MODEL-BASED CONTROL OF A BOTTOM FIRED MARINE BOILER

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Abstract: This paper focuses on applying model based MIMO control to minimize variations in water level for a specific boiler type. A first principles model is put up. The model is linearized and an LQG controller is designed. Furthermore the benefit of using a steam flow measurement is compared to a strategy relying on estimates of the disturbance. Preliminary tests at the boiler system show that the designed controller is able to control the boiler process. Furthermore it can be concluded that relying on estimates of the steam flow in the control strategy does not decrease the controller performance remarkable. *Copyright ©2005 IFAC*

Keywords: Boilers, Dynamic modelling, LQG control, Estimator, Tests

1. INTRODUCTION

The control of marine boilers mainly focuses on minimizing the variation of steam pressure and water level in the boiler, keeping both variables around some given set point. Up till now this task has been achieved using classical SISO controllers. One using the fuel flow to control the steam pressure and one using the feed water flow to control the water level.

A more efficient control can allow smaller water and steam volumes in the boiler implying lower production costs and a more attractive product.

The specific boiler concerned in the present work is a Mission[TM] OB boiler from AI's product range. The boiler is a bottom fired one pass smoke tube boiler. The boiler consists of a furnace and flue gas pipes surrounded by water. In the top of the boiler steam is led out and feed water is injected. This boiler differs from other boiler designs in two ways: it is bottom fired and the flue gas passes straight through.

The challenge in this work is to minimize the variation of water level to allow smaller boiler geometry without compromising pressure performance. The control problem is complicated by the shrink-and-swell phenomenon which introduces non-minimum phase characteristics in the system. This phenomenon is seen when the steam flow or the feed water flow is abruptly increased or decreased.

The purpose of this paper is to verify if MIMO control is suitable for bottom fired one pass smoke tube boilers. Furthermore the benefit of using the expensive and uncertain steam flow measurement compared to an estimate of this disturbance in the control strategy is investigated. The steam flow influence the shrink-and-swell phenomenon which makes knowledge of this quantity crucial in the control problem.

Tests are performed at a full scale Mission[TM] OB boiler.

2. BOILER MODEL

The boiler model is put up using first principles as was done in (Åström and Bell, 2000) for a drum boiler (for detailed information about the

model derivation refer to (Hvistendahl and Solberg, 2004)).

The model consists of two parts, a boiler model and a model of the actuators. Only the boiler model is derived in this paper.

2.1 System Decomposition

The boiler consists of two logically separated parts. One containing the heating part and one containing the water and steam part. The heating part consists of the furnace and the flue gas pipes. The water and steam part consists of all water and steam in the boiler. The two parts are interconnected by the metal separating them i.e. the furnace jacket and the flue gas pipe jackets.

Sub-system Models The boiler is divided into four sub-systems for the purpose of modelling. A block diagram of the boiler using these sub-systems is shown in 1. In the paper $\dot{m}$ denotes mass
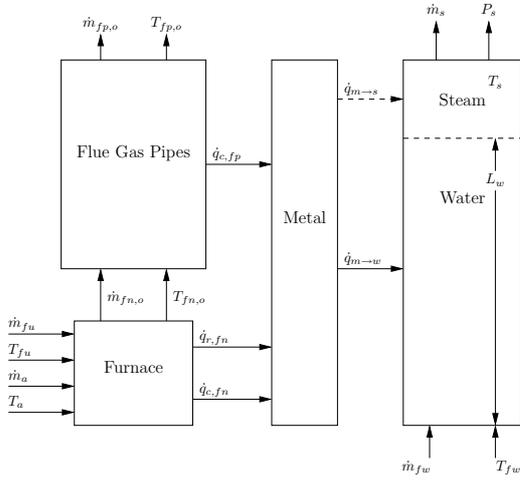

Fig. 1. Block diagram of the four sub-systems in the boiler model. Inputs and outputs are shown in the figure as well as flows and variables connecting the sub-systems.

flow, $\dot{q}$ denotes heat flow, P is pressure, L is level and T is temperature. The subscript a denotes air, fu is fuel, fw is feed water, w is water, s is steam, m is metal, fn is furnace and fp is flue gas pipes.

2.2 The Model Derivation

The derivation is divided into subsections corresponding to the four sub-systems.

The heating part has been divided into four control volumes two in the furnace (one radiation and one convection part) and two in the flue gas pipes (both convection parts). This is done to get a more accurate estimate of the temperature distribution throughout the heating part and to be able to better describe the heat transfer from the flue gas to the metal.

The mean temperature, T_f, in a control volume is set equal to the outlet temperature. The reason for this is that using for instance a bilinear place discretizing method introduces unwanted right half

plane zeros in a linear model. Furthermore the mass flow, $\dot{m}_f$, in a control volume is set equal to the input mass flow. Variations in pressure, P, and specific heat capacity $c_{p,f}$ of the flue gas in the heating part are disregarded whereas the density, ρ_f, variations are considered as these are much larger than variations in pressure and heat capacity in the boiler operating range.

The models of the control volumes in the heating part can be found from the mass and energy balances of each control volume given as:

$$\frac{d\rho_f V}{dt} = \dot{m}_i - \dot{m}_o \qquad (1)$$

$$\frac{d}{dt}\left(\rho_f V c_{p,f} T_o\right) = \dot{m}_i c_{p,f} T_i - \dot{m}_o c_{p,f} T_o - Q \qquad (2)$$

respectively, where Q is the heat delivered to the metal jackets. The subscripts i and o refer to input and output respectively and V is the volume. Combining the two balance equations gives the following equation for the change in output temperature:

$$\frac{dT_o}{dt} = \underbrace{\frac{\dot{m}_i c_{p,f}(T_i - T_o) - Q}{\rho_f V c_{p,f}}}_{h_{\{1,2,3,4\}}} \qquad (3)$$

Before finding the output mass flow $\dot{m}_o$ the change in density ρ_f of the flue gas must be found. The density can be described using the ideal gas equation and is given as:

$$\rho_f = \frac{P M_f}{R(T_o + K)}$$

where the M_f is the molar mass of the flue gas, R is the gas constant and $K = 273.15$ K, see e.g. (Serway and Beichner, 2000). This gives the following equation for the change in density:

$$\frac{d\rho_f}{dt} = \frac{d}{dt}\frac{P M_f}{R(T_o + K)} = -\frac{\rho_f}{T_o + K}\frac{dT_o}{dt}$$

which together with (1) and (3) gives the mass flow:

$$\dot{m}_o = \frac{1}{(T_o + K)c_{p,f}}\left(\dot{m}_i c_{p,f}(T_i + K) - Q\right) \qquad (4)$$

Furnace and Flue gas pipes The models of the four control volumes in the heating part are almost identical and can be described by two equations for each control volume. One for expressing the change in temperature (3) and one for expressing the outlet mass flow being input to the next control volume (4).

For each of the control volumes the heat flow Q of equation 3 is either radiation or convection heat marked $\dot{q}_r$ and $\dot{q}_c$ respectively. Radiation heat $\dot{q}_{r,f1}$ from control volume one is calculated as:

$$\dot{q}_{r,f1} = A_{f1}\alpha_{r,f1}((T_{f1} + K)^4 - (T_m + K)^4)$$

where T_m is the metal temperature, A_{f1} is the volume surface area and $\alpha_{r,f1}$ is the radiation heat transfer constant. Convection heat $\dot{q}_{c,f2}$ from control volume two is calculated as:

$$\dot{q}_{c,f2} = A_{f2}\dot{m}_{f1}^{0.8}\alpha_{c,f2}(T_{f2} - T_m)$$

where $\alpha_{c,f2}$ is the convection heat transfer constant.

Metal The dynamics of accumulated energy in the metal jackets separating flue gas and water/steam can be captured by means of the energy balance. The metal is assumed to have the same temperature in the entire volume as dynamics of thermal conduction for metal are fast. This gives the following model of the metal part:

$$\frac{dT_m}{dt} = \underbrace{\frac{Q_{f\to m} - Q_{m\to w}}{\rho_m V_{m,fj}c_{p,m}}}_{h_5} \tag{5}$$

where $Q_{f\to m} = \dot{q}_{r,f1} + \dot{q}_{c,f2} + \dot{q}_{c,f3} + \dot{q}_{c,f4}$ is the energy flow supplied from the flue gas to the metal and $Q_{m\to w} = A_{mw}(L_w)\alpha_{mw}(T_m - T_s)$ is the energy supplied to the water from the metal. $A_{mw}(L_w)$ is the metal surface area covered by water. Energy supplied to the water steam part above the water surface is considered negligible.

Water/Steam This model has the purpose of describing the steam pressure in the boiler P_s and the water level L_w. The modelling is complicated by the shrink-and-swell phenomenon which is caused by the distribution of steam bubbles under the water surface (this volume is abbreviated V_b). Further the water and steam are assumed only to appear in saturated form.

For the modelling purpose a model of the water and steam part of the boiler as illustrated in figure 2 is used.
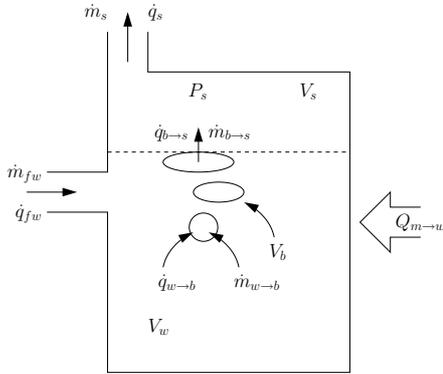

Fig. 2. Model for describing the water and steam part.

The total volume of water and steam in the boiler is given as: $V_t = V_w + V_s + V_b$, where V_w is the water volume, V_s is the volume of the steam space above the water surface and V_b is the volume of the steam bubbles below the water surface.

To capture the dynamics of the water/steam part the total mass and energy balances for the water/steam part are considered. The total mass balance for the water/steam part is given as:

$$\frac{d}{dt}(\rho_s(V_t - V_w) + \rho_w V_w) = \dot{m}_{fw} - \dot{m}_s$$

from which the following expression is found:

$$\underbrace{\left((V_t - V_w)\frac{d\rho_s}{dP_s} + V_w\frac{d\rho_w}{dP_s}\right)}_{f_{66}}\frac{dP_s}{dt} +$$

$$+ \underbrace{(\rho_w - \rho_s)}_{f_{67}}\frac{dV_w}{dt} = \underbrace{\dot{m}_{fw} - \dot{m}_s}_{h_6} \tag{6}$$

The total energy balance for the water/steam part is given as:

$$\frac{d}{dt}(\rho_w V_w(h_w - P_s\nu_w) + \rho_s(V_t - V_w)(h_s - P_s\nu_s) +$$

$$+ \rho_m V_{m,bj}c_{p,m}T_s) = Q_{m\to w} + \dot{q}_{fw} - \dot{q}_s$$

where h is specific enthalpy and ν is specific volume. This leads to the following differential equation:

$$\underbrace{\left(\begin{array}{c}\rho_w V_w\frac{dh_w}{dP_s} + h_w V_w\frac{d\rho_w}{dP_s} + \rho_s(V_t - V_w)\frac{dh_s}{dP_s} + \\ h_s(V_t - V_w)\frac{d\rho_s}{dP_s} - V_t + \rho_m V_{m,bj}c_{p,m}\frac{dT_s}{dP_s}\end{array}\right)}_{f_{76}}\frac{dP_s}{dt} +$$

$$+ \underbrace{(h_w\rho_w - h_s\rho_s)}_{f_{77}}\frac{dV_w}{dt} = \underbrace{\dot{q}_{m\to w} + h_{fw}\dot{m}_{fw} - h_s\dot{m}_s}_{h_7} \tag{7}$$

It should be noticed that the energy in the boiler metal jacket is included in the balance for the water/steam part.

The two equations above only express the pressure and the water volume in the boiler. As the water level of interest in the control problem is given as: $L_w = (V_w + V_b - V_o)/A_{ws}$, another equation is needed for describing the volume of steam bubbles V_b in the water (the water level is measured from the furnace top and V_o is the volume surrounding the furnace and A_{ws} is the water surface area). To do this the mass balances for the steam bubbles and the water are put up as:

$$\frac{d(\rho_s V_b)}{dt} = \dot{m}_{w\to b} - \dot{m}_{b\to s} \tag{8}$$

$$\frac{d(\rho_w V_w)}{dt} = \dot{m}_{fw} - \dot{m}_{w\to b} \tag{9}$$

respectively. The two flows $\dot{m}_{b\to s}$ and $\dot{m}_{w\to b}$ are undetermined. Therefore an empirical equation is introduced. It expresses the amount of steam escaping the water surface as:

$$\dot{m}_{b\to s} = \gamma\frac{V_b}{V_w} + \beta\dot{m}_{w\to b} \tag{10}$$

where β and γ are constants to be estimated. By combining equations 8, 9 and 10 the final differential equation describing the water/steam part can be written as:

$$\underbrace{\left((1-\beta)V_w\frac{d\rho_w}{dP_s} + V_b\frac{d\rho_s}{dP_s}\right)}_{f_{86}}\frac{dP_s}{dt} + \underbrace{(1-\beta)\rho_w}_{f_{87}}\frac{dV_w}{dt} +$$

$$+ \underbrace{\rho_s}_{f_{88}}\frac{dV_b}{dt} = \underbrace{(1-\beta)\dot{m}_{fw} - \gamma\frac{V_b}{V_w}}_{h_8} \tag{11}$$

This equation introduces V_b in the model and hereby the shrink-and-swell phenomenon.

The Nonlinear Model The resulting overall nonlinear model of the boiler can be presented as below.

$$
\underbrace{\begin{bmatrix}
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & f_{66} & f_{67} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & f_{76} & f_{77} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & f_{86} & f_{87} & f_{88} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1
\end{bmatrix}}_{\mathbf{F}(x)}
\underbrace{\begin{bmatrix}
\dot{T}_{f1} \\ \dot{T}_{f2} \\ \dot{T}_{f3} \\ \dot{T}_{f4} \\ \dot{T}_m \\ \dot{P}_s \\ \dot{V}_w \\ \dot{V}_b \\ \dot{T}'_{f4}
\end{bmatrix}}_{\dot{\mathbf{x}}}
=
\underbrace{\begin{bmatrix}
h_1 \\ h_2 \\ h_3 \\ h_4 \\ h_5 \\ h_6 \\ h_7 \\ h_8 \\ h_9
\end{bmatrix}}_{\mathbf{h}(x,u,d)}
\quad (12)
$$

where the first order sensor dynamics of the funnel temperature measurement T'_{f4} are included. The different matrix and vector entries can be found in the equations derived earlier in this section, that is equations 3, 5, 6, 7 and 11.

In practice the steam flow is governed by several valves combined with pipe resistance. Therefore a variable $k(t)$ expressing pipe resistance and valve strokes is introduced. $\dot{m}_s$ is then given as:

$$
\dot{m}_s(t) = k(t)\sqrt{P_s(t) - P_{atm}}
$$

where P_{atm} is the atmospheric pressure and $P_s(t) - P_{atm}$ is the differential pressure over the valve. A parameter estimation has been made to find estimates of the critical parameters in the model such that it reflects the physical boiler as well as possible.

3. CONTROLLER DESIGN

3.1 Strategy

Scheme The control strategy consists of two separate control problems. One main controller, concerned in this paper, in a cascade configuration with two actuator flow controllers for fuel and feed water flow respectively.

Compensator The control strategy is based upon an LQG design. The choice of an LQG design was inspired from a future goal of attempting to implement an MPC (Model-based predictive control) strategy capable of handling limitations in control signals and states. The LQ strategy is comparable to an MPC strategy without constraints. The design is carried out in discrete time. Part of the goal in this control strategy is to compare the benefit of using a steam flow measurement with that of a control strategy relying on estimates of the disturbance. This means that the steam flow disturbance (which is equivalent to the valve stroke k introduced in the model) has to be estimated along with process states. The valve stroke k is the variable determining the load situation of the boiler. A step in k has great influence on the system pressure and water level due to shrink-and-swell effect. A feed-forward in the controller from the valve stroke is presumed to decrease the effects originated from the shrink-and-swell phenomenon. To reconstruct the effect of this feed-forward a good estimate of the valve stroke is needed.

3.2 Model

The model describing the boiler system (12) has the form: $\mathbf{F}(\mathbf{x})\dot{\mathbf{x}} = \mathbf{h}(\mathbf{x}, \mathbf{u}, \mathbf{d})$ where $\mathbf{x}$ is the state vector, $\mathbf{u} = [\dot{m}_{fu}, \dot{m}_{fw}]^T$ is the input vector and $\mathbf{d} = [k, T_{fu}, T_{fw}]^T$ is the disturbance vector. The reason why the air flow $\dot{m}_a$ is not included as an input is that the boiler is constructed with a fixed fuel/air ratio.

Preceding the controller design the model is linearized and the model order is reduced from nine to three leaving the state vector: $\mathbf{x} = [P_s, V_w, V_{sw}]^T$. This new model was found to describe the system sufficiently precisely. The discrete equivalent of the linear model is found and augmented by a model of the actuator controller dynamics resulting in the equation system:

$$
\mathbf{x}_s(k+1) = \boldsymbol{\Phi}_s \mathbf{x}_s(k) + \boldsymbol{\Gamma}_s \mathbf{u}(k) + \mathbf{G}_s \mathbf{d}(k)
$$
$$
\mathbf{y}_s(k) = \begin{bmatrix} \mathbf{y}(k) \\ \mathbf{y}_a(k) \end{bmatrix} = \begin{bmatrix} \mathbf{H}_y & \mathbf{0} \\ \mathbf{0} & \mathbf{H}_a \end{bmatrix} \mathbf{x}_s(k)
$$
$$
= \mathbf{H}_s \mathbf{x}_s(k)
$$

where $\mathbf{y} = [P_s, L_w]$ and $\mathbf{y}_a(k)$ corresponds to outputs from the actuator models.

3.3 Control Law

The goal of the controller is to minimize the variations in the water level L_w and the steam pressure P_s from given set-points. The set-points are constants in normal operation of the boiler hence the purpose is to reject the influence of the disturbances on the two parameters.

The design of the control law follows the principles outlined in (Sørensen, 1995). The goal is to include disturbances in the controller to reject especially the influence of changes in the steam flow valve position k. Furthermore integral action is required to give offset free tracking of the reference. As both disturbances, references and integral action are included in the performance index, the method requires definition of a disturbance model, a reference model and an integral model.

Augmented System Model The original system state vector is now augmented as $\mathbf{x}(k) = [\mathbf{x}_s^T(k), \mathbf{x}_d^T(k), \mathbf{x}_r^T(k), \mathbf{x}_i^T(k)]^T$ giving the model:

$$
\mathbf{x}(k+1) = \begin{bmatrix}
\boldsymbol{\Phi}_s & \mathbf{G}_s\mathbf{H}_d & \mathbf{0} & \mathbf{0} \\
\mathbf{0} & \boldsymbol{\Phi}_d & \mathbf{0} & \mathbf{0} \\
\mathbf{0} & \mathbf{0} & \boldsymbol{\Phi}_r & \mathbf{0} \\
-\mathbf{H}_y & \mathbf{0} & \mathbf{H}_r & \mathbf{I}
\end{bmatrix} \mathbf{x}(k) + \begin{bmatrix}
\boldsymbol{\Gamma}_s \\ \mathbf{0} \\ \mathbf{0} \\ \mathbf{0}
\end{bmatrix} \mathbf{u}(k)
$$
$$
= \boldsymbol{\Phi}\mathbf{x}(k) + \boldsymbol{\Gamma}\mathbf{u}(k) \quad (13)
$$
$$
\mathbf{y}(k) = \begin{bmatrix} \mathbf{H}_y & \mathbf{0} & \mathbf{0} & \mathbf{0} \end{bmatrix} \mathbf{x}(k) = \mathbf{H}\mathbf{x}(k) \quad (14)
$$
$$
\mathbf{e}(k) = \begin{bmatrix} -\mathbf{H}_y & \mathbf{0} & \mathbf{H}_r & \mathbf{0} \end{bmatrix} \mathbf{x}(k) = \mathbf{H}_e\mathbf{x}(k)
$$
$$
\mathbf{x}_i(k) = \begin{bmatrix} \mathbf{0} & \mathbf{0} & \mathbf{0} & \mathbf{I} \end{bmatrix} \mathbf{x}(k) = \mathbf{H}_i\mathbf{x}(k)
$$

A performance index with the purpose of minimizing the errors between reference and output, the integral states and the control signals can be set up as follows:

$$\mathcal{I} = \sum_{k=0}^{\infty} \left(\mathbf{e}^T(k)\mathbf{Q}_{1e}\mathbf{e}(k) + \mathbf{x}_i^T(k)\mathbf{Q}_{1i}\mathbf{x}_i(k) + \right.$$
$$\left. + \mathbf{u}^T(k)\mathbf{Q}_2\mathbf{u}(k) \right)$$

State Feedback Minimizing the performance index results in the well known control law:

$$\mathbf{u}(k) = - \begin{bmatrix} \mathbf{L}_s & \mathbf{L}_d & \mathbf{L}_r & \mathbf{L}_i \end{bmatrix} \mathbf{x}(k) = -\mathbf{L}\mathbf{x}(k)$$

3.4 Estimator

The estimator must reconstruct states not measurable and give a current estimate $\hat{\mathbf{x}}$ of the state vector $\mathbf{x}$. This state estimate will be input to the control law, which becomes $\mathbf{u}(k) = -\mathbf{L}\hat{\mathbf{x}}(k)$.

In the estimator design the two first components $(\mathbf{x_s}(k), \mathbf{x_d}(k))$ of the augmented state vector from equations 13 and 14 are of interest. In the real boiler system both process and sensor noise are present. Including these noise terms a stochastic state space model of system can be presented as:

$$\begin{bmatrix} \mathbf{x}_s(k+1) \\ \mathbf{x}_d(k+1) \end{bmatrix} = \begin{bmatrix} \mathbf{\Phi}_s & \mathbf{G}_s\mathbf{H}_d \\ \mathbf{0} & \mathbf{\Phi}_d \end{bmatrix} \begin{bmatrix} \mathbf{x}_s(k) \\ \mathbf{x}_d(k) \end{bmatrix} +$$
$$+ \begin{bmatrix} \mathbf{\Gamma}_s \\ \mathbf{0} \end{bmatrix} \mathbf{u}(k) + \begin{bmatrix} \mathbf{w}_s(k) \\ \mathbf{w}_d(k) \end{bmatrix} \qquad (15)$$

$$\begin{bmatrix} \mathbf{y}_s(k) \\ \mathbf{y}_d(k) \end{bmatrix} = \begin{bmatrix} \mathbf{H}_s & \mathbf{0} \\ \mathbf{0} & \mathbf{H}_{dy} \end{bmatrix} \begin{bmatrix} \mathbf{x}_s(k) \\ \mathbf{x}_d(k) \end{bmatrix} + \begin{bmatrix} \mathbf{v}_s(k) \\ \mathbf{v}_d(k) \end{bmatrix} (16)$$

where $\mathbf{w}_s(k)$ and $\mathbf{v}_s(k)$ are process noise and measurement noise respectively. Both process and measurement noise are assumed to be uncorrelated zero-mean Gaussian distributed "white" noise sequences. $\mathbf{H}_{dy}$ is a matrix only selecting the temperature disturbances as the steam flow and hence the valve stroke measurement is not available (these temperatures are included in the estimator only to achieve measurement filtering).

3.5 Estimator Gain Design

For derivation of the optimal estimator gain $\mathbf{K}$ see e.g. (Franklin *et al.*, 1998). Here just note that the problem of finding the optimal estimator requires knowledge of the process and sensor noise covariance matrices, $\mathbf{Q}$ and $\mathbf{R}$ respectively.

Here the system described by equations 15 and 16 is considered. Assuming knowledge of $\mathbf{Q}$ and $\mathbf{R}$ the estimator gain can be found.

Covariance Matrices As discussed in (Franklin *et al.*, 1998) knowledge of $\mathbf{R}$ is often given from previous measurements and sensor accuracy whereas $\mathbf{Q}$ is a term accounting for unknown disturbances. The assumption of the process noise being white is often used because it simplifies the resulting optimization problem. Physically the process noise can have any characteristics.

In the present work measurements are available for determining the sensor noise and the covariance matrix $\mathbf{R}$ is designed diagonal containing the variances from the different measurements on the diagonal.

$$\mathbf{R} = \mathrm{diag}([\sigma_z^2(1), ..., \sigma_z^2(p)])$$

where $[\sigma_z^2(1), ..., \sigma_z^2(p)]$ is a vector containing the specific sensor noise variances, where p is the dimension of the measurement vector.

The process noise in the boiler system is regarded as the disturbances, k the steam flow valve stroke, T_{fu} fuel temperature and T_{fw} the feed water temperature. But also unknown disturbances might be present and have to be considered in the design. $\mathbf{w}_d$ is treated as "known" process noise which is changes in the disturbances known to occur. That leaves $\mathbf{w}_s$ regarded as unknown disturbances on the system states. This seems like a reasonable assumption as changes in the disturbance inputs enter the system through the disturbance states.

Of course the variance of $\mathbf{w}_d$ can only be estimated and the problem of the noise being regarded as white still exists. The problem is that changes in the steam flow corresponding to steps from middle load to maximum load are known to occur but these changes can for obvious reasons not be modelled as white noise.

The process noise covariance matrix $\mathbf{Q}$ can now be constructed diagonal with unknown process noise elements corresponding to the system states and reasonable variance expressing disturbance changes corresponding to the disturbance states.

Because of the under determined covariance matrix $\mathbf{Q}$ this is used as a design parameter to achieve the best estimator performance. The matrix is formed as:

$$\mathbf{Q} = \mathrm{diag}([\sigma_{ud}^2(1), ..., \sigma_{ud}^2(n), \sigma_d^2(1), ..., \sigma_d^2(l)])$$

where $[\sigma_{ud}^2(1), ..., \sigma_{ud}^2(n), \sigma_d^2(1), ..., \sigma_d^2(l)]$ is a vector containing variances of the unknown disturbances and the known disturbances respectively. n is the system state dimension and l is the dimension of the disturbance state vector.

3.6 Closed Loop Structure

The closed loop structure of the LQG controller in the form used here is presented in figure 3. Apart from the model matrices the figure contains the estimator gain matrix $\mathbf{K}$ and the feedback gain matrices $\mathbf{L}$'s. $\mathbf{L}_{sd} = [\mathbf{L}_s, \mathbf{L}_d]$ and $\mathbf{I}_y$ is a matrix selecting the outputs P_s and L_w.

The structure of the controller can be found in e.g. (Sørensen, 1995). In this closed loop structure the integral action in the compensator is included in the controller directly on the difference between reference and output signal. Another approach to incorporate the integral action through the estimator is discussed in e.g. (Hvistendahl and Solberg, 2004).

Including the measurement of the steam flow in the controller design is assumed a practicable task and is not illustrated here.
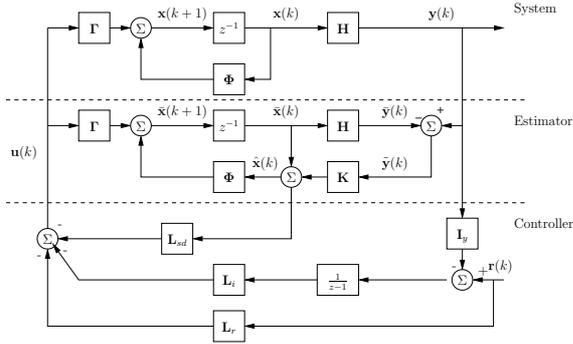
Fig. 3. Closed loop structure of LQG controller.

3.7 Stability

It is well known that an observer reduces the good stability margins exhibited by the LQ controller. For that reason the stability of the designed controllers (with measured steam flow and estimated steam flow) is investigated to insure robustness of the close-loop system and find out if it is necessary to apply LTR (loop transfer recovery).

In figure 4 Nyquist plots of the open-loop system for both controller are shown. From the plots it can be seen that both controllers exhibits good stability margins even with the observer introduced.
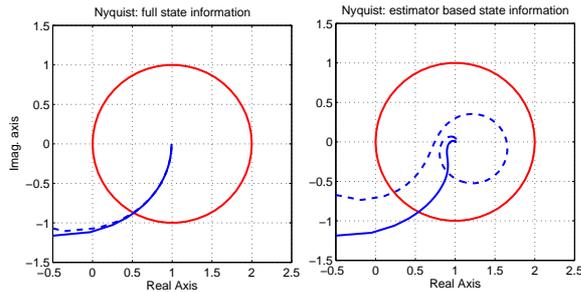

Fig. 4. Nyquist plots of open-loop controlled system based on full state information (left) and estimated state information (right). The solid line represent the controller with measured steam flow and the dashed line the controller with estimated steam flow.

Both controllers showed through simulations to behave and control the system well.

4. RESULTS

Two tests were performed on AI's MISSION™ OB boiler - one for each design. The tests consist of making step changes in the steam valve stroke corresponding to a certain steam flow assuming a pressure of 8 bar in the boiler. The changes are applied with three minutes interval starting at 1700 $\frac{kg}{h}$. The sequence is: 1700-2100-1300-2100-1700 $\frac{kg}{h}$. The test results are shown in figure 5.

From the plots it can be seen that both controllers are able to keep the water level and pressure around the set point. Furthermore it can be seen that there is no remarkable decrease in variations of water level when measuring the steam flow. Whereas performance regarding pressure variations is decreased.
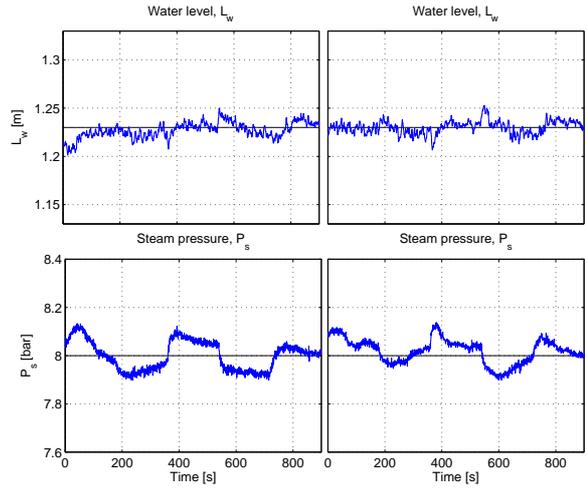

Fig. 5. Verification of controllers and evaluation of estimate of valve stroke k. The plots on the left represent the compensator with estimated k and the plots on the right the compensator with measured k. The top plots show the water level and the bottom plots the steam pressure.

5. DISCUSSION AND FUTURE WORK

It has been verified that model based MIMO control is suitable for control of one specific class of marine boilers (the bottom fired one pass smoke tube boiler). When relying on estimates of the steam flow it was noted that there was no remarkable difference regarding level variations whereas regarding pressure the disturbance is eliminated more slowly. The measurement signals are contaminated by lots of measurement noise corrupting estimates. It is expected that introduction of additional measurement filtering and generation of a better estimate of the disturbance will reduce the influence of the disturbance on the pressure.

Much work still remains in the field of control of marine boilers. The results presented in this paper can be seen as preliminary results. The final goal is to minimize the steam space and water volume in the boiler. To achieve this the final result is expected to use an MPC control strategy as this can handle limitation on states and control signals.

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