Vehicle Dynamics and Simulation

Drivetrain Modelling

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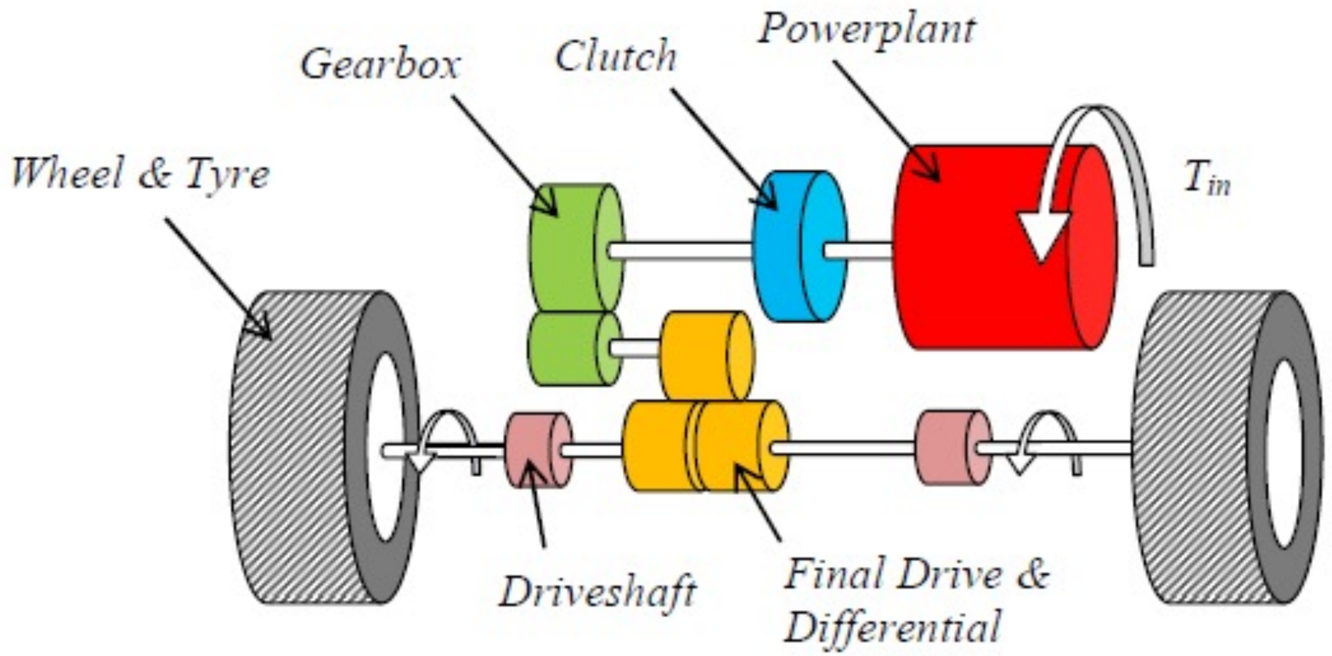
Drivetrain overview

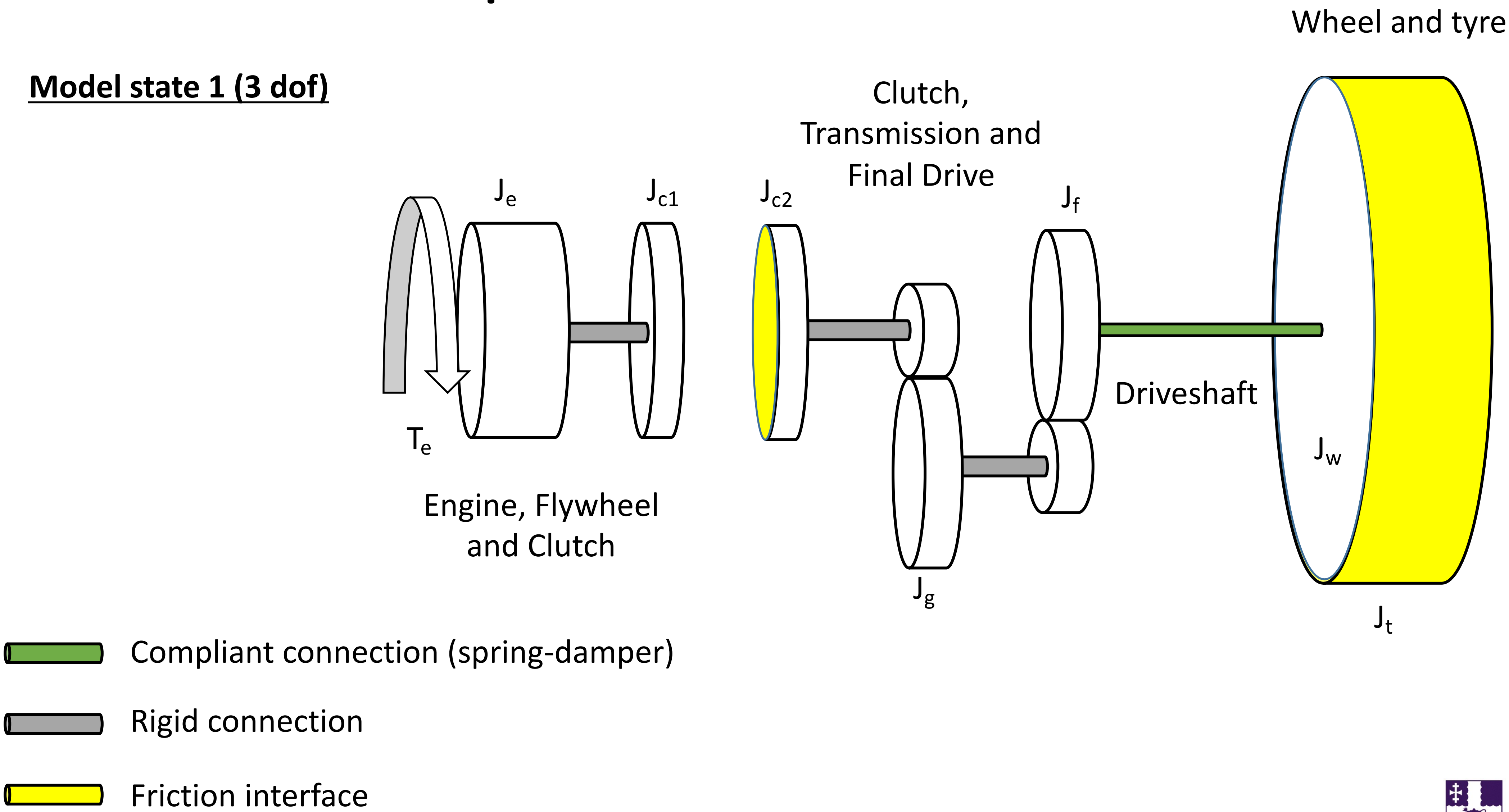
- Drivetrain responsible for optimal delivery of torque.
  - High efficiency
  - Low NVH and driveability targets met.
- Torque introduced at source i.e. powerplant and *flows* through drivetrain for delivery at tyre / road interface.
- For our purposes a relatively simple model is required i.e. time to speed simulation and analysis
  - Drivetrain compliance and damping won't significantly affect results.

Typical drivetrain



Model development

Model state 1 (3 dof)



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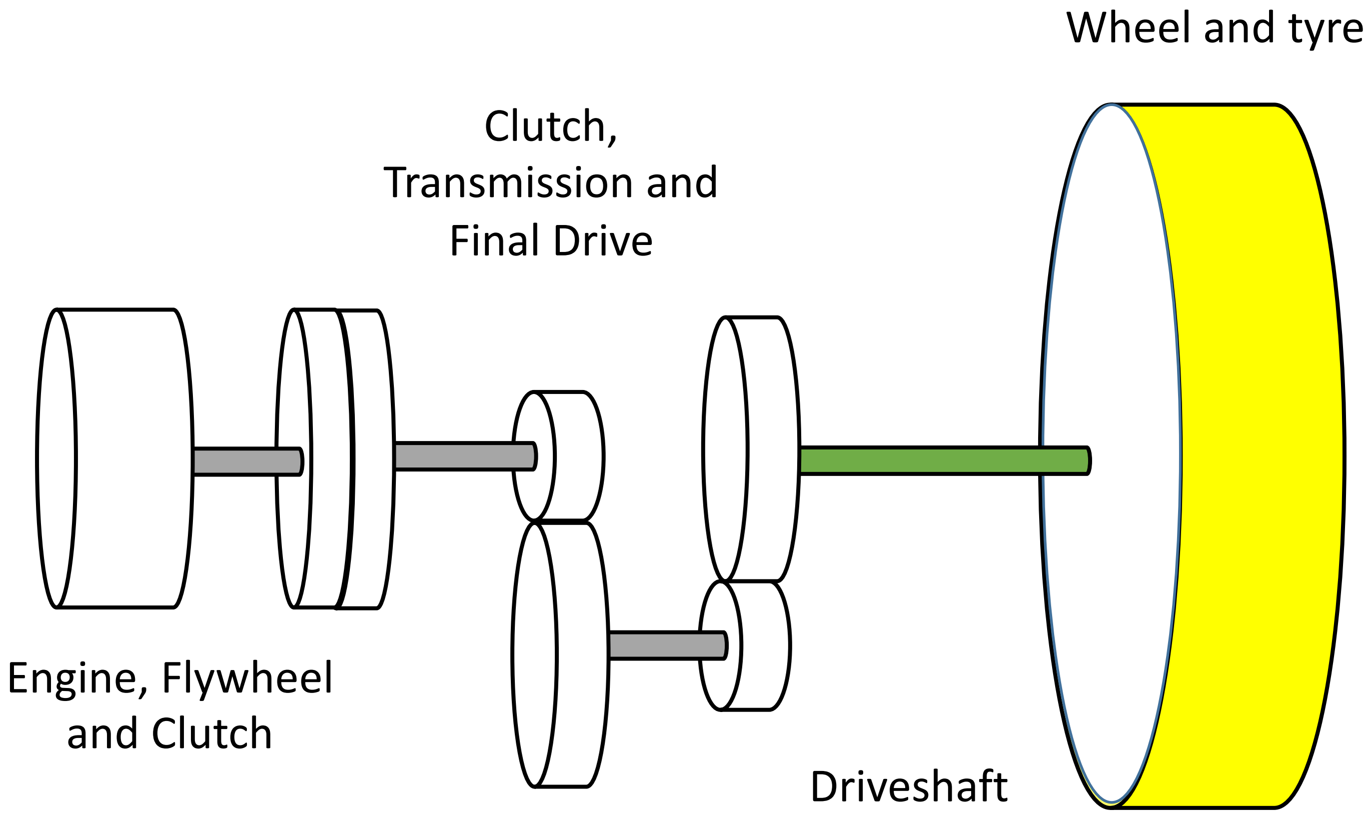
Model development

<u>Model state 2 (2 dof)</u>



Compliant connection (spring-damper)

Rigid connection

Friction interface

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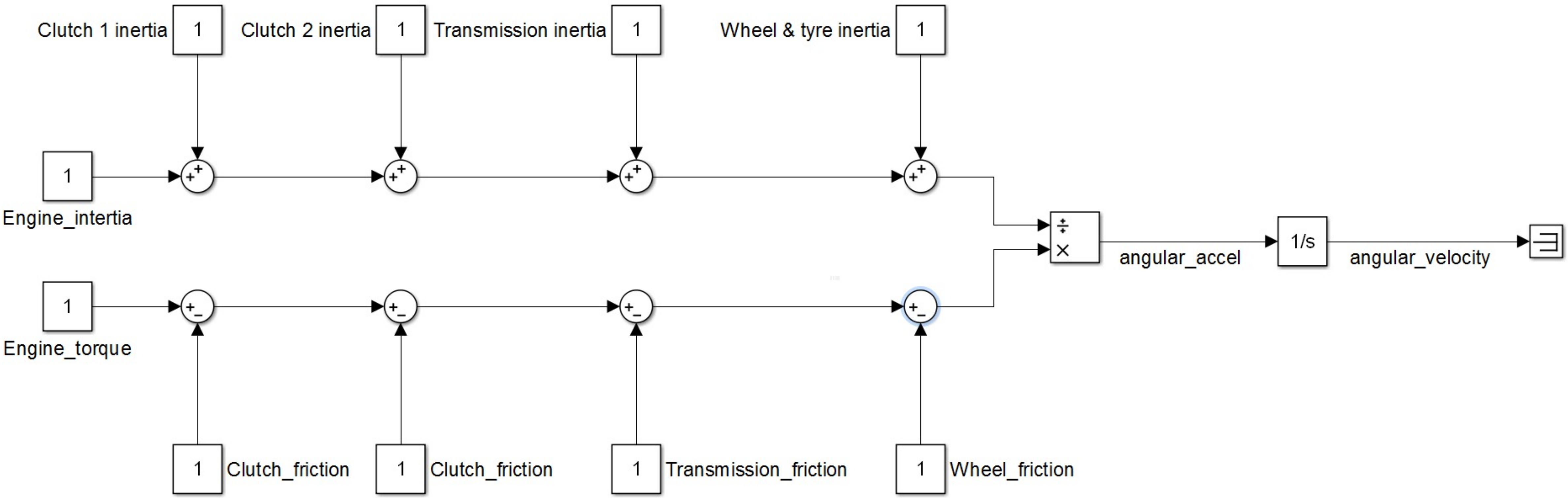
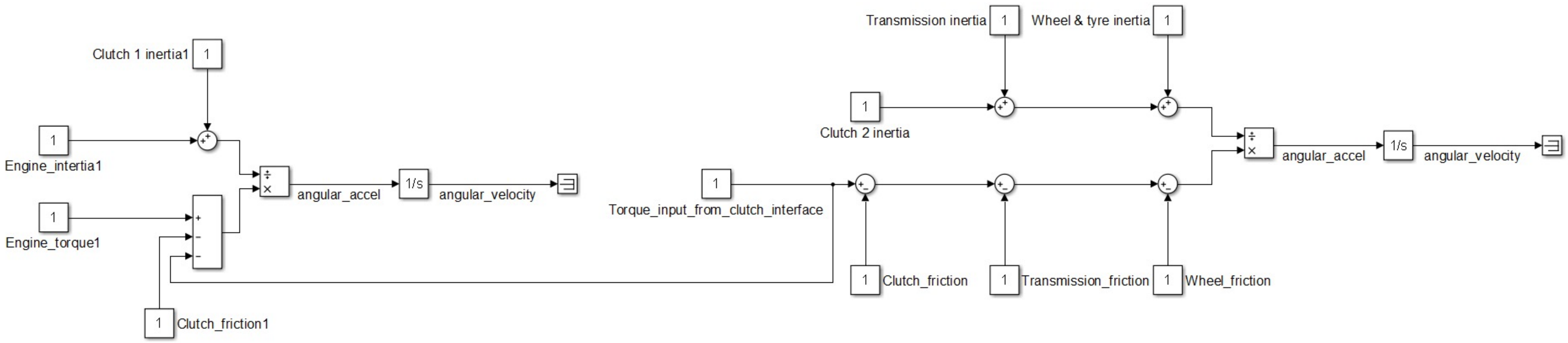
1-DOF


Clutch 1 inertia
1
Clutch 2 inertia
1
Transmission inertia
1
Wheel & tyre inertia
1
1
Engine_intertia
1
Engine_torque
angular_accel
1/s
angular_velocity
1
Clutch_friction
1
Clutch_friction
1
Transmission_friction
1
Wheel_friction

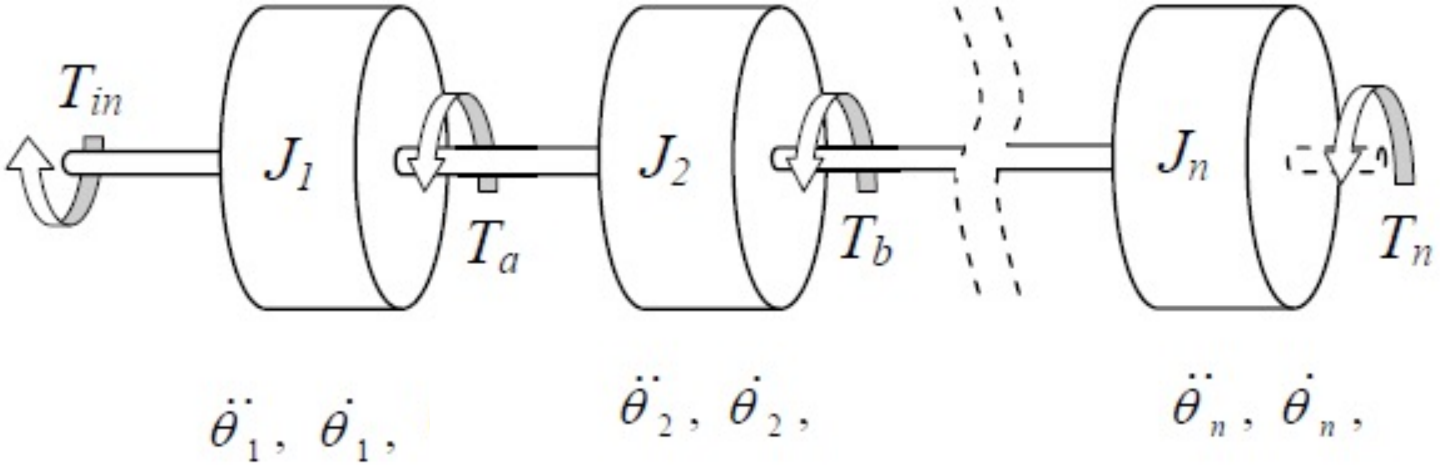
2-DOF



Clutch 1 inertia1
1
Engine_intertia1
Engine_torque1
Clutch_friction1
angular_accel
1/s
angular_velocity
Transmission inertia
Wheel & tyre inertia
Clutch 2 inertia
Torque_input_from_clutch_interface
Clutch_friction
Transmission_friction
Wheel_friction
angular_accel
1/s
angular_velocity

Model development

$$J_1\ddot{\theta}_1 = T_{in} - T_a$$

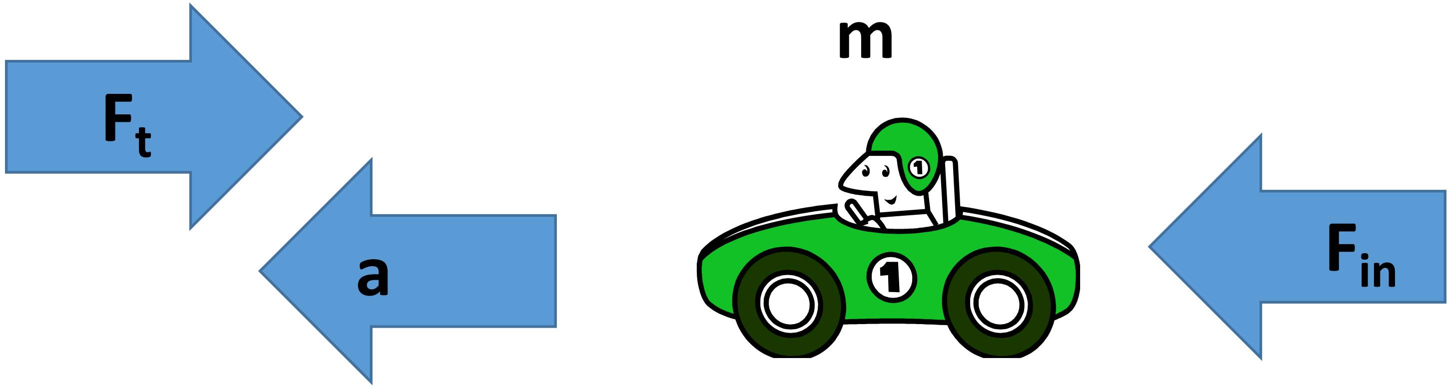


$$T_a = B\left(\dot{\theta}_1 - \dot{\theta}_2\right) + K\left(\theta_1 - \theta_2\right)$$

$$T_b = B\left(\dot{\theta}_2 - \dot{\theta}_3\right) + K\left(\theta_2 - \theta_3\right)$$

Single Inertia Vehicle Model

- Good representation if F_{in} is available.
- Simple to parameterise.
- Good check for validity of more complex multi-dof models.
- Disadvantages;
  - No clutch
  - Friction difficult to model
  - Unable to resolve any drivetrain performance details.



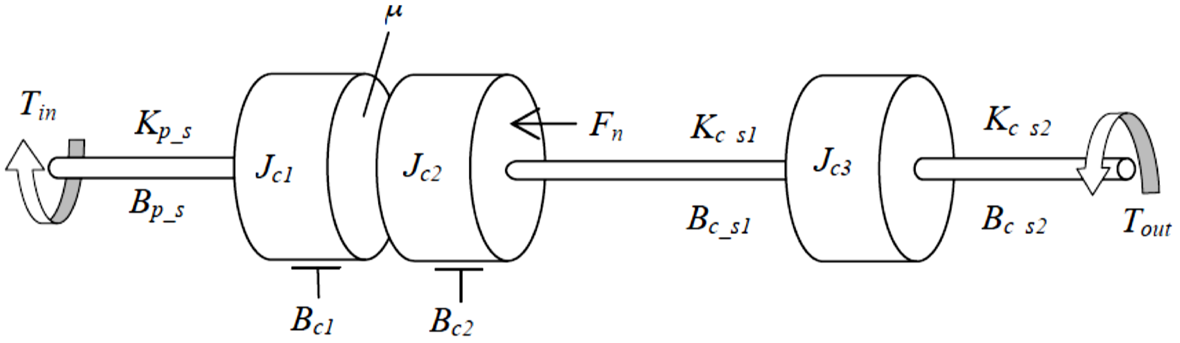
$$\ddot{x} = \frac{F}{m};$$

$$\dot{x} = \int \ddot{x}\,dt;$$

$$x = \int \dot{x}\,dt$$

Engine, Flywheel and Clutch

- Clutch is where the driveline splits and must be represented by some logic.
- Simplest model just deals with transfer of torque.
- More complex models include single or dual mass flywheels with non-linear clutch friction.
- Non-linear effects
  - Lash
  - Spring constants



Unlocked clutch $\omega_{rel} > 0$ or $T_{in} > \mu F_n$ (2 DOF)

$$J_{c1}\ddot{\theta}_{c1} = T_{in} - \mu F_n r_m$$
$$J_{c2}\ddot{\theta}_{c2} = \mu F_n r_m - T_{out}$$

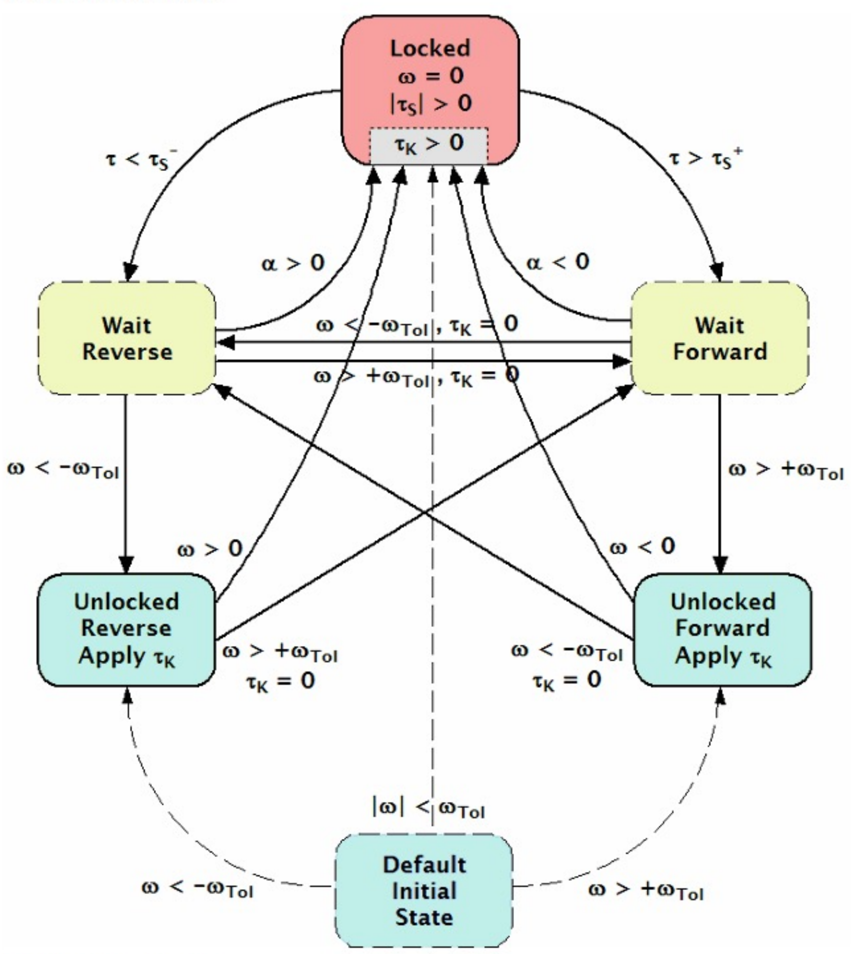
Locked clutch $\omega_{rel} = 0$ or $T_{in} \leq \mu F_n$ (2 DOF)

$$(J_{c1} + J_{c2})\ddot{\theta} = T_{in} - T_{out}$$
$$\theta_{c1} = \theta_{c2}$$

Engine, Flywheel and Clutch

- Clutch switching logic can be complex due to high amplitude, high frequency oscillations that cause switching between locked and unlocked states.
- When $T_{in} > \mu N$ the clutch unlocks and then returns to locked state when $\omega_{rel} = 0$. Zero crossings i.e. oscillating directional changes also cause problems for the solver.
- Stateflow provides an easier way to implement logic.

Clutch state-diagram, typical logic



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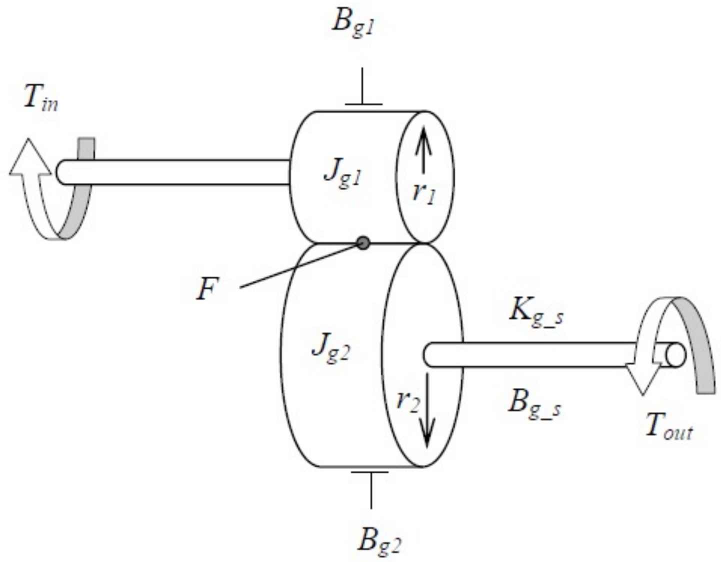
Transmission

- Conceptually very simple however speed synchronisation of input and output shafts and inertia ‘switching’ i.e. 1dof -> 2dof -> 1dof adds complexity.
- Lash non-linearity excites high frequency dynamics that result in very stiff system – can be dealt with by increasing damping.
- Parameterisation can also be troublesome;
  - Different frictions and inertias depending on gear ratio.

Transmission

$$J_1\ddot{\theta}_1 = T_{in} - B_{g1}\dot{\theta}_{g1} - Fr_1$$
$$J_2\ddot{\theta}_2 = Fr_2 - B_{g2}\dot{\theta}_{g2} - T_{out}$$



Perfect gear means that an algebraic constraint exists;

$$r_1\theta_1 = r_2\theta_2$$

Using the equation above, θ_1 can be eliminated (or θ_2)

Subs for $\dot{\theta}_{g1} = \frac{r_2}{r_1}\dot{\theta}_{g2}$ → $J_{g1}\left(\frac{r_2}{r_1}\right)\ddot{\theta}_{g2} = T_{in} - B_{g1}\left(\frac{r_2}{r_1}\right)\dot{\theta}_{g2} - Fr_1$

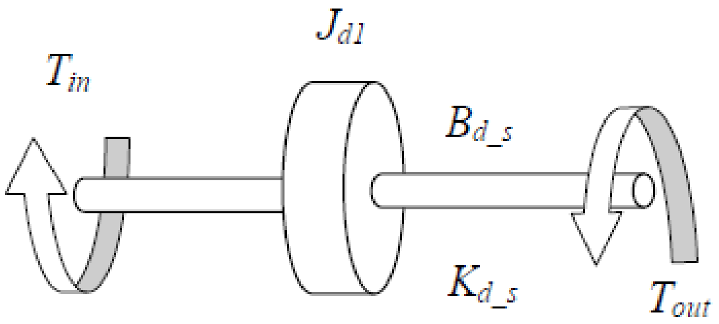
$\times\frac{r_2}{r_1}$ → $J_{g1}\left(\frac{r_2}{r_1}\right)^2\ddot{\theta}_{g2} = T_{in}\left(\frac{r_2}{r_1}\right) - B_{g1}\left(\frac{r_2}{r_1}\right)^2\dot{\theta}_{g2} - Fr_2$

And the complete system described (which can also be arranged for θ_{g1};

$$\ddot{\theta}_{g2} = \frac{T_{in}\left(\frac{r_2}{r_1}\right) - B_{g1}\left(\frac{r_2}{r_1}\right)^2\dot{\theta}_{g2} - B_{g2}\dot{\theta}_{g2} - T_{out}}{J_{g1}\left(\frac{r_2}{r_1}\right)^2 + J_{g2}}$$

Final Drive, Differential and Driveshaft

- Driveshaft propagates torsional oscillation through compliance.
- Simple to model, less so to parameterise.



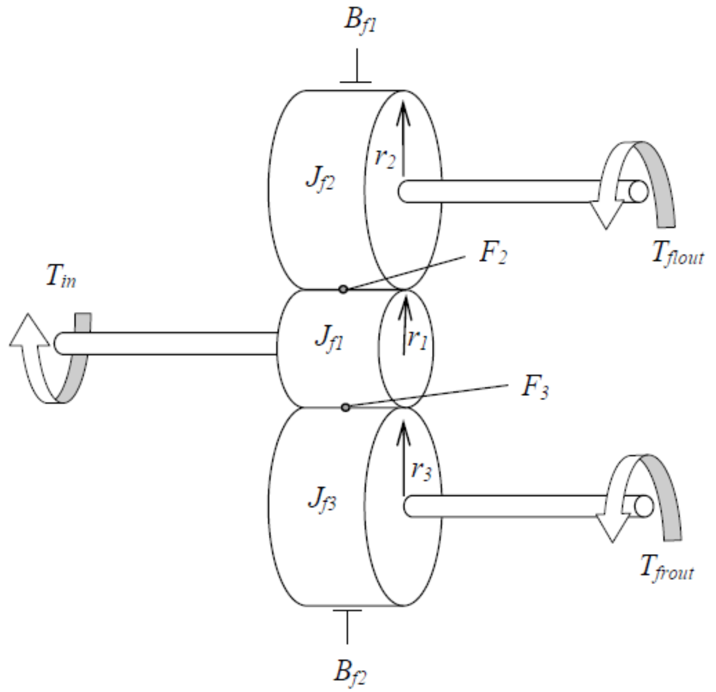
System equations

$$\ddot{\theta}_{d1} = \frac{T_{in} - T_{out}}{J_{d1}}$$

$$T_{out} = K_{d\_s}\left(\theta_{d1} - \theta_{n+1}\right) + B_{d\_s}\left(\dot{\theta}_{d1} - \dot{\theta}_{n+1}\right)$$

Final Drive, Differential and Driveshaft

- Final drive represents final ratio change.
- Can include complex non-linearities such as lash, active/passive friction systems.
- Simple to implement at most basic level.



The three inertias are represented;

$$J_{f1}\ddot{\theta}_{f1} = T_{in} - F_2 r_1 - F_3 r_1$$

$$J_{f2}\ddot{\theta}_{f2} = F_2 r_2 - B\dot{\theta}_{f2} - T_{flout}$$

$$J_{f3}\ddot{\theta}_{f3} = F_3 r_3 - B\dot{\theta}_{f3} - T_{frout}$$

Subject to the constraints;

$$r_1\theta_{f1} = r_2\theta_{f2}$$

$$r_1\theta_{f1} = r_3\theta_{f3}$$

$$T_{flout} = T_{frout}$$

After some algebraic manipulation, to give;

$$\ddot{\theta}_{f2} = \frac{T_{in} - \left(\frac{r_1}{r_2}\right)\left[T_{L2} + T_{flout}\right] - \left(\frac{r_1}{r_3}\right)\left[T_{L3} + T_{frout}\right]}{\left(\frac{r_2}{r_1}\right)J_{f1} + J_{f2}\left(\frac{r_1}{r_2}\right) + J_{f3}\left(\frac{r_2}{r_3}\right)\left(\frac{r_1}{r_3}\right)}$$

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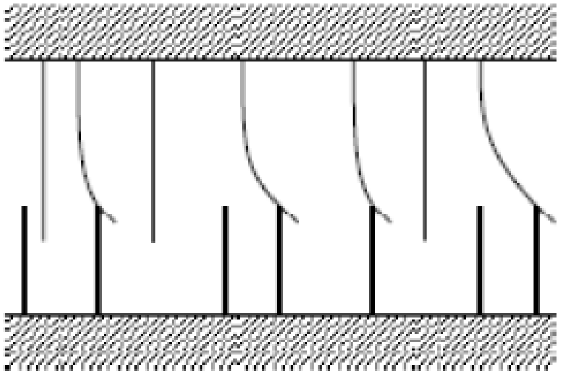
Wheel and Tyres

- LuGre tyre model models the tyre as the sum of solid-to-solid contact and viscous resistance of the lubricant between the tyre and road surface.
- The contribution of each is determined by the relative velocity of the tyre wrt the road.
- The LuGre tyre model remains numerically stable over the range of operation unlike some other models since as $\dot{x}_v \to 0$ so instability occurs since the slip velocity $\dot{x}_{rel} \to -\infty$ i.e. $1 - \frac{r_t \dot{\theta}_t}{\dot{x}_v}$



$$F_b = K x_{b\_rel} + B\left(\dot{x}_{t\_rel}\right) \dot{x}_{b\_rel}$$

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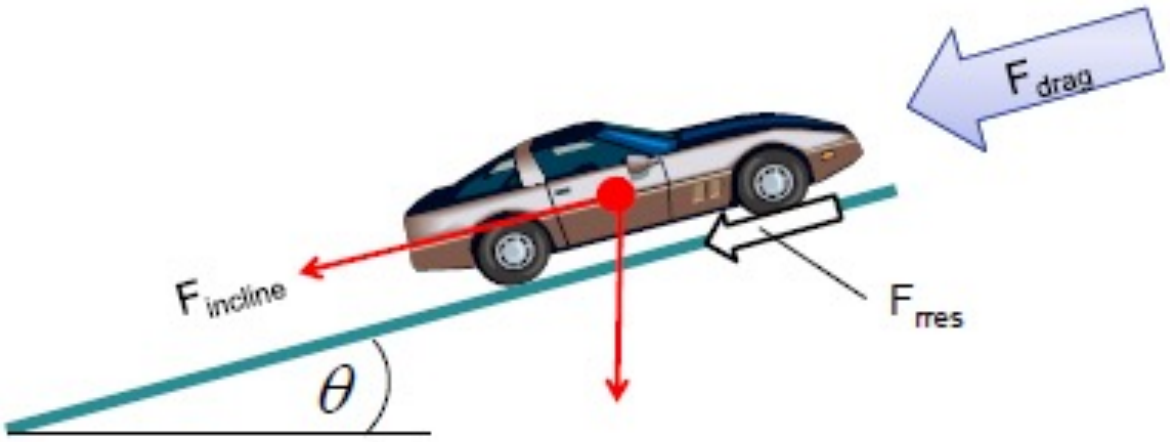
Vehicle Chassis

- Force generated in the tyre contact patch is applied to the vehicle chassis.



$$F_{rr} = mgCr$$
$$F_{inc} = mgsin\theta$$
$$F_d = \frac{1}{2}\rho AV^2 C_d$$

$$T = Fr$$

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