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IFAC-PapersOnLine 49-11 (2016) 511–518

# Optimal Model Predictive Acceleration Controller for a Combustion Engine and Friction Brake Actuated Vehicle

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**Abstract:** This paper investigates how model predictive control can be used to control the acceleration of an over actuated vehicle equipped with a combustion engine and friction brakes. The control problem of keeping appropriate comfort and low energy consumption and simultaneously follow an acceleration reference is described. Vehicle and actuator models are developed and the model predictive controller is tested for an adaptive cruise control cut in scenario in simulation. To be able to quantify the benefit of the proposed model predictive controller, the performance is analyzed and compared with a state of the art PID controller.



*Keywords:* Vehicle Dynamics, Model Based Control, Model Predictive Control, Actuator Coordination, Automotive Control.

## 1. INTRODUCTION

A modern conventional car is an example of a system that is overactuated. To change the speed in the longitudinal direction both friction brakes and combustion engine can be used simultaneously. The friction brake can generate a large negative torque while the combustion engine can generate both negative and positive torques. This makes the car overactuated since the negative torque can be generated by the two different actuators. In modern cars it is also common to have an electric machine which makes the car even more overactuated.

The three properties of the actuators that is of particular interest is the dynamics, controllability and the ranges of the actuators. The dynamics refer to how fast the system responds to a control signal. The meaning of controllability in this context is the expected difference between the requested and received torque on the system from the actuator. The range here refer to the range of torque an actuator can deliver to the system. An overview of the specifications of the actuators is given in Table 1.

Table 1. Overview of the characteristics of the two actuators.

| | Combustion Engine | Friction Brake |
|---|---|---|
| Dynamics | Slow | Fast |
| Controllability | Mediocre | Good |
| Range | [ -small , large] | [-large, 0 ] |

Today the coordination part of control system in cars for the longitudinal propulsion is mostly rule based for the different actuators and the control of the individual actuators is typically done with a PID-controller. The benefits with that solution is the simplicity and the robustness, but the performance is not always optimal.

The goal with the paper is to investigate if it is possible to achieve the same or improved performance with a more sophisticated control structure, a model predictive controller (MPC). An MPC combines the possibility to predict the outcome through an open-loop controller with the stability of a closed-loop controller and gives the optimal solution for a finite horizon optimization problem. Another major benefit of MPC framework is that it can handles constraints in the control signals and states of the system in a very good way. The paper contributes with knowledge in how actuator redundancy should be utilized for best comfort using model-based control.

Many papers have been written about how to optimize the coordination of the actuators and find a global minimum using offline optimization methods. In Lorenzo Serrao [2011], Caiying Shen [2011] and Jinming Liu [2008] dynamic programming (DP) and Pontryagin's maximum principle (PMP) algorithms are presented to illustrate the possible benefits with hybrid electrical vehicles (HEV).

Lorenzo Serrao [2011] and Jinming Liu [2008] have also compared the offline solutions with equivalent consumption minimization strategy (ECMS) which is an instantaneous minimization method and the authors claims that it is possible to implement in real time.


Peer review under responsibility of International Federation of Automatic Control.
10.1016/j.ifacol.2016.08.075

An early MPC approach is used in M.J. West [2003] to control an electric vehicle with multiple energy storage units. The article also describe how zone control can be used in the MPC framework when it is desired to let a variable vary within a given interval. The performance of an MPC for a HEV is compared with both a DP and an ECMS approach in H.A. Borhan [2009]. The conclusion is that the performance is good and there is several advantages such as it is potentially real-time implementable and rather easy to tune.

In Chris Vermillion [2007] and Bjarne Foss [2013] model predictive control allocation (MPCA) is described, an approach to coordinate the actuators for an overactuated system when a specific behavior is desired. The focus in Chris Vermillion [2007] is on how to do this for a system with different limitations and dynamics for the actuators. Karin Uhln [2014] is studying how control allocation can be applied for controling the lateral dynamics for overactuated vehicles, although the papers don't handle any prediction horizon.

In Shengbo Li [2011] it is shown that MPC is used for adaptive speed control in order to minimize energy consumption without sacrificing tracking performance. The use of redundant actuator is however not adressed.

### *1.1 Outline*

The paper is organized as follows. In section 2 the modeling of the vehicle and the actuators are presented. The problem formulation and the MPC algorithm are described in section 3. Section 4 presents the driving scenario and a comparison between the result from the developed MPC and existing PID controller. Finally in section 6 the conclusions are presented.

## 2. MODELING

### *2.1 Actuator modeling*

The controllers internal model of the internal combustion engine (ICE) is represented by a first order system with time constant $T_e$ from input $u_e$ to the output force $F_{\text{eng}}$. A first order system is not optimal to describing the combustion engine but is chosen to keep the complexity of the system down. The dynamic of the combustion engine is also very dependent on the internal states of the engine and cannot be modeled with a higher order system that gives good fit for every case. However we assume that a second order model will improve the performance. The inertia of the powertrain, $F_{\text{pt}}$, is taken into account in the model as the expression in (2) where $J_{\text{eng}}$ is the inertia of the engine, $i$ is the transmission ratio and $a$ is the longitudinal acceleration. The derivation of this is explained in detail in Lars Eriksson [2014].

$$F_{\text{eng}} = \frac{1}{sT_e + 1} u_e + F_{\text{pt,in}} \tag{1}$$

$$F_{\text{pt,in}} = -\frac{J_{\text{eng}} i^2}{r_{\text{w}}} a \tag{2}$$

The friction brake has the ability to convert kinetic energy to heat energy by friction. The brake system builds up a hydraulic pressure during braking, which engages the braking pads to generate a friction force and decelerate the vehicle. When the system requests less braking force on the other hand the pressure must be relieved. That is a significantly faster process than building the pressure. This is why the brake system is modeled as two separate processes as in (3). The model will however be kept the same under one prediction horizon and can only change in the beginning of each time step.

Table 2. Nomenclature used in the paper.

| | |
|---|---|
| $A_a$ | Cross sectional area of car |
| $A_b$ | Contact area of braking pads |
| $a_{\text{ref}}$ | Reference signal in acceleration |
| $C_d$ | Aerodynamic drag coefficient |
| $C_{rr}$ | Rolling resistance coefficient |
| $F_{\text{air}}$ | Longitudinal force from air resistance |
| $F_{\text{brake}}$ | Longitudinal force from brakes |
| $F_{\text{drag}}$ | Air and rolling resistance |
| $F_{\text{eng}}$ | Longitudinal force from engine |
| $F_{\text{pt,in}}$ | Force from powertrain inertia |
| $F_{\text{w,in}}$ | Force from wheels inertia |
| $F_{\text{road load}}$ | Longitudinal force from road load |
| $F_{\text{roll}}$ | Longitudinal force from rolling resistance |
| $F_{\text{slope}}$ | Longitudinal force from slope |
| $g$ | Gravity constant |
| $i$ | Transmission ratio |
| $J_{\text{eng}}$ | Inertia in the engine |
| $J_{\text{w}}$ | Inertia in the wheels |
| $j_{\text{lim}}$ | Jerk limit |
| $k_{\text{cone}}$ | No. of sample before increasing factor |
| $L_b$ | Time delay for brake model |
| $m$ | Mass of car |
| $p$ | Pressure in the braking system |
| $Q_{\text{cone}}$ | Increasing cost for reference deviation |
| $Q_{\text{ref}}$ | Cost for reference deviation |
| $Q_{\text{ref,total}}$ | Total cost for reference |
| $r_b$ | Wheel center to braking pad distance |
| $r_w$ | Wheel radius |
| $s$ | Time derivative operator |
| $T_{b,down}$ | Time constant for brake pressure release |
| $T_{b,up}$ | Time constant for brake pressure build |
| $T_e$ | Time constant for engine model |
| $T_s$ | Sampling time |
| $u$ | Control signal vector |
| $u_b$ | Control signal for brake |
| $u_{\text{b,max}}$ | Maximum braking force |
| $u_{\text{b,min}}$ | Minimum braking force |
| $u_e$ | Control signal for engine |
| $u_{\text{e,max}}$ | Maximum force from the engine |
| $u_{\text{e,min}}$ | Minimum force from the engine |
| $v$ | Longitudinal velocity of car |
| $x$ | State vector |
| $\alpha$ | Slope of road |
| $\epsilon_j$ | Slack variable on the jerk state |
| $\eta_e$ | Time delay for the engine (samples) |
| $\mu$ | Tire to ground friction coefficient |
| $\rho$ | Density of air |
| $\tau_b$ | Torque generated by the brakes |

$$F_{\text{brake}} = \begin{cases} \dfrac{e^{-sL_b}}{sT_{\text{b,up}} + 1} u_b & \text{if} \quad a_{\text{ref}} < a \\ \dfrac{e^{-sL_b}}{sT_{\text{b,down}} + 1} u_b & \text{if} \quad a_{\text{ref}} \geq a \end{cases} \tag{3}$$

The logic that determines which system to use is an estimate on which side of the reference value the actual acceleration is. If the actual acceleration is higher than the reference value it is highly probable that the brake, if used, is going to generate a larger negative torque. For