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Chapter 1

Vehicle Models

1.1 Unicycle model

$$\dot{x} = u_s \cos\theta \tag{1.1}$$
$$\dot{y} = u_s \sin\theta \tag{1.2}$$
$$\dot{\theta} = u_\omega \tag{1.3}$$

1.1.1 Reed Shepp's Car

A Unicycle model can simulate a Reed-Shepp's car with

- $u_s \in \{-1, 0, 1\}$
- Transformation $u_\omega = \frac{\tan\delta}{L}$
- restricting δ to $[-\delta_{max}, \delta_{max}]$ where δ represent the steering angle and $\frac{L}{\tan\delta}$ represents the turning radius. L is the wheel-base.

1.1.2 Dubin's Car

Dubin's car is a simplification of Reed-Shepp's car with $u_s \in \{0, 1\}$

> A unicycle model can simulate a Reed-Shepp's car, Dubin's car and the Single track rear wheel model below when steering is the control input without considering the steering rate. It can also simulate tricycle and differential drive robot. The details are not discussed here explicitly

1.2 Single Track Rear Wheel Model

For a front-wheel driven vehicle with no slip

$$\dot{x} = u_s \cos\theta \tag{1.4}$$
$$\dot{y} = u_s \sin\theta \tag{1.5}$$
$$\dot{\theta} = u_s \frac{\tan\delta_r}{L} \tag{1.6}$$
$$\dot{\delta_r} = u_\delta \tag{1.7}$$

where L is the wheel base and the state represents the rear-wheel, δ_r represents the steering angle at the rear wheels and u_s is the speed of the rear-wheel

The Single track rear wheel model can be further expanded out to include acceleration as

$$\dot{x} = u_s \cos\theta \tag{1.8}$$
$$\dot{y} = u_s \sin\theta \tag{1.9}$$
$$\dot{\theta} = u_s \tan\delta_r / L \tag{1.10}$$
$$\dot{u_s} = u_a (\text{or other first order approximations/reduced order models}) \tag{1.11}$$
$$\dot{\delta_r} = u_\delta \tag{1.12}$$

the control input is $u = [u_a u_\delta]'$ If curvature is the input to the system, the model can be rewritten as

$$\dot{x} = u_s \cos\theta \tag{1.13}$$
$$\dot{y} = u_s \sin\theta \tag{1.14}$$
$$\dot{\theta} = u_s \kappa \tag{1.15}$$
$$\dot{u_s} = u_a (\text{or other first order approximations/reduced order models}) \tag{1.16}$$

where κ indicates the curvature value

1.3 Single Track Front Wheel Model

For a front-wheel driven vehicle with no side slip

$$\dot{x} = u_s \cos(\theta + \delta) \tag{1.17}$$
$$\dot{y} = u_s \sin(\theta + \delta) \tag{1.18}$$
$$\dot{\theta} = u_s \frac{\sin\delta}{L} \tag{1.19}$$
$$\dot{\delta_f} = u_\delta \tag{1.20}$$

where δ_f represents the steering angle at the front wheels wheels. This model can be augmented with acceleration commands instead of speed similar to the equations presented in the rear wheel models.

Furthermore, the models can be rewritten using curvature instead of steering angles or steering rate. Additionally, there are limits on speed, acceleration, steering, steering rate and curvature.

1.4 Model with Slip

Assuming the vehicle can be steered with both front and rear wheels, the model equations with slip β are

$$\dot{x} = u_s \cos(\theta + \beta) \tag{1.21}$$
$$\dot{y} = u_s \sin(\theta + \beta) \tag{1.22}$$
$$\dot{\theta} = u_s \frac{\cos\beta}{l_f + l_r}(\tan\delta_f - \tan\delta_r) \tag{1.23}$$
$$\dot{u_s} = u_a \tag{1.24}$$
$$\beta = \arctan(\frac{l_f \tan\delta_r + l_r \tan\delta_f}{l_f + l_r}) \tag{1.25}$$

where β denotes vehicle slip angle, l_f is the distance of the front wheels to the center of mass, l_r is the distance of the rear wheels to the center of mass, δ_f is the front wheel steer and δ_r is the rear wheel steer

1.4.1 Derivation

From left triangle, applying the sine rule

$$\frac{\sin(\frac{\pi}{2} + \delta_r)}{R} = \frac{sin(\beta - \delta_r)}{l_r} \tag{1.26}$$
$$\implies \frac{l_r}{R} = \sin\beta - \tan\delta_r \cos\beta \tag{1.27}$$

From the triangle on the right, applying the sine rule

$$\frac{\sin(\frac{\pi}{2} - \delta_f)}{R} = \frac{sin(\delta_f - \beta)}{l_f} \tag{1.28}$$
$$\implies \frac{l_f}{R} = \tan\delta_f \cos\beta - \sin\beta \tag{1.29}$$

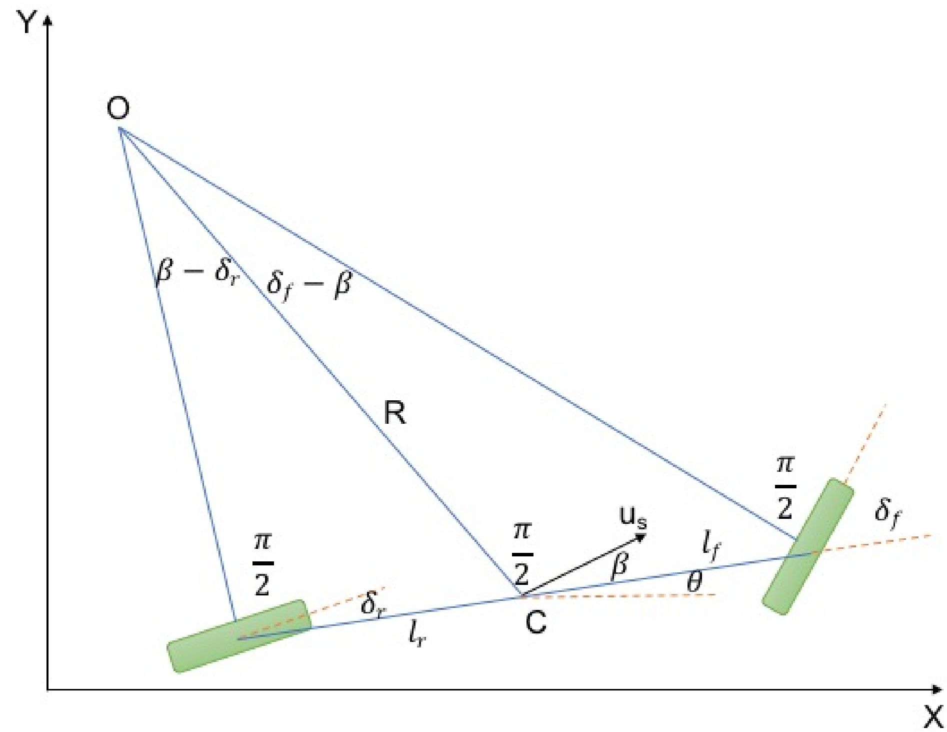

Figure 1.1: Kinematic vehicle model with Slip and front and rear wheel steer. O is the instantaneous center of rotation

Divide equations (1.27) and (1.29), we have

$$\frac{l_r}{l_f} = \frac{\sin\beta - \tan\delta_r\cos\beta}{\tan\delta_f\cos\beta - \sin\beta} \tag{1.30}$$

$$\implies \frac{l_r}{l_f} = \frac{\tan\beta - \tan\delta_r}{\tan\delta_f - \tan\beta} \tag{1.31}$$

$$\implies \tan\beta = \frac{l_r\tan\delta_f + l_f\tan\delta_r}{l_f + l_r} \tag{1.32}$$

Now we know that $\dot{\theta} = \frac{u_s}{R}$. To derive R, we use (1.27). Substituting $\tan\beta$ derived above into 1.27, we have

$$\frac{l_r}{R} = \cos\beta(\tan\beta - \tan\delta_r) \tag{1.33}$$

$$\implies \frac{l_r}{R} = \cos\beta(\frac{l_r\tan\delta_f + l_f\tan\delta_r}{l_f + l_r} - \tan\delta_r) \tag{1.34}$$

$$\implies \frac{l_r}{R} = \frac{\cos\beta}{l_f + l_r}(l_r\tan\delta_f - l_r\tan\delta_r) \tag{1.35}$$

$$\implies \frac{1}{R} = \cos\beta\frac{\tan\delta_f - \tan\delta_r}{l_f + l_r} \tag{1.36}$$

$$\implies \dot{\theta} = \frac{u_s}{R} = u_s\cos\beta\frac{\tan\delta_f - \tan\delta_r}{l_f + l_r} \tag{1.37}$$

As far the equations for x and y go, it is straightforward to see that

$$\dot{x} = u_s\cos(\theta + \beta) \tag{1.38}$$

$$\dot{y} = u_s\sin(\theta + \beta) \tag{1.39}$$

$$\tag{1.40}$$

1.5 Dynamic Lateral and Longitudinal Model

1.5.1 Lateral Dynamics

The lateral dynamics is given by

$$ma_y = m\ddot{y} + u_{sx}\dot{\theta} = F_{yf} + F_{yr} + F_{bank} \tag{1.41}$$

$$I_z\ddot{\theta} = l_fF_{yf} - l_rF_{yr} \tag{1.42}$$

where F_{yf} and F_{yr} denote the lateral tire forces at the front and rear wheels, m is the mass of the vehicle, u_sx is the longitudinal speed of the vehicle at CG.

> Slip is the difference in angle between velocity vector at the wheel and the steering angle

For small slip angles (i.e. lower speeds ¡ 35-40 mph typically for nominal friction conditions i.e no sleet, water, ice etc), they are approximated by (for front wheel drive) a linearly as.

$$F_{yf} = 2C_{\alpha f}(\delta - \theta_{vf}) \tag{1.43}$$

$$F_{yr} = 2C_{\alpha r}(-\theta_{vr}) \tag{1.44}$$

where $C_{\alpha f}$ and $C_{\alpha r}$ represent the cornering stiffness of each of the front and rear wheels respectively. $F_{bank} = mg\sin\phi$, where ϕ is the bank angle, g is the acceleration due to

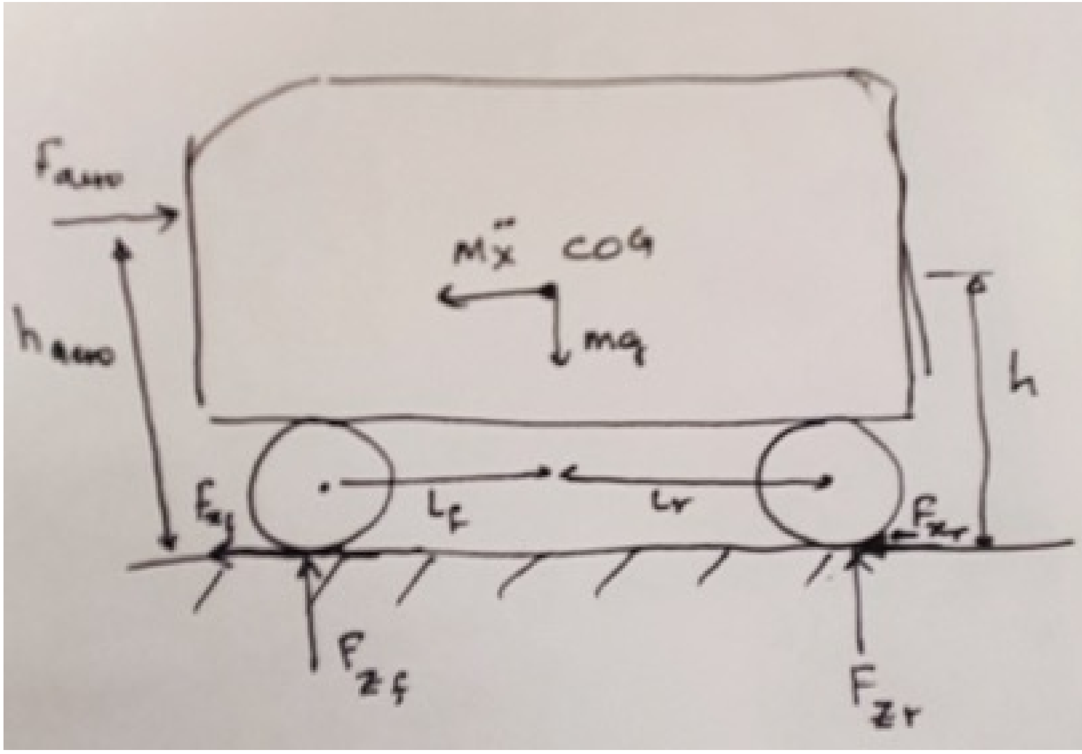

Figure 1.2: Longitudinal forces on the vehicle

gravity The velocity vector at the wheels are given by

$$\tan\theta_{vf} = \frac{u_s y + l_f\dot{\theta}}{u_s x} \tag{1.45}$$

$$\tan\theta_{vr} = \frac{u_s y - l_r\dot{\theta}}{u_s x} \tag{1.46}$$

Using small angle approximations, the above equations can be further reduced to

$$\theta_{vf} = \frac{\dot{y} + l_f\dot{\theta}}{u_s x} \tag{1.47}$$

$$\theta_{vr} = \frac{\dot{y} - l_r\dot{\theta}}{u_s x} \tag{1.48}$$

1.5.2 Longitudinal Dynamics

Ignoring the pitch of the vehicle i.e. assuming flat surface, the longitudinal dynamics is given by

$$m\ddot{x} = F_{xf} + F_{xr} - F_{aero} - R_{xf} - R_{xr} \tag{1.49}$$

where F_{xf} and F_{xr} denote the longitudinal forces at the front and rear wheels respectively, F_{aero} denotes aerodynamics drag force, R_{xf} and R_{xr} denote rolling resistance at the front and rear wheels Aerodynamic drag force is given by

$$F_{aero} = \frac{1}{2}\rho C_d A_F (V_x + V_{wind})^2 \tag{1.50}$$

where ρ is the density of air, C_d is the aerodynamic drag coefficient, V_x is the longitudinal speed of the vehicle, V_wind is the wind velocity and A_F is the frontal area of the vehicle. The longitudinal tire forces are a function of slip, normal load on the tire and the friction coefficient. The longitudinal slip ratio is defined as

$$\sigma_{long} = \frac{r_{eff}\omega_{wheel} - V_x}{V_x} \text{BRAKING} \tag{1.51}$$

$$\sigma_{long} = \frac{r_{eff}\omega_{wheel} - V_x}{r_{eff}\omega_{wheel}} \text{ACCELERATION} \tag{1.52}$$

where ω_{wheel} is the rotational speed of the wheel and r_{eff} is the effective tire radius. When the slip is small, the longitudinal forces are given by

$$F_{xf} = C_{\sigma f}\sigma_{xf}$$
$$F_{xr} = C_{\sigma r}\sigma_{xr}$$

where $C_{\sigma f}$ and $C_{\sigma r}$ represent longitudinal stiffness at front and rear wheels.

The rolling resistance R_{xf} and R_{xr} are approximated using the coefficient of friction and the normal loading force at the front wheel F_{zf} and rear wheel F_{zr} as

$$R_{xf} + R_{xr} = f(F_{zf} + F_{zr})$$

. When the vehicle is travelling along a surface with no gradient, F_{zf} and F_{zr} are in turn calculated by taking moments about the contact points of the front and rear wheels as (see Figure 1.2)

$$F_{zf}(l_f + l_r) + F_{aero}h_{aero} - m\ddot{x}h - mgl_r = 0 \tag{1.53}$$

$$F_{zr}(l_f + l_r) - F_{aero}h_{aero} + m\ddot{x}h - mgl_f = 0 \tag{1.54}$$

The above equations can be straightforwardly extended to the case when the vehicle is pitching while maintaining contact with the ground i.e when the vehicle is moving on an inclined surface. (the terms that correspond mg will have additional component along and perpendicular to the surface)

1.5.3 Error models

With respect to the road

Let's say the objective is to follow the center line of the road. At any given point let θ_{des} be the headong of the centerline. The lateral acceleration desisred at this point would be

$V_x\theta_{des}$. The lateral acceleration of the vehicle in the inertail coordinates is $a_y = \ddot{y} + V_x\theta$ where V_x is the longitudinal speed of the vehicle in the body frame and θ is the heading of the vehicle. The error equations under the assumption of constant longitudinal speed can be written as

$$\dot{e_1} = \ddot{y} + V_x(\dot{\theta} - \dot{\theta_{des}}) \tag{1.55}$$

$$\dot{e_2} = \dot{\theta} - \dot{\theta_{des}} \tag{1.56}$$

where $e_1 = \dot{y} + V_x(\theta - \theta_{des})$ and $e_2 = \theta - \theta_{des}$ The error dynamics can now be obtained using (1.41). The model now can be used for developing steering control law the objective of which would be to stabilize the above system.

1.6 Linearization

Let's say a dynamic system is defined by

$$\dot{x} = f(x, u) \tag{1.57}$$

where x is the state and u is the control input The linearized system is given by

$$\dot{\delta x} = \frac{\partial f}{\partial x}\delta x + \frac{\partial f}{\partial u}\delta u + H.O.T \tag{1.58}$$

H.O.T (Higher order terms) are generally neglected under the assumption that δx and δu are small. Linearization is generally used for stability analysis and for designing controllers for reference and trajectory tracking (example LQR, MPC etc)

First define $\delta x = x - x^*$ and $\delta u = u - u^*$ where x^* and u^* represent the state and control inputs at equilibrium or define the operating points i.e. trim conditions. The derivation proceeds as follows

$$\dot{\delta x} = \dot{x} - \dot{x^*} \tag{1.59}$$

$$\implies \dot{\delta x} = f(x, u) - f(x^*, u^*) \tag{1.60}$$

$$\implies \dot{\delta x} = f(x^* + \delta x, u^* + \delta u) - f(x^*, u^*) \tag{1.61}$$

Taylor series expansion can now be used to derive the equations above.

H.0.T some times are used to a second degree in methods like DDP (Differential Dynamic Programming).We will discuss more about this in another chapter.

Example

Let's take the example of the simplified single track rear wheel model. The equations for the simple car model are given by

$$\dot{x} = u_1 \cos\theta$$

$$\dot{y} = u_1 \sin\theta$$

$$\dot{\theta} = u_1 \frac{\tan u_2}{L}$$

where u_1 is the speed control input and u_2 is the steering control input with L as the wheel base.

Using the derivation, let's linearize the system about the state $[0, 0, pi/4]$ and control $[1, 0]$

$$\dot{\delta x} = 0 \times \delta x + 0 \times \delta y - (u_1 = 1) \times \sin(\theta = \frac{pi}{4}) \times \delta\theta + \delta u_1 \tag{1.62}$$

$$\dot{\delta y} = 0 \times \delta x + 0 \times \delta y + (u_1 = 1) \times \cos(\theta = \frac{pi}{4}) \times \delta\theta + \delta u_1 \tag{1.63}$$

$$\dot{\delta\theta} = 0 \times \delta x + 0 \times \delta y + 0 \times \delta\theta + \frac{\tan(u_2 = 0)}{L} \times \delta u_1 + (u_1 = 1) \times \frac{\sec^2(u_2 = \frac{pi}{4})}{L} \times \delta u_2 \tag{1.64}$$

In matrix form it can be represented as

$$\begin{bmatrix} \dot{\delta x} \\ \dot{\delta y} \\ \dot{\delta\theta} \end{bmatrix} = \begin{bmatrix} 0 & 0 & -\frac{1}{\sqrt{2}} \\ 0 & 0 & \frac{1}{\sqrt{2}} \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \delta x \\ \delta y \\ \delta\theta \end{bmatrix} + \begin{bmatrix} 1 & 0 \\ 1 & 0 \\ 0 & \frac{2}{L} \end{bmatrix} \begin{bmatrix} \delta u_1 \\ \delta u_2 \end{bmatrix} \tag{1.65}$$

The controllability rank condition is not met by the linearized system. However the original nonlinear model i.e. the simple car model is Small Time Locally Controllable (STLC) - we will talk about this later

Chapter 2

Control

2.1 PID

2.2 LQR

Linear Quadratic Regulator is an optimal controller for linear systems and a quadratic cost function. It drives the states to zeros while keeping the control input small. It can be extended to

1. Regularization of non-linear system to a non-zero fixed point
2. Penalize rate of change of control inputs i.e for example jerk
3. Trajectory tracking for non-linear systems
4. Linear Time Varying (LTV) systems
5. Stochastic Systems
6. Affine systems

2.2.1 Problem formulation

The finitie horizon LQR problem is to find control inputs that minimizes

$$J = \sum_{k=0}^{N} (x_k^T Q x_k + u_k^T R u_k)$$

, with $Q \succ 0$ and $R \succ 0$, Q and R are positive semi-definite matrices, subject to

$$x_{k+1} = Ax_k + Bu_k$$

When N is inf, the problem is infinite horizon problem. Since Q and R are positive semi-definite, any x or u that are not zeros will yield a positive cost by definition.

2.2.2 Examples

Leader following

For this problem, let's consider kinematic models in 1D. (The 2D motion can be reduced to under the assumption that the leader and the ego do not have any lateral movement and move on a straight road and appropriately attaching a coordinate system). Assuming our prediction module is perfect, we will model the leader as

$$s_{k+1}^{leader} = s_k^{leader} + v_k^{leader}\delta t \tag{2.1}$$
$$v_{k+1}^{leader} = v_k^{leader} \tag{2.2}$$

v_k is the longitudinal speed of all vehicles and is known at all time steps (from an on-board perception module). The ego or the self vehicle that is controlled is modeled as

$$s_{k+1}^{ego} = s_k^{ego} + v_k^{ego}\ \delta t + 0.5\ a_k^{ego}\ \delta t^2 \tag{2.3}$$
$$v_{k+1}^{ego} = v_k^{ego} + a_k^{ego}\ \delta t \tag{2.4}$$
$$a_{k+1}^{ego} = u_k^{ego} \tag{2.5}$$

where u_k is the control input to our system. The goal is to follow the leader while maintaining a distance greater than or equal to L_{sep}. Let's redefine the state of our system as

$$s_{k+1} = s_{k+1}^{leader} - s_{k+1}^{ego} - L_{sep} \tag{2.6}$$
$$v_{k+1} = v_{k+1}^{leader} - v_{k+1}^{ego} \tag{2.7}$$
$$a_{k+1} = -u_k^{ego} \tag{2.8}$$

Rewriting the matrix form with $x_k = \begin{bmatrix} s_k \\ v_k \\ a_k \end{bmatrix}$, the system is defined by

$$x_{k+1} = Ax_k + Bu_k \tag{2.9}$$

with

$$A = \begin{bmatrix} 1 & \delta t & 0.5\delta t^2 \\ 0 & 1 & \delta t \\ 0 & 0 & 0 \end{bmatrix}$$

and

$$B = \begin{bmatrix} 0 \\ 0 \\ -1 \end{bmatrix}$$

. The goal is to drive the system states to zeros while minimizing the control effort. In this formulation, the control effort gets penalized twice as it is part of the state as well. So Q and R should be chosen carefully.

> Notice that in the formulation, there is nothing preventing ego vehicle from crossing the leader and then maintaining L_{sep} apart from the control cost. There is also nothing preventing the system from commanding high acceleration or deceleration between consecutive steps $\implies$ uncomfortable ride. (which could happen based on the initial conditions)

One way to address the second problem is to increase entries in R i.e. penalize control effort a lot but this will also imply slow convergence and still nothing prevents it from giving large δu. To address this issue one can penalize δu in the cost function. The problem can be reformulated with δu as the control input i.e rate of change of acceleration as control input. The equations are now

$$s_{k+1} = s_{k+1}^{leader} - s_{k+1}^{ego} - L_{sep} \tag{2.10}$$
$$v_{k+1} = v_{k+1}^{leader} - v_{k+1}^{ego} \tag{2.11}$$
$$a_{k+1} = a_k + u_{k-1}^{ego} - u_k^{ego} \tag{2.12}$$
$$\implies a_{k+1} = a_k - \Delta u_k \tag{2.13}$$

where $\Delta u_k = u_k^{ego} - u_{k-1}^{ego}$ In matrix form the A and B matrices are transformed to

The typical way to go about penalizing change in control inputs for this problem is to initially start with x as two states (position and speed), then augmenting the state vector with control input as the third state (position, speed and acceleration/control input) and a change in the control input (change in acceleration) as the control input.

Trajectory tracking

Problem The trajectory tracking problem is to compute a sequence of control input $u_1, u_2, ..., u_{H-1}$ so that a state sequence $x_0^*, x_1^*, ..., x_H^*$ is attainable such that $x_{t+1} = f(x_t, u_t)$ while minimizing the cost function $\sum_{i=0}^{H-1}(x_i - x_i^*)^T Q(x_i - x_i^*) + (u_i - u_i^*)^T R(u_i - u_i^*)$ where $u_i^*, i = 0toH-1$ is a control input that ensures feasibility i.e. $x_{t+1}^* = f(x_t^*, u_t^*) \forall t \in \{0, 1, ..., H-1\}$

Example The trajectory tracking problem for non-linear systems can be transformed to LQR for Linear Time Varying case via linearization. If the input trajectory is not feasible, the computed solution will not visit the desired states. Let's once again consider the example of cruise control. The objective is to regulate the longitudinal speed of the vehicle using LQR. In this case, it is sufficient to consider longitudinal dynamics alone. Let the reference speed to be tracked be given by

$$v(t) = 10|\sin(0.4\pi t)| \tag{2.14}$$

2.2.3 Derivation

Dynamic Programming

Hamiltonian

2.2.4 Relation to H_2 controller

2.3 MPC

2.3.1 Stability

2.4 Other

2.4.1 Feedback Linearization

2.4.2 Lyapunov Controller

2.4.3 Notes on Stability

Chapter 3

Path Planning

Chapter 4

Trajectory Planning

Chapter 5

Behavior Planning

Chapter 6

Architectures

Chapter 7

General Notes

7.1 Kinematics and Dynamics

7.2 Probability

7.3 Geometry

7.4 Optimization

7.5 Control

7.6 Programming & Algorithms