VEHICLE DYNAMICS PROJECT

DRIVELINE AND ENGINE CONTROL

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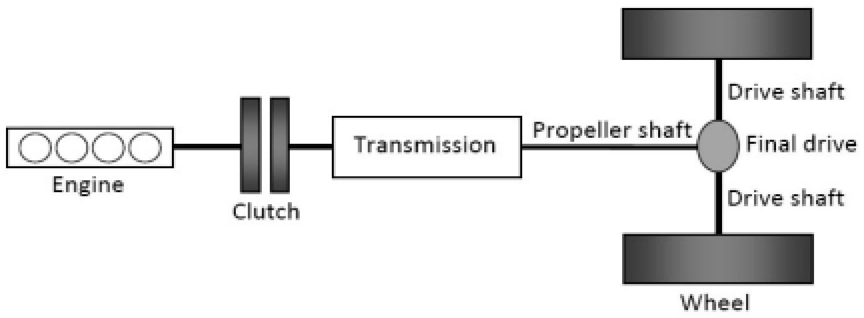
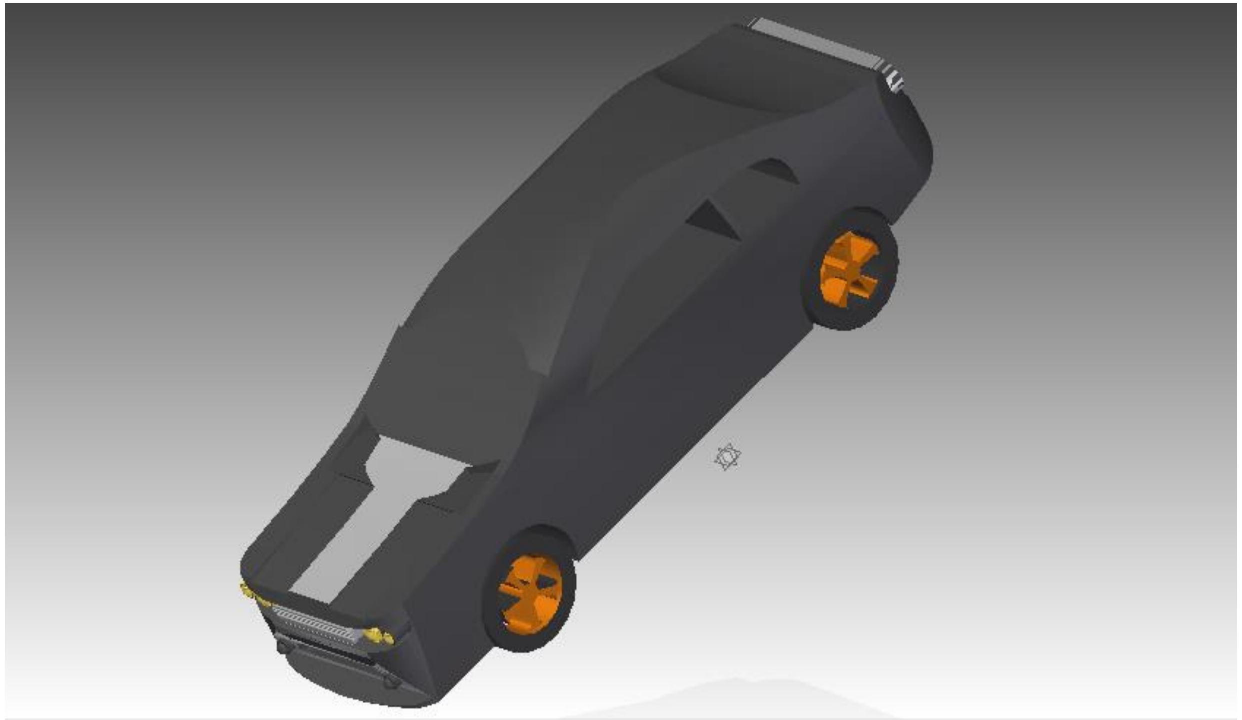
INTRODUCTION TO DRIVELINE

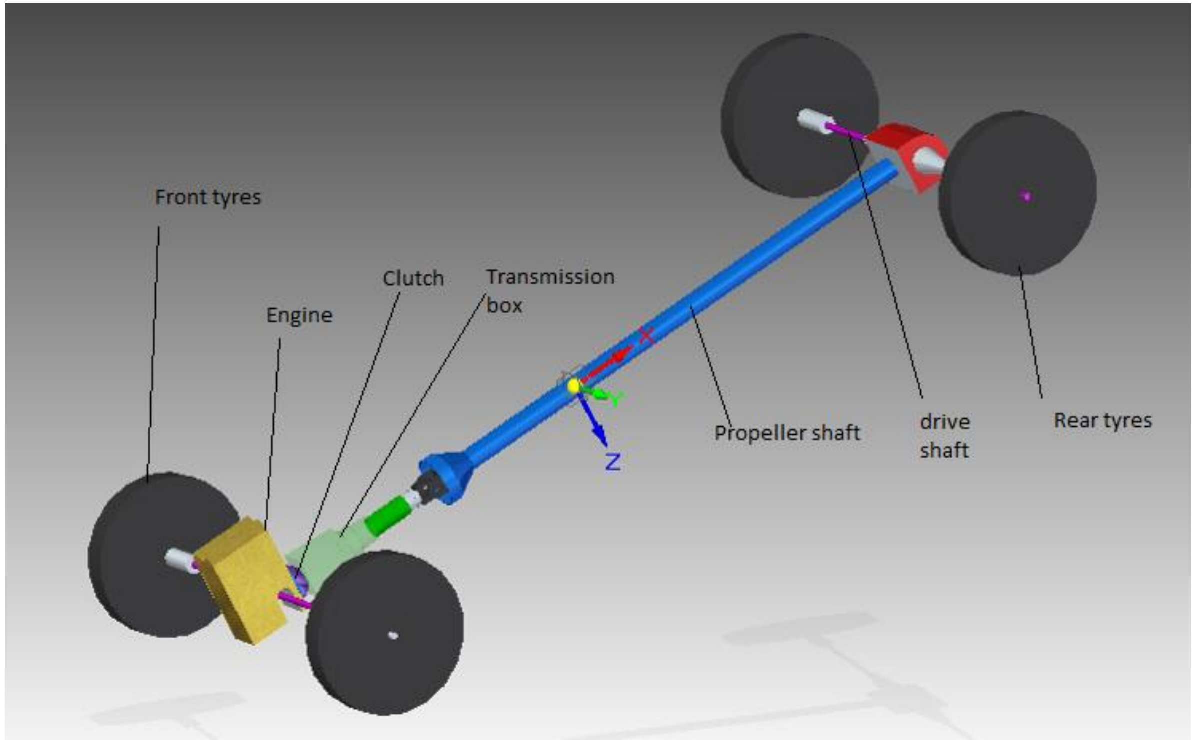
- A driveline is the part of a motorized vehicle which connects the engine and transmission to the wheel axles.
- In order to transmit this torque in an efficient way, a proper model of the driveline is needed for controller design purposes, with the aim of lowering emissions, reducing fuel consumption and increasing comfort.
- It can be rear drive, front drive or four wheel drive.
- Schematic of driveline:



CAD model

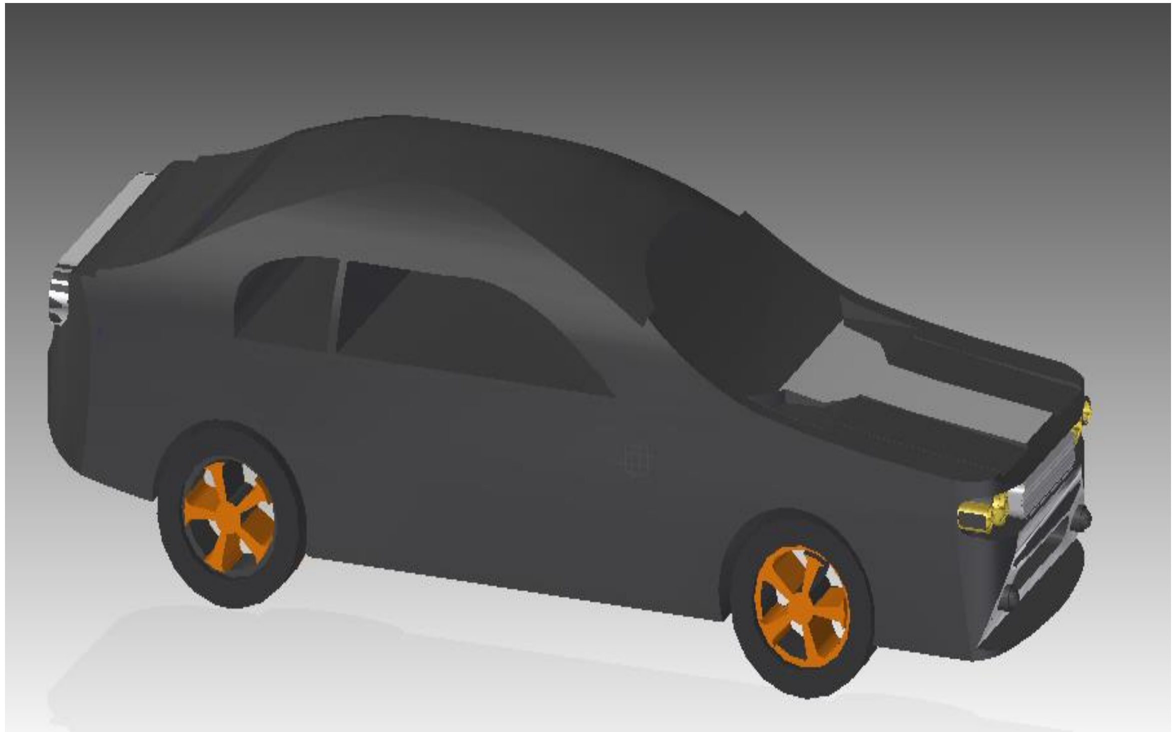


CAD model of car(Dodge Challenger SRT8)



CAD model of driveline

CAR model- car(Dodge Challenger SRT8)



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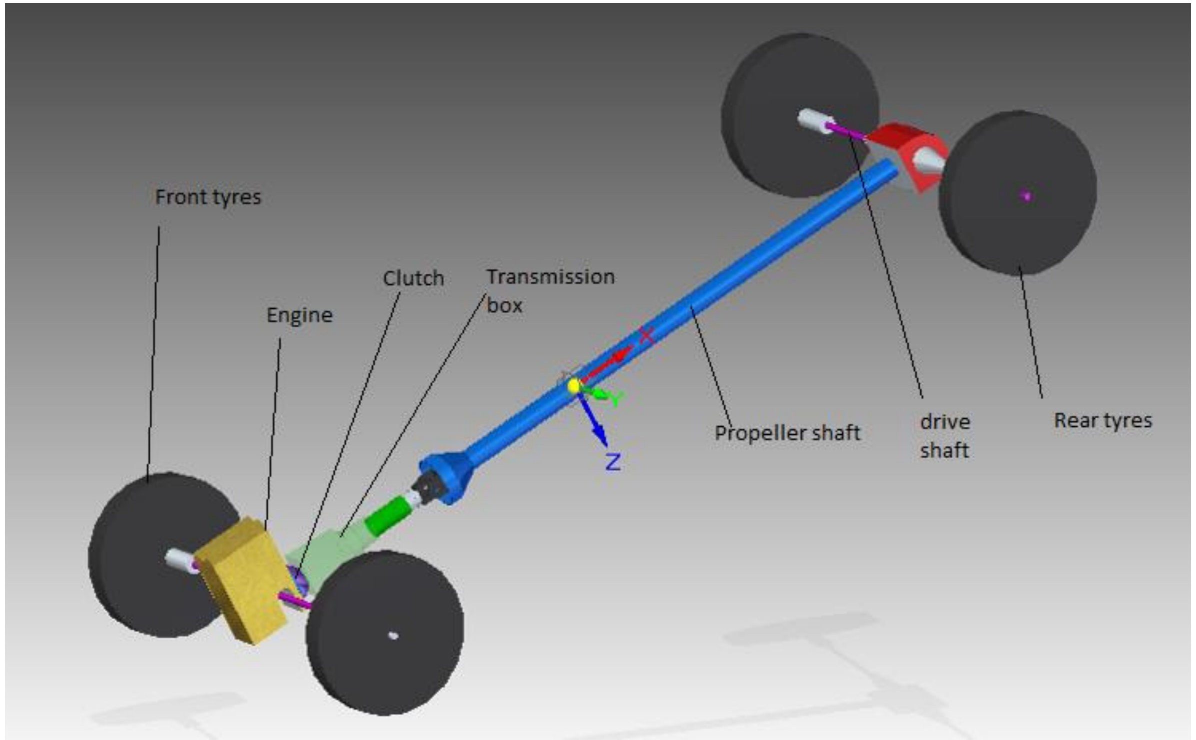
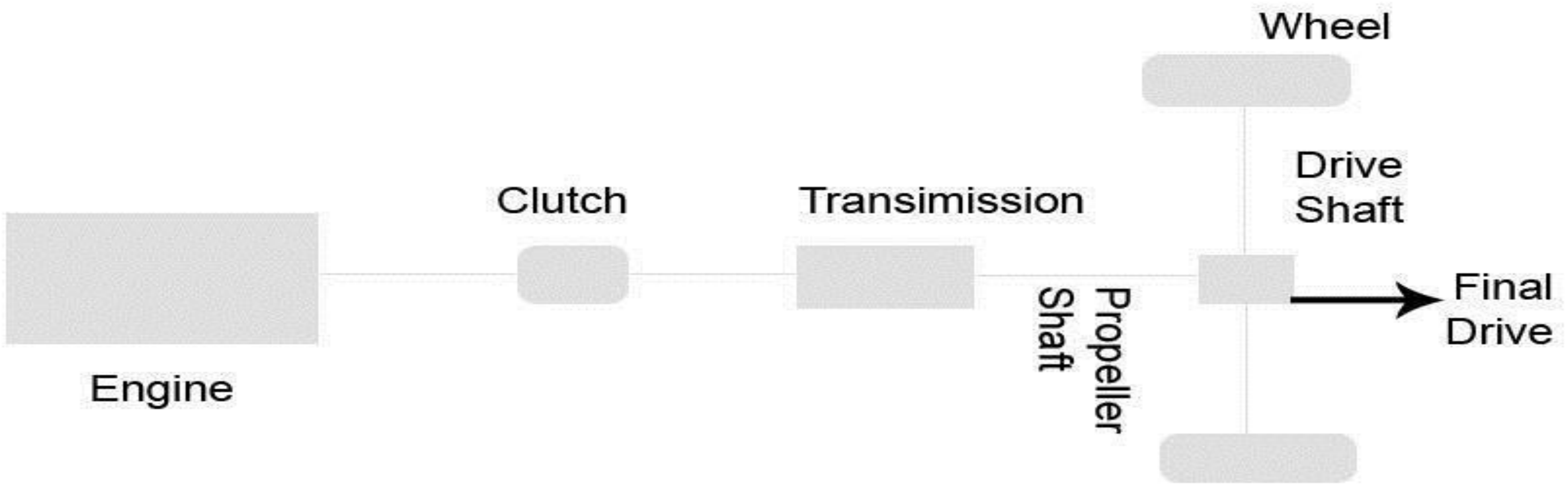
COMPONENTS OF DRIVELINE

The components of driveline are

- Engine
- Clutch
- Transmission
- Shafts
- Wheels

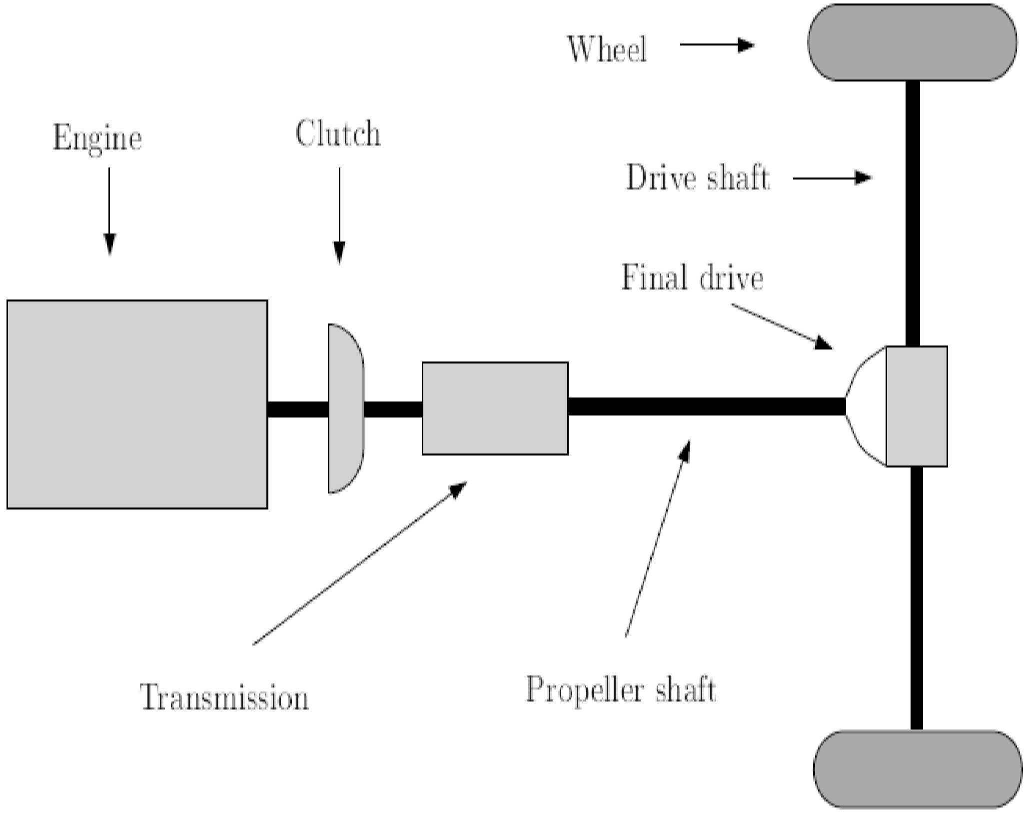


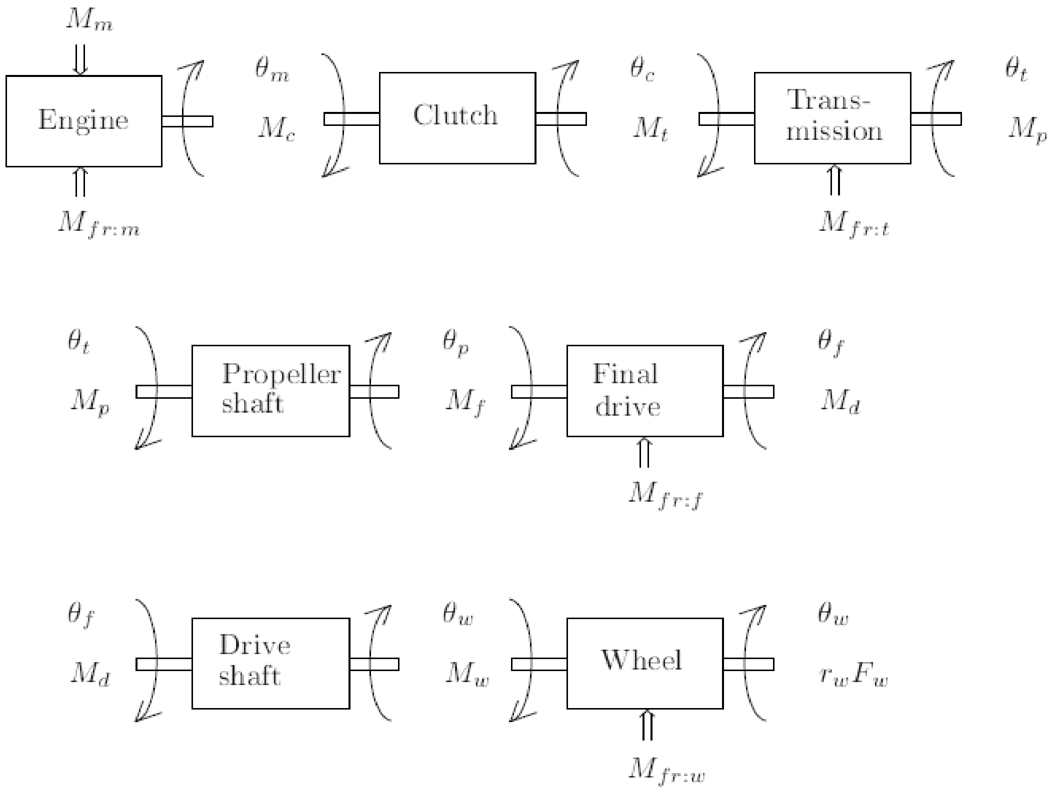
Simplest model : Flexible drive shaft model



This Picture shows the driveline of heavy truck driveline.
Fundamentals equation of driveline will be derived by using the generalized Newton's Second law of motion.
Relations between inputs and outputs will be described for each part in the given figure.

Schematic of Driveline





Parameters used in mathematical model

Driving torque: Mm

External load from Clutch: Mc

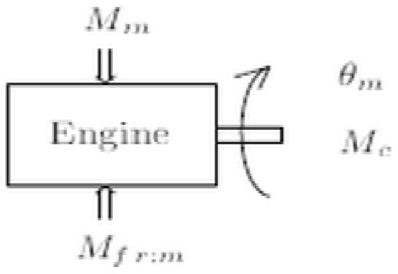
Moment of Inertia of the engine: Jm

Angle of flywheel: theta m

Conversion ratio of transmission: i t

internal friction torque of transmission: Mf r:t

- **Engine :**The Output torque of the engine characterized by the driving torque (M_m) resulting from the combustion, the internal friction form the engine ($M_{fr:m}$) and the external load from the clutch (M_c).Newtons's second law of motion gives the following model where J_m is the mass moment of interia of the engine and the is $\theta_m$the angle of the flywheel.



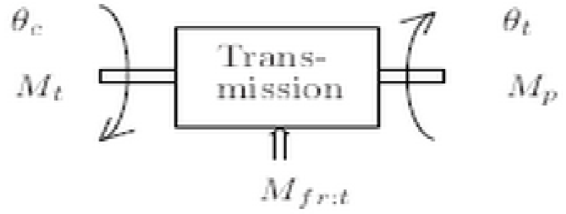
$$J_m\ddot{\theta}_m = M_m - M_{fr:m} - M_c$$

- **Clutch:** A friction clutch found in vehicles ins equipped with a manual transmission consists of a clutch disk connecting the flywheel of the engine and the transmission input shaft .When clutch is engaged and no internal friction is assumed ,M_c=M_t is obtained .The transmission torque is a function of the angular difference (Θ_m-Θ_c)and the angular velocity difference over the clutch

$$M_c = M_t = fc(\theta_m - \theta_c , \dot{\theta}_m - \dot{\theta}_c)$$

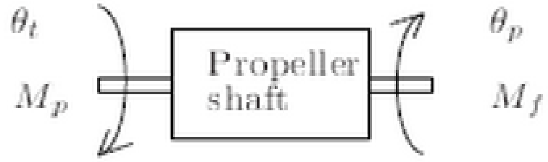
- Transmission: A transmission has a set of fears ,each with a conversion ratio i_{t}. This gives the following relation between the input and the output torque of the transmission where the internal frcition torque of the transmission is labeled $M_{fr:t}$. The reason for considering the angle difference $\Theta_c - \Theta_t i_t$ is the possibility of having the torsional effects in the transmission.

$$M_p = f_t(M_t, M_{fr:t}, \Theta_c - \Theta_t i_t, \Theta_c - \Theta_t i_t, i_t)$$



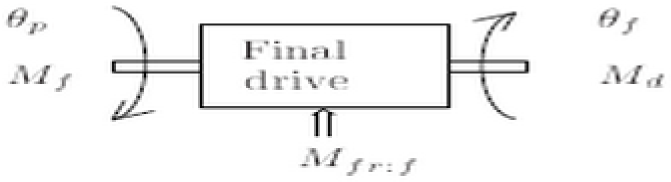
- Propeller Shaft: The Propeller shaft connects the transmission's output shaft with the final drive .No friction is assumed ($M_p = M_f$) giving the following model of the torque input to the final drive

$$M_p = M_f = fp(\Theta_t - \Theta_p, \Theta_t - \Theta_p)$$



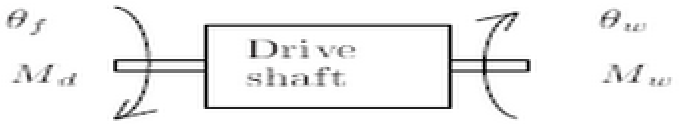
- Final Drive : The final drive is characterized by a conversion ratio I_f in the same way as for the transmission .The following relation for the input and output torque holds

$$M_d = f_f(M_f, M_{fr:f}, \theta_p - \theta_f i_f, \dot{\theta}_p - \dot{\theta}_f i_f, i_f)$$



- Drive Shafts : The drive shafts connect the wheels to the final drive .Here it is assumed that the wheel speed is the same for the 2 wheels. Therefore, the drive shafts are modeled as one shaft .When the vehicle is turning and the speed differs between the wheels ,both drive shafts have to be modeled .No friction ()gives the model equation

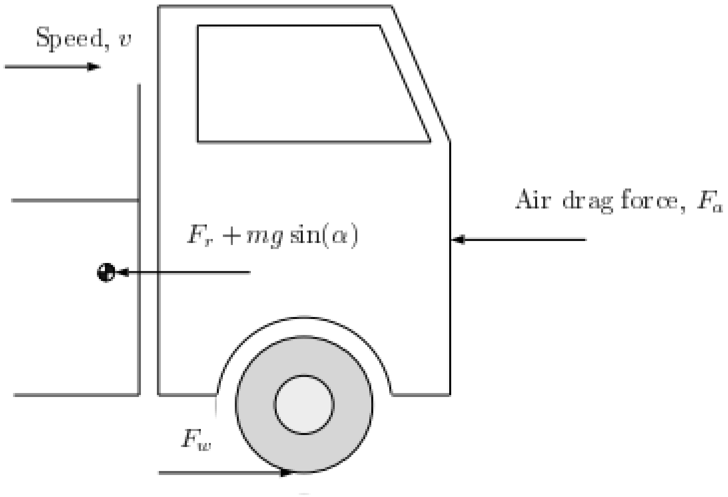
$$Mw=Md=fd(\Theta_f-\Theta_{w,}\ \Theta_f-\Theta_w\ )$$



Wheels:

- For vehicle with mass, m and velocity

Frictional force F_w is given by



$$F_w = m\dot{v} + F_a + F_r + mgsin(\alpha)$$

Where air drag force(F_a) is

$$F_a = \frac{1}{2} * c_w A_\alpha \rho_\alpha v^2$$

Rolling resistance(F_r) is

$$F_r = m(c_{r1} + c_{r2} v)$$

Gravitational force is $mgsin(\alpha)$

where α is slope of the road.

- Resulting torque

$J_w\ddot{\theta} = M_w - F_w r_w - M_{fr:w}$

Where J_w is the mass moment of inertia of the wheel,$M_{fr:w}$ is the friction torque.

By using above equations

$$(J_w + mr_w^2)\ddot{\theta} = M_w - M_{fr:w} - \frac{1}{2} * C_w A_\alpha \rho_\alpha r_w^3 \dot{\theta}_w^2 - r_w m(c_{r1} + c_{r2} r_w \dot{\theta}_w)$$

$$-r_w m(c_{r1} + c_{r2} r_w \dot{\theta}_w)\text{-}r_w m g \sin\alpha$$

Final equation of torque for wheel

Mathematical model

Two types of mathematical models are analyzed:

1. Drive shaft model

Flexible drive shaft, all other components are rigid

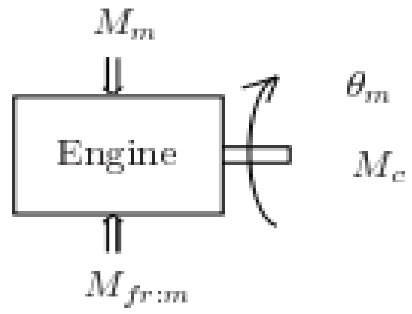
2. Linear Clutch drive shaft model

Flexible drive shaft and Clutch, all other components are rigid

Flexible drive shaft model

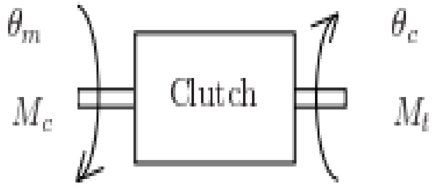
Engine: Driving torque from combustion (Mm) and internal friction from the engine (Mf r:m) and the external load from clutch(Mc).

Force and moment balance gives:



Clutch : The clutch is assumed to be stiff which gives the following equations for the torque and the angle.

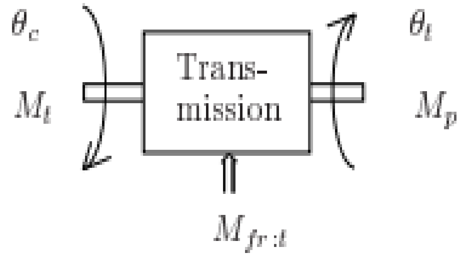
$$M_c = M_t, \quad \theta_m = \theta_c \text{ ----------------------- } \quad (1)$$



Transmission: The transmission is described by one rotating inertia. The friction torque is assumed to be described by a viscous damping coefficient. The model of the transmission,

$$\theta_c = \theta_t i_t \quad \text{---------}(2)$$

$$J_t\ddot{\theta}_t = M_t i_t - b_t\dot{\theta}_t - M_p \quad \text{----------}(3)$$

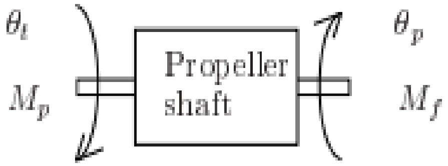


From equations 1, 2, 3,

$$J_t\ddot{\theta}_m = M_c i_t^2 - b_t\dot{\theta}_m - M_p i_t \quad \text{-----------}(4)$$

Propeller Shaft: The propeller shaft is assumed to be stiff. Hence following equations can be written as,

$$M_p = M_f \,,\, \theta_t = \theta_t \text{---------}(5)$$



Final drive: In the same way as for the transmission, the final drive is modelled by one rotating inertia J_f. The friction torque is assumed to be described by a viscous damping coefficient b_f. The model of the final drive, is

$$\theta_p = \theta_f i_f \text{ ------------ } (6)$$

$$J_f \ddot{\theta}_t = M_f i_f - b_f \dot{\theta}_f - M_d \text{ ----------}(7)$$

From equations 5 and 6,

$$J_f \ddot{\theta}_t = M_p i_f^2 - b_f \dot{\theta}_f - M_d i_f \text{ ----------}(8)$$

From equations 8 and 2,

$$J_f \ddot{\theta}_m = M_p i_f^2 i_t - b_f \dot{\theta}_m - M_d i_f\, i_t \text{------------}(9)$$

From equations 9 and 4,

$$(J_t i_f^2 + J_f) \ddot{\theta}_m = M_p i_t^2 i_f - b_t \dot{\theta}_m\, i_f^2 - M_d i_f\, i_t \text{ -- } b_f \dot{\theta}_m$$

- **Drive shaft:** It is modelled as damped torsional flexibility having stiffness k and internal damping c

$$M_w = M_d = k(\theta_f - \theta_w) + c(\dot{\theta}_f - \dot{\theta}_w) = k(\frac{\theta_m}{i_f\, i_t} - \theta_w) + c(\frac{\dot{\theta}_m}{i_f\, i_t} - \dot{\theta}_w)$$

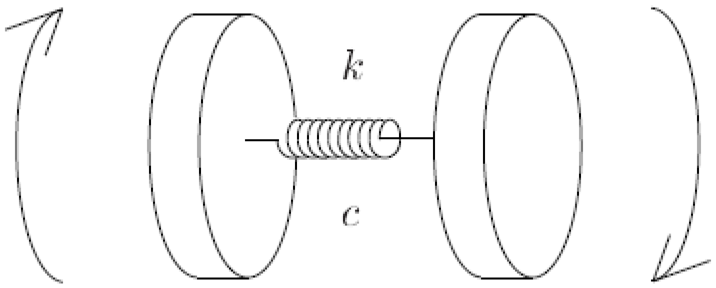
By replacing M_d in final drive equation,

$$(J_t i_f^2 + J_f)\, \ddot{\theta}_m = M_c i_t^2 i_f^2 - b_t \dot{\theta}_m\, i_f^2 - b_f \dot{\theta}_m - k\left(\theta_m - \theta_w i_f\, i_t\right) - c(\dot{\theta}_m - \dot{\theta}_w\, i_f\, i_t)$$

- **Wheel:** As discussed earlier , the dynamics of wheel is given by

$$(m r_w^2 + J_w)\, \ddot{\theta}_w = k\left(\frac{\theta_m}{i_f\, i_t} - \theta_w\right) + c(\frac{\dot{\theta}_m}{i_f\, i_t} - \dot{\theta}_w) - b_w \dot{\theta}_w - 0.5 c_w A_a \rho_a r_w^3 \dot{\theta}_w^2 - r_w m(c_{r1} + g sin\alpha)$$

The drive shaft model:



$$(J_m + \frac{J_t}{i_t^2} + J_f/i_t^2 i_f^2)\ddot{\theta}_m = M_m - M_{fr:m} - (\frac{b_t}{i_t^2} + b_f/i_t^2 i_f^2)\dot{\theta}_m - k(\frac{\theta_m}{i_f\, i_t} - \theta_w)/i_f\, i_t - \mathrm{c}(\frac{\dot{\theta}_m}{i_f\, i_t} - \dot{\theta}_w)/i_f\, i_t$$

$$(\mathrm{m}r_w^2 + J_w)\,\ddot{\theta}_w = k\left(\frac{\theta_m}{i_f\, i_t} - \theta_w\right) + \mathrm{c}(\frac{\dot{\theta}_m}{i_f\, i_t} - \dot{\theta}_w) - (b_w + mc_{r2}r_w^2)\dot{\theta}_w - 0.5c_w A_a \rho_a r_w^3 \dot{\theta}_w^2 - r_w \mathrm{m}(c_{r1} + gsin\alpha)$$

State space formulation

- Input to open-loop drive system is $u = M_m - M_{fr:m}$ (Difference between the driving torque and the fiction torque).
- Possible physical state variables are torques, angle differences, and the angular velocity of any inertia.
- The state space equation is $\dot{x} = Ax + Bu + Hl$

Where A, B, H, x, and l are defined for drive shaft model and for the clutch and drive-shaft model.

State space formulation for linear drive shaft model

- $x_1 = \frac{\theta_m}{i_t i_f} - \theta_\omega$
- $x_2 = \dot{\theta}_m$
- $x_3 = \dot{\theta}_\omega$
- $l = r_\omega m(c_{r1} + gsin(\alpha))$

This gives A=

$$\begin{matrix} 0 & 1/i & -1 \\ -k/iJ_1 & -(b + c/i^2)/J_1 & \mathrm{c}/iJ_1 \\ k/J_2 & \mathrm{c}/iJ_2 & -(c + b_2)/J_1 \end{matrix}$$

- B=[0 $1/J_1$ 0]^T
- H=[0 0 $-1/J_2$]^T

Where

$$i = i_t i_f$$

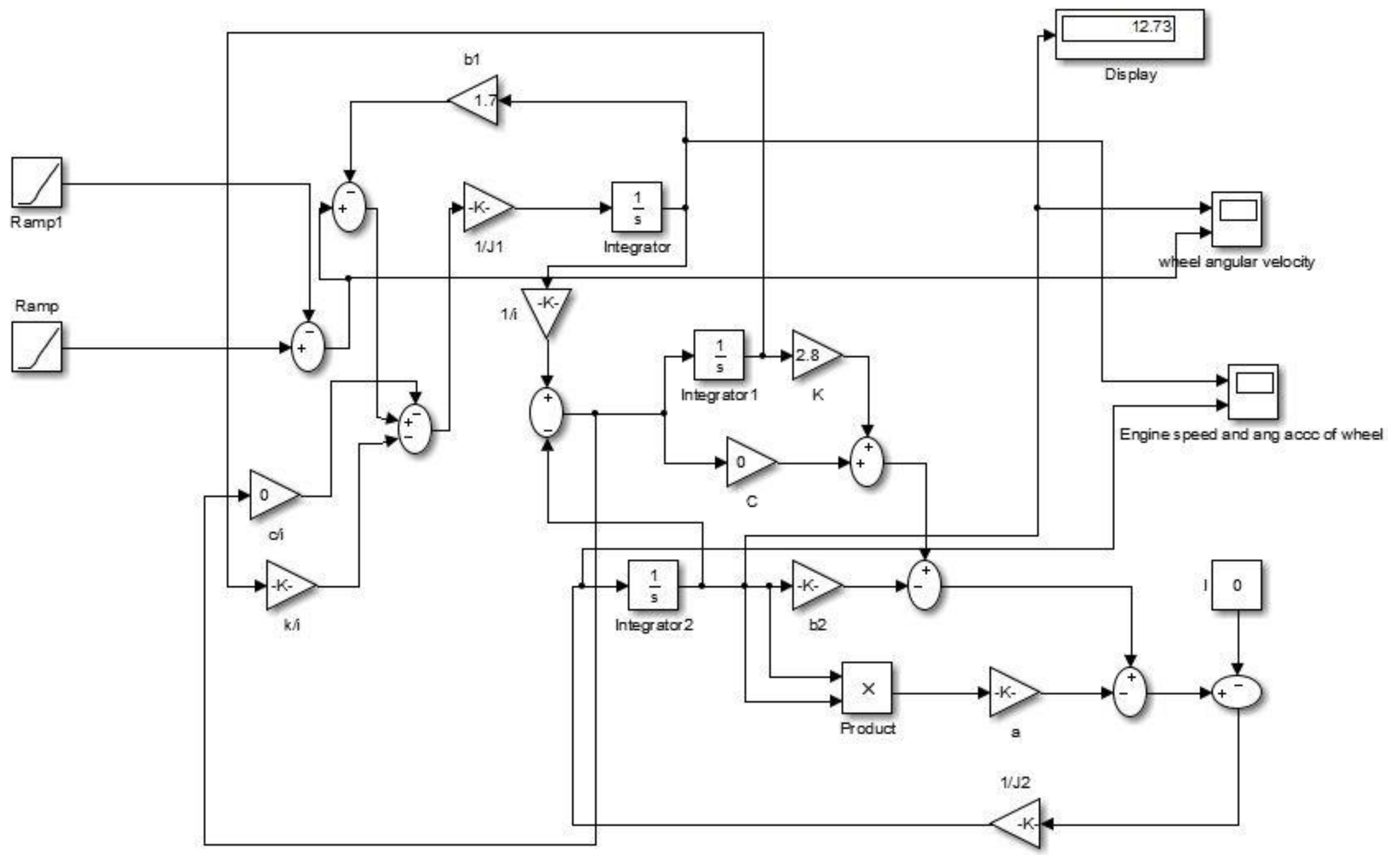
$$J_1 = J_m + \frac{J_t}{{i_t}^2} + \frac{J_f}{{i_t}^2 i_f^2}$$

$$J_2 = J_w + m r_w^2$$

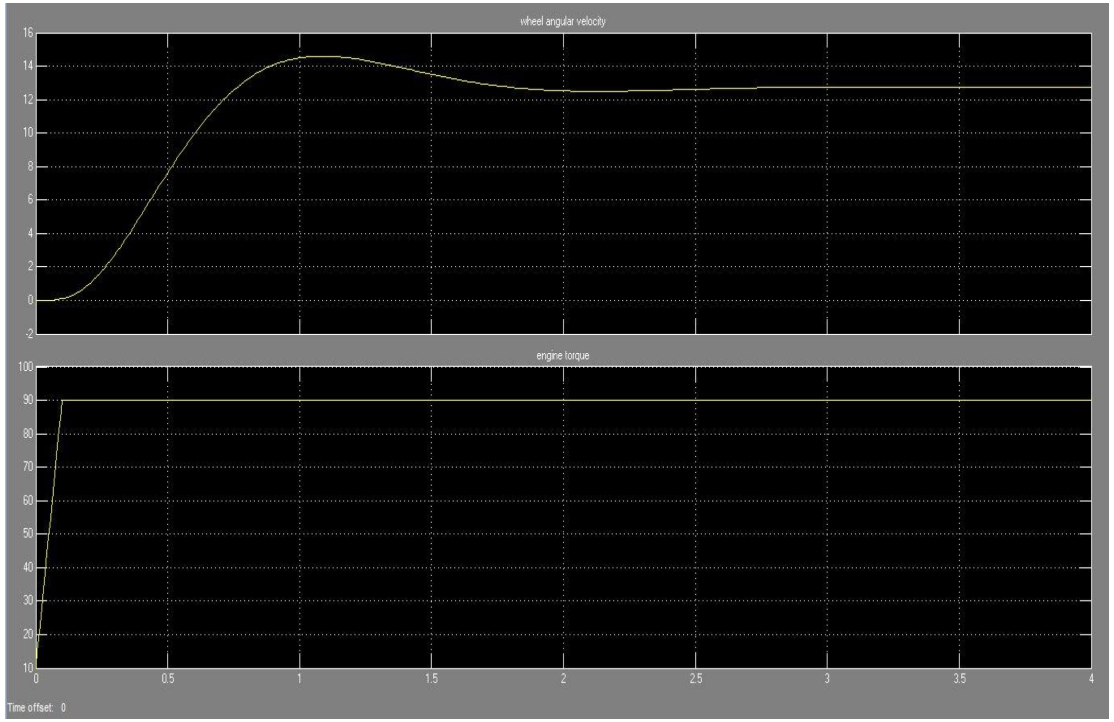
$$b_1 = \frac{b_t}{{i_t}^2} + \frac{b_f}{{i_t}^2 i_f^2}$$

$$b_2 = b_w + m c_{r2} r_w^2$$

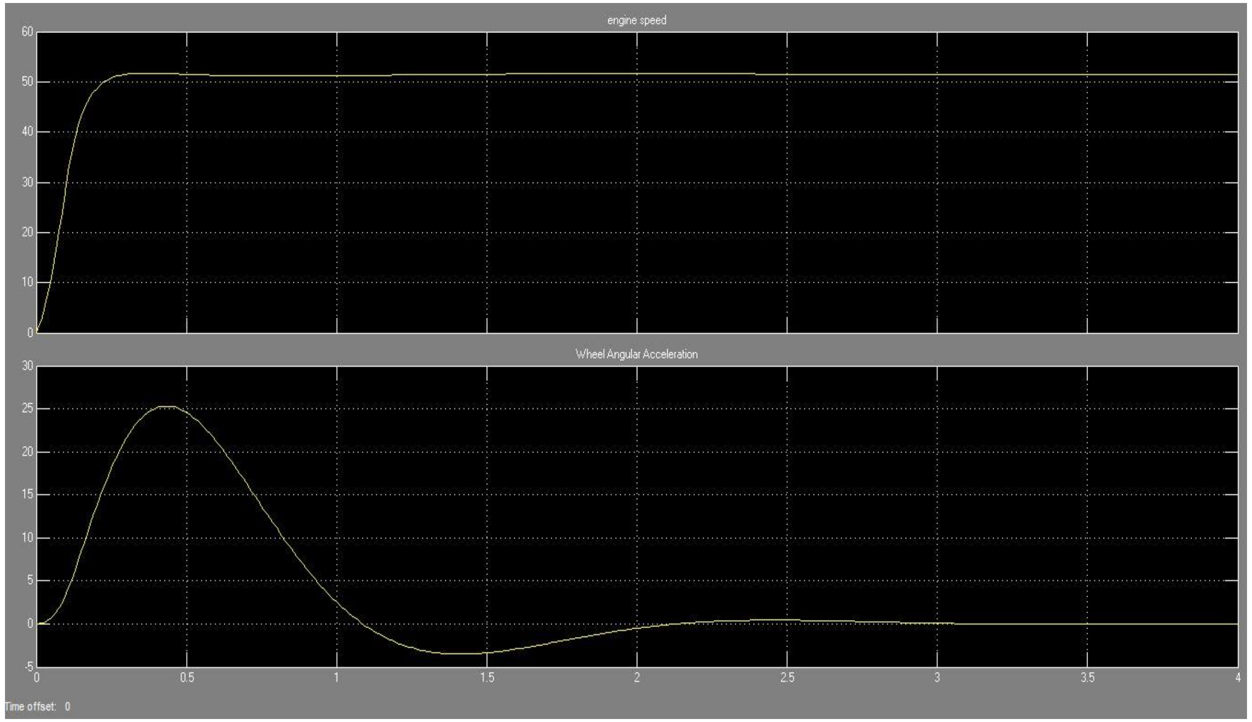
SIMULINK flexible drive shaft model



Ramp response

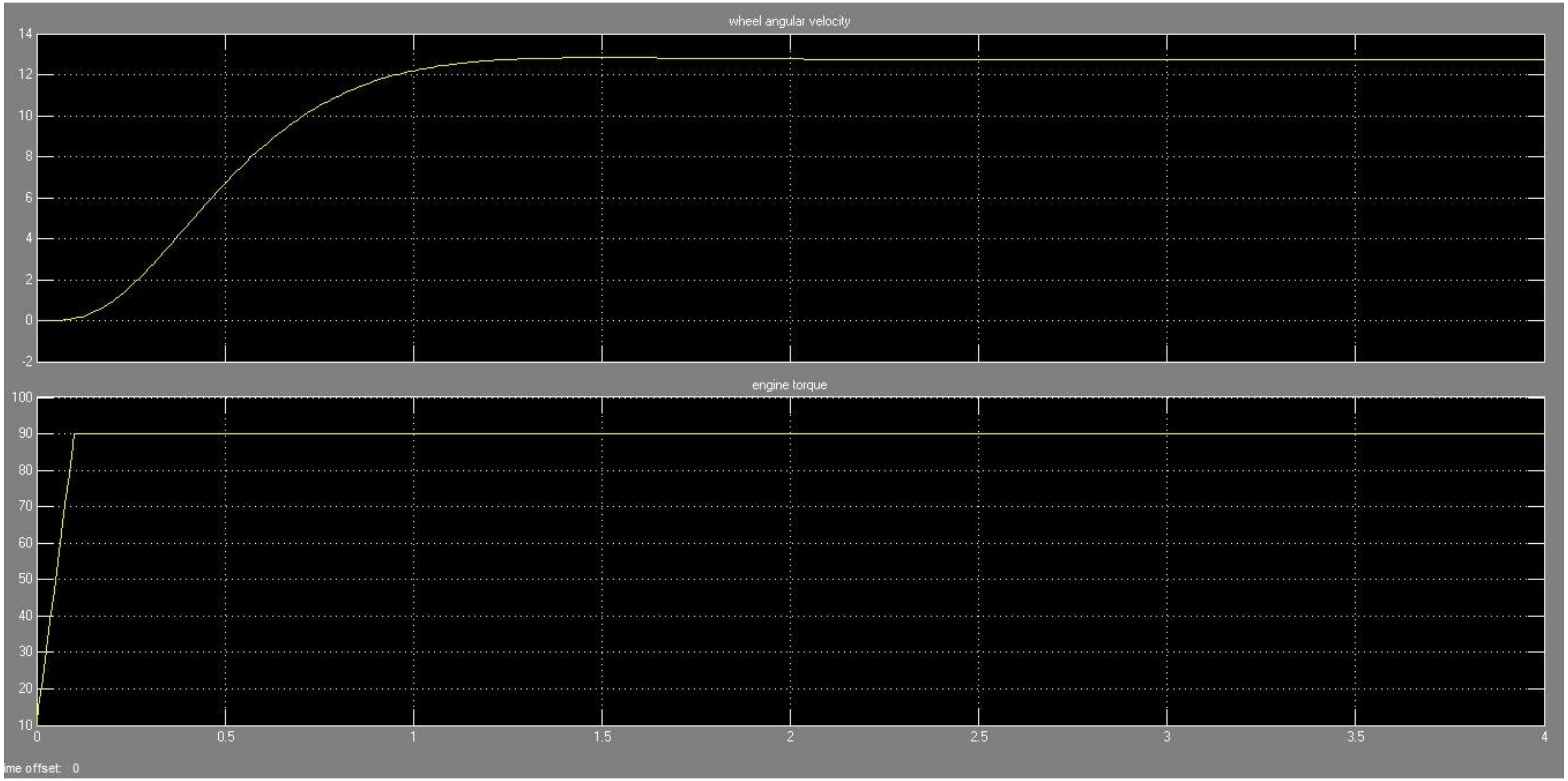


INPUT TORQUE,WHEEL SPEED



ENGINE SPEED,WHEEL ACCELERATION

CONTROL USING PD



Linear flexible clutch and drive shaft

A model with linear clutch flexibility and torsional flexibility in drive shaft is derived by repeating the process as in flexible drive shaft model with the difference that the model for clutch is a flexibility with stiffness Kc and damping coefficient Cc.

$$M_c = M_t = K_c(\theta_m - \theta_c) + C_c\left(\dot{\theta}_m - \dot{\theta}_c\right) = K_c(\theta_m - \theta_t i_t) + C_c\left(\dot{\theta}_m - \dot{\theta}_t i_t\right)$$

Engine inertia is given by

$$J_m\ddot{\theta}_m = M_m - M_{fr:m} - (K_c(\theta_m - \theta_t i_t) + C_c\left(\dot{\theta}_m - \dot{\theta}_t i_t\right))$$

Equation describing the transmission is given by

$$J_t\ddot{\theta}_t = i_t\left(K_c(\theta_m - \theta_t i_t) + C_c\left(\dot{\theta}_m - \dot{\theta}_t i_t\right)\right) - b_t\dot{\theta}_t - M_p$$

Substituting Mp

$$\left(J_t + \frac{J_f}{{i_f}^2}\right)\ddot{\theta}_t = i_t\left(K_c(\theta_m - \theta_t i_t) + C_c\left(\dot{\theta}_m - \dot{\theta}_t i_t\right)\right) - (b_t + \frac{b_f}{{i_f}^2})\ \dot{\theta}_t - M_d/i_f$$

Drive shaft equation is same as previous case.

The Clutch and driveshaft model

- $J_m\ddot{\theta}_m = M_m - M_{fr:m} - (k_c(\theta_m - \theta_t i_t) + C_c(\dot{\theta}_m - \dot{\theta}_t i_t))$
- $\left(J_t + \frac{J_f}{i_f^2}\right)\ddot{\theta}_t = i_t\left(k_c(\theta_m - \theta_t i_t) + c_c\left(\dot{\theta}_m - \dot{\theta}_t i_t\right)\right) - \left(b_t + \frac{b_f}{{i_f}^2}\right)\dot{\theta}_t - 1/i_f(K_d\left(\frac{\theta_t}{i_f} - \theta_w\right) + c_d(-\dot{\theta}_w + \dot{\theta}_t i_t))$
- $(J_w + mr_w^2)\ddot{\theta}_w = K_d\left(\frac{\theta_t}{i_f} - \theta_w\right) + c_d\left(\frac{\dot{\theta}_t}{i_f} - \dot{\theta}_w\right) - (b_w + C_{r2}r_w)\dot{\theta}_w - \frac{1}{2} * \left(c_w A_a r_w^3 \rho_a \dot{\theta}_w^2\right) - r_w m(C_{r1} + gsin(\alpha))$

State space formulation of the linear Clutch and drive-shaft model

- $x_1 = \theta_m - \theta_t i_t$
- $x_2 = \frac{\theta_t}{i_f} - \theta_\omega$
- $x_3 = \dot{\theta}_m$
- $x_4 = \dot{\theta}_t$
- $x_5 = \dot{\theta}_\omega$

- A is given by the matrix:

$$\begin{matrix} 0 & 0 & 1 & -i_t & 0 \\ 0 & 0 & 0 & 1/i_f & -1 \\ -k_c/J_1 & 0 & -C_c/J_1 & C_c i_t/J_1 & 0 \\ k_c i_t/J_2 & -k_d/i_f J_2 & C_c i_t/J_2 & -(C_c i_t^2 + b_2 + \frac{c_d}{i_f^2})/J_2 & C_d/i_f J_2 \end{matrix}$$

- B=[0 0 $1/J_1$ 0 0]^T
- H=[0 0 0 0 -1/J_2]^T

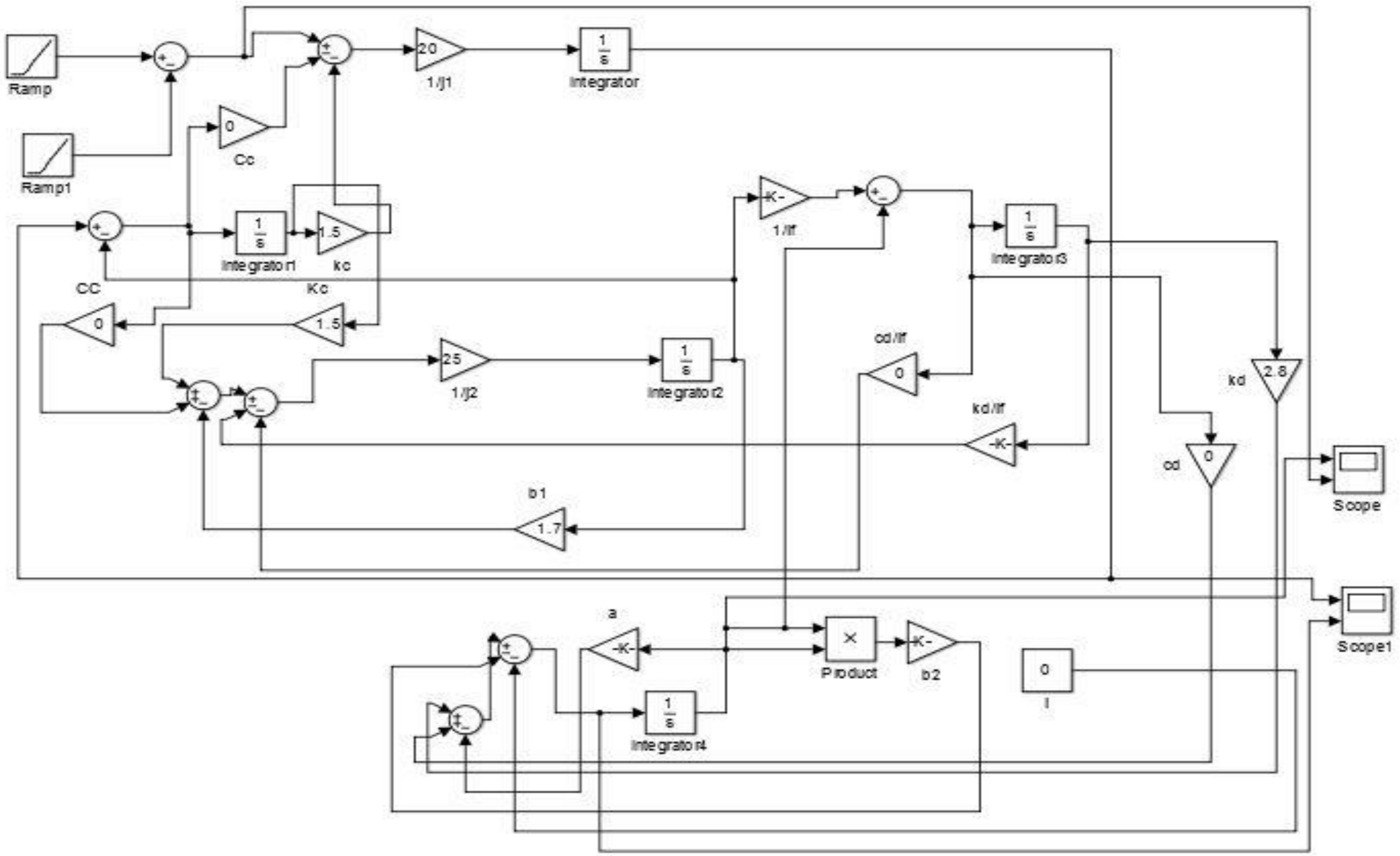
Where

$$i = i_t i_f$$
$$J_1 = J_m$$
$$J_2 = J_t + J_f / i_f^2$$
$$J_2 = J_w + m r_w^2$$

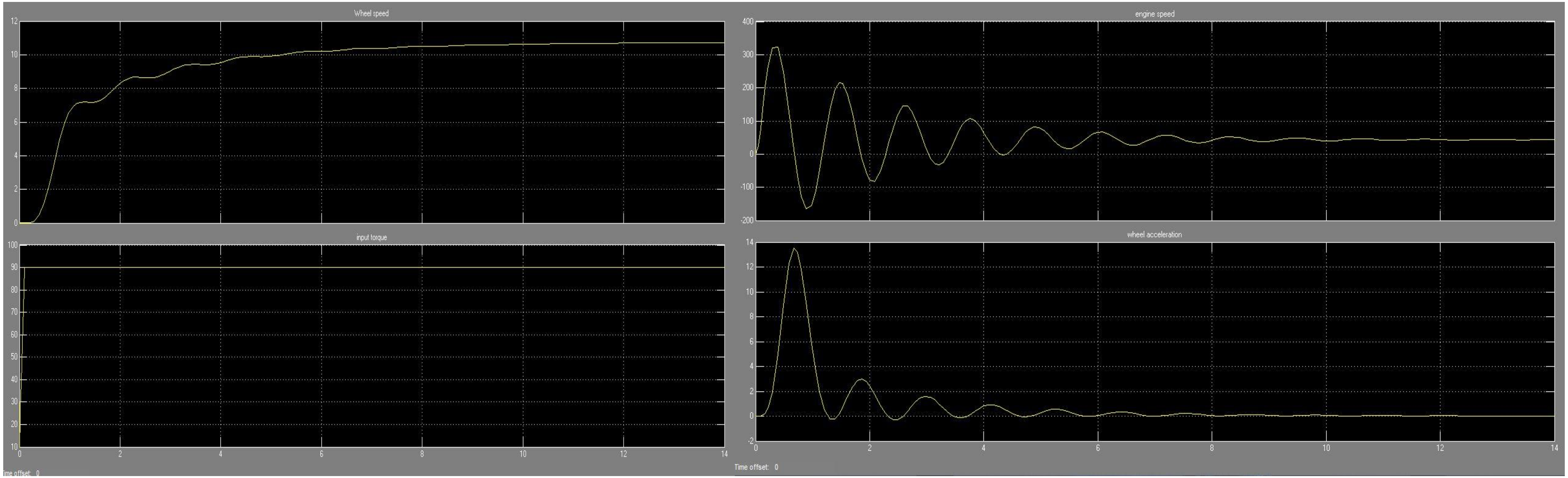
$$b_2 = b_t + \frac{b_f}{i_f^2}$$
$$b_3 = b_w + c_{r2} r_w$$

- $x_1 = \theta_m - \theta_t i_t$
- $x_2 = \left(\frac{\theta_t}{i_f} - \theta_w\right)$
- $x_3 = \dot{\theta}_m$
- $x_4 = \dot{\theta}_t$
- $x_5 = \dot{\theta}_w$

SIMULINK flexible linear clutch and drive shaft model



RAMP RESPONSE



INPUT TORQUE,WHEEL SPEED

ENGINE SPEED,ANG ACC OF WHEEL

Control

Disturbance description:

- Disturbance from the road is assumed to be described by slow varying load l and disturbance v.
- Another disturbance n is a disturbance acting on the input of the system. This disturbance is caused by the firing pulses of the driving torque.

$$\dot{x} = Ax + Bu + Bn\ + Hl + Hv$$

Sensor :

- Sensor output is defined as combination of states given by matrix C.

y= Cx +e

Performance output:

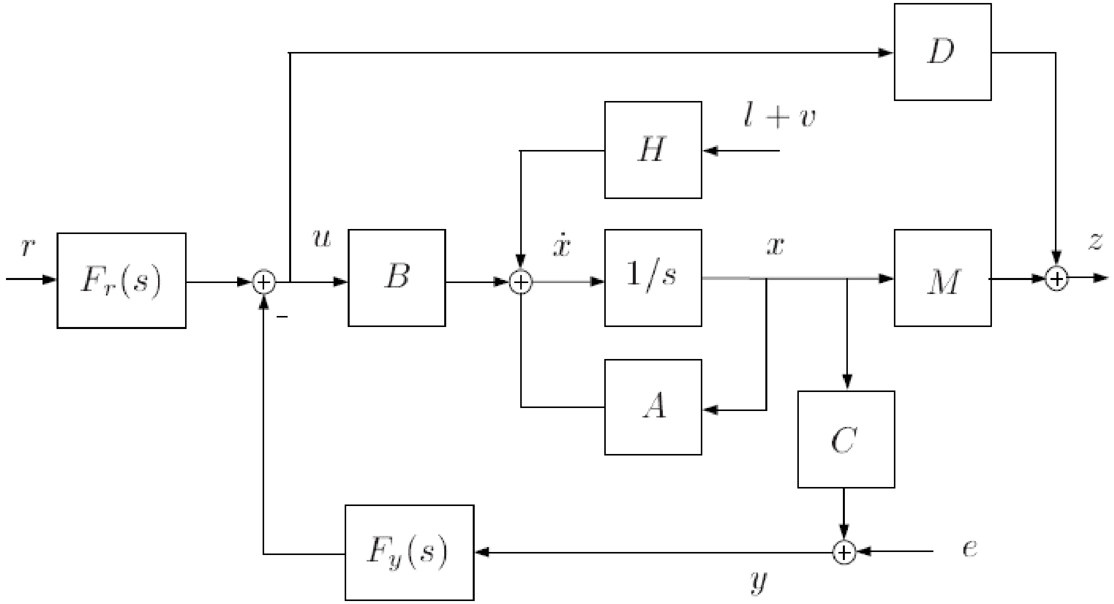
Z = Mx +Du (M and D can be defined as desired)

The control signal 'u' is assumed to be linear function of states.

$$u = \ l_o r\ -\ K_c x^{\wedge}$$ K_c is state feedback matrix

$State\ estimates$ $\dot{x}^{\wedge} = Ax^{\wedge} + Bu + \ K_f(y - Cx^{\wedge})$ K_f is Kalman gain

Control diagram



- F_r *and* F_y *are controllers*
- Identifying the controllers from control diagram
- $F_y(s) = K_c\,(sI - A + K_f C + BK_c)^{-1} K_f$
- $F_r(s) = l_o\,(1 - K_c(sI - A + K_f C + BK_c)^{-1} B)$

Conclusions

- Mathematical model for driveshaft model and linear clutch driveshaft model are created in Simulink.
- Wheel speed characteristics are studied and controlled using PID controller.
- Overshoot decreased and Settling time decreased.

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