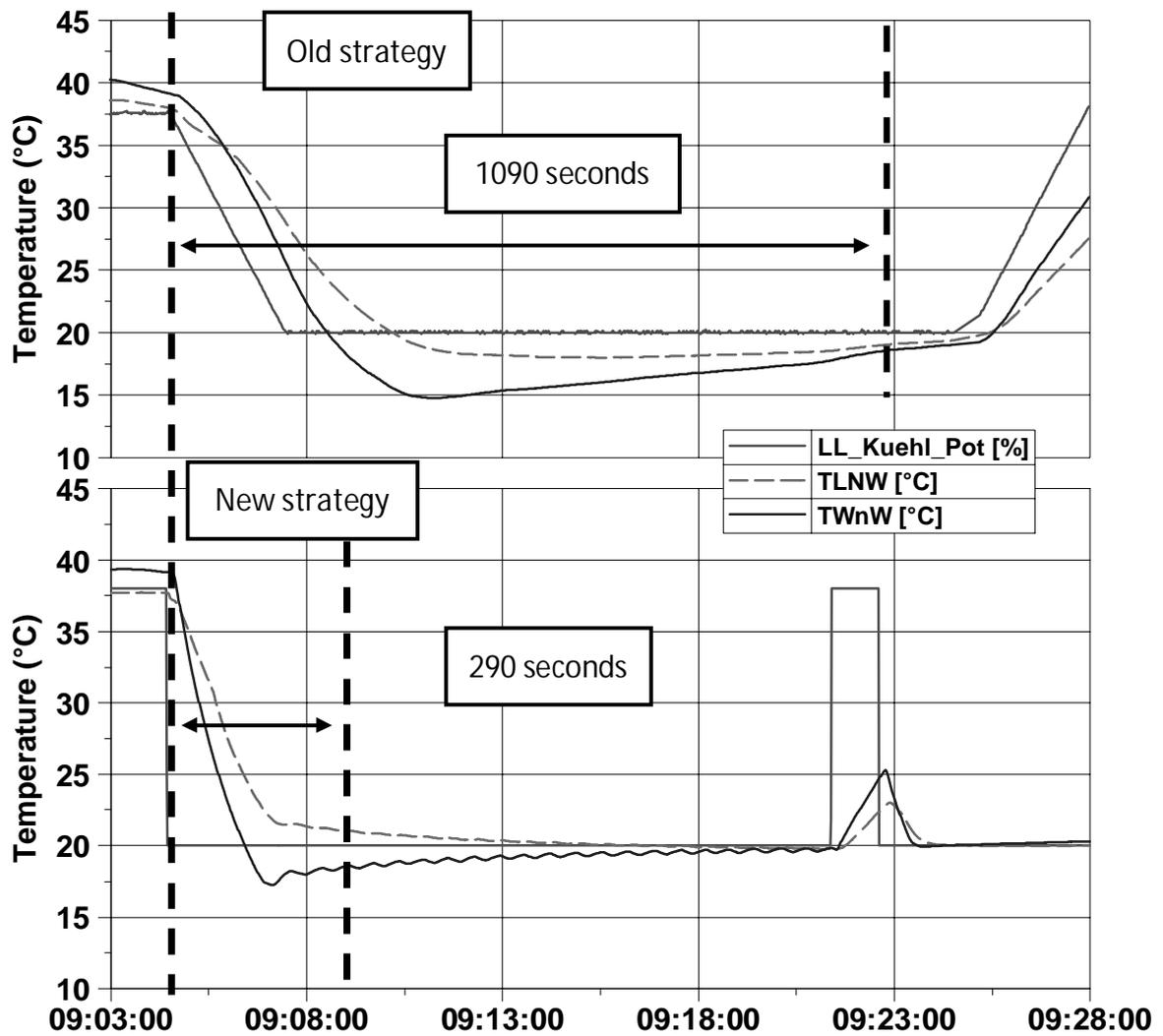
Figure 21: Comparison old/new strategy, step down (MP12-MP3)

The new strategy in much faster and has no undershoot of the air temperature, $TLnW$. Here the system in stable in 290 seconds instead of 1090 seconds with the old strategy.
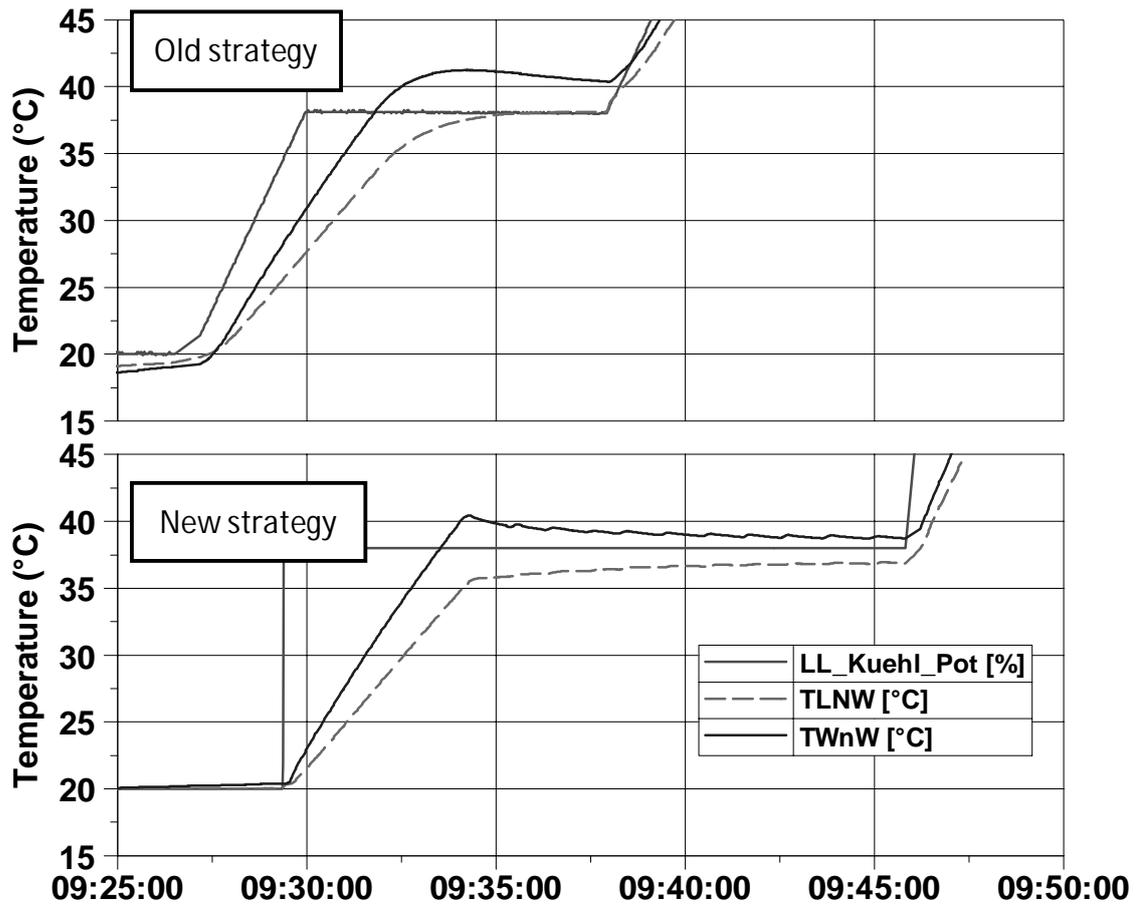
Figure 22: Comparison old/new strategy, step up (MP14-MP5)

In this case, the new strategy does not fit the requirements. The set point is not reached with the new strategy or would take too much time. In this case the old strategy works better. This problem is mainly due to the fact that the heater is controlled only with a proportional governor with a gain that decreases when the air temperature gets close to the set point. To solve this problem the heater has to be controlled also with the PI governor.

The improvement realized regarding response time can be seen in <u>Appendix 7</u>. Without considering the MP5, the stabilization times are reduced by 40%.

2.3.2. Modifications of the new strategy and second set of experiments

As said in the previous paragraph, the PI governor is now also connected to the heater.
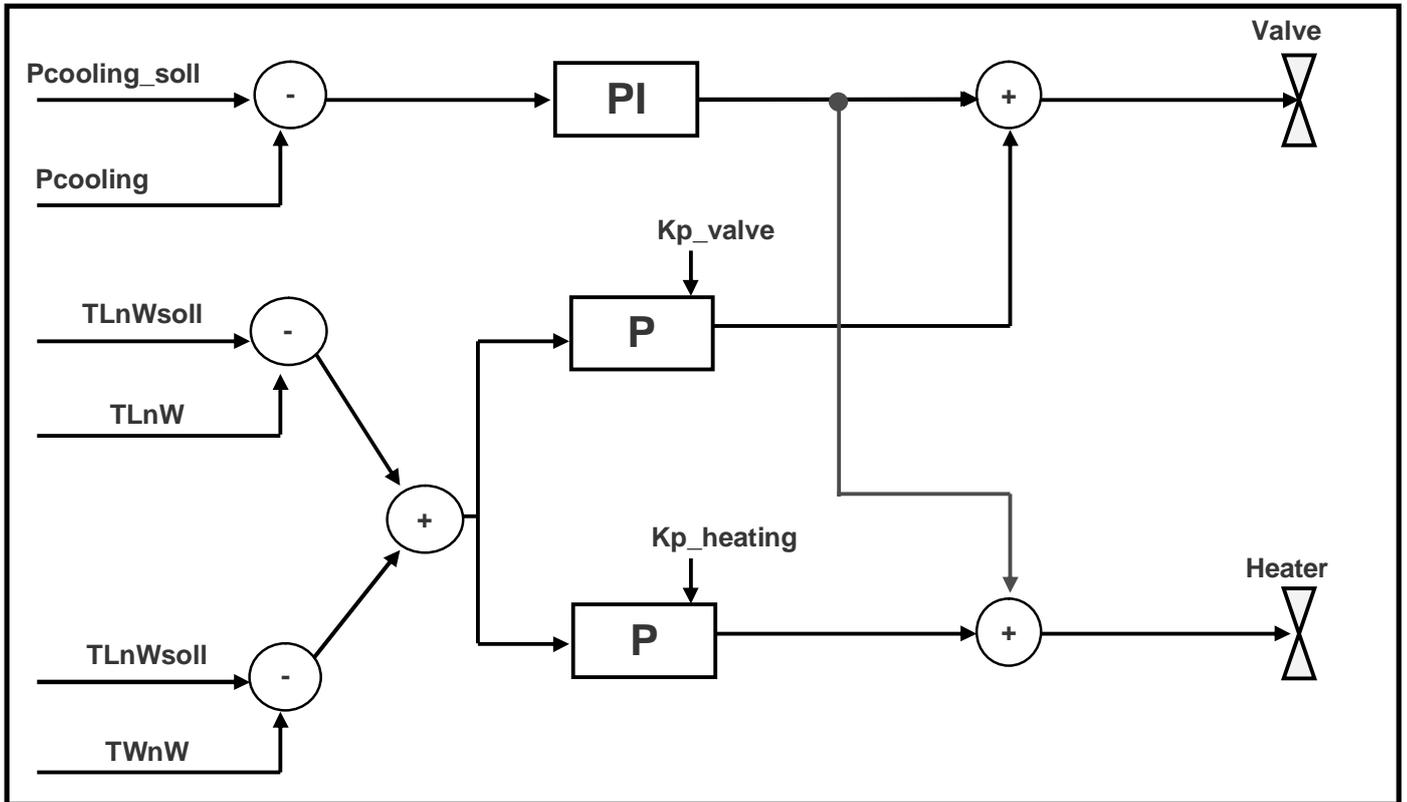
Figure 23: Modification of the new control strategy

This model has also been implemented in ASCET® in order to be simulated. Now the same PI governor controls the heater and the valve and the gains and time constants have been adapted to the new governor. The heater works with a negative output of the PI governor and the valve works with a positive output.
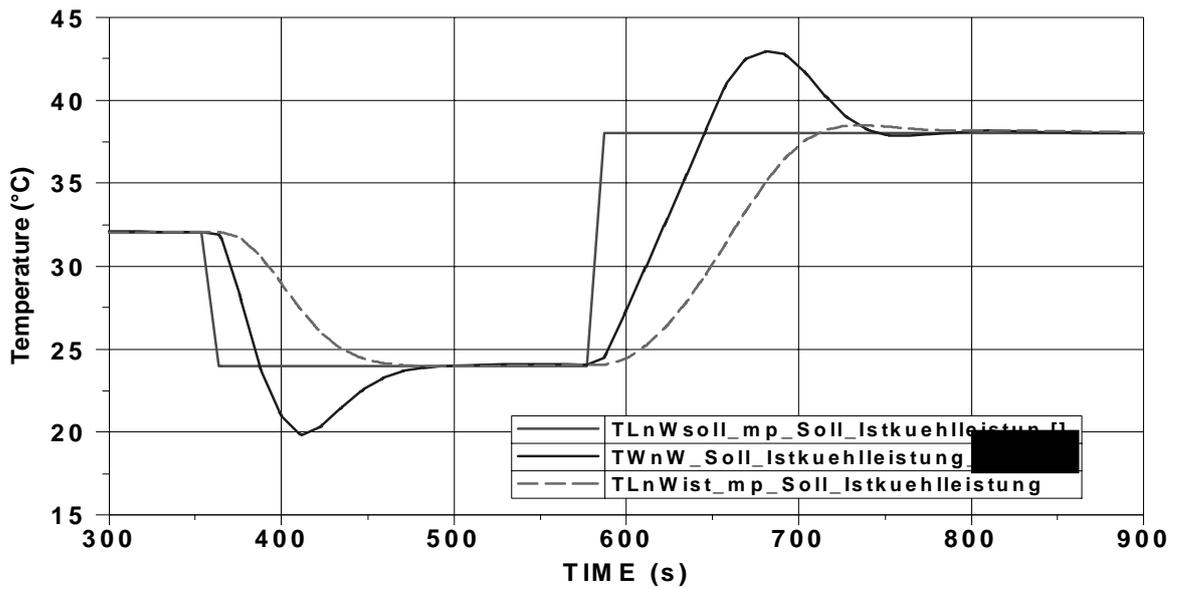
Figure 24: Simulation results of the new strategy improved

With the modifications the model works better in both conditions, step up and down.

A first series of test with the real system have been made in order to figure out which set of parameters will fit the requirements. The different parameters to modify are the gain and integral time constant of the PI governor, the I part limitations.

The fact that the PI governor work for both heater and valve changes the way the system reacts. Having the same governor parameters to control two different systems, makes calibration more cumbersome. Compromises have to be made in order to get satisfying results.

Figure 25 show the result of a step from the operating point 2 to 1 (cf. table 6).
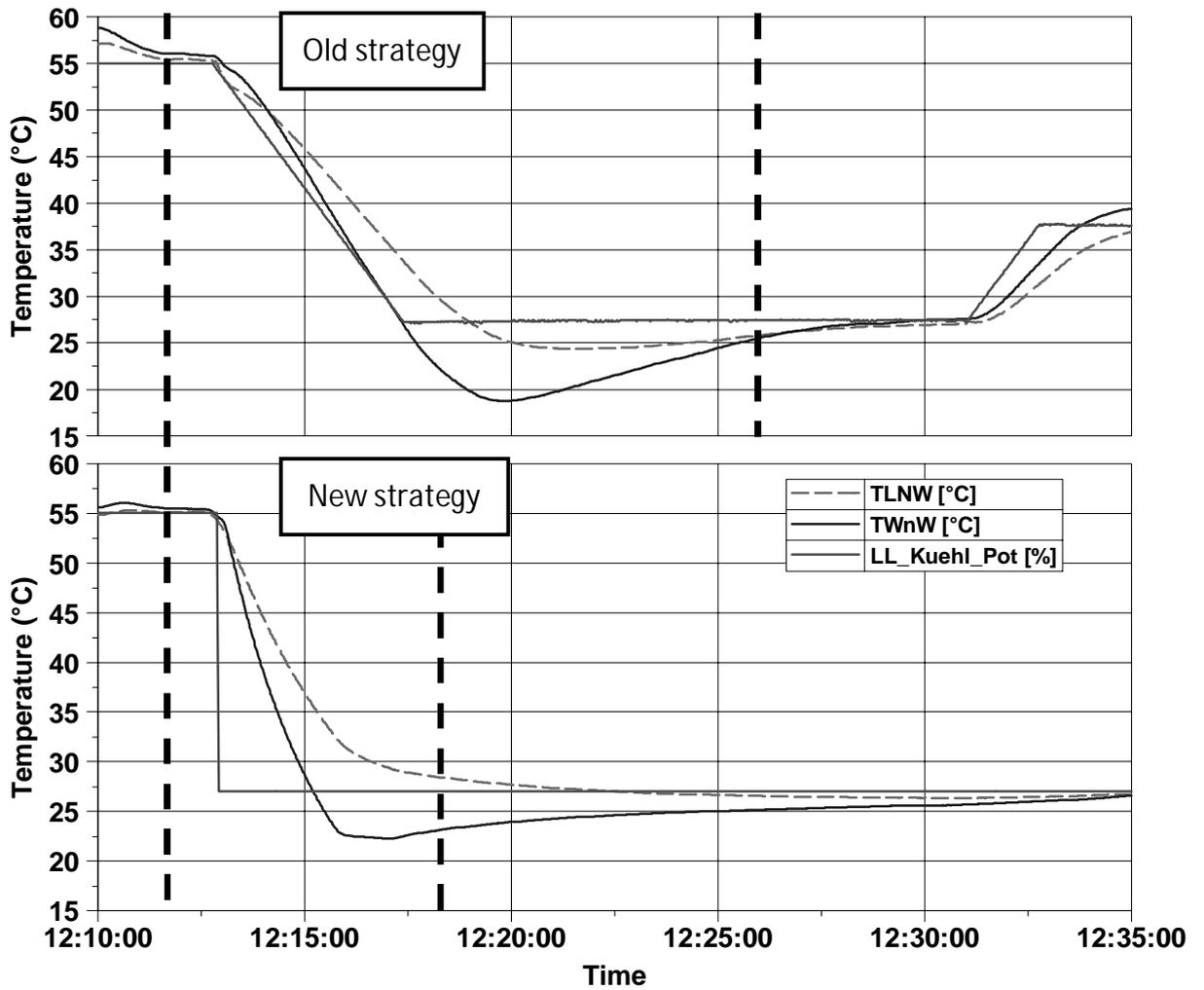
Figure 25: Step down, comparison old vs. new strategy modified

Even with the modifications, the new strategy still works better than the original one. In this case the old strategy has a stabilization time of about 14 minutes with an undershoot of 2.7°C. The modifications lead to a stabilization time of 6 min and a small undershoot of 0.6°C, which is acceptable because the temperature stays within the +/-1°C.

The connection of the heater to the PI leads to new problems.
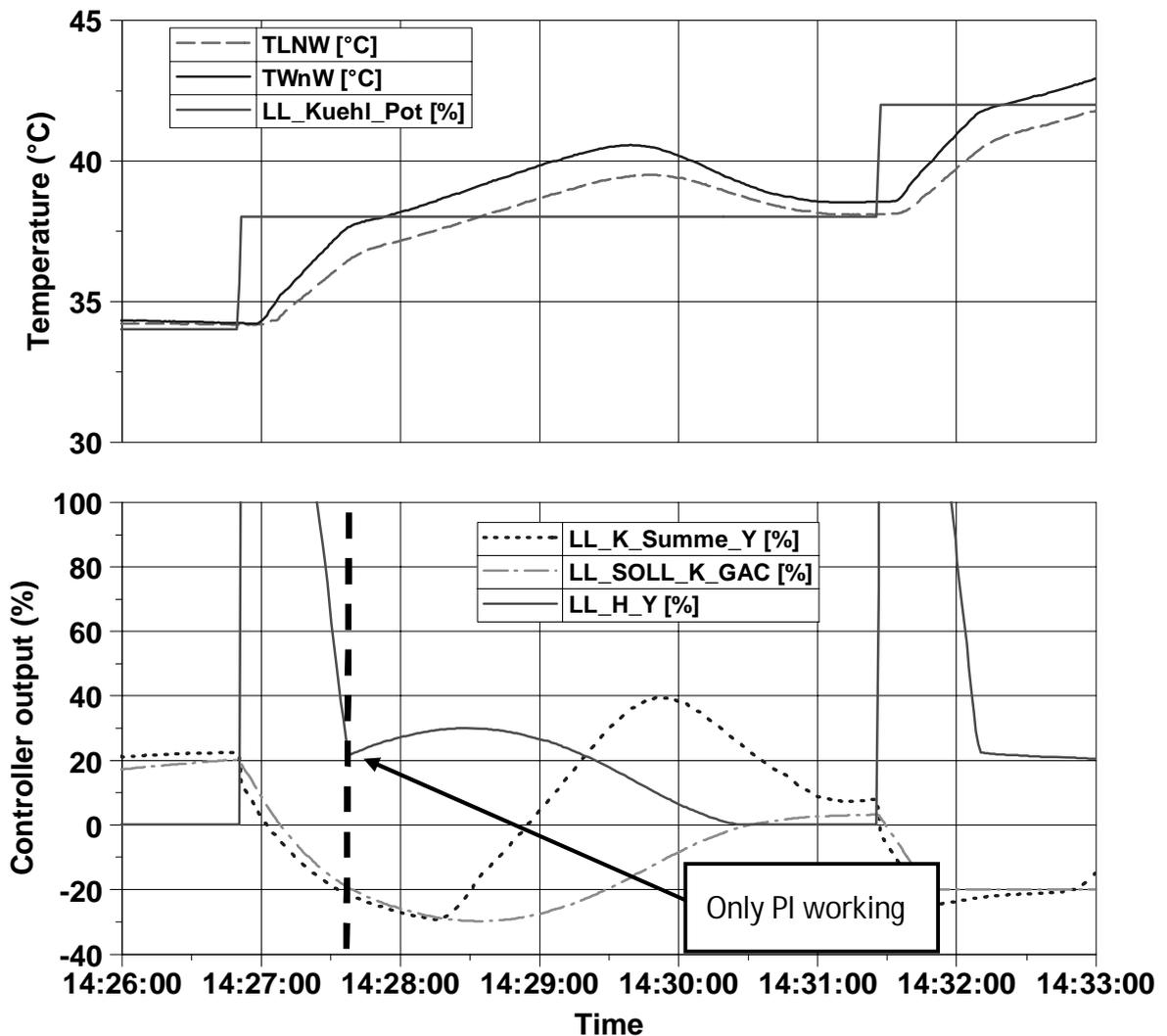
Figure 26: Step up, new strategy with modification (582 MP11)

The first graph shows the water and air temperature and the second the controller outputs. On the first graph, there is a big overshoot, 1.5°C, of the air temperature, $TLnW$. This overshoot is due to the I part of the PI governor, dotted line in green (LL_SOLL_K_GAC). The I part grows while the temperature is under the set point but takes too much time to go down to 0 again to stop heating. To compensate the proportional controller of the valve increases and the temperature $TLnW$ decreases. The stable point is reached when the I part of the controller reaches 0.

Solutions to avoid this problem would be to make the I part works faster but this would lead to oscillations or instability, the I part limitation can also be changed. By reducing the negative limit from -30% to -20%,the time the I part needs to go down to 0 and the duration of the overshoot are reduced.

More advanced control strategies, not available on ADAPT®, can be use to avoid this situation. I part initialization and the ARW method, (cf. chapter 1.1.2), are adapted strategies for this problem.

These methods have been tried "manually" with ADAPT® in order to evaluate the possibilities. To simulate the I part initialization, the I part is set to 0 when the temperature set point step occurs and to simulate the ARW method, the I part is frozen and the ARW method let it work only when the controller output is smaller than 30%.
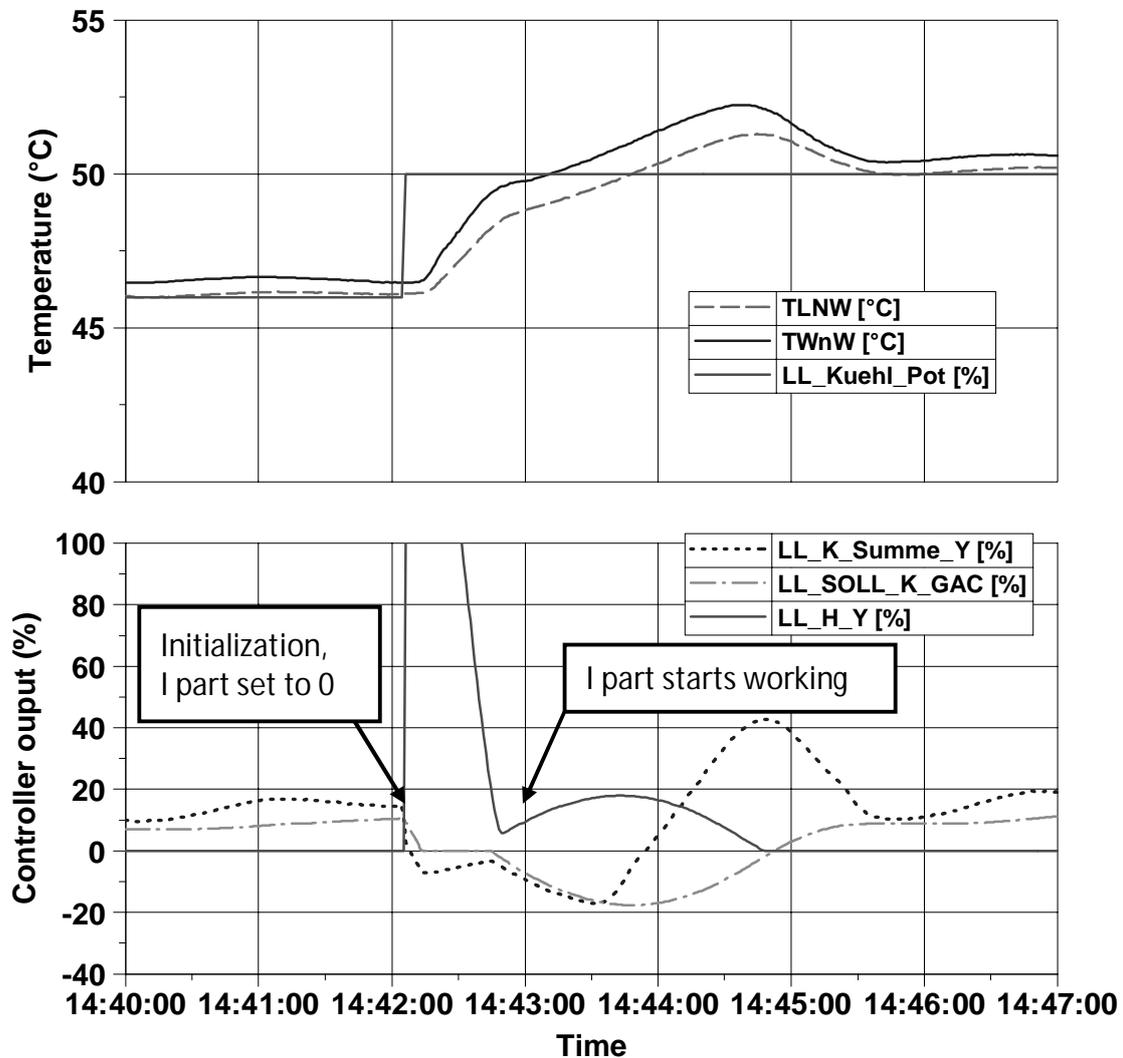
Figure 27: Step up, new strategy with modification, "fake" ARW (582 MP11)

On this graph, the I part grows to a smaller value, less than -20%, compare to the -30% of the precedent case. This allows the controller output to decrease faster to 0 and so on to reduce the duration of heating when there is no need.

In conclusion this solution works in every condition faster than the original configuration. This leads to reductions of cost and faster experiments. Further investigations might be needed in order to improve more the behavior of the system. The last investigation to do would be to try this control strategy with other test benches equipped with different engines and different AirCon systems with different parts (heater more powerful, bigger heat exchanger) in order to see how to adapt and tune the parameters of the controller.

3. Development of a new governing strategy of the AirCon system used with TCM®

3.1. Introduction with the TCM® software

TCM®, Test Cell Manager, is a software used in several test benches and was developed by FEV Gmbh. It is used in modern engine test benches to control every actuator and measure all the variables needed. It is made to replace several systems, control units by one unique computer.

TCM® uses Matlab Simulink® models integrated with the TCM® software module, ModelHandler. In the following, two models are used, one to simulate the system and one to control the system. The "simulator" model will use the controller outputs, for the heater and the valve, to calculate the temperature needed for the controller model, $TLnW$ and $TWnW$. The simulator model uses the equations described in chapter 1.3.
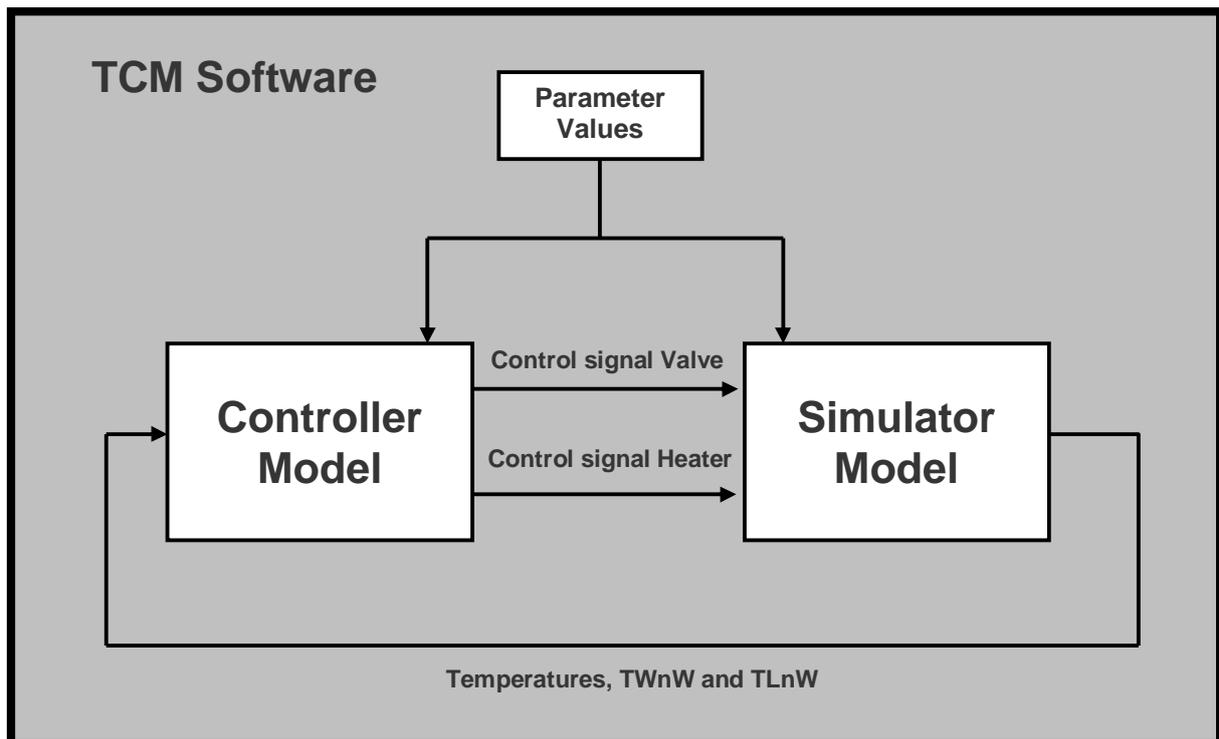
Figure 28: TCM system description

The simulator model is only necessary to verify if the controller works in the software since there is no test bench available to make experiments.

3.2. New governing strategy and ASCET® simulation

With TCM®, the control strategies are only limited to what Matlab Simulink® can do. The governing strategy use now two PIDT1, one for the valve and one for the heater and two proportional governors.
The PIDT1 uses the ARW method and the initialization of the I part.
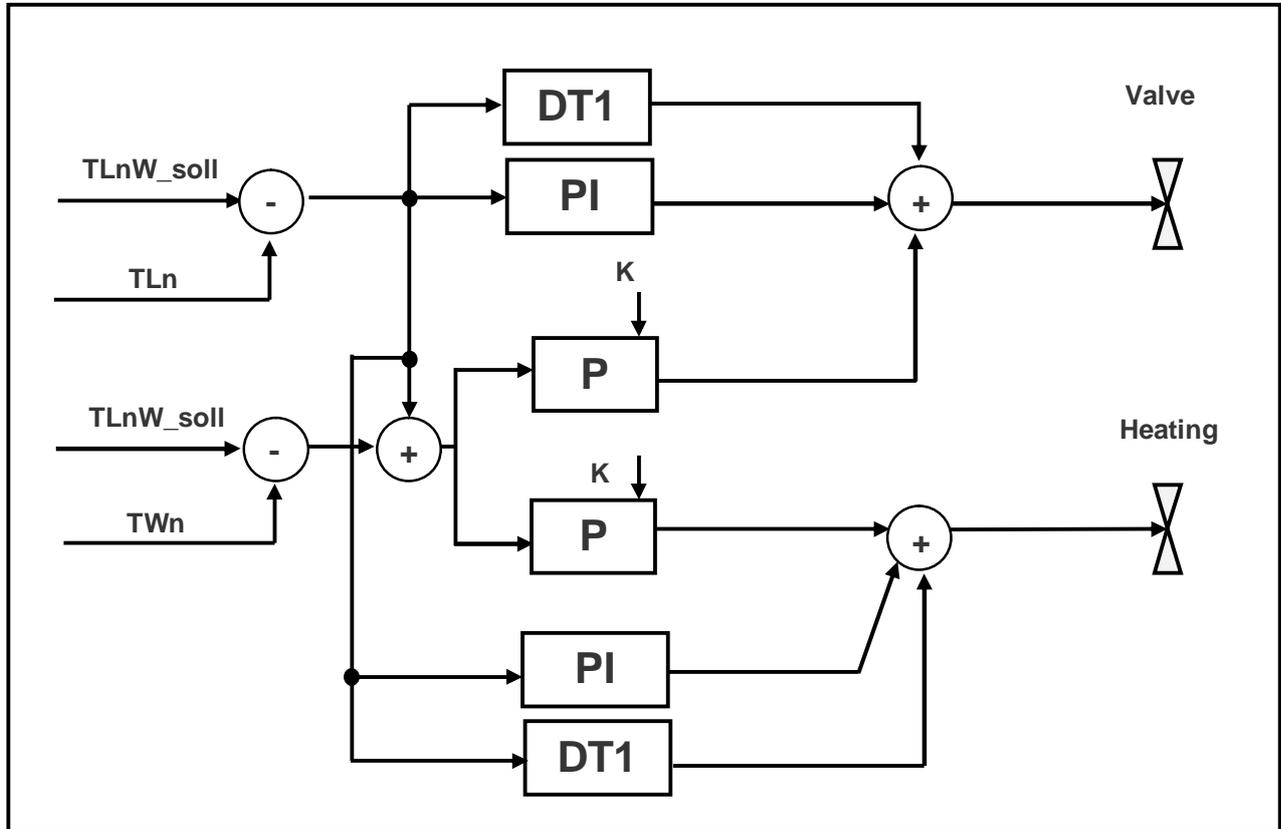
Figure 29: TCM governing strategy

With the TCM® software the governing strategy is more advanced. The deviation for the PIDT1 is now the air temperature deviation, $TLnW\_soll - TLnW$, and all the coefficient of the controller, gains, time constants, ARW method parameters, are now depending of the previous deviation and on the air mass flow.

The Simulink® model of the controller can be seen in Appendix 8 (the controller of the heater and of the valve being identical, just the first one is displayed). On the Simulink® diagrams, all the input channel, measured variables form the test bench, have their name starting with TCM_In_, all the output channels, (measured channels), are situated after a "gain" of 1 and named TCM_Out_, and all the parameters, called attributes, have their names starting with TCM_Att_.
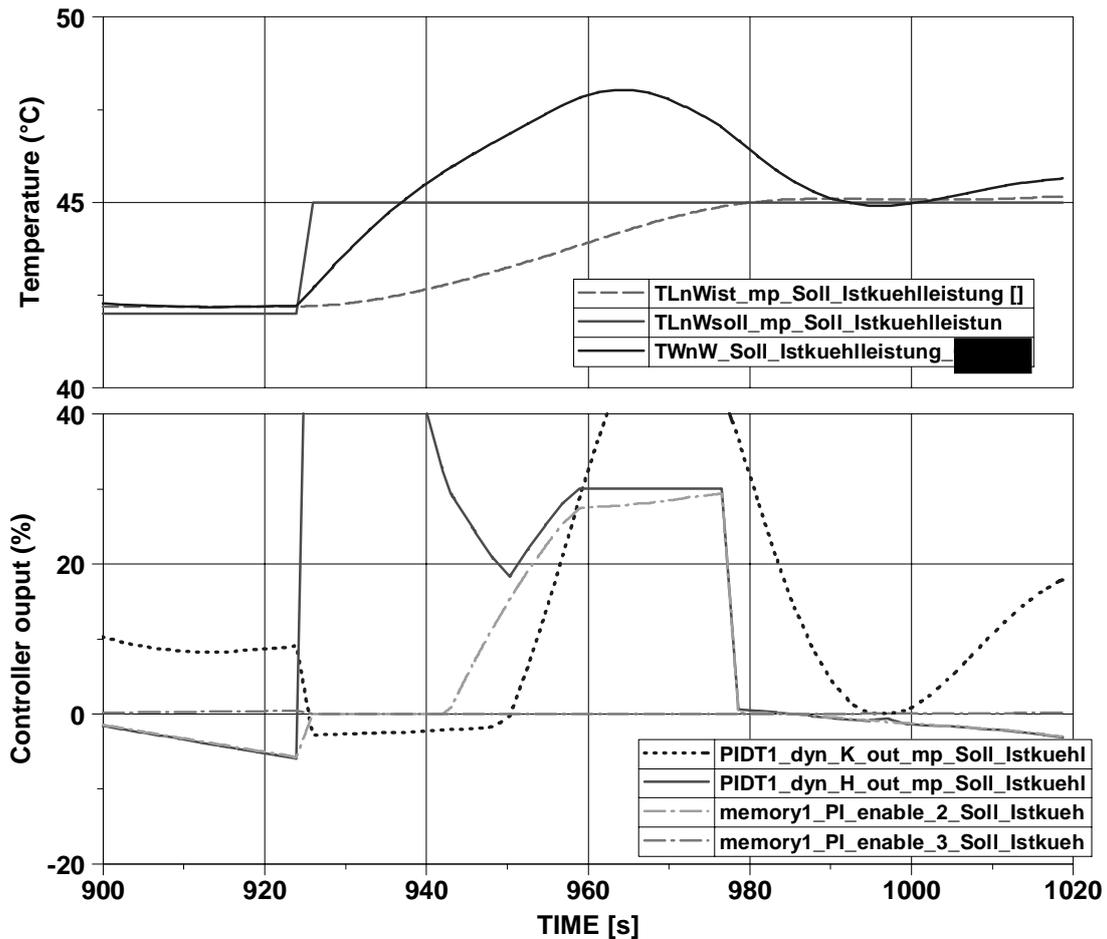
Figure 30: Step up, TCM® model simulation

On this graph, the way how the new controller is working can been seen. The controller output of the heater is displayed in red and its I part in dotted orange (memory1) and the controller output of the valve is displayed in blue with its I part in dotted turquoise (memory2).

On the second graph, there are two I part initializations. One at the set point step to put the controller in good conditions, and the second one to avoid simultaneous heating and cooling when the step point is reached. The I part initialization bloc can be seen in Appendix 8 figure 43.

The I part being around 30% when the temperature reaches the set point means that the heater will work at 30%. Since there is no more difference between the temperature and the set point the controller output is not changing anymore so the I part needs to be reinitialized to avoid cooling and heating in stationary conditions.

The tuning of the controller parameters offers us the possibilities to calibrate and adapt the controller for every situation.

The initialization is done under certain specifics conditions. These conditions are important in order not to initialize the I part when the controller is working "hard", this could lead to instability.

In the case of the heater, it occurs when the air step point changes, (step up), that means a positive air deviation, (higher than a threshold, in this case 2°C), and the water temperature, $TWnW$, higher than the air temperature, $TLnW$. This describes the first initialization.

The second initialization, when the air temperature gets close to the set point, is done when the air deviation is smaller than 0.1°C and the water temperature, $TWnW$, higher than the air temperature, $TLnW$.

These operations control a bit with a value of 0 or 1 with enable or not the I part initialization.

3.3. Implementation of the Simulink® models in TCM®

The link between the two Simulink® models is made using the outputs and inputs of each model. The inputs of the simulator are the engine speed and torque, the air mass flow and the controller signals and the water, and air temperatures, $TWnW$ and $TLnW$ are the outputs. The controller model uses these temperatures to calculate the deviations and then calculate the controller outputs for the valve and the heater.

The parameters of the controllers can be modified directly in the software using tables. The input parameters are the engine speed and torque and the air mass flow. These parameters, which are normally measured, are set manually by the user.
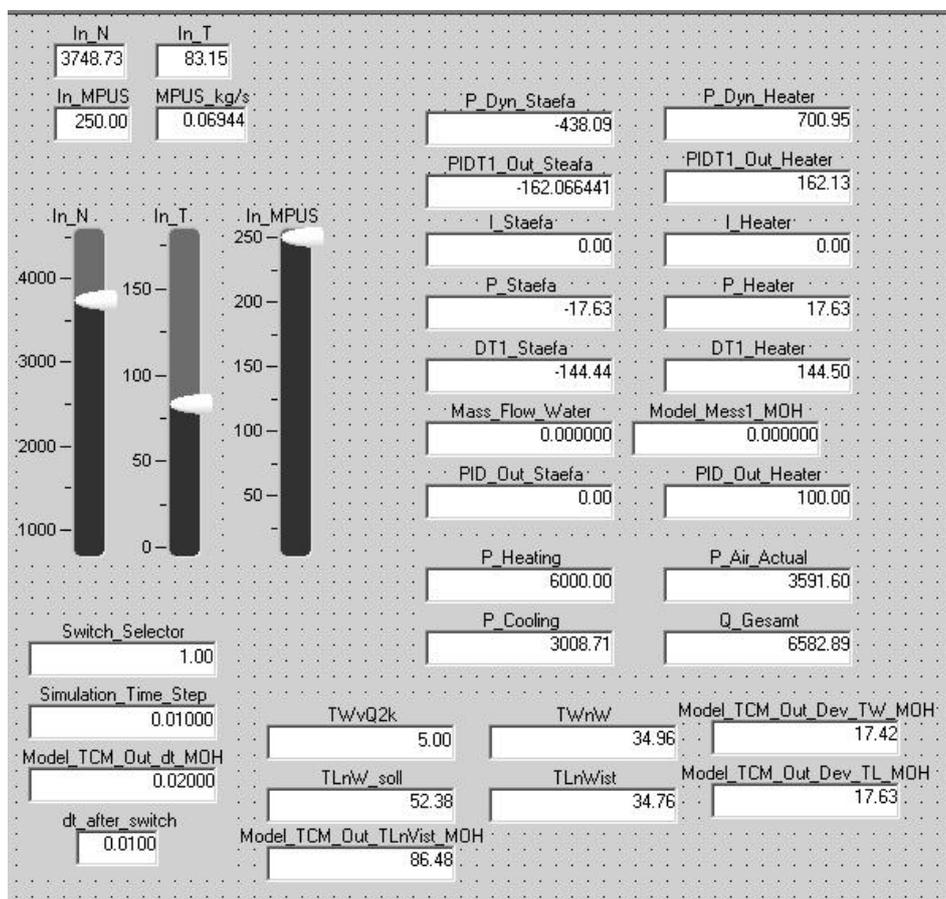
Figure 31: TCM® window with parameters and measured variables
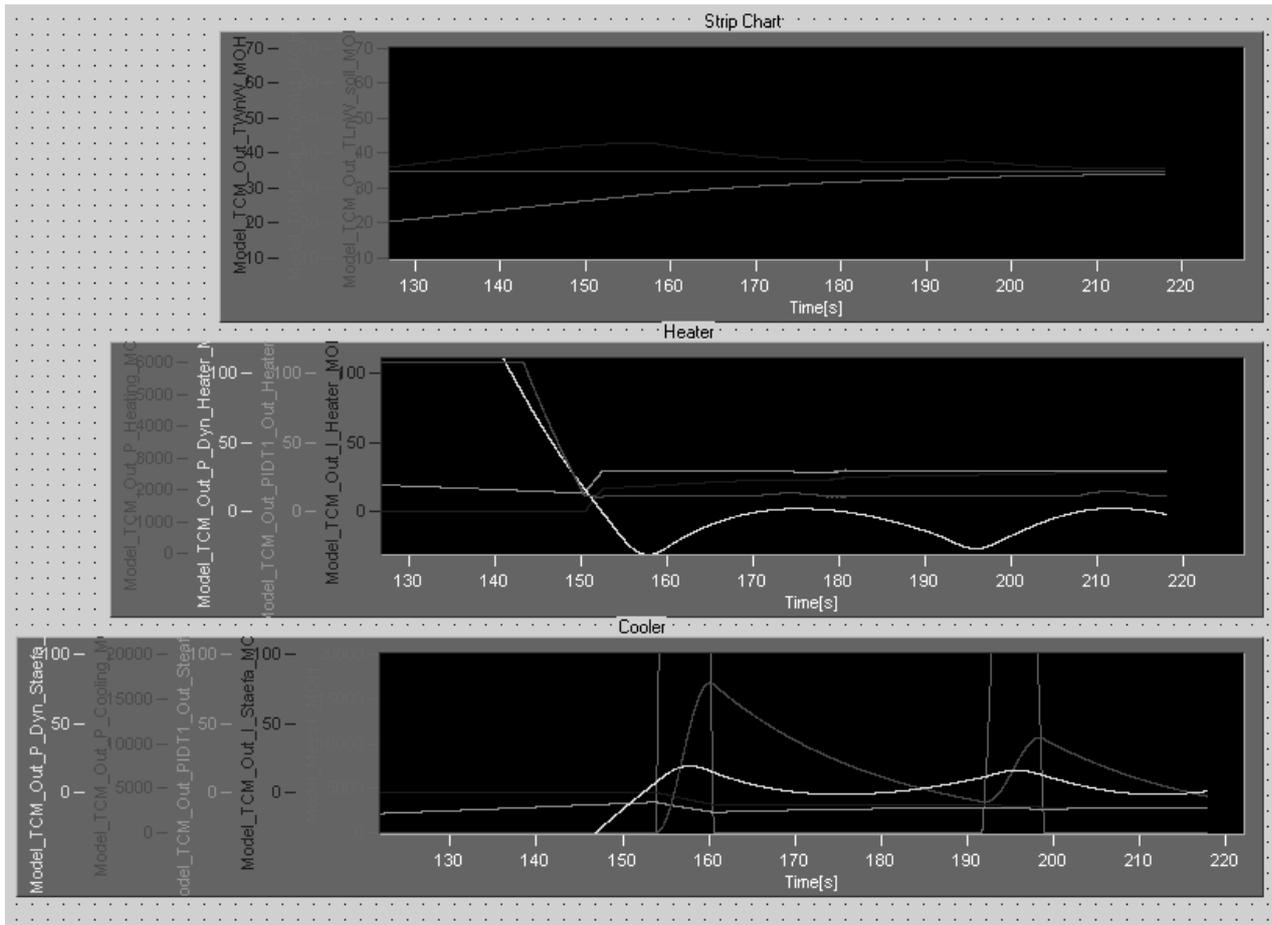
Figure 32: Test of the controller with TCM®

The first graph shows the air and water temperatures, the second graph is for the heater controller and the third one for the valve controller. The proportional controllers are displayed in yellow, the PIDT1 in green, the I parts in blue and the cooling and heating power in red. On this graph, the controllers are working and the air temperature, $TLnW$ in magenta, reaches the set point, in red. The ARW method, (for the heater, second graph), can be seen working around 150 seconds when the proportional controller output goes down and when the I part start working.

4. Development of a new AirCon hardware system

4.1. Motivation for developing a new AirCon system

With equation 1.8, the power exchanged by the air at the heat exchanger 1 is:

$$P_1 = \dot{m}_l \cdot cp_{air}(TLvW - TLnW)$$

By comparing the temperature maps of TLnV and TLnW, appendices 2 and 3, TLnV is always larger than TLnW so the intake air only needs to be cooled down. The use of a heater is being questionable. The idea is then to only cool the air through a heat exchanger and use the high energy available due to the high temperature of the air down-stream compressor to heat up.

With this new idea the system becomes much simpler, and therefore cheaper, and easier to control. With less parts and less mass of fluid the inertia of the system is reduced significantly so the dead times are smaller and the system would become much faster.

This new system is inspired by the cooling circuits used for water and oil cooling on some engine test benches (picture from "Engine Testing Theory and Practice", A.J. Martyr and M.A. Plint).
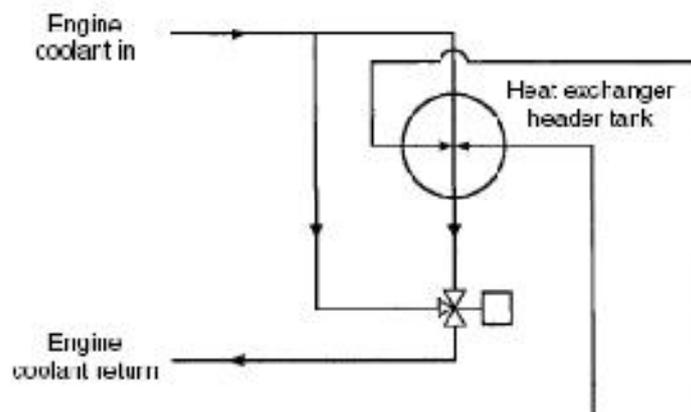
Figure 33: Water cooling system on engine test bench

The system uses a three-way valve to mix the hot and cold flows. The heating of the fluid is done only by bypassing the heat exchanger. The temperature of the fluid can be adjusted by the position of the three-way valve.

4.2. Introduction to the AirCon new system

The new AirCon system uses the same principle as in <u>figure 33</u>. The flow downstream compressor is split in two. One goes directly from the compressor to the mixing valve, or three-way valve, and the second goes through a heat exchanger to be cooled down. The valve is adjusted to mix the hot and cold flows in order to get the required air temperature.
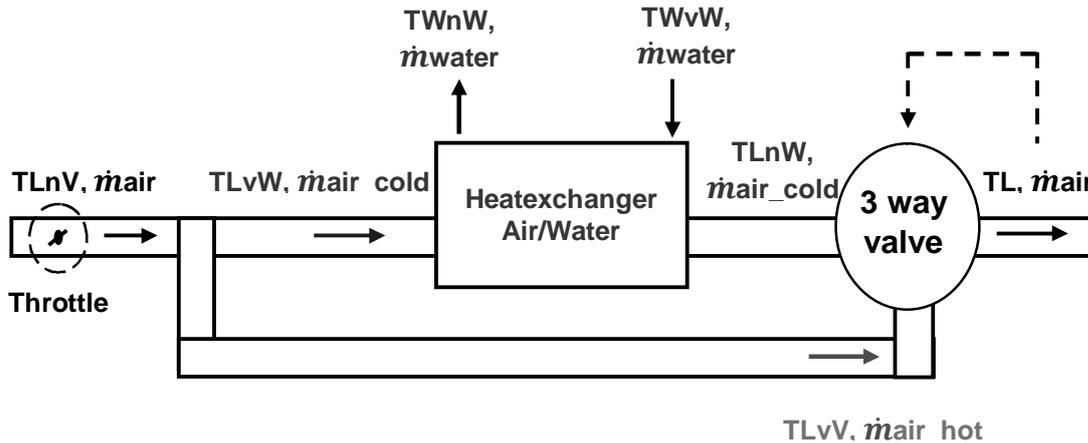
Figure 34: Schema of the new AirCon system

The equations used to calculate the model are similar to those used in <u>chapter 1.3</u>. First the temperatures $TLvW$ and $TLvV$ are calculated using the power losses due to convection. The <u>equation 1.3</u> is used
.

$$TLvW = TLnV - \frac{hS}{\dot{m}_l \cdot cp_{air}}((TLnV - x) - T_{air})$$

In the case of $TLvW$ is parameter values are kept the same: $hS = \textbf{2.8}\,W/K$, $x = \textbf{16°}C$ and $T_{air} = \textbf{20°}C$.
For $TLvV$:

$$TLvV = TLnV - \frac{hS}{\dot{m}_l \cdot cp_{air}}((TLnV - x) - T_{air})$$

The pipe between the junction and the valve being longer the losses are higher. The parameters values are adapted to plausible values: $hS = \textbf{6}\,W/K$, $x = \textbf{21°}C$ and $T_{air} = \textbf{20°}C$.

As in the previous models the temperature of the cold water entering the heat exchanger is the constant and set to 5°C.

In this case the cold water is not controlled. It has to be set to a value in order to be able to fit the requirements in the worst conditions, maximum power operating point, with the highest air mass flow and the highest temperature.

Assuming the worst case where all the air flow goes through the "cold way", ($\dot{m}_{air\_cold} = \dot{m}_{air}$ and $\dot{m}_{air\_hot} = 0\ kg/s$), and $TL = TLnW$, the power exchanged at the heat exchanger is:

$$\dot{m}_{air} \cdot c_{pair}(TLnV - TL) = \dot{m}_{water} \cdot c_{pwater}(TWnW - TWvW)$$

At maximum power, the parameters have the following values:
$\dot{m}_{air} = 300\ kg/h,\ TLnV = 140°C$ and $TL = TLnW = 55°C$.

$TWnW$ and $TLnW$ are related like in the previous models with a transfer function:

$$\frac{TLnW(s)}{TWnW(s)} = \frac{1}{(1 + \tau \cdot s)}$$

In stationary conditions and in the model $TLnW = TWnW$.

Having all the values, a minimum water flow can be calculated for the model:

$$M_{water} = \frac{\dot{m}_{air} \cdot c_{pair}(TLnV - TL)}{c_{pwater}(TWnW - TWvW)} = \frac{300.1005.(140 - 55)}{4180.(55 - 8)} \approx 130\ kg/h$$

To keep a safety margin, the cold water mass flow is set to **150** kg/h in the model. Another solution could be to consider also regulating the water flow with an extra valve, in order to save some more energy. But with this extra valve another controller has to be implemented, which increases significantly costs without offering significant improvements.

The water temperature down-stream heat exchanger, $TWnW$, is calculated using the heat exchange formula at the heat exchanger.

$$\dot{m}_{air} \cdot c_{pair}(TLnV - TL) = \dot{m}_{water} \cdot c_{pwater}(TWnW - TWvW)$$
$$TWnW = TWvW + \frac{\dot{m}_{air} \cdot c_{pair}(TLnV - TL)}{\dot{m}_{water} \cdot c_{pwater}}$$

The final air temperature, TL, is calculated using the temperatures $TLnW$ and $TLvV$ and the hot and cold air mass flows.
By assuming no losses in the three-way valve the intake temperature can be calculated with the following equation:

$$M_{air}TL = \dot{m}_{air\_cold}TLnW + \dot{m}_{air\_warm}TLvV$$

So:

$$TL = \frac{\dot{m}_{air\_cold}TLnW + \dot{m}_{air\_warm}TLvV}{\dot{m}_{air}}$$

To simplify the equation a new parameter is introduced,

$$alpha = \frac{\dot{m}_{air\_cold}}{\dot{m}_{air}}$$

Assuming that the valve has a linear behavior, alpha can be seen as the valve opening. The valve fully opened, alpha=1, meaning that all the flow goes through the cold way and the valve fully closed, alpha=0, meaning that all the flow goes through the hot way.

The temperature equation becomes:

$$TL = alpha.TLnW + (1 - alpha)TLvV$$

In the model, the different values which are supposed to be real or measured, are implemented with first order filters to makes the system more realistic.

4.3. Development and simulation of the new AirCon system

The regulation of the valve will be made in the model using the valve opening alpha. A desired value and calculated value, ("real" values) of alpha are compared to get the deviation of the controller.
The desired value of is calculated with the air temperature set point:

$$alpha_{soll} = \frac{TLvV - TL_{soll}}{TLvV - TLnW}$$

The actual air temperature, TL, will be calculated with the PID output. The actual value of the valve can then be calculated using the TL value:

$$alpha = \frac{TLvV - TL}{TLvV - TLnW}$$

The controller will work with the deviation: $alpha_{soll} - alpha$.
It is first chosen to work with a PI controller to be able to use the system with ADAPT®. The system being much simpler and faster, the results should be satisfying with a correct calibration.
The system has much smaller dead times, the ARW method is now not needed anymore and the I part initialization which was needed to avoid heating and cooling is also not needed.
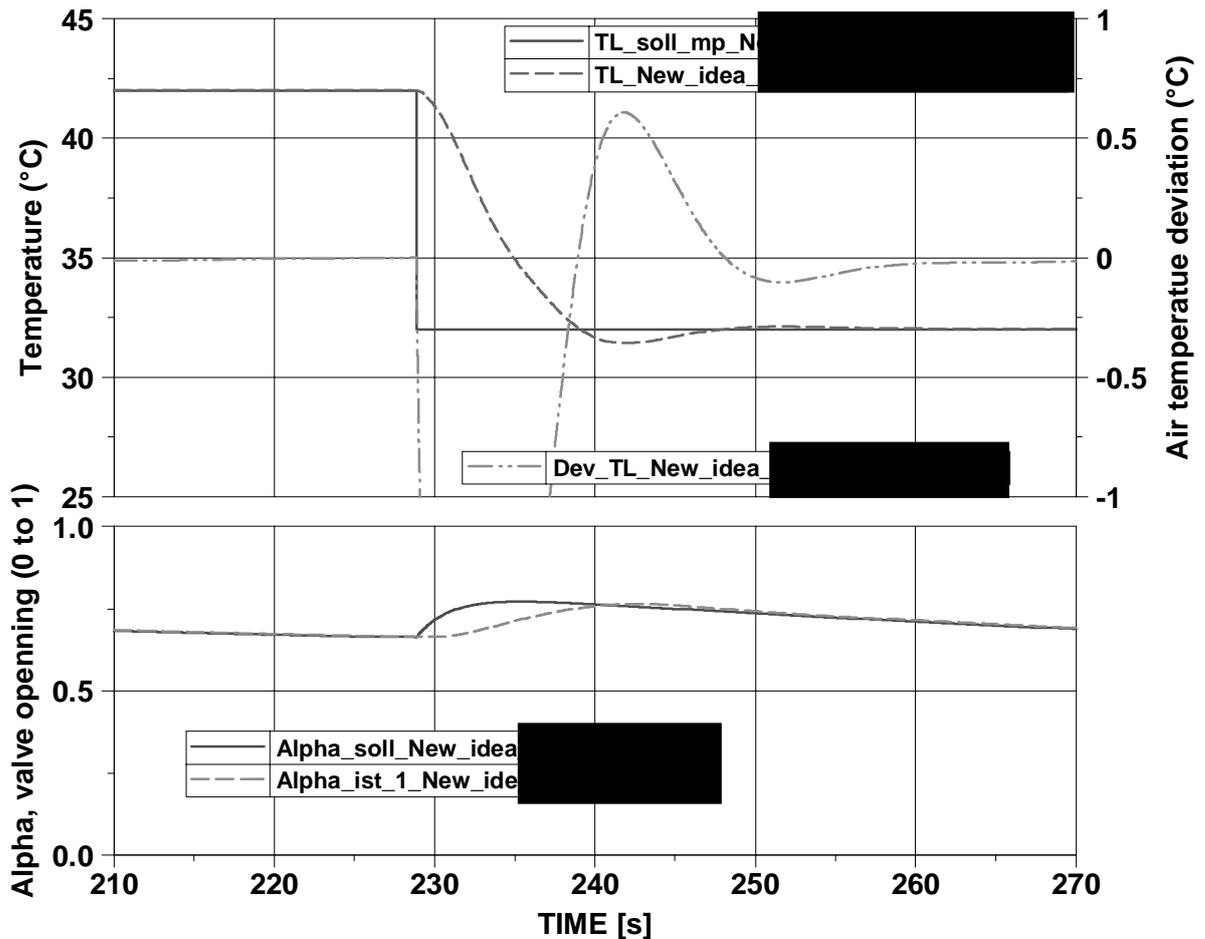
Figure 35: Step down, New AirCon system

This graph shoes how fast this new system is. In this case the stabilization time is less than 10 seconds. In similar conditions the simulation model developed in chapter 3.2 takes more than 100 seconds.

The second graph shows how by opening the valve, bigger alpha value, the air temperature drops. Then the alpha value adapts to the new operating point with the new temperatures.

The main disadvantage of this new system is that by introducing a new component on the air path, it changes the pressure losses. The pressure losses are set on the test bench with a throttle, cf. figure 34. This throttle is set up in order to simulate the pressure losses over by the intercooler, normally present on the engine. The pressure loss over the throttle is, depending on the engine, usually between 80 and 120 mbar, at full load.

To be able to use the new system, the valve pressure has to be calculated to know what must be the new set up for the throttle.

To calculate this pressure drop, the valve documentation from <u>LDM</u> is used.

$$\Delta p = \left(\frac{Q_n}{K_v \cdot \mathbf{5141}}\right)^2 \frac{\rho_n \cdot T_1}{p_2}$$

Δp is the pressure drop in MPa, Q_n is the air mass flow in m³/h, ρ_n is the gas density under normal conditions in kg/m³, T_1 is the temperature upstream valve in K, p_2 is the pressure downstream valve in MPa and K_v pressure losses coefficient in m³/h.

The following calculations are done with an experiment from a 2.2 liter turbocharged diesel.
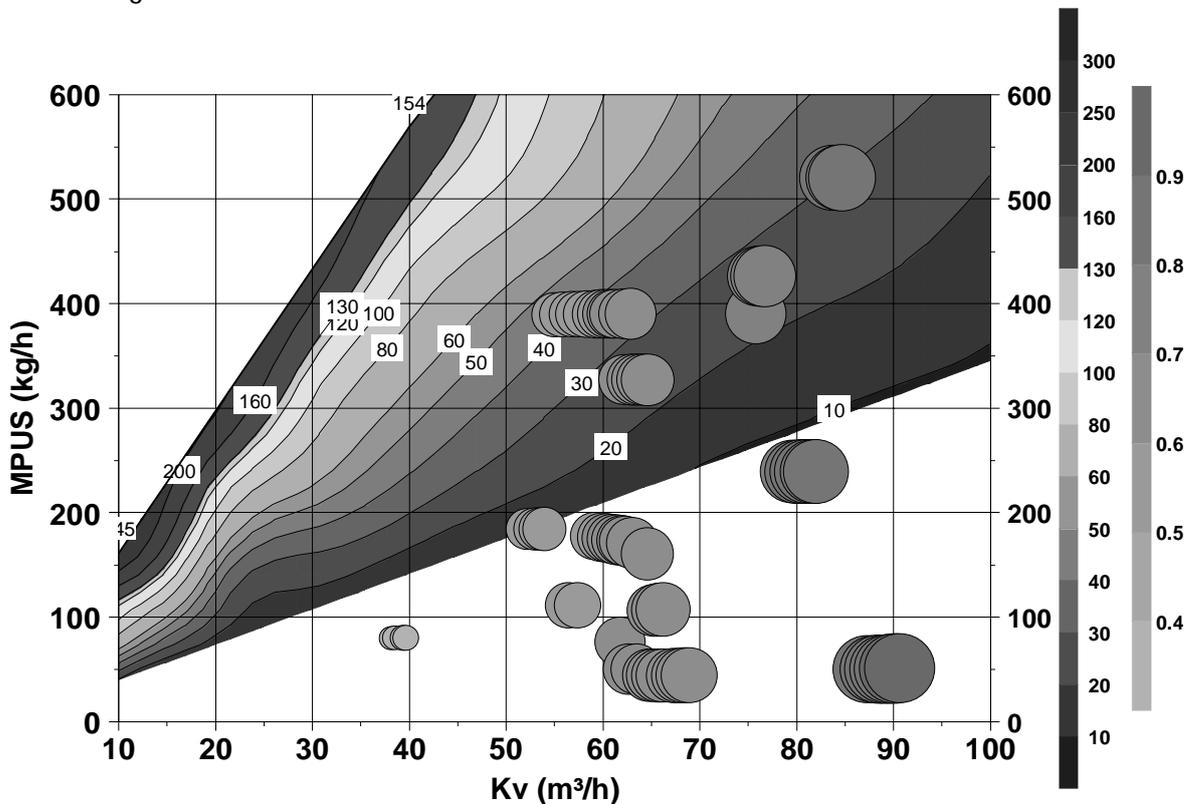
Figure 36: Pressure losses (mbar) for the three-way valve

On this graph, the pressure losses, (blue to red) are display in function the intake air mass flow, MPUS and K_v which is equivalent to the valve opening. $K_v = \mathbf{100}$ means a valve fully opened. The simulated operating points of the valve are displayed with circles. These operating point have been calculated simulating engine operating that cover the whole engine map.

These circles are in a zone where the pressure losses due to the valve are less than 50 mbar. The pressure losses being less than the ones created with the throttle, the valve can be used on the air path. The throttle just needs to be readjusted. To control the throttle to correct automatically the total pressure drop could also be investigated.
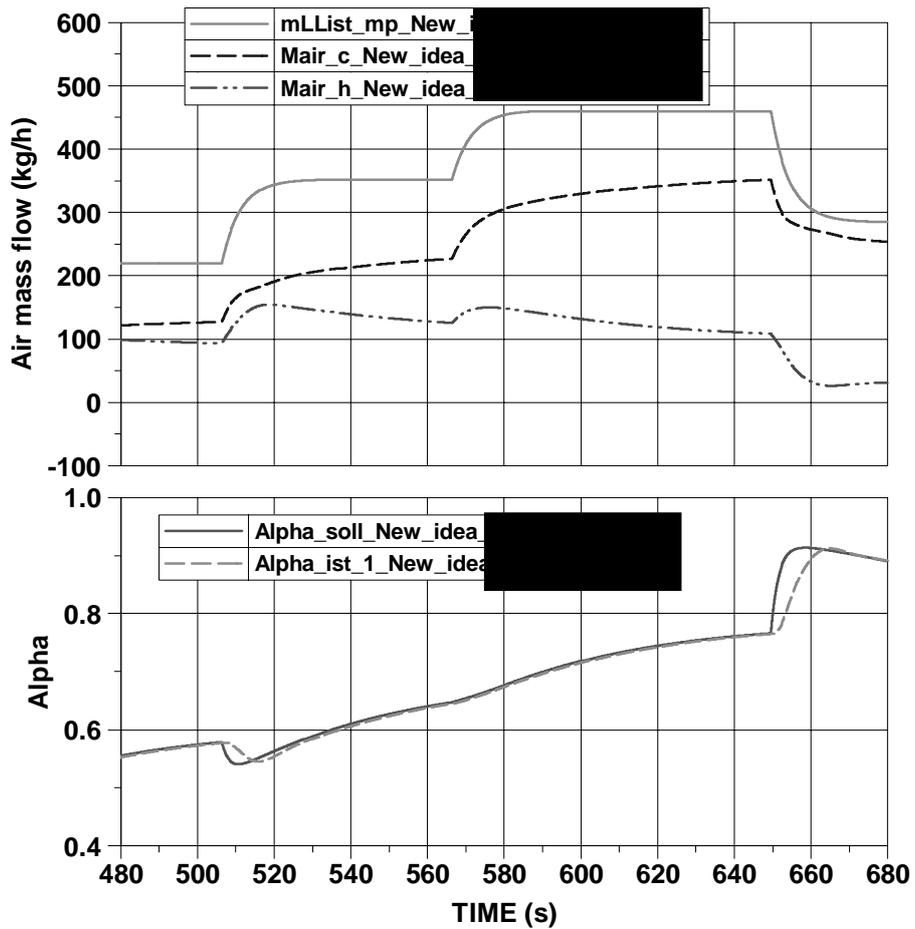
Figure 37: Air mass flow regulated

The first graph shows the air mass flows, cold, hot and global. On the second graph, the alpha value is changing constantly. Temperature changes being really slow and the system being fast, it can adapt the air mass flow to keep a constant output air temperature.

This system has few chances to be realized and tested because of the variable pressure drop it is creating.

5. Conclusion and future prospects

The improvements realized on the test bench controller have improved significantly the stabilization time of the intake air temperature. It is not anymore the slowest parameters. Further improvements can now be made on other controllers to improve stabilization times.

The governing strategy developed on ADAPT® has also to be tested and adapted on other test benches with different configurations and with different engines.

Once a test bench is equipped with the TCM® system in combination with an AirCon system, the Simulink® model of the controller, already implemented in the ModelHandler of TCM®, has to be tested with a real engine and calibrated in order to get the performances of the system.

The new AirCon system developed in chapter 4 shows promising performances which could induce big cost savings concerning engine testing and the system itself, being cheaper with fewer parts and less maintenance, but the pressure being too different from the pressure losses in a real engines, this system has few chances to be implemented on a test bench.

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[4]   "Instrumentation for Process Measurement and Control", Norman A. Anderson

[5]   "Fundamentals of Process Control Theory" Research Triangle Park, N.C., Instrument Society of America, 1981

[6]   DIN 19226 Part 1 to 6 "Leittechnik: Regelungstechnik und Steuerungstechnik" (Control Technology). Berlin: BeuthVerlag

[7]   "International Electrotechnical Vocabulary", Chapter 351: Automatic control. IEC Publication 50

[8]   "Fundamentals of Heat and Mass Transfer", Frank P. Incropera and David P. Dewitt

List of symbols, abbreviations and translations

Abbreviations	Description/translation	Unit
S	Exchange area	m^2
C_p	Heat capacity	$J/kg.K$
C_{pwg}	Heat capacity of the mixture 70% water and 30% glycol	$J/kg.K$
C_{pwk}	Heat capacity of the cold water	$J/kg.K$
DTQ2w	Temperature difference TWvH-TWnW	$°C$
DTQ3w	Temperature difference TWvW-TWvH	$°C$
DVT_gekoppelt	Coupled deviation, DVT_TL+DVT_TLW	$°C$
DVT_TL	Air temperature deviation, TLnWsoll-TLnW	$°C$
DVT_TLW	Water temperature deviation, TLnWsoll-TWnW	$°C$
h	Convective heat transfer coefficient	$W/m^2.K$
Heizleistung	Heating power	W
K_i	integral gain, $K_i=1/T_i$	$s-1$
K_p	Proportional gain	
Kuhlleistung	Cooling power	W
LL_H_REF	intake air temperature set point correct (ramp)	$°C$
LL_H_Y	Controller output heater	%
LL_Heiz_Pot	Controller output heater	%
LL_K_Summe_Y	Controller output valve	%
LL_K_Y	Controller output valve	%
LL_Kuehl_Pot	Air intake temperature set point	$°C$
LL_soll	Air intake temperature set point	$°C$
LL_soll_Des	Air intake temperature set point	$°C$
LL_SOLL_K_GAC	I part of the governor	%
M_{air}	Total air mass flow (New AirCon system)	kg/s
M_{air_cold}	Warm air mass flow (New AirCon system)	kg/s
M_{air_hot}	Hot air mass flow (New AirCon system)	kg/s
MDRE	Engine torque	Nm
ml	Intake air mass flow	kg/s
MPUS	Intake air mass flow	kg/h
Mwater	Water mass flow (New AirCon system)	kg/s
mwg	Mass flow of the mixture 70% water and 30% glycol	kg/s
mwk	Mass flow of the cold water	kg/s
NMOT	Engine speed	rpm
Pheat	Power from the electric heater	W
Q	Heat transfer (Power)	W
Staefa	Also used to name the valve	
T	Temperature	$°C$

Abbreviations	Description/translation	Unit
T_{air}	Temperature of the ambient air	$°C$
T_d	Derivative time constant	s
T_i	Integral time constant	s
T_L	Air temperature down-stream the three-way valve	$°C$
TLnV	Air temperature down-stream compressor	$°C$
TLnW	Air temperature down-stream heat exchanger	$°C$
TLvV	Warm air temperature up-stream the three-way valve	$°C$
TLvW	Air temperature up-stream heat exchanger 1	$°C$
TWLK	Water temperature down-stream heater	$°C$
TWnQ2k	Cold water temperature up-stream heat exchanger 2	$°C$
TWnW	Water temperature down-stream heat exchanger 1	$°C$
TWvH	Water temperature up-stream heater	$°C$
TWvQ2k	Cold water temperature down-stream heat exchanger 2	$°C$
TWvQ2w	Water temperature up-stream heat exchanger 2	$°C$
TWvW	Water temperature up-stream heat exchanger 1	$°C$
VWLLK	Cold water mass flow	l/min
τ	Filter time constant	s

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Appendices

Appendix 1: Hardware part specifications

Part	Specification
Engine	1,3 liter turbocharged diesel engine
Test bench	N10
Heat exchanger Air/Water	AlfaLaval 40L
Heat exchanger Water/Water	AlfaLaval 20L
Electric Heater	Power: 6 kW
Auxiliary pump (auxiliary water circuit)	Grundfos, UPS 32-80 180

Appendix 2: Air temperature down-stream compressor (TLnV)
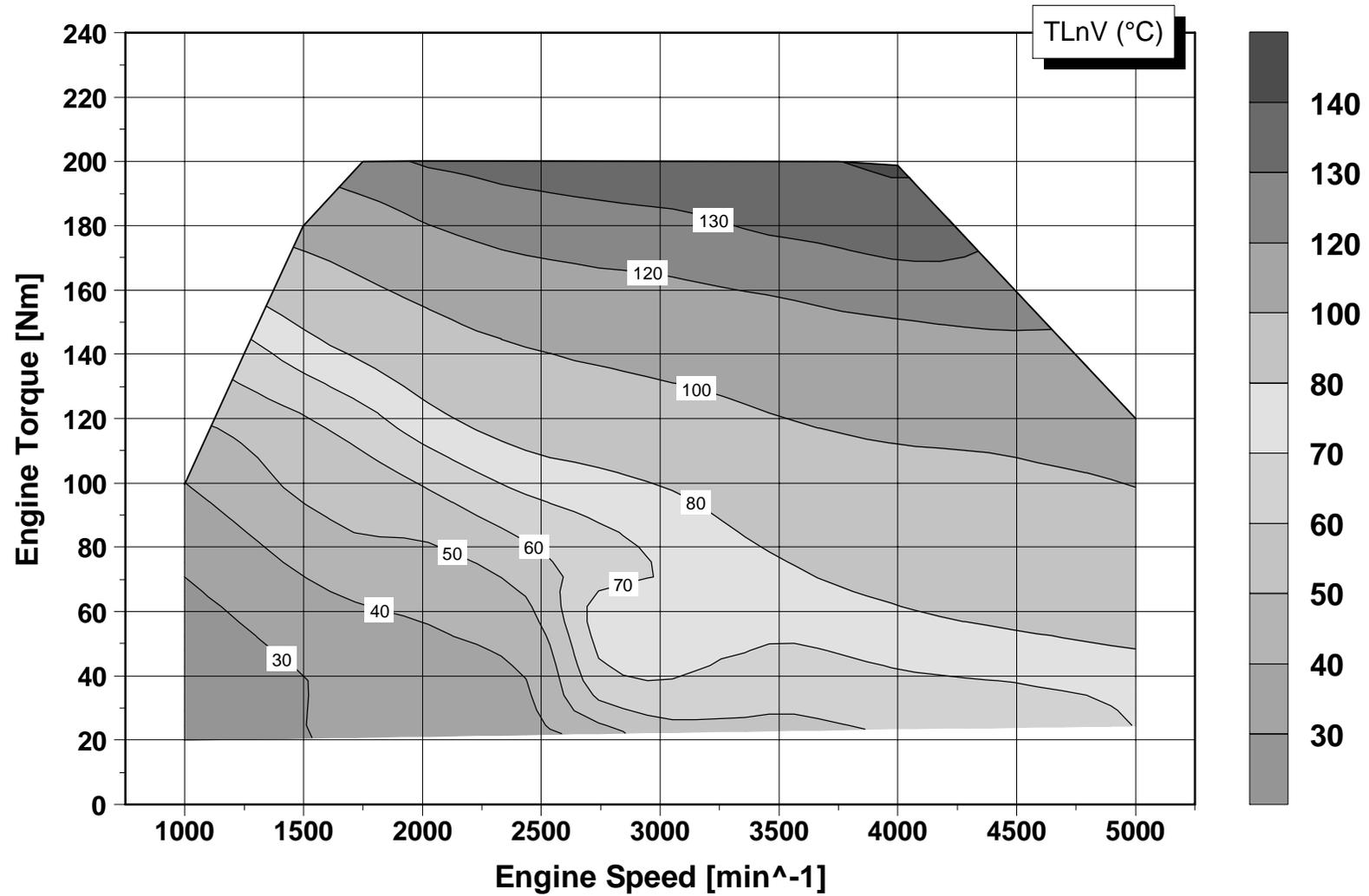
Appendix 3: Air temperature set point down-stream heat exchanger (TLnV)
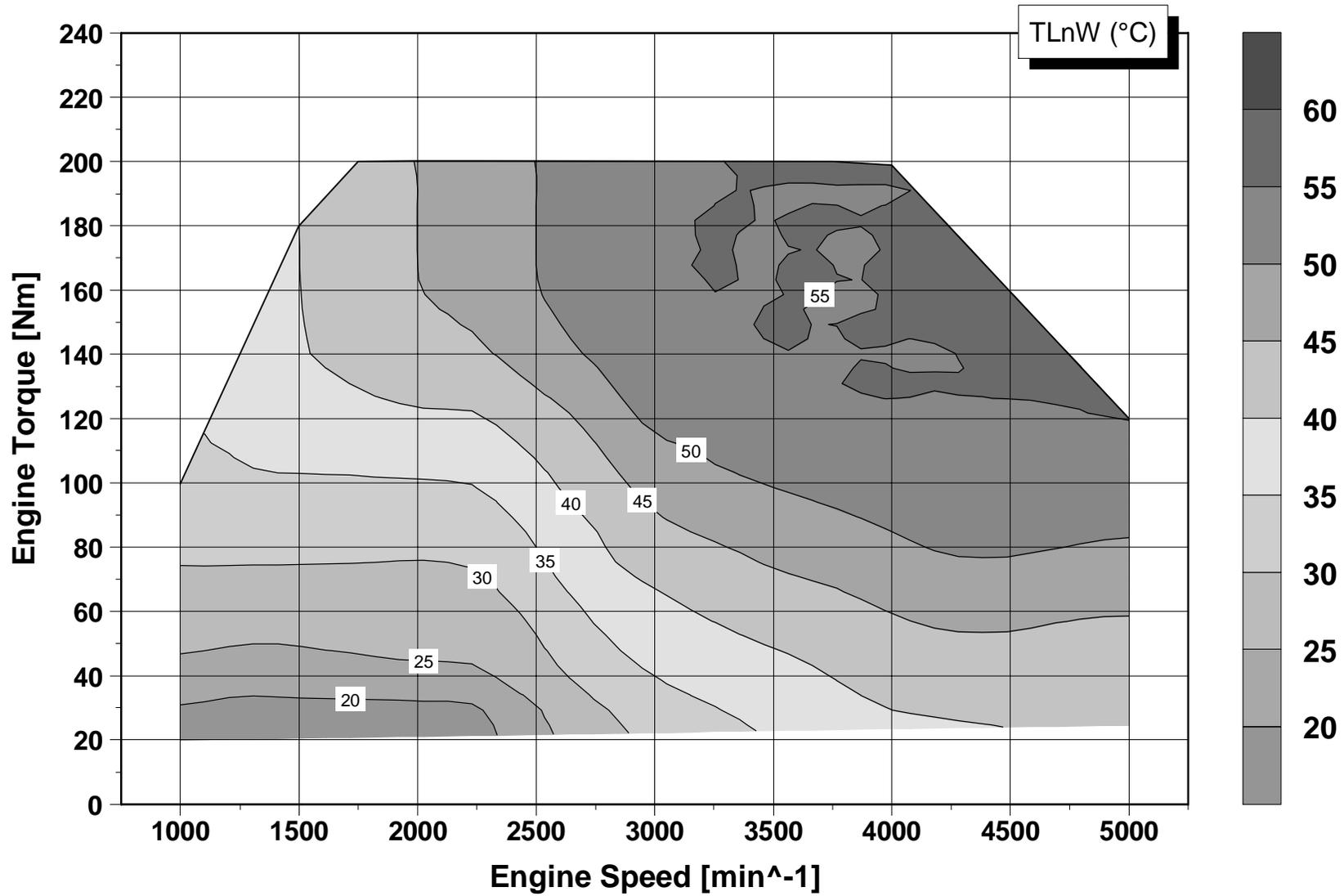
Appendix 4: Intake air mass flow
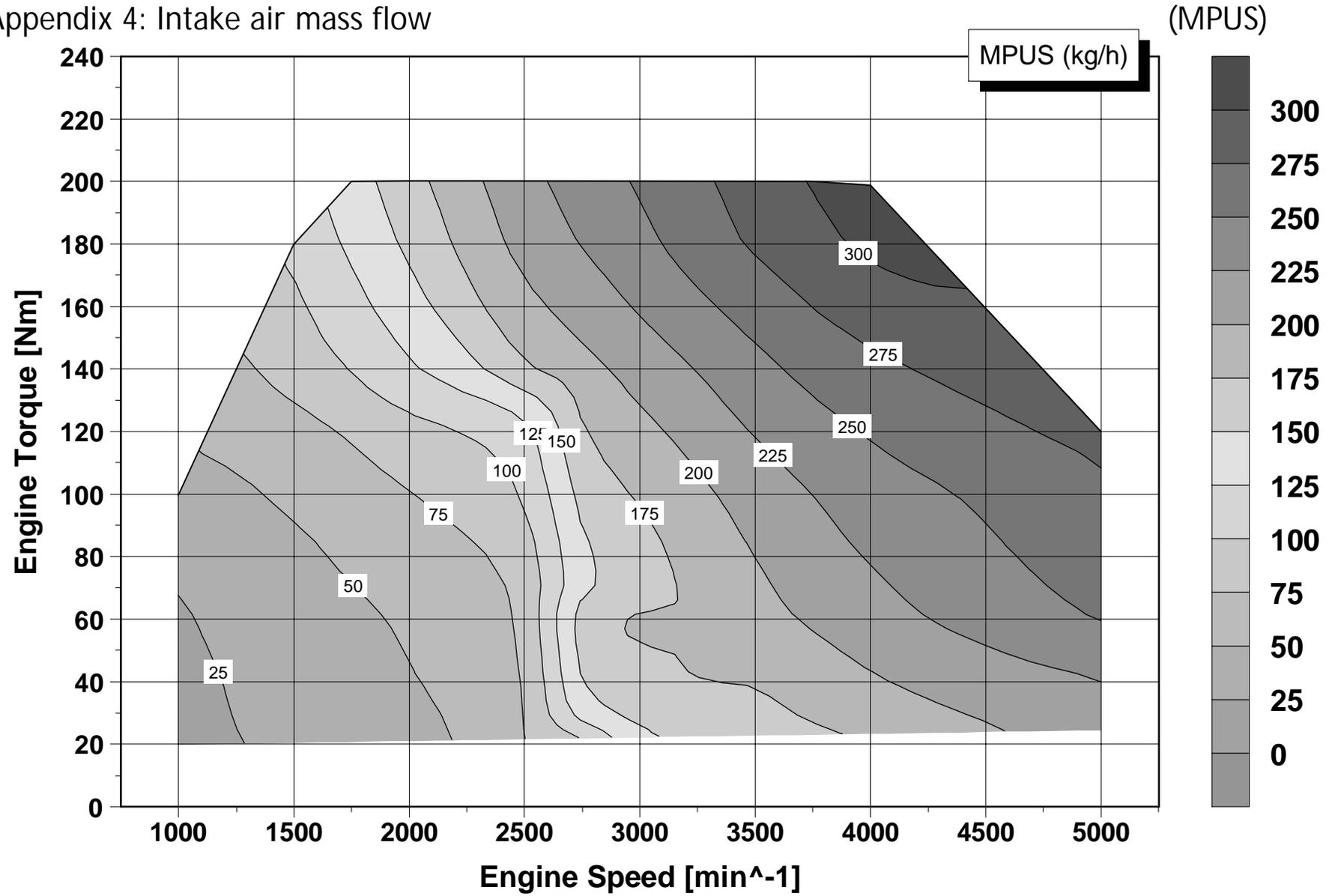
Appendix 5: ADAPT® display window
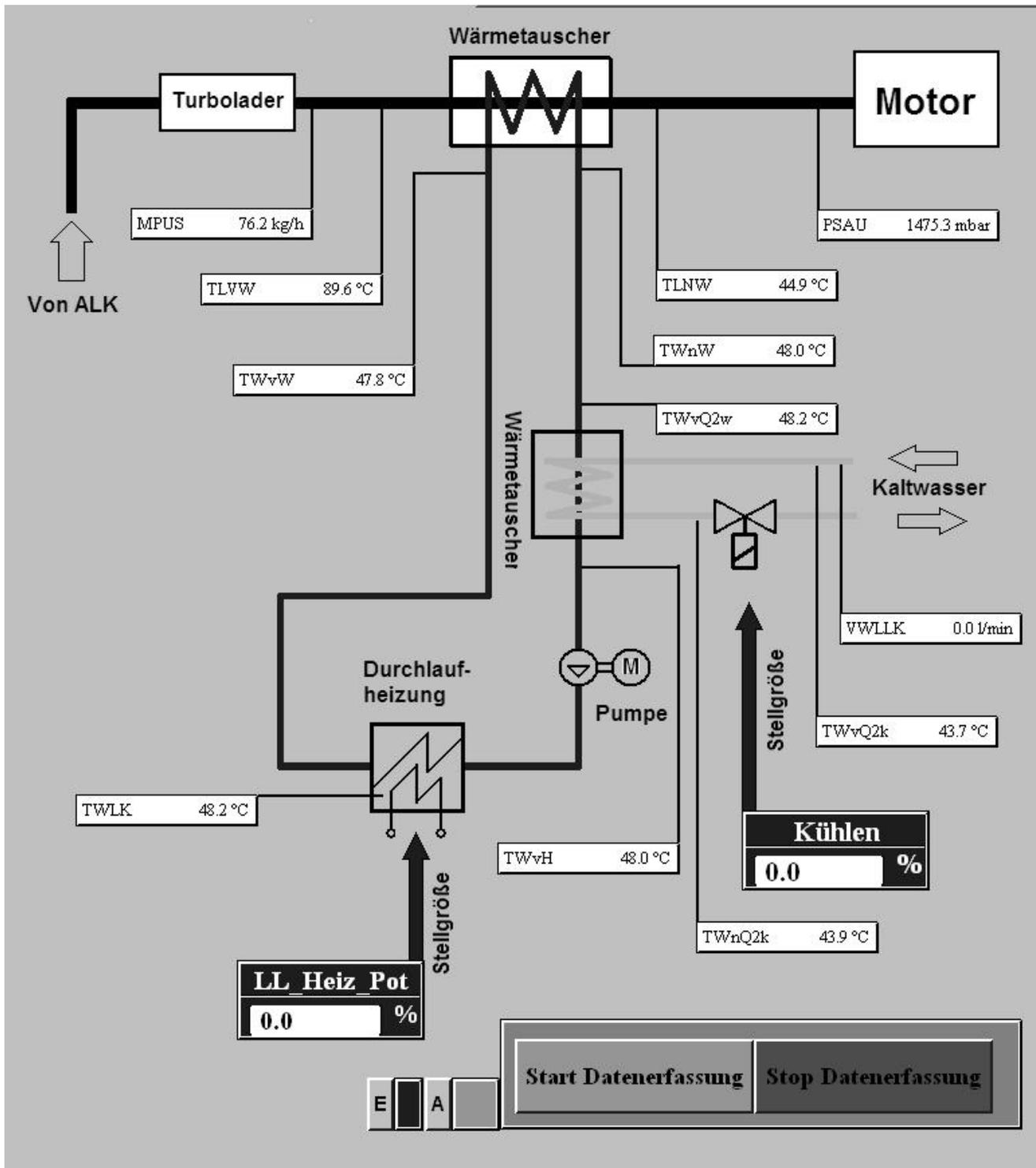
Appendix 6: Valve flow characteristics
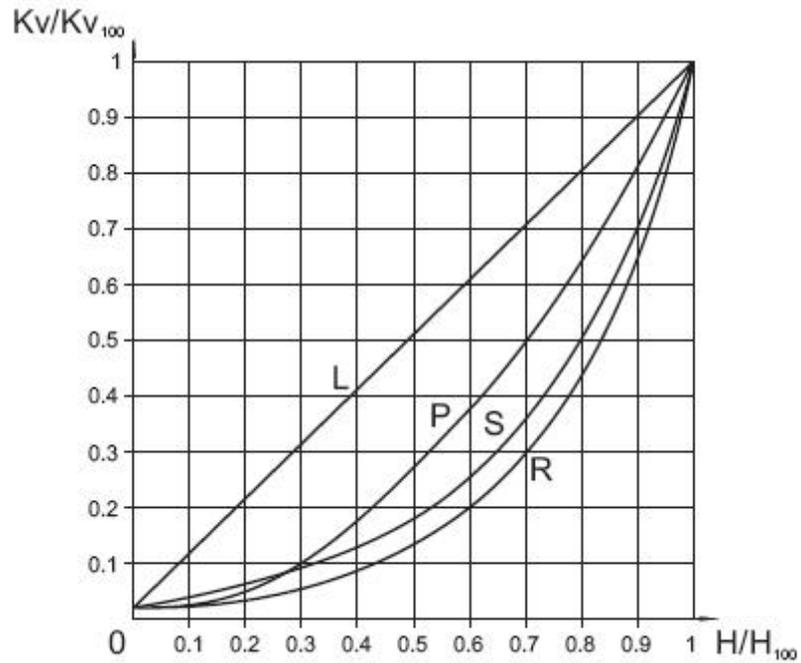
L - linear characteristic

$$Kv/Kv_{100} = 0.0183 + 0.9817 \cdot (H/H_{100})$$

R - equal-percentage characteristic (4-percentage)

$$Kv/Kv_{100} = 0.0183 \cdot e^{(4 \cdot H/H_{100})}$$

P - parabolic characteristic

$$Kv/Kv_{100} = 0.0183 + 0.9817 \cdot (H/H_{100})^2$$

S - LDM spline® characteristic

$$Kv/Kv_{100} = 0.0183 + 0.269 \cdot (H/H_{100}) - 0.380 \cdot (H/H_{100})^2$$
$$+ 1.096 \cdot (H/H_{100})^3 - 0.194 \cdot (H/H_{100})^4$$
$$- 0.265 \cdot (H/H_{100})^5 + 0.443 \cdot (H/H_{100})^6$$

Appendix 7: Time response comparison, old/new governing strategy

set point change	Engine set point before	Engine set point after	temp. set point before	temp. set point after	Delta temp.	Step	New governor strategy			Old governor strategy			Improvement
	RPM/Nm	RPM/Nm	°C	°C	°C		response time -/+1°C (sec)			response time -/+1°C (sec)			
1->2	3000	1500	55	27	-28	down			351			889	**60.5%**
	170	45											
2->3	1500	2000	27	38	11	up			304			219	**-38.8%**
	45	100											
3->4	2000	800	38	20	-18	down			287			1087	**73.6%**
	100	0											
4->5	800	1400	20	38	18	up	No data, time too long					394	
	0	100											
5->6	1400	2500	38	50	12	up			139			289	**51.9%**
	100	155											
6->7	2500	3500	50	55	5	up			50			155	**67.7%**
	155	200											
7->6	3500	2500	55	50	-5	down			60			79	**24.1%**
	200	155											
6->8	2500	1000	50	24.7	-25.3	down			520			883	**41.1%**
	155	37										Average=	**40.02%**

Appendix 8: Bloc diagrams of the controller
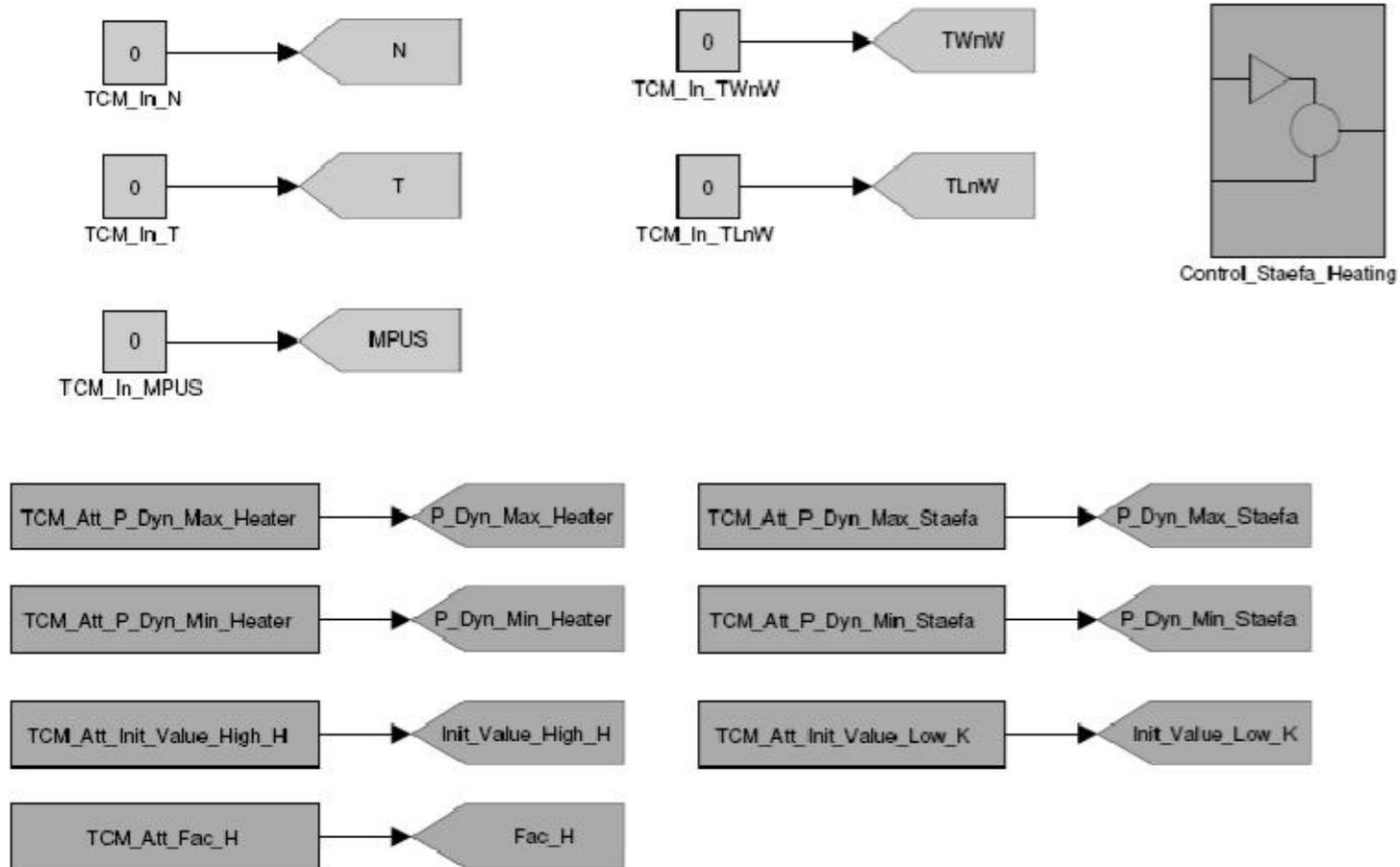
Figure 38: Inputs and attributes definitions
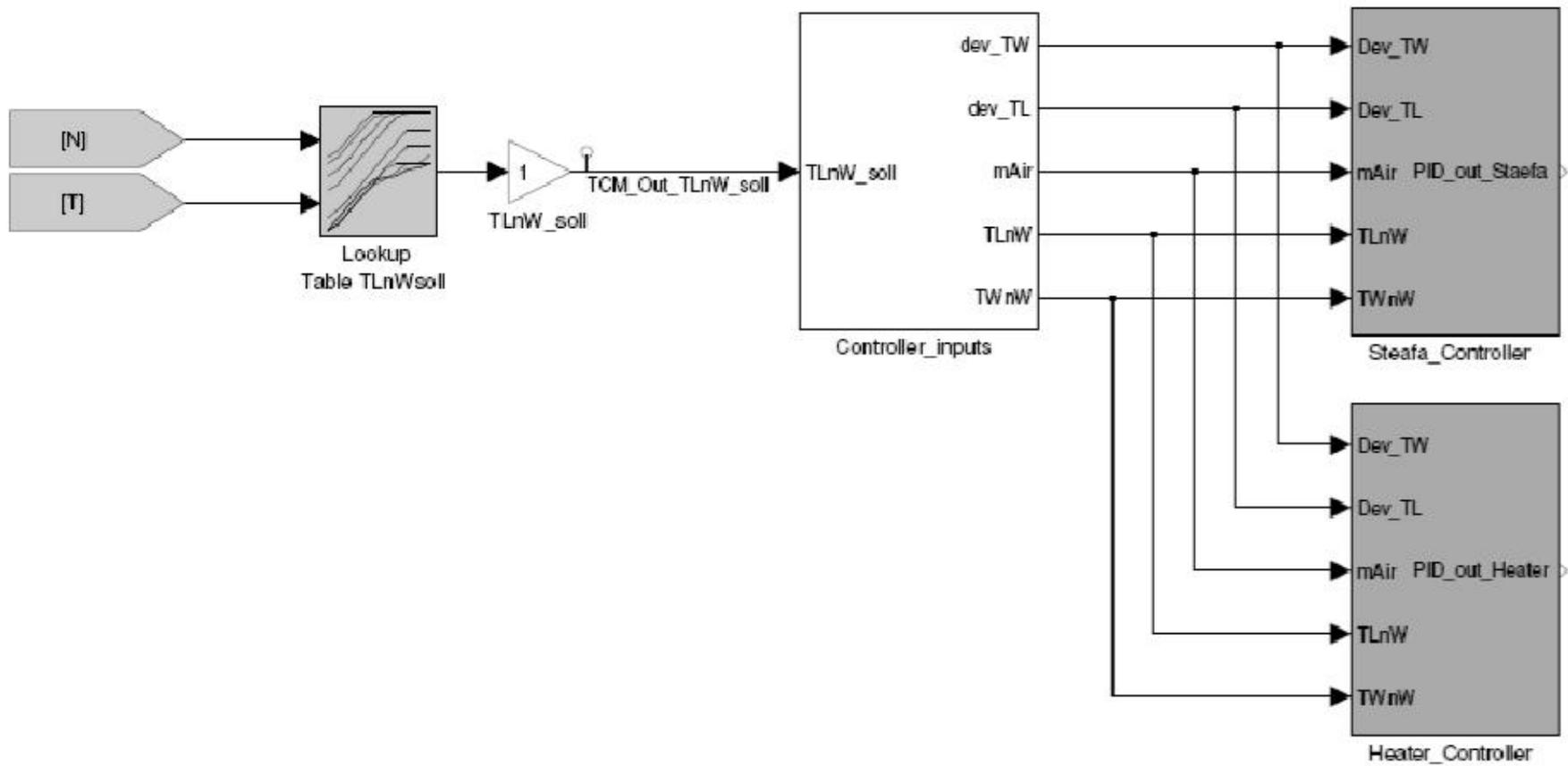
Figure 39: Controller structure
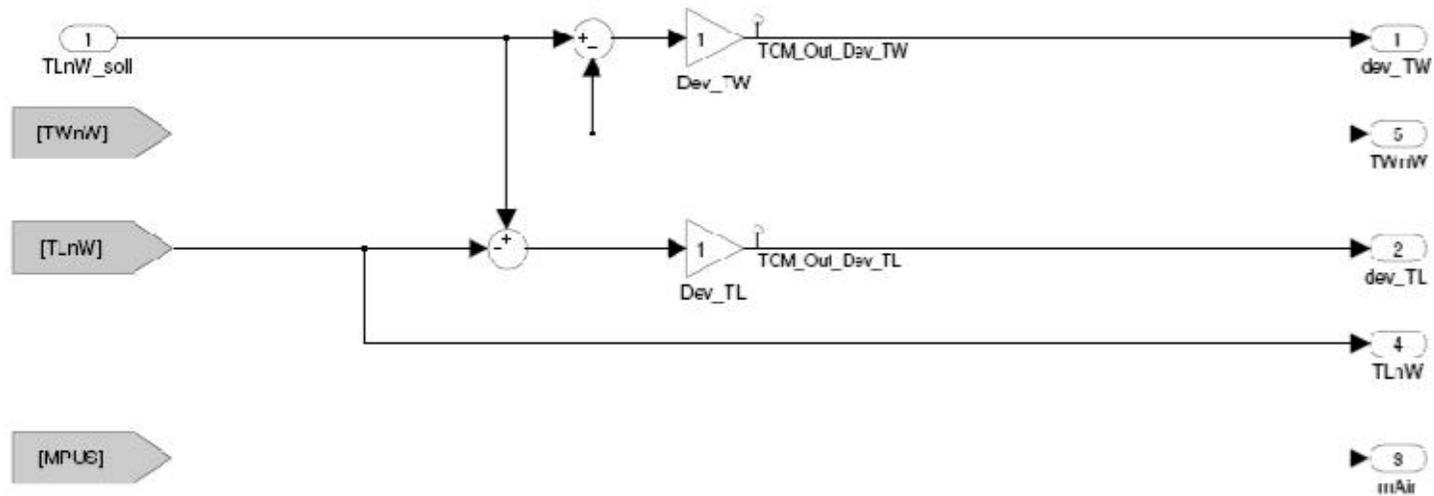
Figure 40: Inputs, and deviation calculation
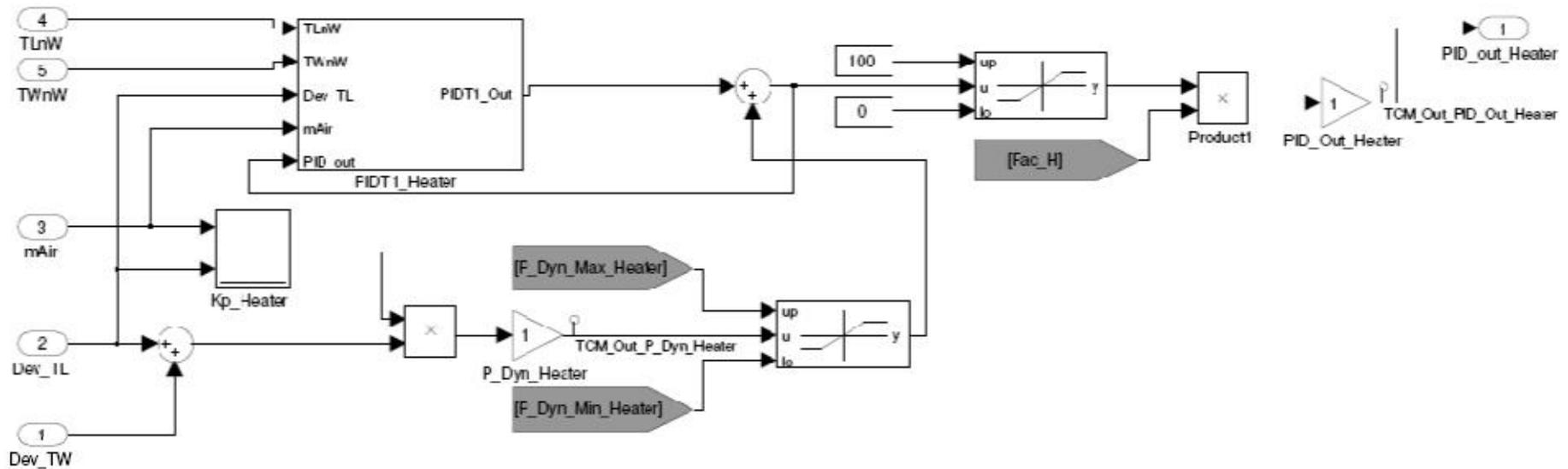
Figure 41: Heater controller
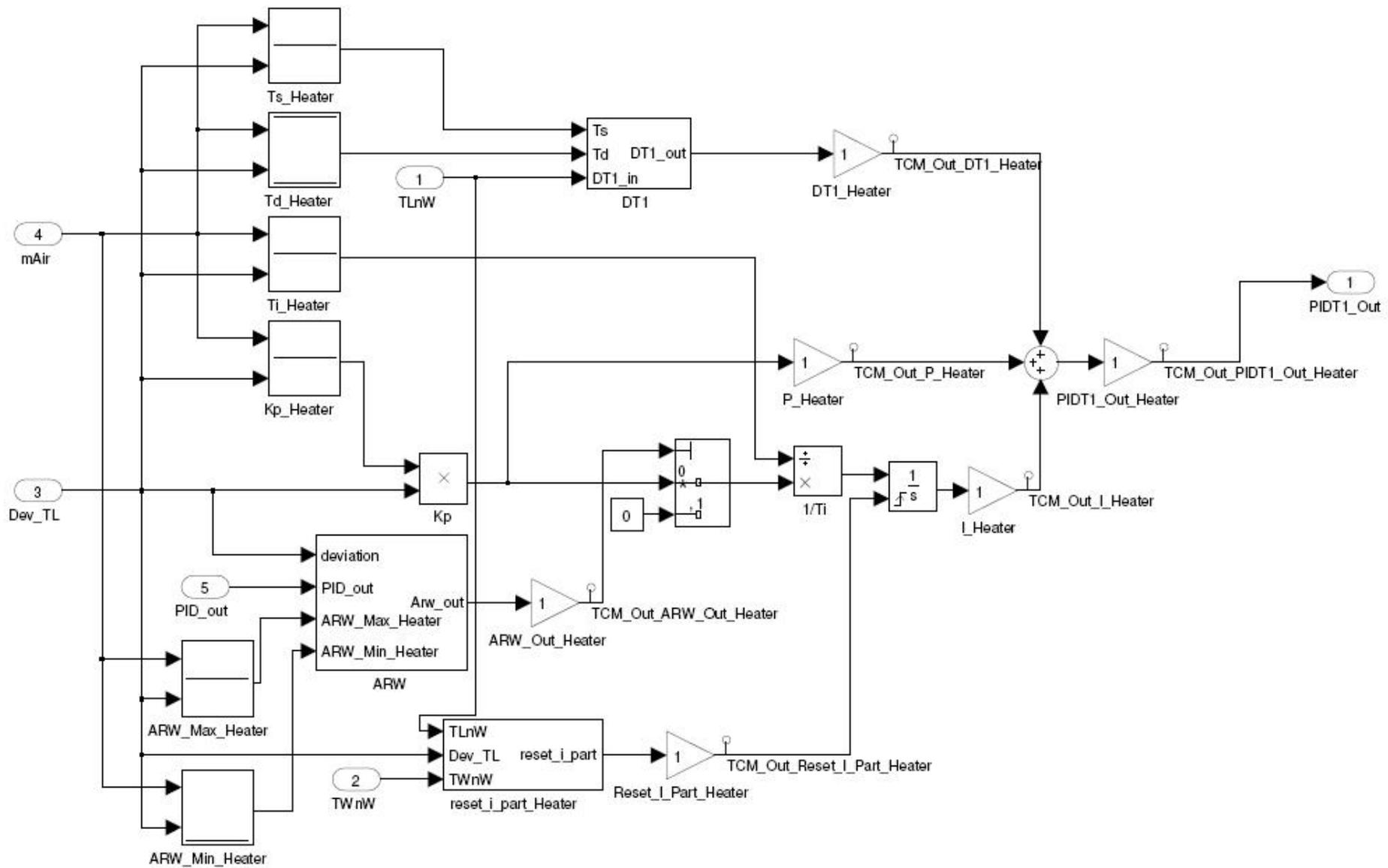
Figure 42: PIDT1 controller of the heater
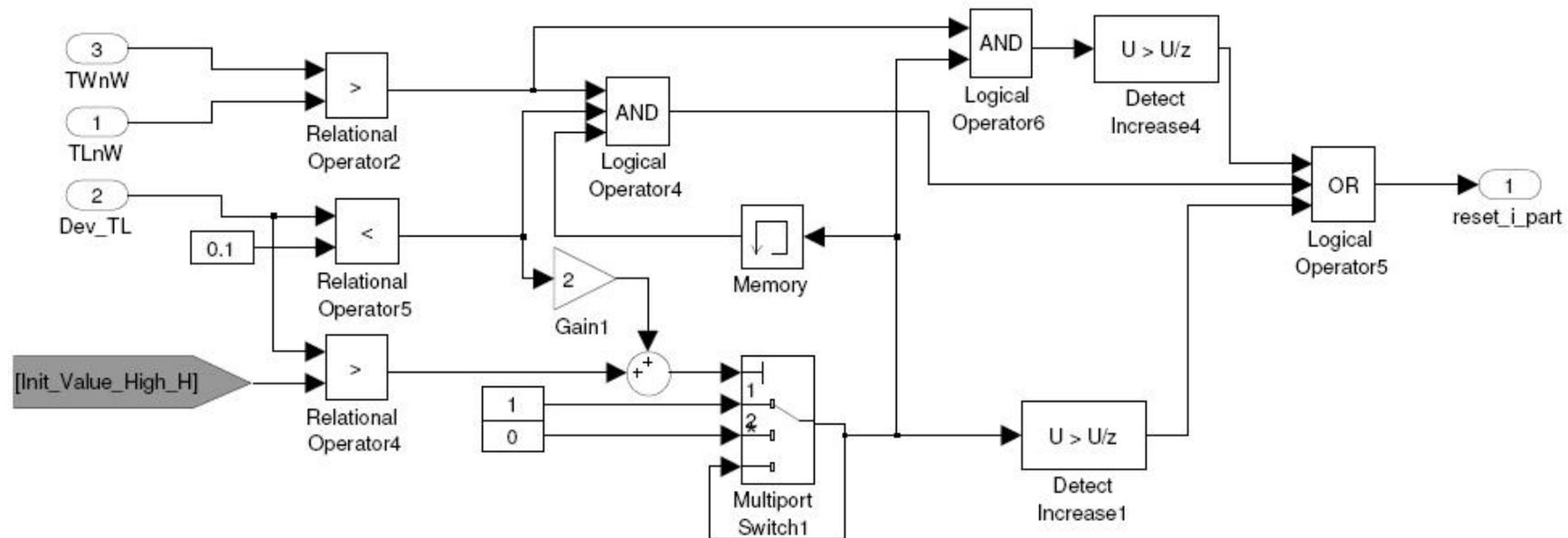
Figure 43: I part initialization bloc diagram

Figure 44: ARW bloc