STEAM BOILER DRUM TRANSIENT RESPONSE AND DISTURBANCE REJECTION

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Steam boilers used in critical applications (turbo-blowers, compressors, steam-driven drums) are required to continue operation even in high load disturbances. At the same time, the disturbances frequently force the parameters out of the safety margins, which lead to boiler shutdown. One of these parameters is the steam drum level, whose response is, in the following paper, fully modeled to allow simulation and insight. Finally, a new control approach is proposed, which increases disturbance rejection.

Keywords: steam boiler, drum level, mathematical model, disturbance rejection

Nomenclature (in context):
Boiler: industrial installation used to generate steam by burning fuel; also, part of this installation situated in the fire's vicinity, where most of the boiling process occurs;
Economizer: water to exhaust gas heat exchanger, used to recover additional heat from low-temperature exhaust gas;
Drum: cylindrical container of special construction, able to withstand high pressure, in which the phase separation between water and steam takes place in a boiler; the water level inside the drum is shortly called drum level;
Superheated steam: steam with 0% humidity, usually with a temperature above water's boiling point at the corresponding pressure; dry steam;
MPC (Model Predictive Control): control strategy that uses a running mathematical model of the process evaluated at future time steps;

1. Introduction

Controlling the drum level of steam boilers has always been a critical point in operation. This measurement is, for any steam boiler, part of the general interlock, which stops the fuel supply. For gas boilers, this requires purging of the fire chamber, which further delays restarting and increases the severity of the shutdown.

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Various control strategies, like using the steam output flow as feedforward to the water flow, have the downside of relying on flow measurements, that tend to be both inaccurate and prone to malfunctions [1]. Moreover, during high load disturbances, the same feedforward system generates commands which push the system further away from normal operation [2], [3]. Complex techniques, like MPC or neural network implementations, require both a highly-dependable process model, which is hard to achieve, and high computational power, which is not always available [4], [5], or views only certain moments of the installation's functioning cycle [6].

Strong research in this area, of K.J. Åström and R.D. Bell [7], has provided a mathematical model for the loop, backed by a good model of the whole system. Further development of this model [8], [9] which also include a superheater and a reduced order model of the turbine, but increase the complexity, would allow complete modelling of the whole loop, but, as the studies themselves conclude, their process implementation is not desirable.

The following approach will provide a simplified alternative where the main parameter to be handled is the boiler pressure. The role of the modelling is to provide a test ground for proving control methods, not necessarily to model the process in all detail.

An experienced human operator knows that, for example, as the pressure lowers the water in the boiler will "swell", and as the pressure rises the water will "shrink" [10]. A mathematical model of the boiler water circuits, considered as **heat exchangers – h.e.** – (economizer, boiler and drum) was constructed in order to use this insight in a quantified amount. All system types have been considered as fairly equivalent in terms of processes of energy, mass and volume exchange.

2. Modelling a water heat exchanger

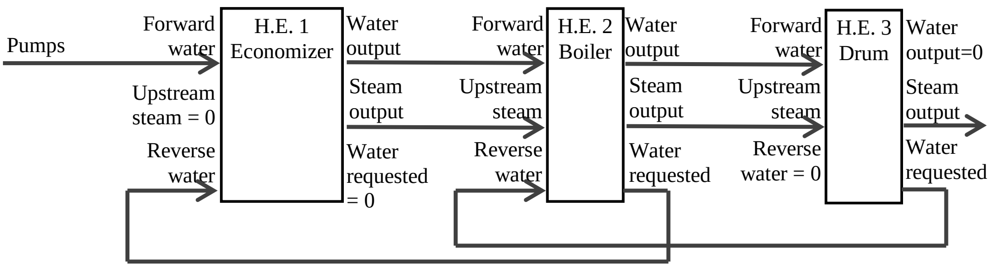

Fig. 1. Mass transfer system for 3 coupled heat exchangers

Each of the water to exhaust gases h.e. can be described by the same processes:

a. Inlet water is heated up to a temperature equal or below the boiling point;

b. After reaching boiling point, further heat transforms water into steam. Phase change also leads to a significant increase in volume;

c. Any steam entering the h.e. can be considered as added heat, in quantity of the difference in enthalpy between the steam phase and existing water phase;

d. Any steam formed in the h.e. or coming from the inlet pipe has a transit time before reaching the outlet;

e. All water and steam properties (e.g. enthalpy, boiling temperature, density, …) are linked to pressure and/or temperature, according to non-linear characteristics. Some have been modeled through interpolation tables, some through polynomial interpolations;

f. The heat exchanger’s internal volume is fixed. All excess water is taken out through the outlet port to the downstream h.e. . If the volume of the water and steam immersed in water is lower than the internal volume and there is another h.e. downstream, the necessary water in order to occupy the available volume will be transferred from downstream (reverse flow);

g. There is a static pressure difference between the drum, boiler and the economizer, due to the mounting position;

2.1. Heat exchange

Intake heat is formed by the added (or subtracted) heat from the water inside the h.e.. It has 4 components:

a. Intake water (“forward water flow”), described by its mass flow m_{in_fw} and temperature t_{in_fw}, being the water received from upstream, with a corresponding enthalpy, h_{in_fw}. For the first h.e., this is the water received from the pumps;

b. Reverse flow water, water received from downstream h.e., when it is available, when the current water volume decreased below the h.e.’s internal volume. It is described by mass flow m_{in_rv} and temperature t_{in_rv}, with a corresponding enthalpy, h_{in_rv};

c. Steam received from the upstream h.e.. It is described by mass flow m_s, with a corresponding enthalpy, h_s;

d. Heat received directly from fuel combustion (radiation and convection), P_{in}.

The enthalpy of all input flow is then linked with the enthalpy of the water inside the h.e., h_{st} [12]:

$$\frac{dQ}{dt} = \frac{dm_{in_{fw}}}{dt} * \left(h_{in_{fw}} - h_{st}\right) + \frac{dm_{in_{rv}}}{dt} * \left(h_{in_{rv}} - h_{st}\right) + \frac{dm_s}{dt} * \left(h_s - h_{st}\right) + P_{in} \quad (1)$$

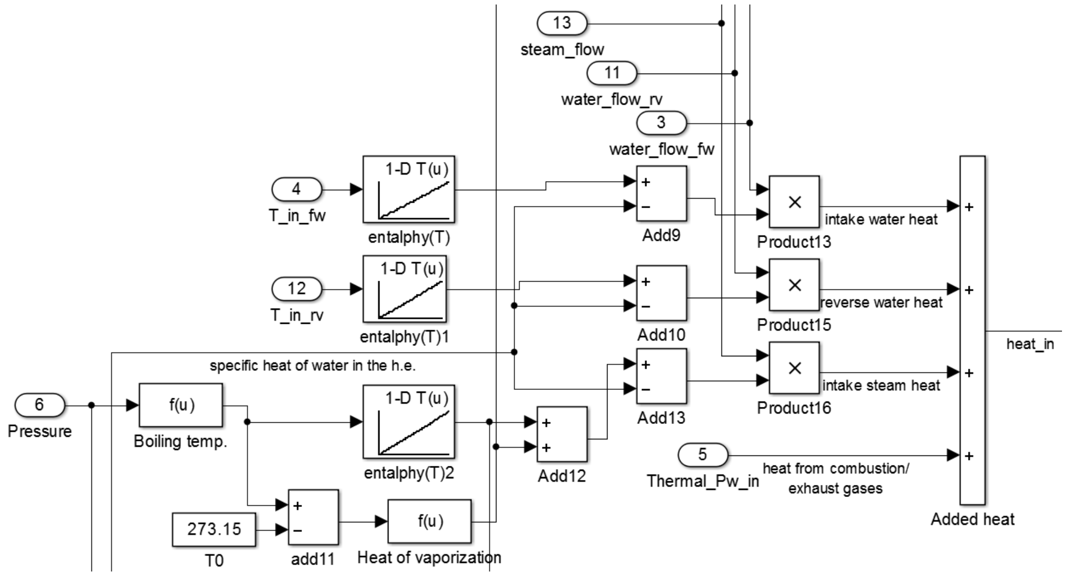

Fig. 2. Matlab model of intake heat computing system

The heat intake system model is drawn in Fig. 2. All water inside the h.e. is considered to have the same temperature, with the temperature variation occurring instantly. The water will heat up to boiling point temperature, consuming the necessary heat. All remaining heat (if any) will be used for boiling.

$$Q_{h\_max} = (h_{boil} - h_{st}) * m_{st} \tag{2}$$

$$\frac{dT}{dt} = \frac{dQ_h}{dt} * \frac{1}{m_{st} * c_{st}} \tag{3}$$

$$\frac{dm_s}{dt} = \frac{dQ_{vap}}{dt} * \frac{1}{\mathrm{h_{boil}}} \tag{4}$$

where

a. $Q_{h\ max}$ is the maximum heat for heating, to raise the water temperature up to the boiling point;

b. h_{boil} is the boiling enthalpy;

c. h_{st} and m_{st} are the enthalpy and mass of the water inside the h.e.;

d. Q_h is the heat for heating the water up to the boiling point, limited by $Q_{h\ max}$;

Q_{vap} is the heat for boiling.

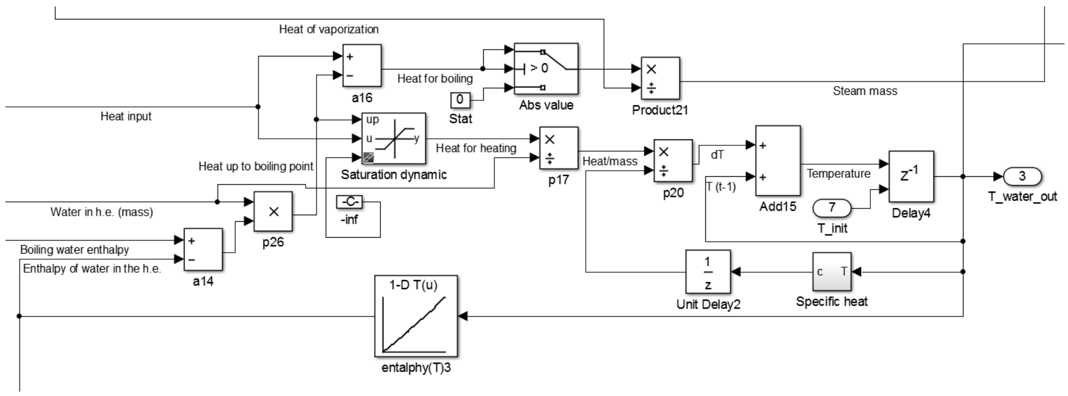

Fig. 3. Matlab model of heat distribution to heating and boiling water

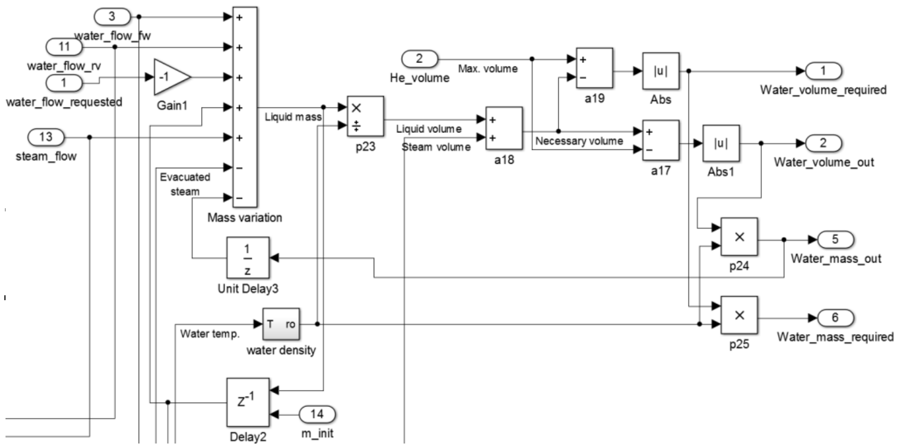

Fig. 4. Matlab model for computing the water mass and volume inside the heat exchanger

2.2. Volume and mass exchange

The water volume inside the h.e. is computed by taking into account all the variation sources:

a. Intake water flow m_{in_fw};
b. Reverse water flow from downstream h.e., m_{in_rv}, if any and if required;
c. Water flow requested by upstream h.e., m_{out_rq};
d. Steam flow mass entering the h.e., m_s;
e. Steam leaving the h.e., m_{sout};
f. Water leaving the h.e., m_{out}.

g. Intake water flow m_{in_fw};
h. Reverse water flow from downstream h.e., m_{in_rv}, if any and if required;
i. Water flow requested by upstream h.e., m_{out_rq};
j. Steam flow mass entering the h.e., m_s;
k. Steam leaving the h.e., m_{sout};
l. Water leaving the h.e., m_{out}.

$$\frac{dm_{st}}{dt} = \frac{dm_{in_{fw}}}{dt} + \frac{dm_{in_{rv}}}{dt} - \frac{dm_{out_{rq}}}{dt} + \frac{dm_s}{dt} - \frac{dm_{sout}}{dt} - \frac{dm_{out}}{dt} \tag{5}$$

All inlet steam is considered condensed. If the heat provided by the inlet steam is enough to vaporize again in the current h.e., it will be again boiled. This solution was required in order to take into account the initial heating stage, when the drum is heated primarily by the steam vaporized in the boiler, while the water inside is at a temperature below boiling point.

The steam volume, computed before, is added to the computed liquid volume to form the necessary volume of the water-steam content. This may be higher than the inside volume of the h.e., so all surplus water is evacuated through the outlet port. In order to simplify interpretation, both the water volume and mass are computed. Steam bubbles in the h.e. are considered as transitory volume [7], [13], [14].

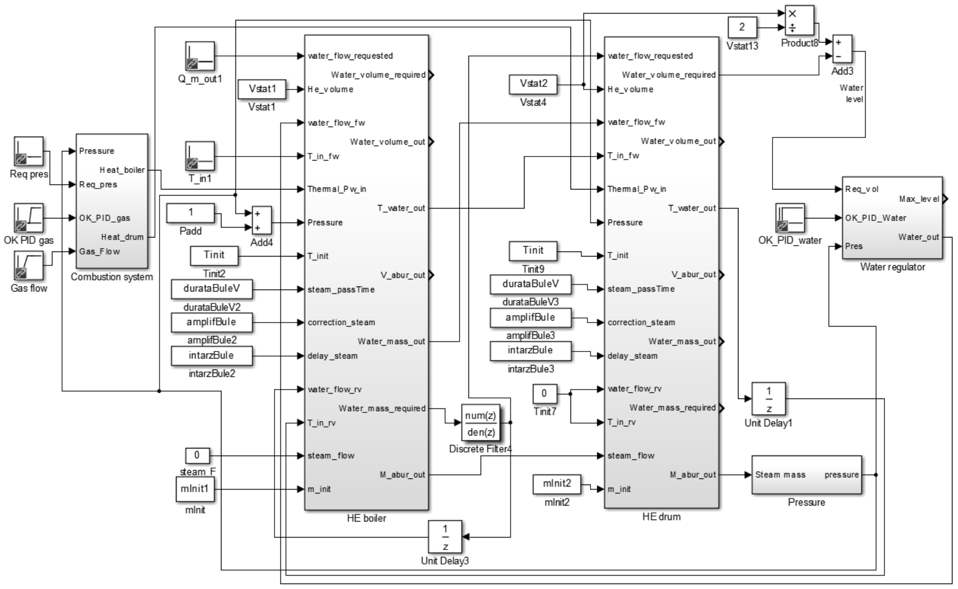

Fig. 5. Simulink model of both heat exchangers with combustion, water and pressure systems

If the water-steam content has a volume lower than the h.e. internal volume, the difference is forwarded to the "Water required" mass and volume outputs, which will be connected to the downstream "Water flow requested" port, if it is available. This is the case for the economizer and boiler h.e., which can take water from downstream.

All the 3 systems discussed before are integrated into a single function block, including filtering and one sample-time delays. This block can model any of the heat exchangers.

3. Modelling the complete loop

The following steps were taken into account in order to model the complete drum level loop:

a. Only the drum and boiler were modeled. Since there is no phase exchange in the economizer, its effect has been neglected. The 2 blocks have been interconnected according to the designations described above;

b. The boiler data and operating point were taken from a 50t/h, 36bar saturated steam boiler, fired on natural gas;

c. The pressure was taken from the steam output flow (square root of the flow), like on most steam consumers;

d. Combustion power (inlet heat) was initially supplied from a ramp, in order to start the boiler with low power, to prevent high oscillations. After the boiling process stabilizes, the power is gradually increased up to reaching the desired working pressure. After reaching the working pressure, the control is passed over to a PID controller. Thermal inertia was simulated through a low pass filter. 75% of the heat is delivered to the boiler, 5% is considered as absorbed by the drum [11];

e. The drum is initially filled with water up to 25%. Like in a real boiler start-up, the level rises fast as the water starts to boil. As it boils, the level lowers back to a point at which intake water is required. In this moment, a PID loop controller starts to regulate the water intake flow, with the drum level at 0 as its target. High disturbances require high speed response from the water intake, above what can be attained on an actual installation, at least without damaging the pumping system. A rate limiter was put in path of the control system, to model the intermediate flow control loop;

f. The boiler level is taken from the output "waterflow_required" of the drum h.e., as described above, which gives the volume of water required to fill the drum. The relation between volume and level is not linear, as the drum is a cylinder:

$$\frac{V}{L} = r^2 * \arccos\left(\frac{r-h}{r}\right) - (r-h) * \sqrt{(2*r*h-h^2)} \tag{6}$$

where V is the water volume, L is the length of the drum, r is the radius of the drum and h is the water level. In order to simplify execution, the results have been numerically pre-calculated and then interpolated. A complete ramp-up to

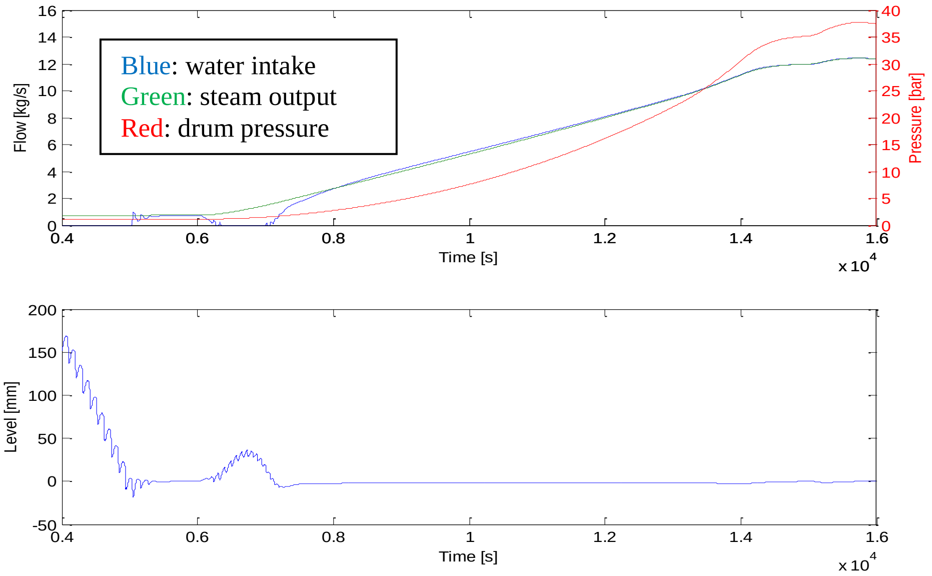

Fig. 6. Boiler startup and automatic control handover

working pressure can be seen in Fig. 6.

At 15500 seconds' steam pressure control is handed over to the PID regulator, which can be seen as another oscillation in the pressure and flow curves. A small change in level can be observed at the moment the fuel flow starts to rise, as more steam evaporates. Operating point conditions (36 bar, 40t/h) are achieved after around 14000 seconds. Both PID loops, water and pressure control, are functional.

4. Pressure disturbance rejection

The operation of this boiler requires high stress caused by the consumer, which often changes its steam requirements. Because of this, a solution was required to keep the boiler level, which was the first affected, in as tight limits as possible during high disturbances. The fastest PID regulator proposed could still not handle the task and it stressed too much the pumping system during normal operation, so a disturbance rejection control was sought after. A simulation of the behavior during a moment when the consumer is demanding more steam, thus reducing the pressure, can be seen in Fig. 7.

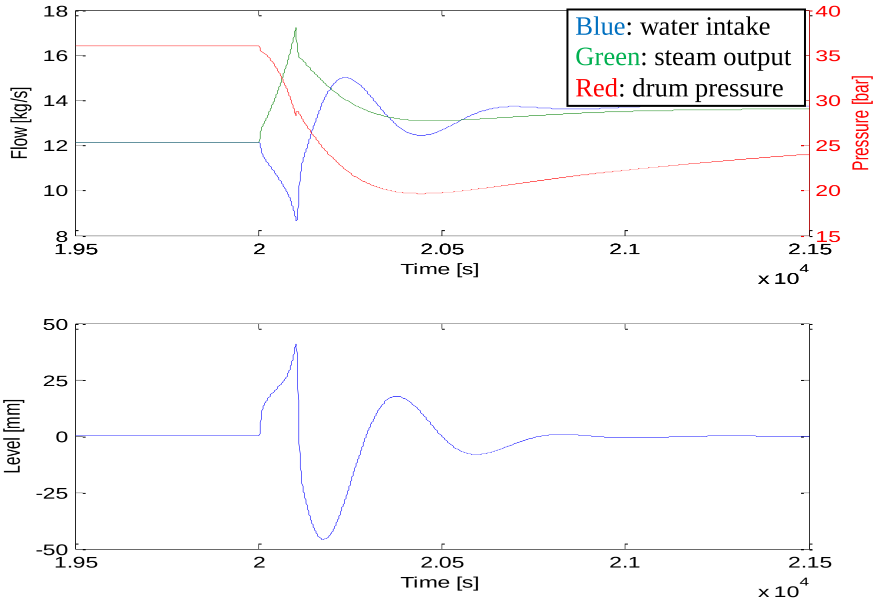

Fig. 7. Load disturbance and PID response

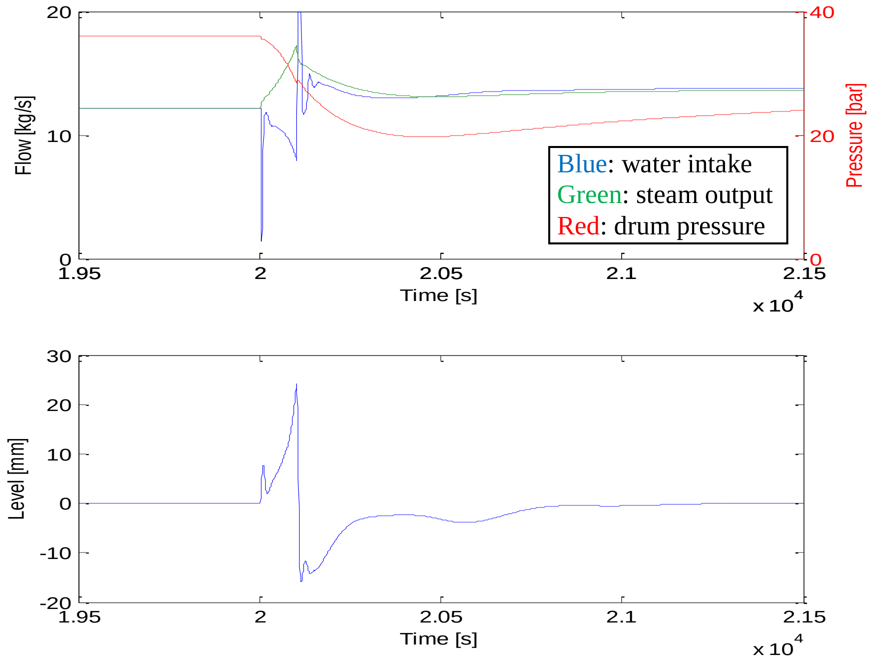

Fig. 8. Level response with feedforward under the same disturbance conditions

As the consumer demands more steam, pressure is quickly lost. The decrease in pressure increases the steam output from 12 to a peak value of 17 kg/s. The PID regulator responds to this, but is still too slow, as the pressure dips it is lowering the water volume in a not so rapid pace. The water level has peaked at 45 mm, with a return dip at -47 mm. As the pressure stabilizes, the level continues to oscillate. In order to cope for this behavior, a feedforward system was designed taking into account the pressure. The proposed solution was to use the second derivative of the pressure squared and signed:

$$dp_2 = \frac{d^2p}{dt^2} \tag{7}$$

$$Q_{ff} = (dp_2)^2 * sign(dp_2) * \; k_{ff} \tag{8}$$

where dp_2 is the second derivative (calculated) of the pressure, Q_{ff} is the water flow calculated for feed-forward and k_{ff} is the scaling factor, chosen after testing. This calculated water flow is added to the command given to the water flow regulator. Squaring the derivative means it will not be very prominent during normal operation, which was tested to be less useful, but it will be stronger during high disturbances. The proposed solution is tested in simulation, Fig. 8, with very good results. With the same disturbance, the level peaked at 27 mm, with a return peak lower than 20 mm. The disturbances were rejected by almost 50%, keeping the boiler in operation.

5. Implementation and results

The above described system was implemented on the 36 bar 50t/h boiler, with the form described above. The results can be seen in Figs. 9 and 10, in a series of tests performed during the drying run of the boiler.

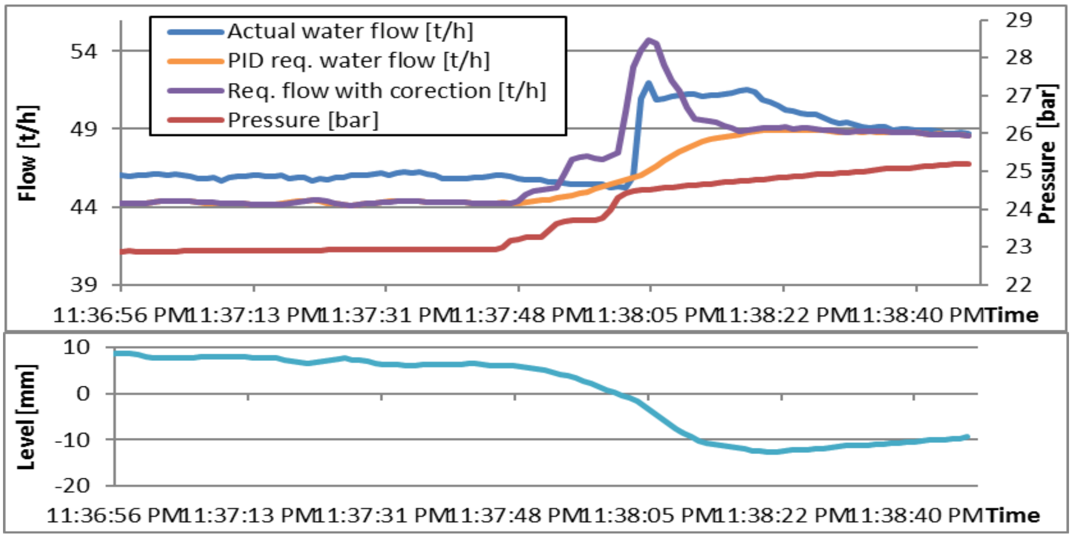

Fig. 9. Actual loop response to pressure increase

As it can be observed, the additional water flow increase due to the feedforward algorithm reduced the level drop. A pressure increase of more than 2 bar in approximately 10 seconds, which might be considered fast, led to only 20mm of variation. The PID controller increased the required flow by no more than 2 t/h, lower than the dead band of 3t/h of the flow controller, while the feedforward system demanded an additional 10 t/h during the rise phase, definitely making an impact.

The other case, which is more frequently found in practice during boiler or load failures, of pressure loss, is shown in Fig. 10.

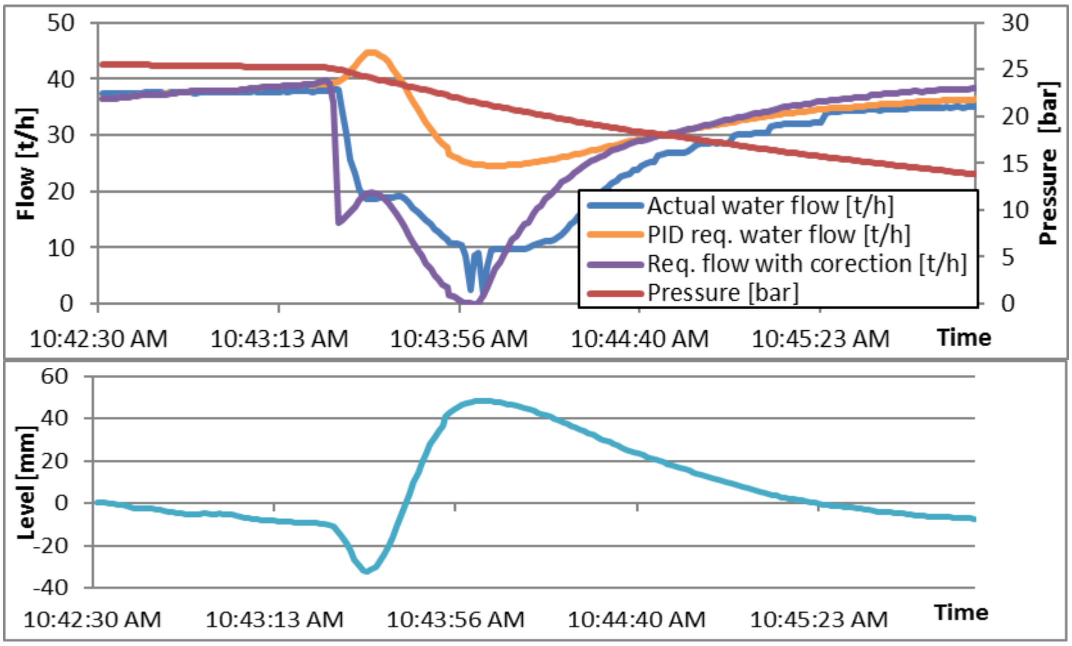

Fig. 10. Actual loop response to critical pressure loss

As it can be observed, the loss in pressure was severe enough to create a considerable increase in the level, of more than 70mm, even with the level reduced to -30mm by the initial feedforward response. The water intake was almost stopped, greatly reducing the level variation, while the PID controller was unable to predict the added variation caused by the change in pressure. Actually, due to the lower level before the change in pressure and to the initial dip caused by the feedforward algorithm, the PID controller increases the required water flow, which is obviously the wrong behavior.

6. Conclusions

The proposed feedforward strategy is, since 09.2015, functional on two 50t/h superheated steam boilers. As it can be seen in the above tests, the results are impressive as the loop handles shut-downs and startups fully automatic, under the highest disturbance levels. Other boilers to which access can be granted for on-line edit will soon follow.

The model behind the results offered a fast solution to test various control strategies on steam boilers. Having flexibility in mind, it can be further expanded to include a single or dual-pass economizer, or to divide the boiler into smaller, more accurate elements (1 or 2 radiation stages, convection stage). A great improvement would be the addition of a super-heater model, which will allow full steam cycle modelling, including the injection temperature.

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