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Multi-parametric control and optimisation of a small scale CHP

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The current work presents an application of multi-parametric control to a scheduling of a cogeneration unit in a small hotel. The cogeneration system is described, along with the mathematical model. The latter has been set in the simulation environment of gPROMS. Different response dynamics are assumed for each of the model’s components, while the overall system response is monitored. The two subsystems of the engine and the heat recovery are adequately modeled and dynamic response of the system control variables is investigated. Based on the aforementioned control model, controllers will be designed at a later stage, tested and implemented in this system.

1. Introduction

Small scale cogeneration applications (μCHP) are becoming extremely important in order to achieve energy efficiency goals, especially in the buildings sector. The decentralized production of electricity could also reduce electrical transmission and distribution congestion and alleviate utility peak demand problems. A number of manufacturers worldwide are developing natural gas fired cogeneration devices for single to multiple family residential buildings as well as for small hotels, employing fuel cell based engines, internal combustion engines, and Stirling engines (Knight and Ugursal, 2005). This is partly in response to the financial incentives in the European Union promotes often promoted via dissemination projects.

The presented contribution uses a model of a fossil fueled internal combustion engine cogeneration system. The system under study consists of a natural gas powered 4-stroke engine with power output from 5 to 20 kW of electricity and 10 to 43 kW of heat. It includes an electrical generator and a waste heat recovery system. Power is produced in a reciprocating engine, while the useful heat is extracted from the exhaust gases and the engine’s cooling circuit, via a secondary heat exchanger network. The power is supplied to the grid, while the heat is utilized locally to serve space heating purposes of a hotel.

An important part of achieving high energy efficiency is optimal operation. The current work focuses on the application of model predictive control within the frame of a scheduling problem. The controller has the task to automatically determine which actions should be taken to minimize the operational costs of meeting the hotel electricity

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and heat demands, subject to operational constraints. The model predictive control (MPC) strategy is designed to:

- Take into account the degree of freedom offered by heat storage possibilities;
- Incorporate predictions on electricity and heat demands;
- Incorporate the dynamics and constraints of installed generators and storages.

Scheduling is based on solving at each control step an optimization problem over a prediction horizon minimizing the operating cost subject to system dynamics – including constraints on states, actions, and outputs. At each control step the optimization yields a sequence of actions optimizing expected system behavior over the horizon. Actions (control inputs) are implemented by the system until the next control step, after which the procedure is repeated with new system measurements.

The scheduling problem has been implemented by employing GAMS software. Due to the prediction horizon it can take benefit of knowledge it may have over the future, e.g. forecasted energy demand based on historical data of energy consumption patterns. The focus of the work concerning this case study has been on minimizing daily operational costs of a μCHP system with a prediction horizon of up to several days (Konstantinidis, 2008). The scheduling problem can of course be used for larger horizons and heat and electricity generation units.

The model presented here has been previously implemented in Modelica (Videla and Lie, 2007) for a single-family residential CHP system, based on an Internal Combustion Engine mean-value model (Guzella and Onder, 2004). The current model is set in the gPROMS. Different response dynamics are assumed for each of the model components, while the overall system response is monitored. The two subsystems are adequately modeled and dynamic response of the system control variables is investigated. The system on/off dynamics is investigated in terms of different load conditions.

The system transition times between various states of operation are investigated and control rules are proposed. Further, a multi-parametric control model is in place allowing for the optimization of the plant operation by taking into account the time variability of fuel and electricity prices, as well as meeting electricity/heat demands. The dynamic model follows scheduling in order to optimally satisfy the electricity and heat demand of the hotel. Based on the aforementioned control model, controllers will be designed at a later stage, tested and implemented in this system.

2. μCHP Model

The Ignition Combustion (IC) engine model described here is a mean-value model, without spatially varying variables. The dynamic behavior of the components is represented by ordinary differential equations. The sub-models are continuous control oriented models (COM), neglecting the discrete cycles of the engine and assuming that all processes and effects are spread out over the engine cycle. The sub-models of the cogeneration system are the engine and the heat recovery system. The engine is subdivided into the air system, the fuel system, the torque and power generation system, the engine thermal system and the exhaust emission system. The heat recovery system is comprised by a set of two interconnected heat exchangers, capturing the residual heat from the engine jacket and from the exhaust gases.

2.1 Throttle valve

The valve has been modeled as isenthalpic. The air flow (Eq.1) through the throttle area A_α, is a function of the pressure difference (p_{in} and p_{out} respectively), the discharge coefficient, c_d, and the upstream temperature. A_α is a function of the stem position.

$$\dot{m} = c_d A_\alpha \frac{p_{in}}{\sqrt{R\vartheta_{in}}} \psi\left(\frac{p_{in}}{p_{out}}\right) \tag{1}$$

$$\psi\left(\frac{p_{in}}{p_{out}}\right) = \begin{cases} 1/\sqrt{2} & \text{for } \dfrac{p_{out}}{p_{in}} < \dfrac{1}{2} \\ \sqrt{\dfrac{2p_{out}}{p_{in}}\left[1-\dfrac{p_{out}}{p_{in}}\right]} & \text{for } \dfrac{p_{out}}{p_{in}} \geq \dfrac{1}{2} \end{cases} \tag{2}$$

2.2 Intake manifold

The intake manifold of volume V has been considered isothermal, with a dynamic mass balance for the mass flows entering and exiting and a steady state energy balance.

$$\frac{dp}{dt} = \frac{R\vartheta}{V}\left[\dot{m}_{in} - \dot{m}_{out}\right] \tag{3}$$

$$\vartheta = \vartheta_{in} \tag{4}$$

2.3 Cylinders

The cylinders have been modeled as steady-state pumps aspirating a mass of fluid m_α proportional to the angular velocity ω_e. V_d is the displacement volume, ρ_m is the mixture density, N is the number of revolutions per cycle and η_{vol} the volumetric efficiency. With subscripts e, β and φ the mass flows of the exit gases, the combustion air and the fuel are depicted. λ and σ_o are the normalized and the stoichiometric ratio, respectively.

$$\dot{m}_\alpha = \rho_m \dot{V} = \rho_m \frac{V_d}{2N\pi}\eta_{vol}\omega_e \;,\quad \dot{m}_e = \dot{m}_\alpha \quad \dot{m}_\beta = \lambda\sigma_o\dot{m}_\varphi \quad \dot{m}_\alpha = \dot{m}_\beta + \dot{m}_\varphi \tag{5}$$

The steady state energy balance and the engine efficiencies are written below, along with the approximations for the torque and the electric output of the engine:

$$\dot{H}_{out} = \dot{H}_{in} + \Delta_c H^o \dot{m}_\varphi - \dot{W}_m - \dot{Q}_{g,\omega} \qquad \dot{W}_m = \omega_e T_e \tag{6}$$

$$\begin{gathered} \eta_e = \frac{p_{me}}{p_{m\varphi}} = \frac{T_e 4\pi}{m_\varphi H_\ell} \quad \dot{m}_\varphi = \frac{m_\varphi \omega_e}{4\pi} \\ p_{me} = \frac{T_e 4\pi}{V_d} \qquad p_{m\varphi} = \frac{H_\ell m_\varphi}{V_d} \end{gathered} \tag{7}$$

$$T_e = p_{me}\frac{V_d}{4\pi}\text{, with } T_e = \left(e p_{m\varphi} - p_{me0f} - p_{me0g}\right)\frac{V_d}{4\pi} \text{ (Willan's approximation)} \tag{8}$$

$$ep_{m\varphi} = e\frac{H_\ell \dot{m}_\beta (t-\tau)4\pi}{\lambda\sigma_0\omega_e(t-\tau)V_d} \tag{9}$$

$$\Theta_e \frac{d\omega_e}{dt} = T_e - T_\ell \quad , \quad \frac{dT_\ell}{dt} = \frac{1}{\tau_\ell}\left[-T_\ell + \frac{P_{elec}}{\omega_e \eta_{elec}}\right] \tag{10}$$

where p_{me} and $p_{m\varphi}$ are the break and the fuel mean effective pressures, H_l is the lower heating value of the fuel, p_{me0f} and p_{me0g} the engine friction and the pump losses (after Guzella and Onder, 2004), Θ_e is the inertial term of the flywheel, T_e and T_l are the mean engine and the load torque (with dominant time constant τ_l) and finally P_{elec} and η_{elec} are the power delivered and the conversion energy of the generator.

2.4 Exhaust manifold

The exhaust manifold is modeled by steady state mass and energy balances, at constant pressure:

$$p_e = p_{e,fixed} \quad , \quad \dot{H}_{out} = \dot{H}_{in} - \dot{Q}_{exh} \tag{11}$$

2.5 Engine internal heat recovery system

The heat produced by the engine is extracted by cooling water circulation jacket, and between the engine wall and the engine block. Heat is lost also the atmosphere by the engine block. Heat transfer is modeled through the following balances where the subscripts w, c, b and oil depict wall, cooling water, engine block and oil:

$$\frac{d\vartheta_w}{dt} = \frac{\dot{Q}_{g,w} - \dot{Q}_{w,c}}{c_w m_w} \,,\quad \frac{d\vartheta_c}{dt} = \frac{c_c \dot{m}_c(\vartheta_{ci} - \vartheta_{co}) + \dot{Q}_{w,c} - \dot{Q}_{c,b}}{c_c m_c} \quad ,\quad \frac{d\vartheta_b}{dt} = \frac{(\dot{Q}_{i,f} + \dot{Q}_{b,a}) + \dot{Q}_{c,b}}{c_b m_b + c_{oil} m_{oil}} \tag{12}$$

Moreover, the heat fluxes between the wall and the cooling water, $Q_{w,c}$ and between the cooling water and the engine block, $Q_{c,b}$, are given by the following expressions

$$\dot{Q}_{w,c} = \alpha_c A_c\left[\vartheta_w - \frac{\vartheta_{eo} + \vartheta_{ei}}{2}\right] \quad , \quad \dot{Q}_{c,b} = \alpha_b A_b\left[\frac{\vartheta_{eo} + \vartheta_{ei}}{2} - \vartheta_b\right] \tag{13}$$

while the heat flux due to friction and to the ambient are given by:

$$\dot{Q}_{if} = \frac{p_{me0f} V_d \omega_e}{4\pi} \quad \dot{Q}_{b,a} = A\alpha(\vartheta_b - \vartheta_\alpha) \tag{14}$$

2.6 External heat recovery system

The heat recovery system (two interconnected heat exchangers) is modeled as adiabatic. A diagram of the gPROMS model is presented in Figure 1. The engine and the cooling water circuits may be seen, as well as the storage tank for the heat storage.

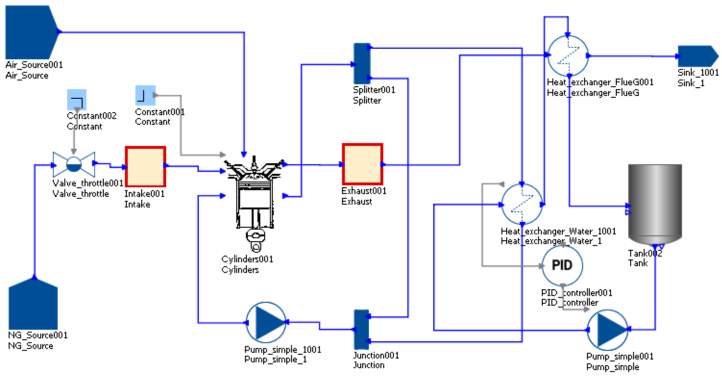

Figure 1: The μCHP model as implemented in the gPROMS simulation environment

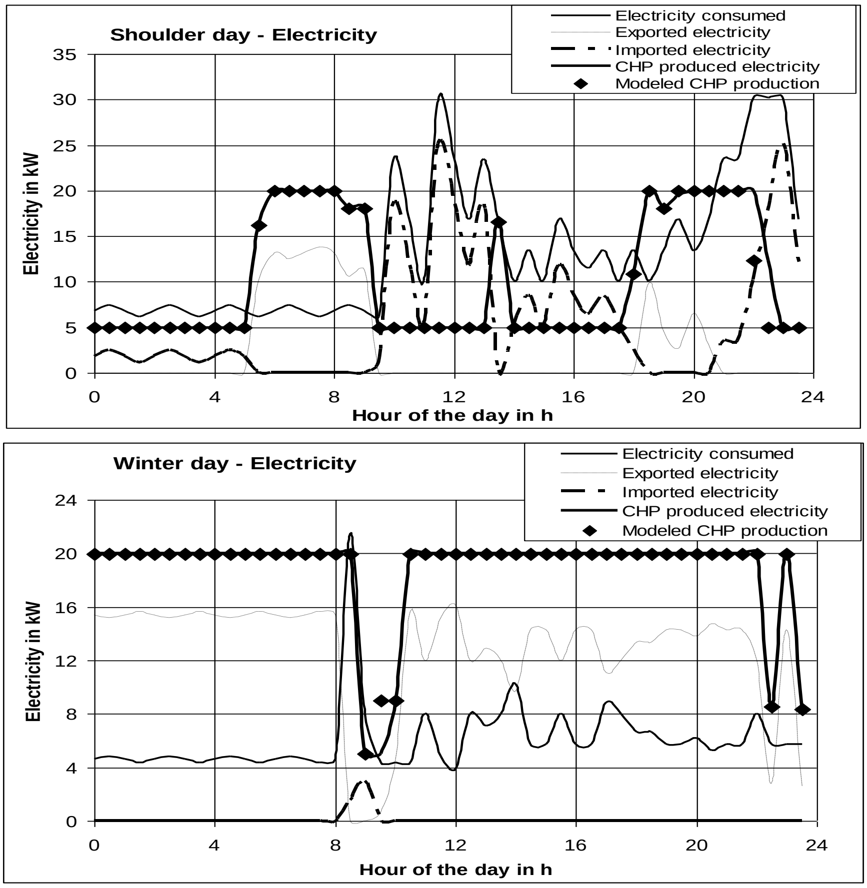

Figure 2: Optimal CHP electricity produced and modelled CHP output for a shoulder and a winter day

3. Simulations

For calculating the heat and electricity demand of the case study, the scheduling MINLP problem for the µCHP serving a small hotel has been implemented in GAMS, assuming a load equivalent to 50 customers and constant fuel and electricity costs. The results from two cases of 24-h prediction horizon are presented in Figure 2 for a shoulder and a winter typical day (Konstantinidis, 2004). The predictions of electricity and heat consumptions are estimated by scaled previous day's consumptions, using next day's customer load, weather conditions and heat stored.

4. Conclusions

The current paper presents a MPC simulation model of a µCHP system for a hotel application. The described method consists of two main parts – identification of the electricity and heat demands, and simulation of the dynamic system operation. The electricity production is optimized to minimize the operating costs. The model is being further developed to implement the MPC controllers.

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