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BOILER FIRING CONTROL DESIGN USING MODEL PREDICTIVE TECHNIQUES [1]

Wen Tan [*], **Tongwen Chen** [**,2], **Horacio J. Marquez** [**]

[*] *Department of Automation, North China Electric Power University, Beijing, China*
[**] *Department of Electrical & Computer Engineering, University of Alberta, Edmonton, AB, Canada T6G 2V4*

Abstract: PI controllers are typically used in industry for boiler firing rate control for their simplicity and ease in tuning; this is the case with the utility boilers at Syncrude Canada's northern Alberta plant. However, instability often occurs in cases of large (load) disturbances, primarily due to firing rate limit constraints. Using a Syncrude Canada's utility boiler as an example, we attempt to redesign the firing rate controller. We show that stability and performance of the closed-loop system can be improved to some extent by properly designed PID controllers. For further improvement, we adopt a model predictive control (MPC) scheme which is capable of handling the firing rate constraints directly; a simple MPC algorithm is implemented on a nonlinear simulation package, and significantly better results are achieved. *Copyright ©2002 IFAC*.

Keywords: Boiler control, model predictive control, PID design, control applications.

1. INTRODUCTION

Drum boilers are used a lot in industry in order to generate steam; their dynamics and control are widely studied, see, for example, (Dukelow, 1991; Maffezzoni, 1997; Astrom and Bell, 2000; Pellegrinetti and Bentsman, 1994). A drum boiler control system usually consists of the following subsystems: combustion control, drum level control, steam temperature control, and furnace draft control. It is well-known that in a drum boiler, the drum level can be efficiently controlled by a three-element level control structure, the steam temperature by a cascade temperature control structure, and the furnace draft pressure by a forced and induced draft control system. These controllers for the subsystems are typically designed separately.

The objective of a boiler combustion system is to ensure that the boiler generates sufficient amount of steam to meet certain steam load demand. Notice that boilers are usually connected with other components/systems and steam demands are usually determined by these systems: For examples, in a unit power plant, a boiler is typically connected to a turbogenerator, and the steam demand on the boiler should follow the electrical power demand associated with the turbogenerator; in a co-generation plant, several boilers are connected to a main header; to maintain the main header pressure, the steam demand on each boiler is distributed from the total steam demand on the main header. So boiler combustion control is often designed together with the connected systems. In a unit power plant, the control of both the boiler combustion and the electrical power generated by the turbogenerator is usually referred to as the *coordinated control*; while in a co-generation plant, the combustion control has two parts: steam control (to regulate the steam flow) and pressure control (to regulate the main header pressure). Since boiler combustion is a slow process, while the demand for steam is usually required to be met as fast as possible, these conflicting facts make the boiler combustion controller design a challenging problem,

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[2] Corresponding author: Tel: (780)4923940; Fax: (780)4921811; Email: tchen@ee.ualberta.ca

especially in cases when there exist physical limit constraints on the firing rate.

Our paper is motivated by a practical situation with Syncrude Canada's utility plant in Fort McMurray, Alberta: Because of the firing rate limit constraints, the existing PI type of combustion controllers exhibit unstable behavior when large disturbances are present in the system. Improving stability and performance of the firing rate control system is the goal of this investigation.

Some discussion of the overall system at the Syncrude utility plant is necessary. The plant currently has three utility boilers, three CO-type boilers, and two once-through steam generators (OTSG). The total steam generated by the boilers is gathered in a 900# header, whose pressure should be maintained at 6.306MPa for normal plant operation. Steam at other levels of pressure (4.24MPa, 1.0682MPa and 0.3584MPa) is obtained by four letdown stations, namely, from 900# header to 600# header, from 600# header to 150# header, from 600# header to 50# header, and from 150# header to 50# header. A simple diagram of the steam system in the utility plant is shown in Figure 1.

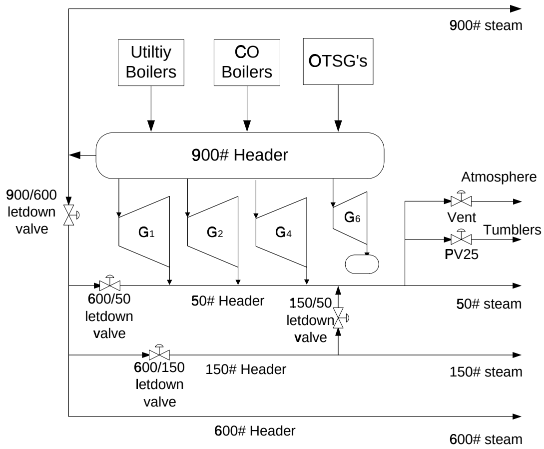


Fig. 1. A simple diagram of steam system in the Syncrude utility plant

For this co-generation plant, the utility boilers are used to regulate the 900# header pressure, while the CO-type boilers are operated in the steam regulation mode. The combustion controller for each boiler is composed of two parts: a master firing controller which generates a firing rate command, and a fuel-air flow controller which generates desired fuel flow rate and air flow rate according to the firing rate command. The existing master firing rate controller for the utility boilers is of PI type, with the parameters manually tuned. As we commented earlier, the closed-loop system exhibits instability when there are large steam demands – its performance in regulating the 900# header pressure in these cases is unacceptable. We show in this paper that we can improve the firing rate control performance by introducing PID type firing rate controllers and fine tuning them. Furthermore, to directly handle the firing rate constraints, we show that a simple model predictive control (MPC) strategy is most effective.

2. MODELING FOR BOILER FIRING CONTROL

To facilitate the control design, we will derive a simple model to be used in the design of the boiler master firing controller.

We note that in the utility plant the CO-type boilers must burn out the coker-off gases from other processes; they are operated with fixed steam load. The OTSG's must burn out the waste gases from the gas turbines; their loads are usually fixed too. So the contributions from CO-type boilers and OTSG's to the 900# header pressure can be ignored. To regulate the 900# header pressure, we need to consider only the utility boilers. Furthermore, since the three utility boilers in the plant are of the same capacity and are operated in parallel with equal load, we need to study just one of them. Based on the operational conditions, we assume that the air and fuel flow controller functions sufficiently well so that we can treat the firing rate as the input to the boiler.

Near operating conditions of the utility boiler, the transfer function from the firing rate, denoted B_{UB}, to the steam flow rate, denoted D_{UB}, can be approximated as a first-order system with a time delay:

$$\Delta D_{UB} = \frac{k_{UB}}{T_{UB}s+1} \mathrm{e}^{-\tau_{UB}s} \Delta B_{UB}. \tag{1}$$

The dynamics of a header can be simply expressed by

$$\frac{dP_{header}}{dt} = \frac{\text{inlet steam} - \text{outlet steam}}{\text{header capacity}},$$

where P_{header} is the header pressure. Based on this we obtain the equation for the 900# header:

$$\Delta P_{900} = \frac{\Delta D_{Boiler} - \Delta D_{900}}{C_{900}s}. \tag{2}$$

Here, C_{900} is the capacity of the 900# header, D_{900} is the total steam demand for the 900# header, including steam demand from electricity generation and other headers, D_{Boiler} is the total steam generated by the boilers. By the argument above, we have $\Delta D_{Boiler} \approx \Delta D_{UB}$.

Replacing ΔD_{Boiler} in (2) by ΔD_{UB} and then eliminating ΔD_{UB} by considering (2), we arrive at the following model for the firing control design:

$$\Delta P_{900} = G\Delta B_{UB} - G_d \Delta D_{900} \tag{3}$$

with

$$G = \frac{k_{UB}}{(T_{UB}s+1)C_{900}s} e^{-\tau_{UB}s},$$
$$G_d = \frac{1}{C_{900}s}.$$

At the operating point where each utility boiler generates 90.6kg/s steam, the parameters involved in the above model are estimated as follows:

$$k_{UB} = 4, \quad T_{UB} = 80, \quad \tau_{UB} = 12, \quad C_{900} = 441.$$

As we mentioned earlier, firing rate limits are important constraints to be considered in control design; these reflect physical characteristics of the boiler which prevents the firing rate from responding as fast as desired. These limits are given by

$$-0.16/60 \le \dot{B}_{UB} \le 0.16/60,$$
$$0 \le B_{UB} \le 1.$$

The above rate limit and saturation constraints are critical for controller design and closed-loop performance.

3. PID CONTROL DESIGN

First, we will try to introduce a derivative term in the existing PI controller and re-tune the resultant PID controller for firing rate control. There are two reasons for doing so: First, we would like to see how much improvement is possible using just PID controllers, which are still quite implementable in practice; second, this sets up a performance benchmark for more advanced control scheme such as the MPC technique which we will study in the next section.

Since the utility boilers are used to maintain the 900# header pressure, the main objective of the firing rate control design is to reject the disturbance, namely, the steam demand at 900# header. Note that the model in (3) is an integrating process. Unlike the case of stable processes, there are few methods for tuning PID controllers for integrating processes. Nevertheless, we have found three methods in the literature which can be directly used here. Table 1 shows the PID parameters tuned by these methods.

Table 1. PID parameters tuned by three different methods

	Tuning Parameters	K_p	T_i	T_d
P-P	$M_r = 4$	3.322	222	51.2
W-C	$\zeta = 0.707, \beta = 0.7$	2.591	732.4	13.3
T-L-T	$\lambda = 0.1$	5.86	162	40.5

The P-P method (Poulin and Pomerleau, 1996) uses the maximum peak resonance (M_r) of the closed-loop system as a specification. A higher M_r indicates that the system is less damped and has larger overshoots. The W-C method (Wang and Cluett, 1997) has two tuning parameters: ζ, the damping factor, and β, the time constant of the desired control system. The T-L-T method (Tan *et al.*, 1998) has one parameter λ which reflects the trade-off between the system's time-domain performance and robustness. The tuning parameters shown in Table 1 are recommended by the references.

Figures 2 shows the responses of the closed-loop systems with the three PID controllers due to a step disturbance of magnitude 1, where the linear model in (3) is used and the controller output (firing rate) constraints are ignored. The PID controller tuned by the W-C method has a very weak integral action so the output takes a long time to return to its setpoint. The T-L-T method has the largest proportional gain and integral action, and thus has the best disturbance rejection ability; however, the controller response is very aggressive. In fact, if we consider the controller rate limit constraints, none of the three PID controllers shown in Table 1 will make the closed loop system stable.

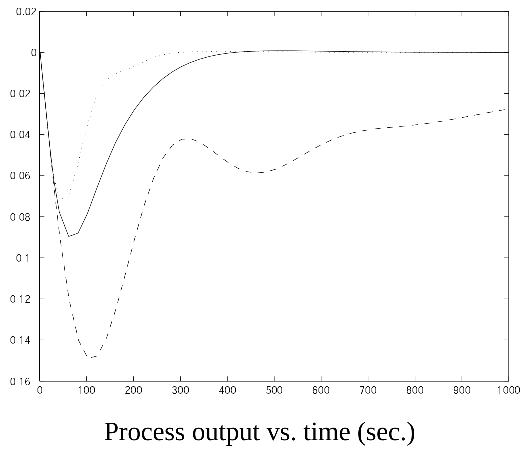


Fig. 2. PID control (solid: P-P; dotted: T-L-T; dashed: W-C)

If we use anti-windup PID configuration (Hanus, 1980) for the PID controllers shown in Table 1, the closed-loop systems become stable but the responses are too oscillatory to be acceptable. One can detune the PID parameters in order to get better responses; however, due to space considerations, we will not include the results in this case.

4. MODEL PREDICTIVE CONTROL DESIGN

In the previous section we mentioned that anti-windup PID controllers can be detuned to accommodate the firing rate constraints. However, two major problems exist:

1. (1) Since the PID parameters are detuned because of constraints, the performance may be poor in cases such as small load disturbances, where the constraints are not active.
2. (2) The anti-windup scheme does not work well for very large (load) disturbances.

In order to take full advantage of the boiler capacity, we need to design a firing rate controller which achieves optimal performance whether the constraints are active or not, and no matter how large the load disturbances are. This calls for a method which *explicitly* handles the constraints in controller design; the model predictive control framework is a good candidate for this purpose.

Model predictive control (MPC) was originally proposed in the process control community and has had wide applications in the process control industry (Camacho and Bordons, 1999); MPC is, perhaps, the most general way of posing process control problems in the time domain, and the only technique accepted in the process industry for handling multivariable control systems. The relevance of MPC in this project lies in two aspects: First, it is capable of directly incorporating input and output constraints, and slew rate constraints in the optimal control law design; second, there exists an efficient solution technique – quadratic programming (QP) – for the resultant optimization problem.

For our application, we adopt a special MPC technique, the so called DMC (dynamic matrix control) approach proposed by Cutler and Ramaker (1980). In this method, open-loop step responses are required for the algorithm, which can be obtained easily by either sampling continuous-time models, or by identification in discrete time. The step response model to be used has the following form:

$$y(t) = \sum_{i=1}^{\infty} g_i \Delta u(t-i), \qquad (4)$$

where y and u are the output and input in discrete time, g_i are the step response coefficients, and Δ is the discrete-time difference operator: $\Delta u(t) = u(t) - u(t-1)$. Note that we have selected the sampling period to be 2 seconds, which is quite reasonable in view of the DCS system currently in use with the Syncrude plant. In order to have a fair comparison with the PID controllers in Section 3, we will use the step response model obtained by directly sampling the continuous-time model discussed in Section 2 in the DMC design.

The objective of DMC is to drive the output as close to the setpoint as possible in a least-squares sense with the possibility of the inclusion of a penalty term on the control moves, i.e., at sampling time t, we need to minimize the following cost function to obtained $\Delta u(t+j-1)$ $(j = 1, 2, \cdots, m-1)$:

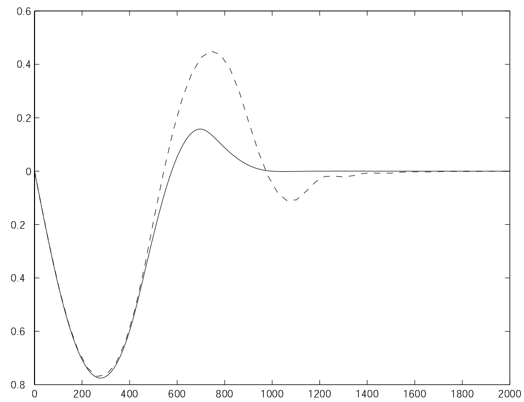
$$\begin{aligned} J(t) = &\sum_{j=1}^{p} \Gamma_y [\hat{y}(t+j|t) - w(t+j)]^2 + \\ &\sum_{j=1}^{m} \Gamma_u [\Delta u(t+j-1)]^2. \end{aligned} \qquad (5)$$

Here $\hat{y}(t+j|t)$ is the j-step prediction of the output, $w(t)$ is the reference trajectory, p is the output prediction horizon, m is the input prediction horizon, Γ_y is the output weighting, and Γ_u is the input weighting. When the optimal control moves are computed, only the first $(\Delta u(t))$ is in fact implemented; and at the next sampling time $t+1$, the optimization process is repeated – a receding horizon control scheme. In the special case when there are no constraints, the solution to the optimization problem can be obtained analytically, and the controller can be implemented easily in a feedback form. However, if there are input or output constraints, the solution requires a QP solver, which is available in, e.g., the MPC Toolbox associated with Matlab.

In our design, we choose the following parameters:

$$p = 100, \quad m = 10, \quad \Gamma_y = 1, \quad \Gamma_u = 1.$$

For a step (load) disturbance of magnitude 1 (using the linear model), we see in simulations that closed-loop responses for the MPC/DMC controller with constraints is almost the same as the PID controller tuned by the P-P method. However, for small and large disturbances, the MPC controller is much better than the PID controller, see, e.g., Figure 3 for a step disturbance of magnitude 2. These clearly show that the MPC controller handles the constraints effectively, and good performance is achieved regardless of the magnitudes of disturbances.



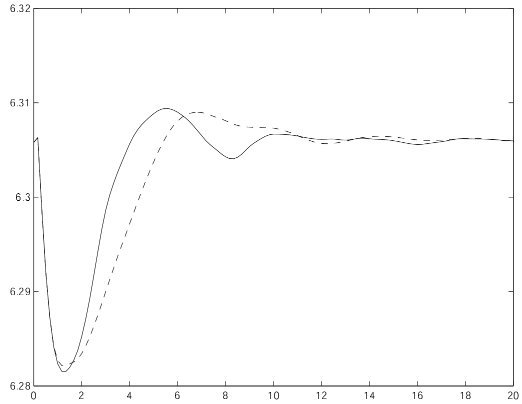
Process output vs. time (sec.)

Fig. 3. MPC (solid) and PID (dash) control

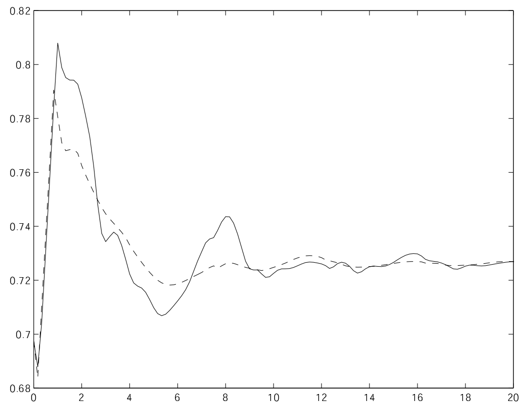
In the Syncrude boiler system, it is a fact that the steam demand (load disturbance) is measured. A natural question to ask is whether the performance of the closed-loop system can be improved if the disturbance model is incorporated in the DMC design. The answer is no; the reason is due to the firing rate constraints: Since the initial response of the controller has already reached its maximum capacity, advance prediction of the disturbance using the model will not help in speeding up the response.

5. CLOSE-TO-REAL SIMULATION

The simulation results reported in the preceding section were based on the linear model discussed in Section 2; while they serve the purpose of illustrating possible performance improvement by re-turning PID controllers and by designing MPC controller in the boiler firing control system, extensive testing on the actual Syncrude boiler system is desirable in order to justify the performance and robustness properties of the designed controllers, the reason being that the actual system is a complex nonlinear system and the linear model is just an approximation of the real plant at a certain operating point. Unfortunately, such real tests are not possible at this stage, mainly due to safety reasons in the utility plant. However, Syncrude has available a complex and nonlinear simulation package called SYNSIM (Rink *et al.*, 1996), which is built in Matlab/Simulink environment and has undertaken thorough tests to closely reflect the real plant. In this section, we report the testing results done on SYNSIM for the performance and robustness of the PID and DMC controllers designed earlier.



(a) 900# Header Pressure(MPa) vs. time(min.)

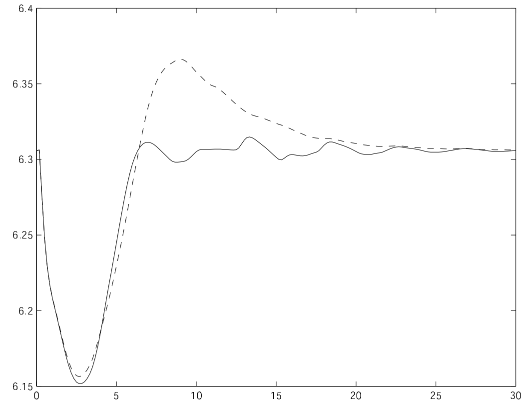


(b) Utility Boiler Firing Rate vs. time(min.)

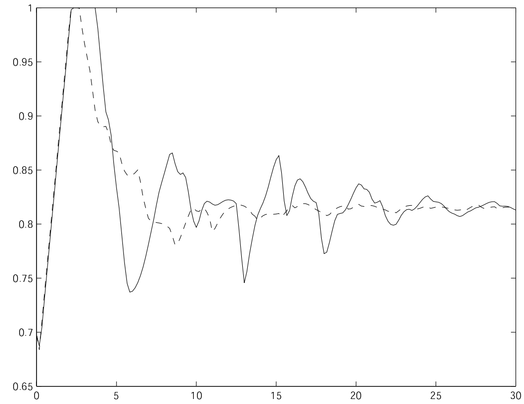
Fig. 4. Comparison of MPC and PID controllers for a 6.3kg/s steam demand change on the 900# header (solid: MPC; dashed: PID)

Figures 4 and 5 show the responses of the 900# header pressure for a step steam demand on 900# header with small and large magnitudes, respectively. The operating condition is specified as follows:

- Two utility boilers, three CO-type boilers, and two OTSG boilers are on-line, with each utility boiler generating steam at 90.6kg/s, each CO-type boiler at 69.4kg/s, and each OTSG at 12.8kg/s.



(a) 900# Header Pressure(MPa) vs. time(min.)



(b) Utility Boiler Firing Rate vs. time(min.)

Fig. 5. Comparison of MPC and PID controllers for a 25.2kg/s steam demand change on the 900# header (solid: MPC; dashed: PID)

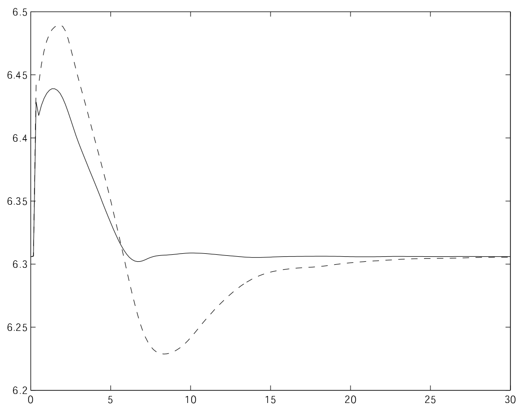
- The total steam load for the 900# header is 415kg/s.
- The total steam loads for the 600#, 150# and 50# headers are 88kg/s, 16.5kg/s, and 215kg/s, respectively.
- The total electricity generated by the steam turbines is 230.2MW.

The responses resemble the simulation results we reported in the preceding sections based on the linear model. The DMC controller has very good performance though the model used in DMC design is a simple step response one.

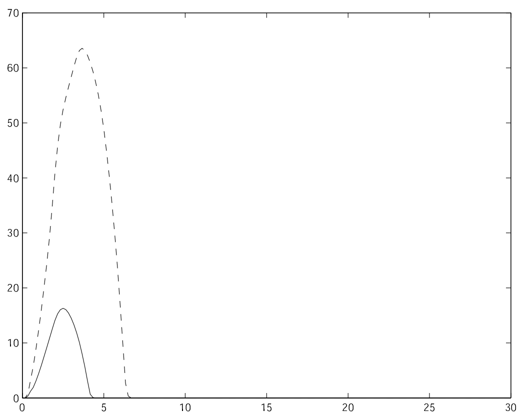
Sudden large changes on the steam demand that we considered could come from failures found in some subsystems in the plant. For example, in the current plant operation, an OTSG or a steam turbine might trip due to some unknown reasons; since an OTSG generates steam at 900# level, while a steam turbine consumes 900# steam, a trip would cause a sudden large change on 900# steam demand. More specifically, at the operating point, an OTSG trip amounts to a 12.8kg/s steam demand increase on 900# header. To maintain system stability in such cases, we need the utility boilers to increase or decrease the steam generation as fast as possible. The response of the utility boiler firing rate controller shows that the DMC controller can take advantage of the full capacity of the utility boiler; the time required for the DMC controller to bring back the 900# header pressure is about half of that for the PID controller. The larger the sudden steam change is, the more advantageous the DMC controller is.

We have conducted simulations on SYNSIM for an OTSG trip and a CO-type boiler trip. The responses of the 900# header pressure for the case of an OTSG trip and a CO-type boiler trip are similar to Figure 5, hence they are omitted to save space.

A steam turbine trip amounts to a steam demand decrease on 900# header; Figures 6 show the case of one steam turbine trip. Since the turbine extracts 83.8kg/s steam, the loss of the steam load is so large that it cannot be handled by two utility boilers. The vent system at 50# header is invoked to send some steam to the atmosphere. As shown in Figure 6(b), using the MPC controller, less steam is sent to the atmosphere, compared with the PID controller; which is quite desirable for environmental considerations. We also observe that although there is less steam sent to the atmosphere, the MPC controller has a considerably better performance than the PID controller.



(a) 900# header pressure (MPa) vs. time (min.)



(b) Vent steam vs. time (min.)

Fig. 6. MPC (solid) and PID (dash) control for a steam turbine trip

We remark that the current PI firing rate controller used in the plant is unable to maintain the system stability in case of a CO-type boiler trip or a steam turbine trip. The SYNSIM testing results clearly show the advantages of the DMC controller over PID type of controllers in the extreme situations.

6. CONCLUSIONS

In this paper, we studied PID and MPC controllers for firing rate control of an industrial boiler in Syncrude Canada's utility plant in Fort McMurray, Alberta. Extensive testing showed that both controllers can improve plant stability and performance due to sudden change in steam demand; but we highly recommend the MPC controller which has superior performance and stability and handles controller constraints effectively.

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