Vehicle Dynamics and Simulation

Drivetrain Dynamics

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Note

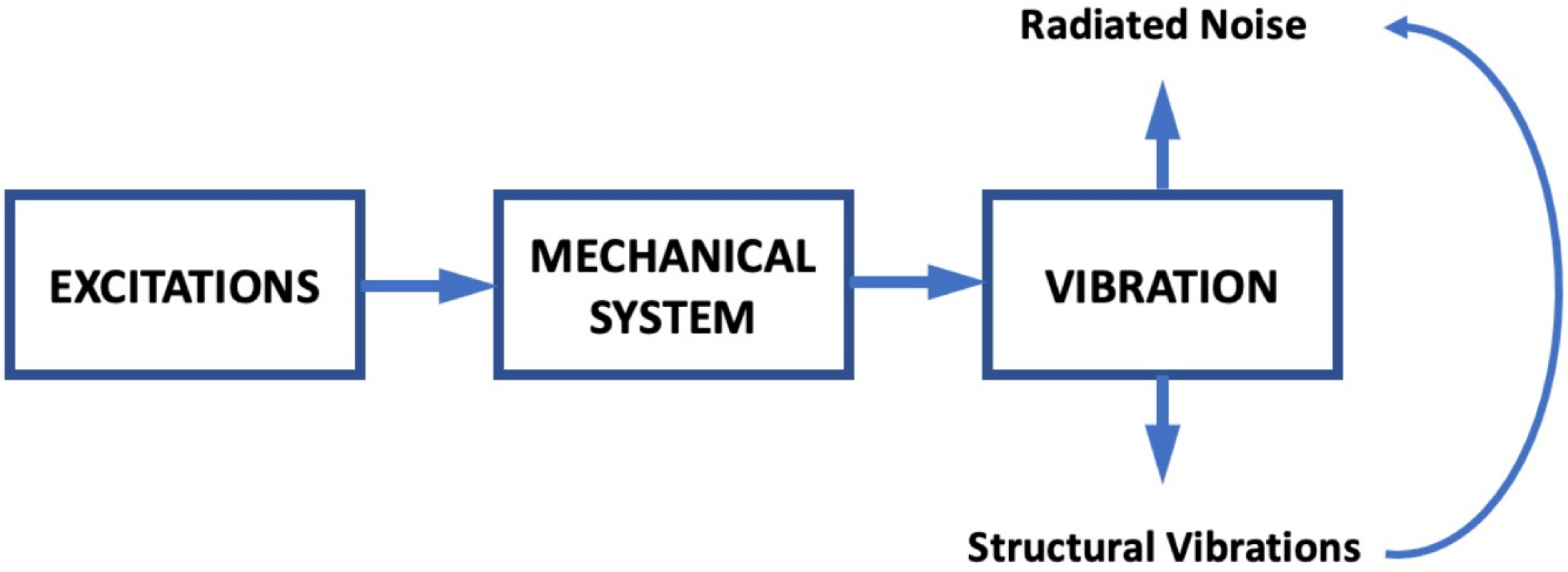
- The test is on Tuesday at 2pm
- Location is SM109 computer lab
- There will be **no** online session (contact me before 9am on the day if you are unable to attend)

- Main focus;
  - Application of fundamentals to Drivetrain Dynamics
  - Drivetrain dynamic model use.

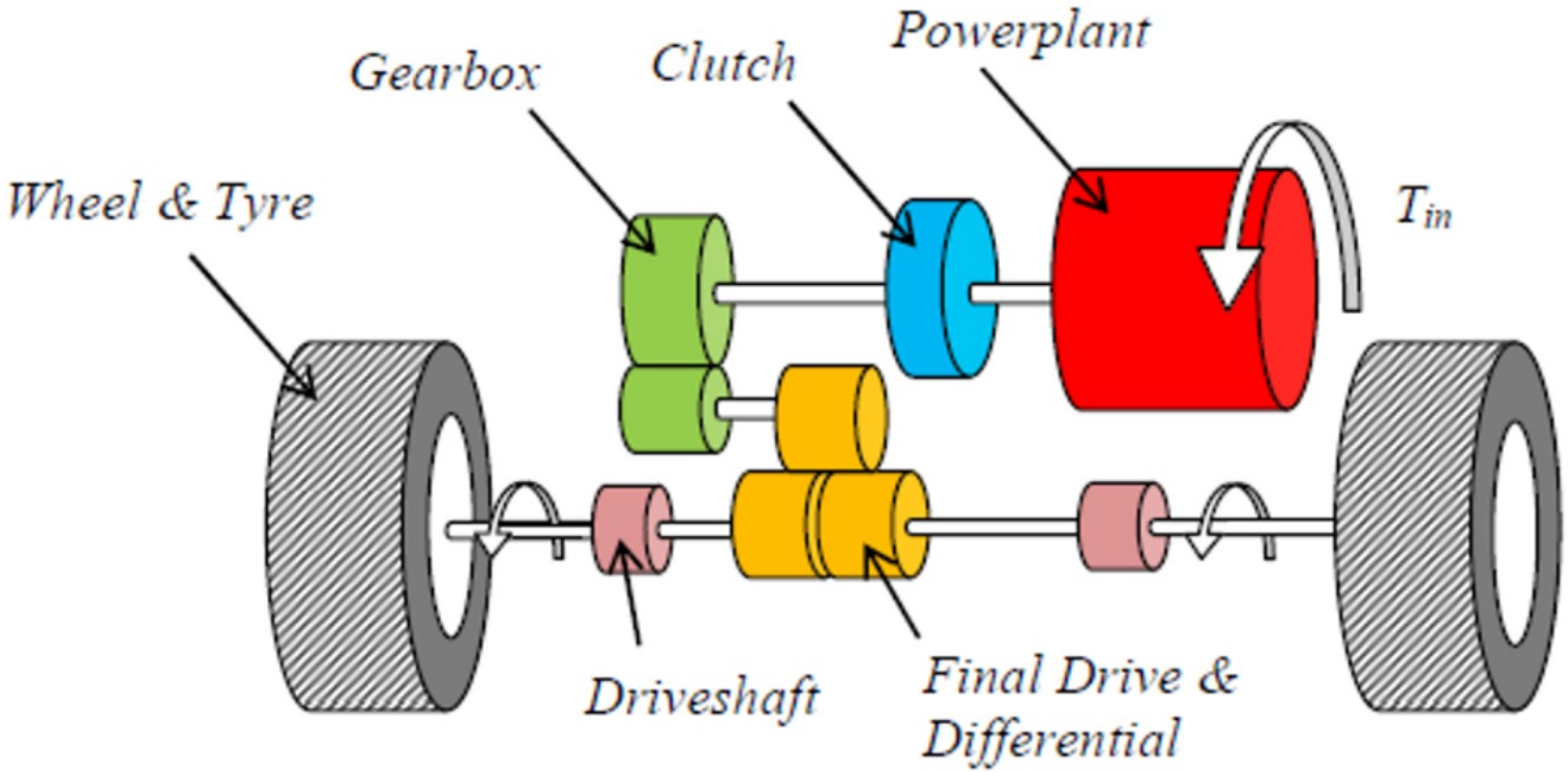
Lecture Overview

- Drivetrain as a vibrational system
- Torsional drivetrain model
- Excitation sources
- Driveline Components
- Vibration analysis

The Drivetrain System

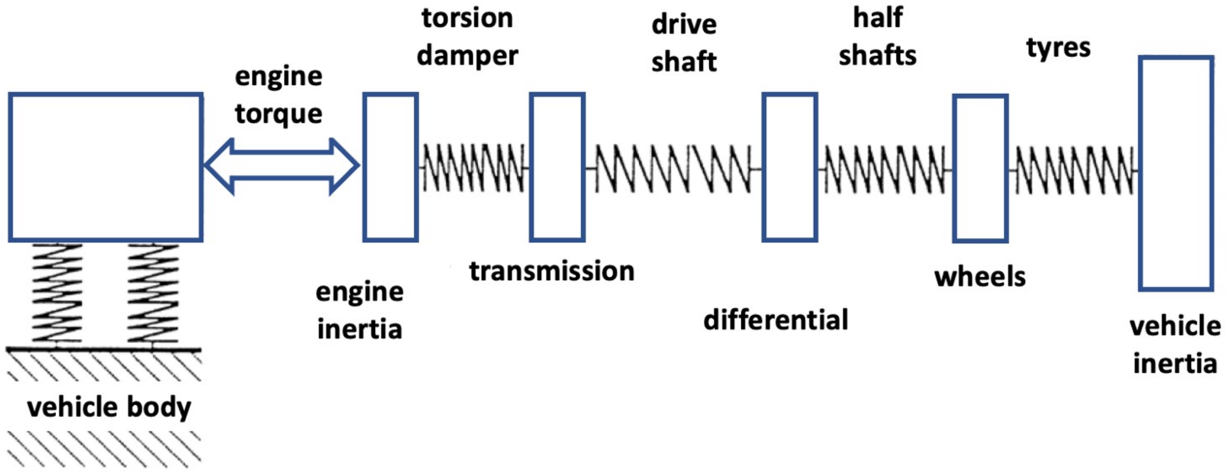


The Drivetrain System



The Drivetrain System

- The drivetrain can be represented as a number of springs and masses
- Each mass is isolated as a point-mass
- These types of models are known as lumped parameter models
- Control of vibration is achieved by;
  - Reducing excitation
  - Changing stiffness and damping (resonance frequencies)



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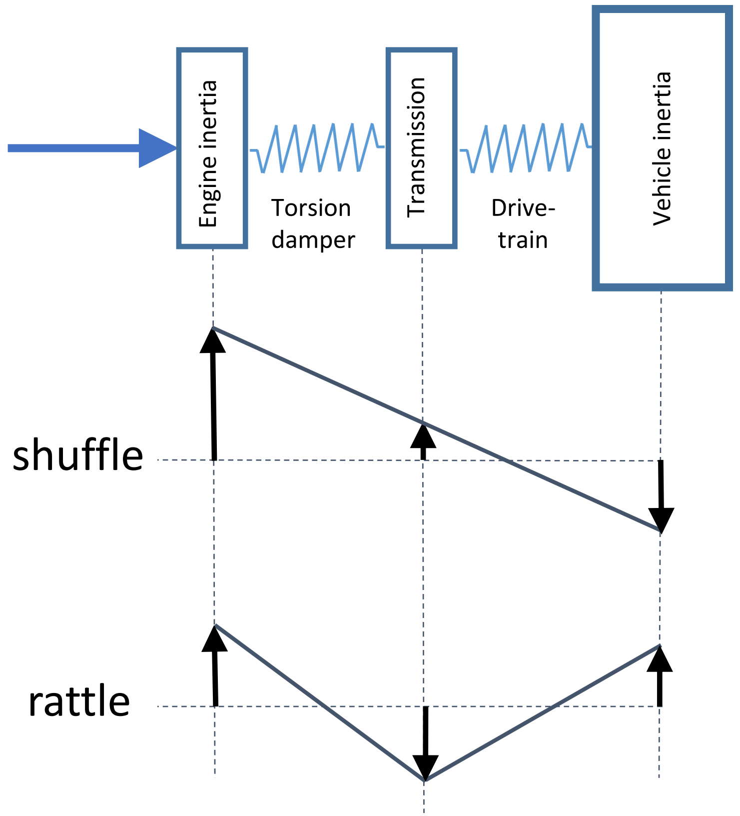
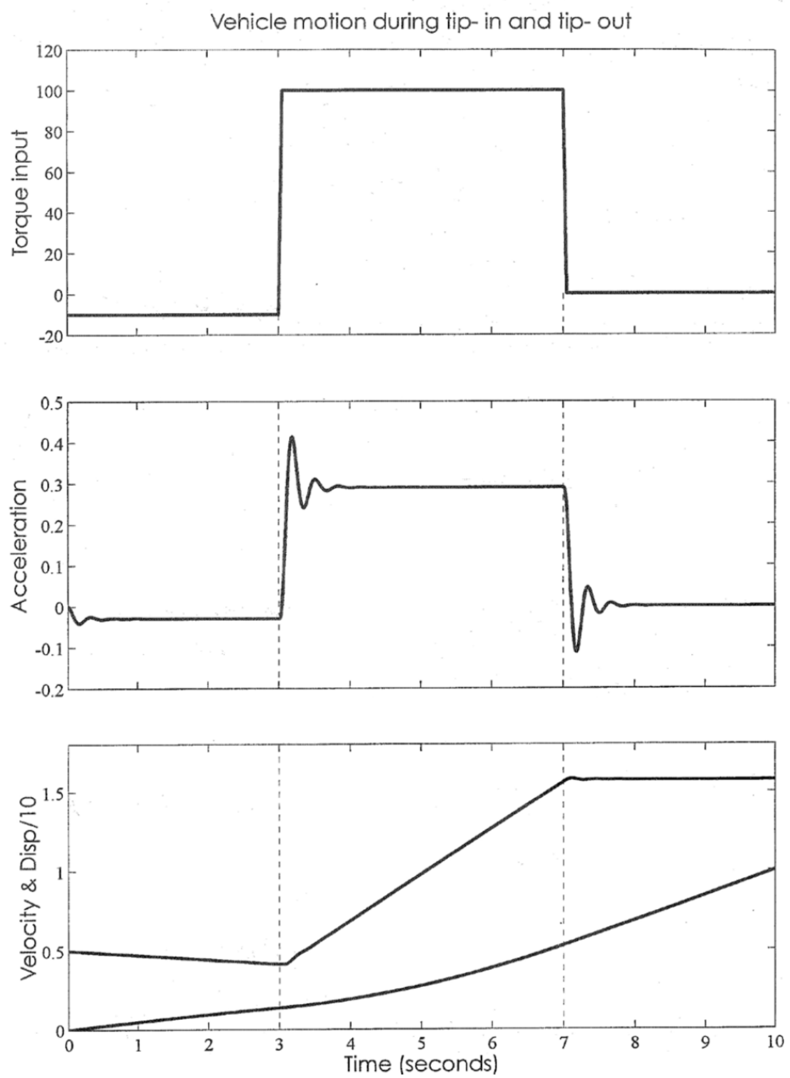
The simplest useful model

- Three main modes;
  - Shuffle (4-12 Hz)
  - Rattle (40 – 80 Hz)
  - Rigid body rotation
- And others;
  - Boom (interior compartment)
  - Judder (low frequency on clutch engagement)
  - Clonk (lash in driveline)

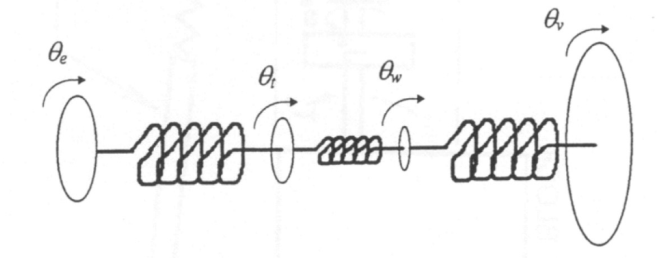


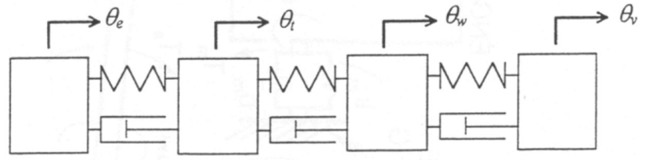
Response to step input

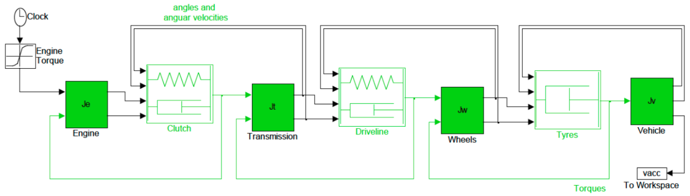


More advanced models

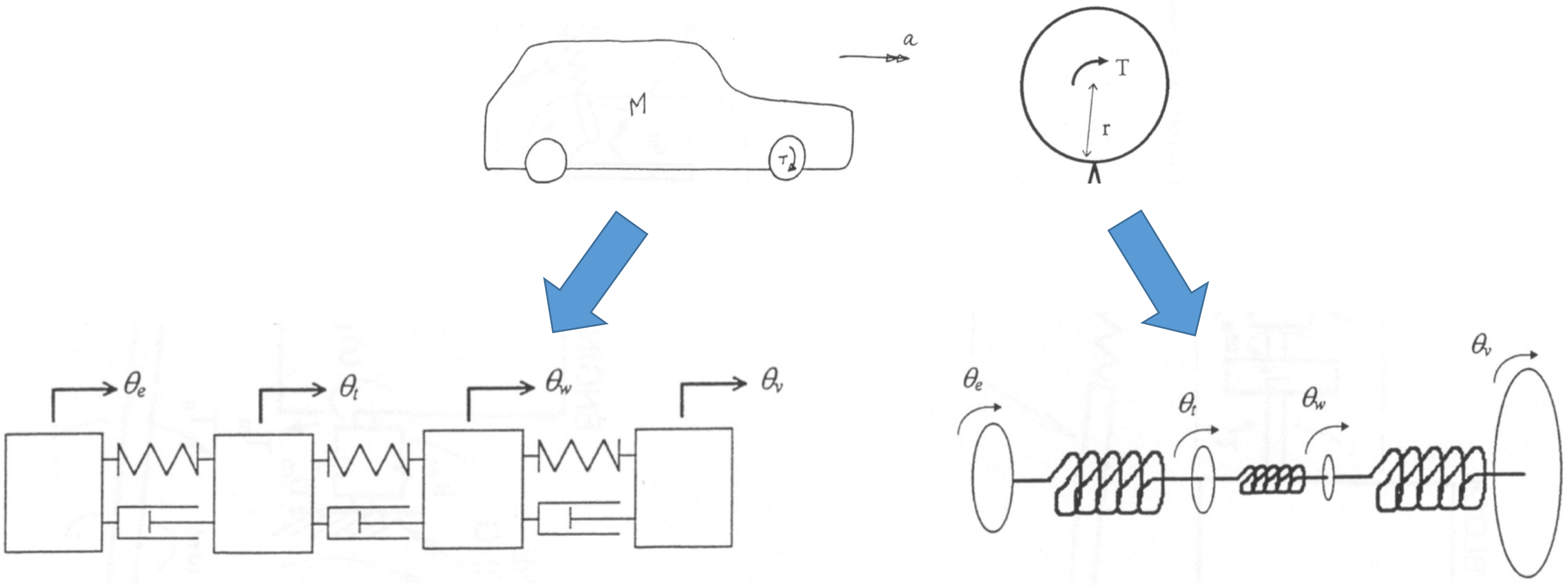
- Three, four, six, …, twenty! mass models
  - Complexity driven by requirements
- Can add masses to suit
- Other components in the drivetrain are important
  - Clutch
  - Differential
  - Actuators and related controls



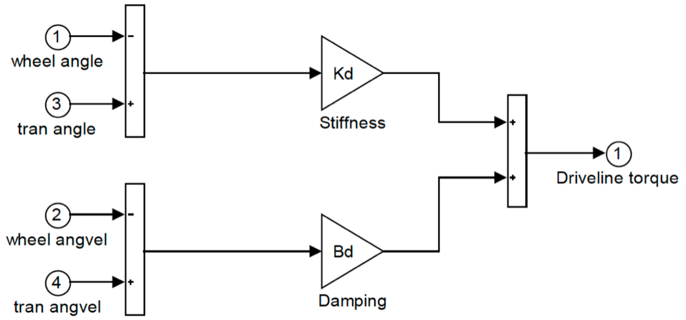




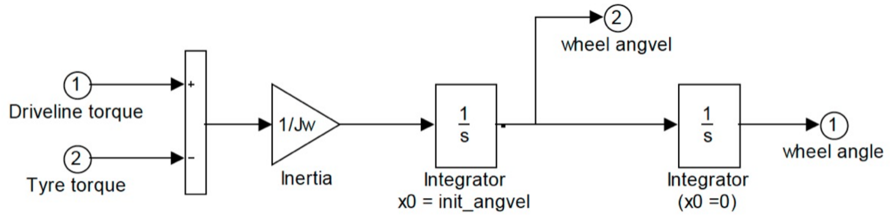
Rotation or Translation?



Standardised (Simulink) components



Spring-damper



Mass

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Sources of excitation

- Main sources of excitation
  - Combustion
  - Step-in and step-out
  - Electric machine
  - HEV transitions
  - Cylinder deactivation
- Transmission
  - Gear meshing
  - Ratio changes
- Driveshaft
  - Universal joints
- Tyres and wheels
  - Runout
  - Mass imbalance
  - Stiffness variations

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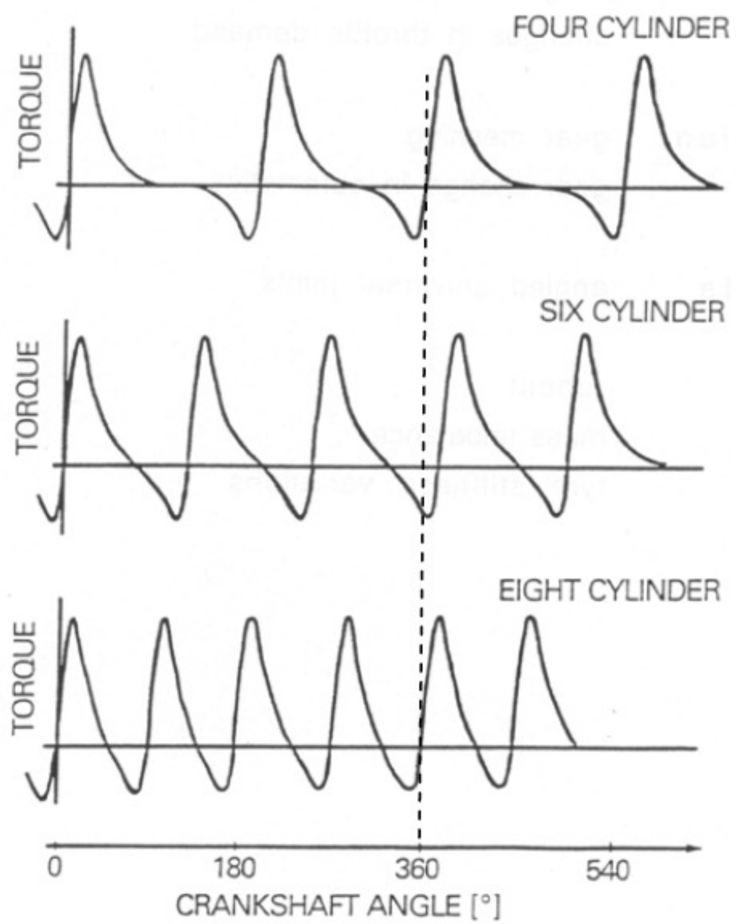
Excitation - Combustion

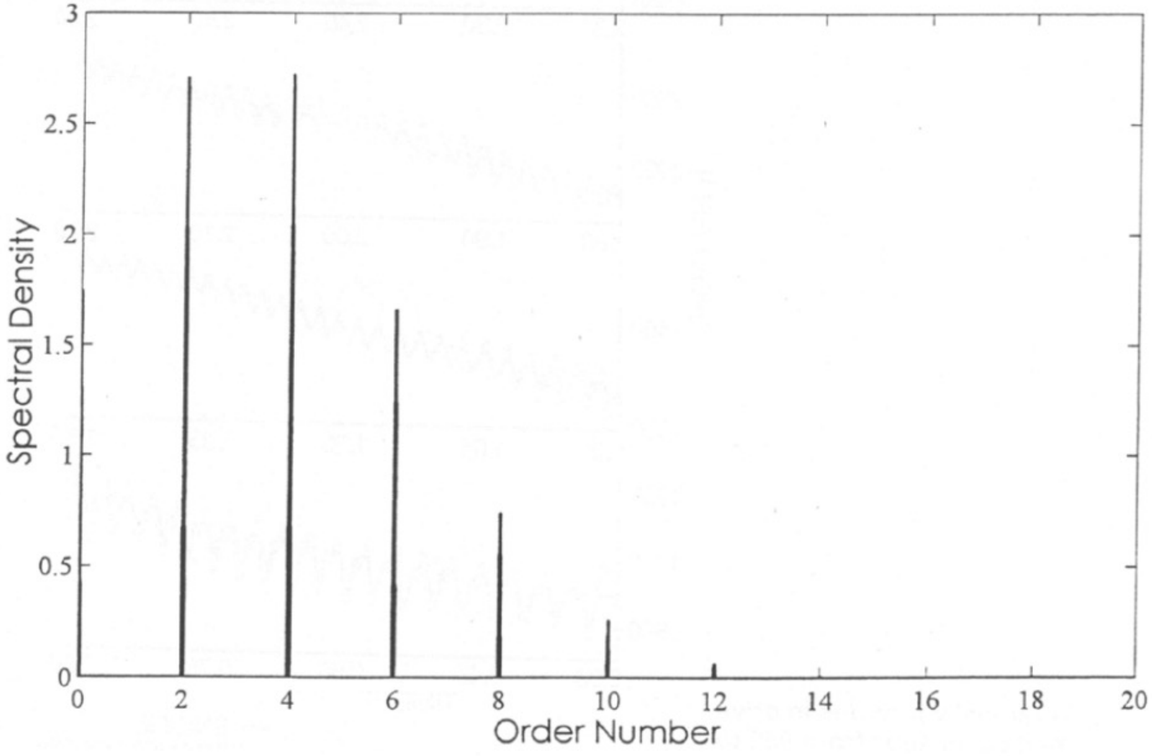
- Firing of engine causes torque pulses
- Varies in frequency
  - Number of cylinders
  - Engine speed
- Other events
  - Valve opening and closing (x1 per rotation)
  - Injector opening and closing (x1 per cycle)



Excitation - Combustion

- Spectral analysis (FFT) to look at frequency content of measured response
- Can normalise spectral analysis result with rotational speed
  - Shows how response relates to engine speed
- Can also create interaction plot / Campbell diagram to show response relative to excitation (engine).

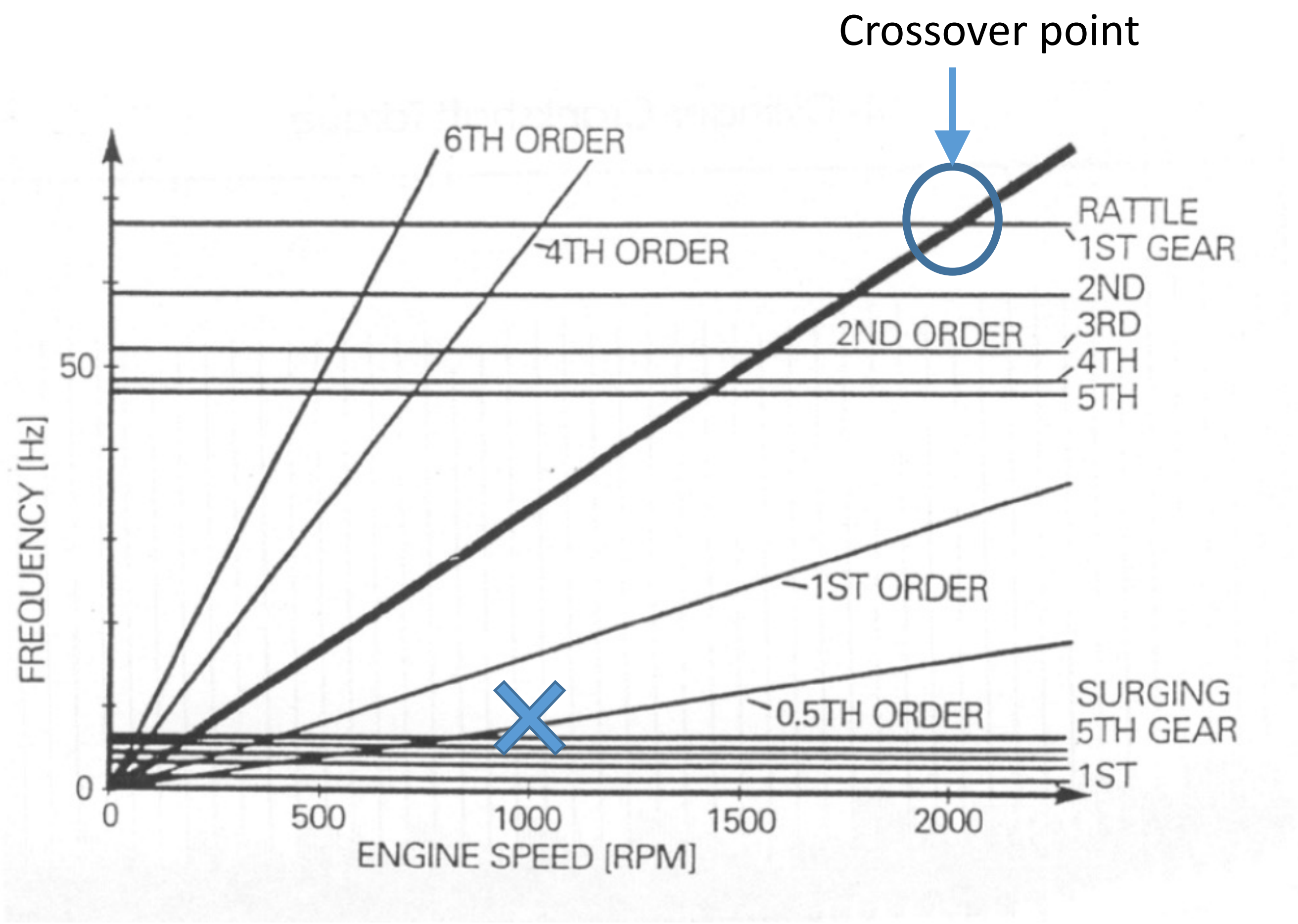
Engine order vibration



Each peak in the diagram above represents a frequency described in terms of the order i.e. frequency/rotational frequency

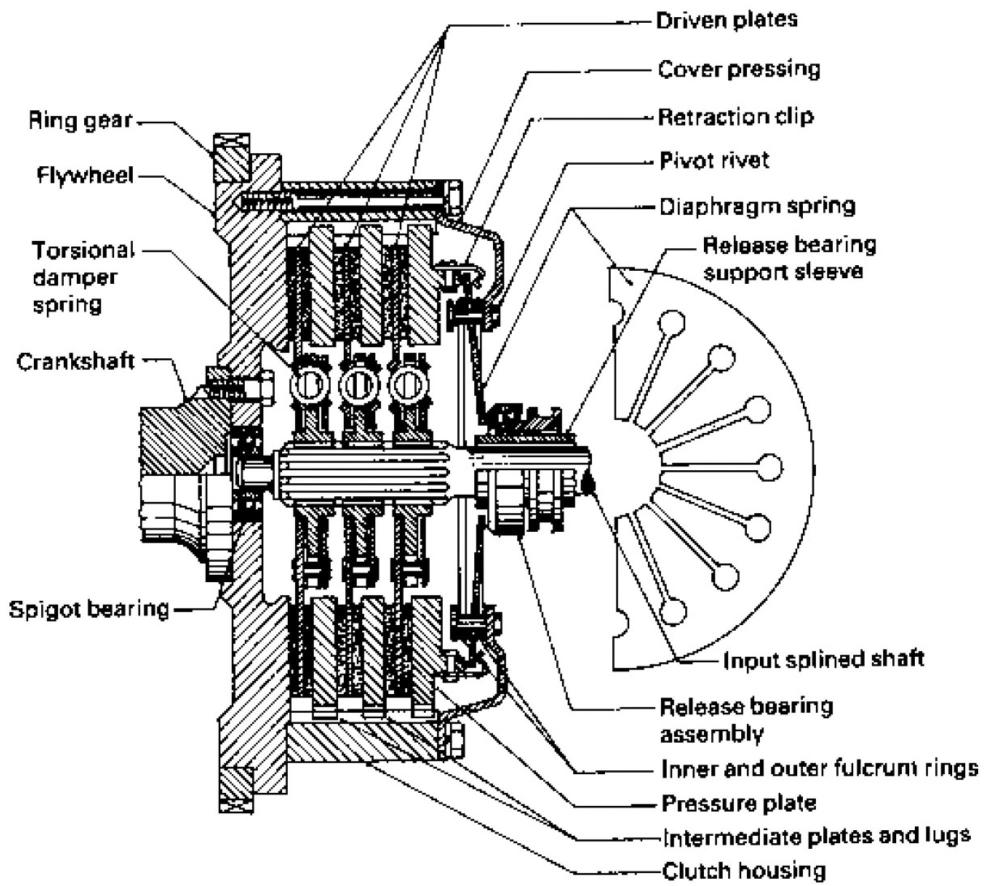
Campbell Diagram

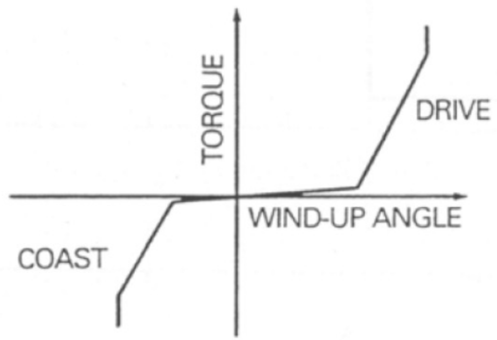
- Order is frequency / rotational frequency
- 1st order is 1:1 mapping e.g. at 1000 rpm = 1000 / 60 cycles per second (Hz).
- Where the ‘order’ crosses the ‘resonance’ line is the point of max vibration.



Torsion Damper

- Used as first line of defence against excitation in the driveline.
- Different physical arrangements.
- Nonlinear spring rate with some damping.





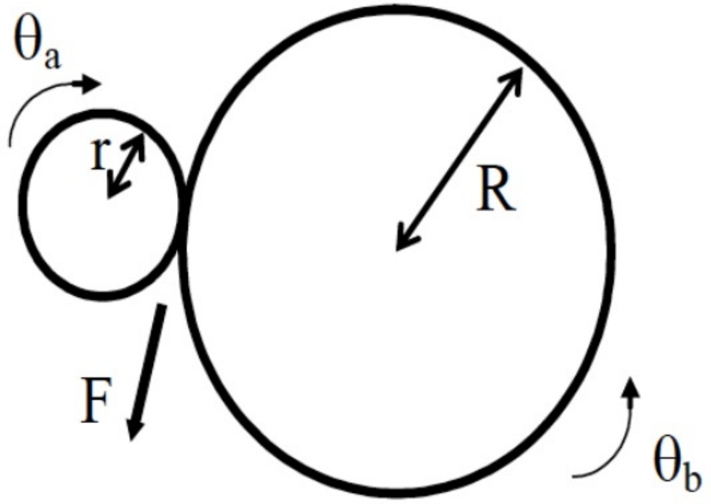
Gearbox

- Can be modelled as a single pair (and 'switched').
- Output torque is calculated (T_a is input torque);

$$T_b = GT_a$$

- With the gear ratio, G given by;

$$G = R/r$$



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Simple Drivetrain Model

- Three inertias
- Compliantly connected by rotational springs
- Engine inertia accel;

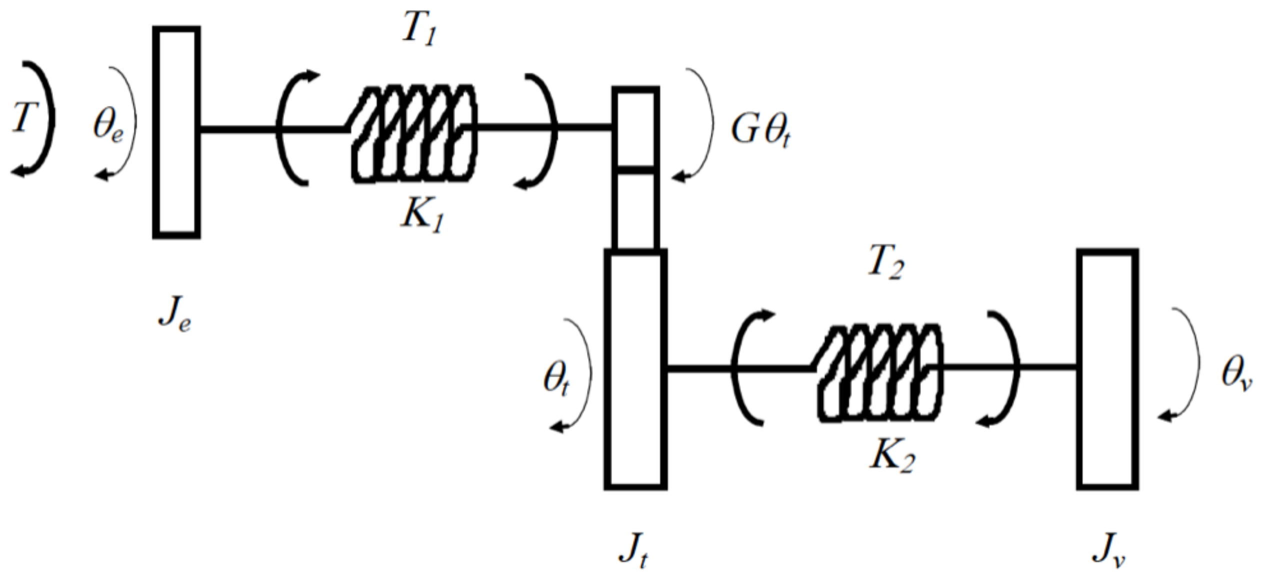
$$J_e\ddot{\theta}_e = T - T_1$$

- Transmission inertia accel;

$$J_t\ddot{\theta}_t = GT_1 - T_2$$

- Vehicle inertia accel;

$$J_v\ddot{\theta}_v = T_2$$



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Simple Drivetrain Model

- Torque transmitted between engine and transmission;

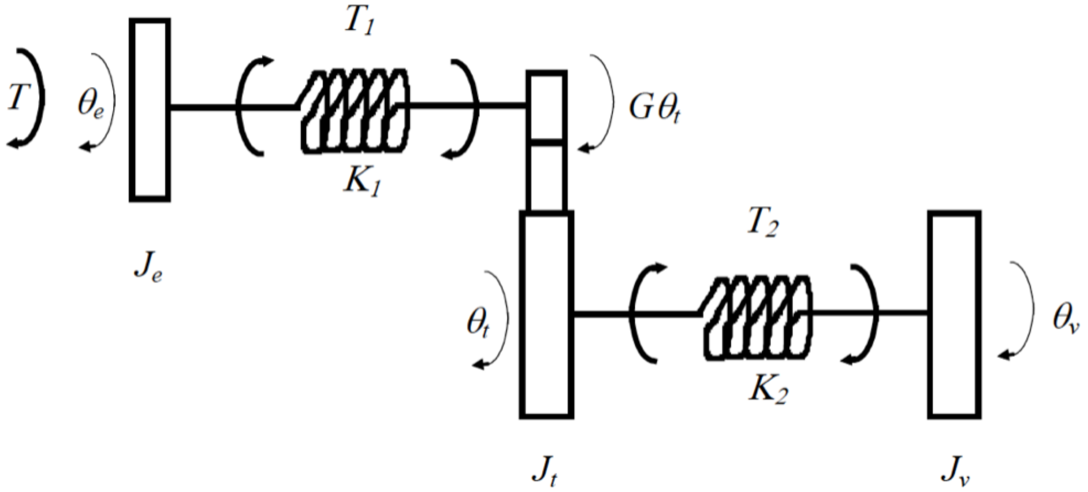
$$T_1 = K_1(\theta_e - G\theta_t)$$

- Torque transmitted between transmission and vehicle;

$$T_2 = K_2(\theta_t - \theta_v)$$

- So that;

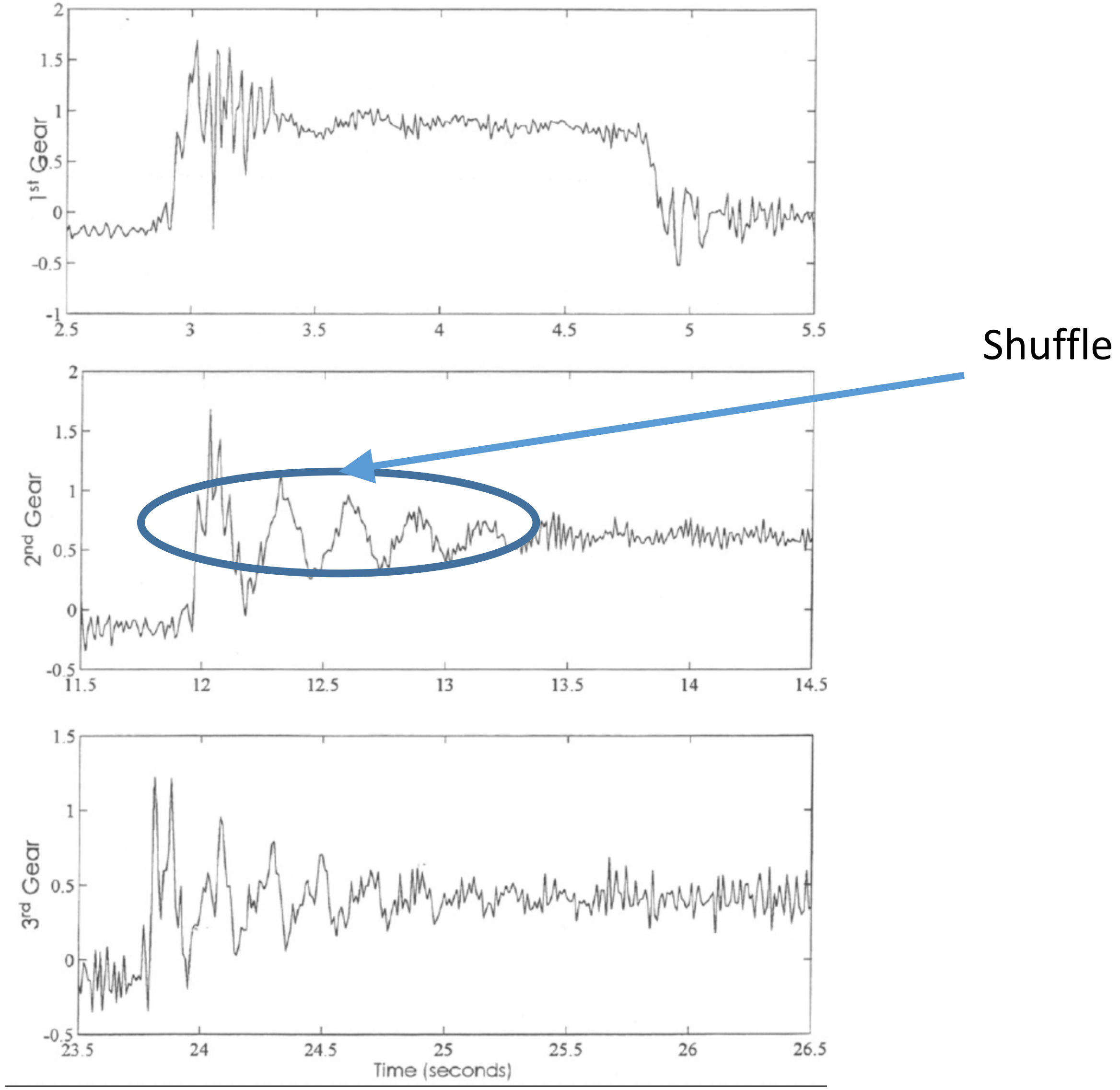
$$J_e\ddot{\theta}_e = T - K_1\left(\theta_e - G\theta_t\right)$$
$$J_t\ddot{\theta}_t = GK_1\left(\theta_e - G\theta_t\right) - K_2\left(\theta_t - \theta_v\right)$$
$$J_v\ddot{\theta}_v = K_2\left(\theta_t - \theta_v\right)$$



- A less messy formulation is (see notes for derivation);

$$J_e^*\ddot{\theta}_e^* = GT - K_1^*\left(\theta_e^* - \theta_t\right)$$
$$J_t\ddot{\theta}_t = K_1\left(\theta_e^* - \theta_t\right) - K_2\left(\theta_t - \theta_v\right)$$
$$J_v\ddot{\theta}_v = K_2\left(\theta_t - \theta_v\right)$$

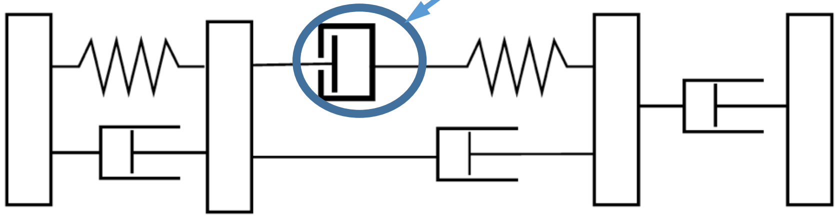
Measured Response (Mondeo 1.8l)



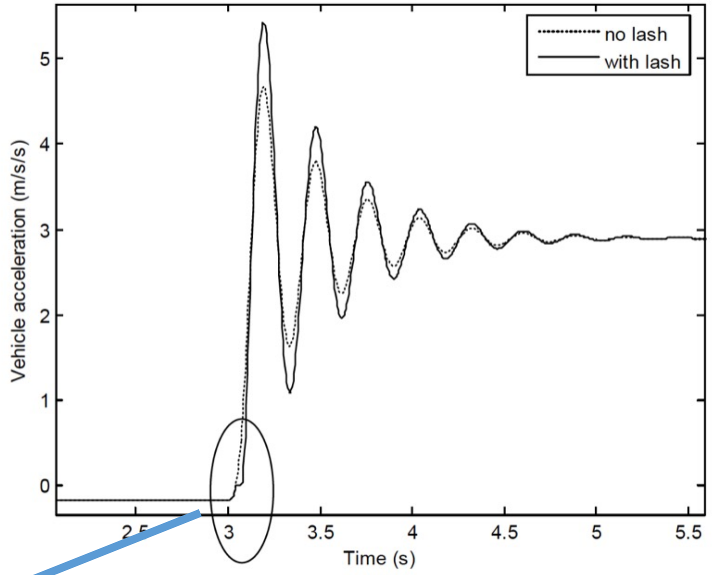
Driveline Nonlinearities

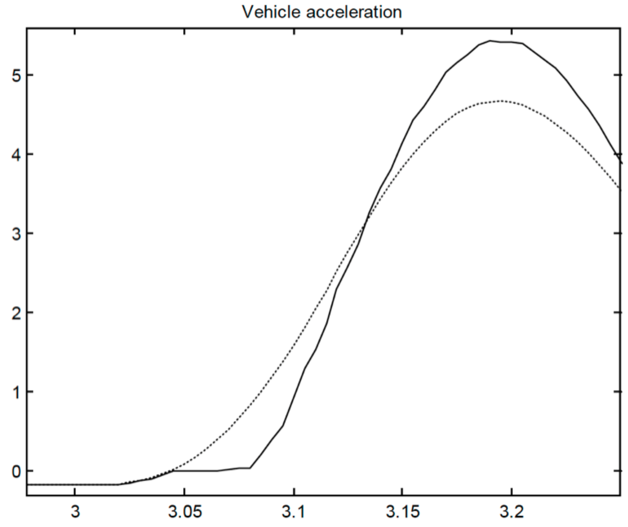
- There are many sources of nonlinearity.
- Typically include 'lash' as most significant.
- Lash element available in Simulink

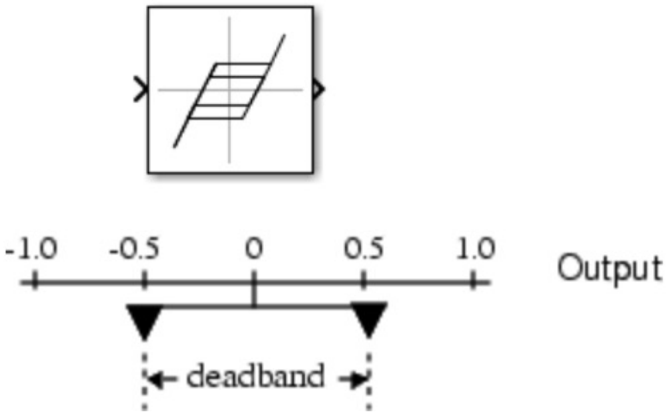
Lash element



Step input

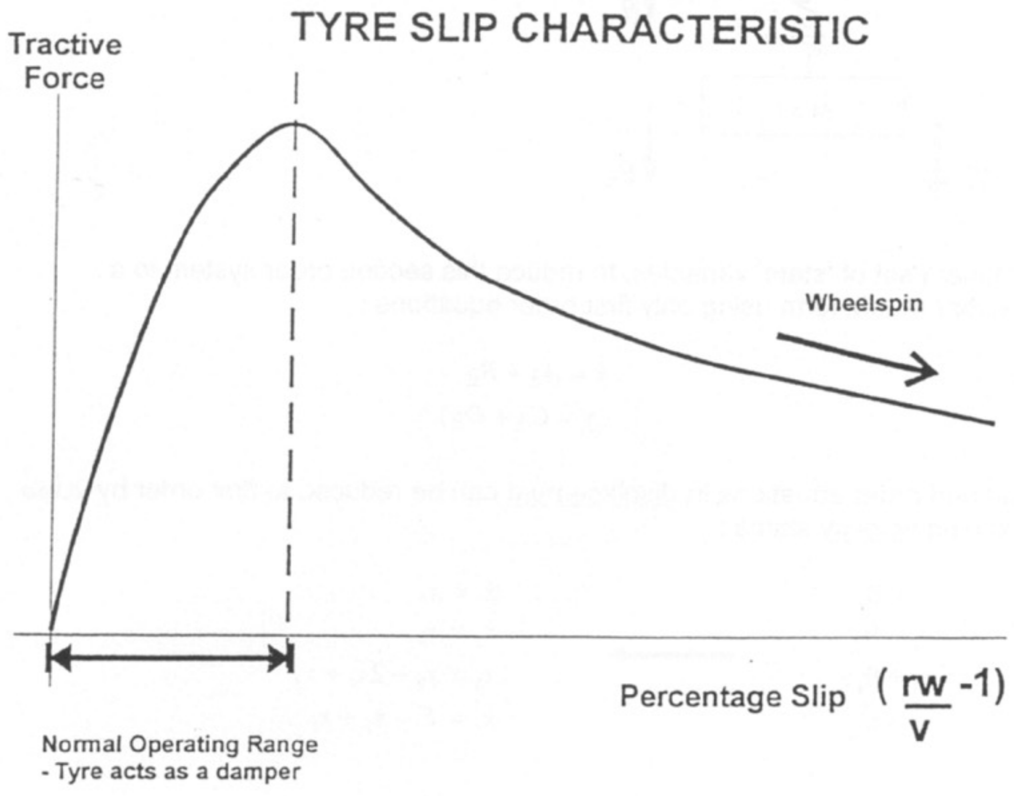






A Simple Tyre Model

- Longitudinal force in acceleration is generated through contact patch.
- The amount of force generated is proportional to ‘slip’ i.e. velocity difference between tyre (translation) and vehicle (translation).
- Sharply rises, reaches a peak then falls. Stiction vs viscous friction.



Vibration Analysis

Create state matrix

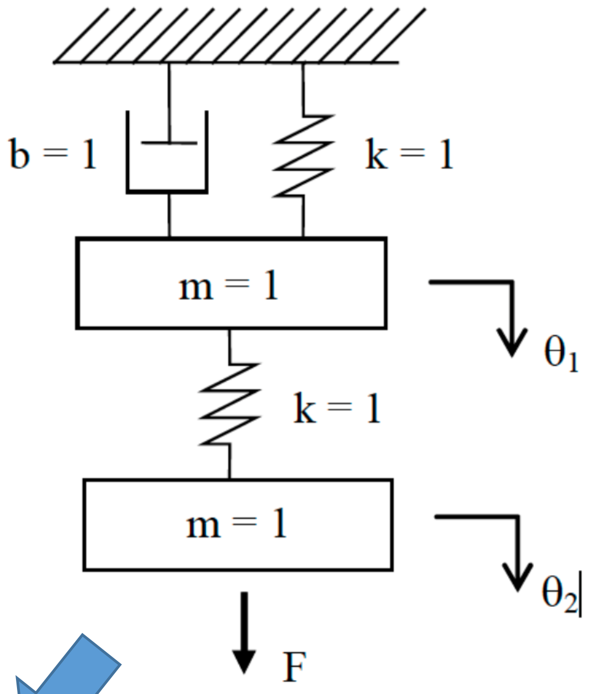
$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{bmatrix} = \begin{bmatrix}0 & 0 & 1 & 0\\ 0 & 0 & 0 & 1\\ -2 & 1 & -1 & 0\\ 1 & -1 & 0 & 0\end{bmatrix}\begin{bmatrix}x_1\\ x_2\\ x_3\\ x_4\end{bmatrix} + \begin{bmatrix}0\\ 0\\ 0\\ 1\end{bmatrix}\mathbf{F}$$

$$\dot{\mathbf{x}} = A\mathbf{x} + B\mathbf{u}$$

Create output matrix

$$\mathbf{y} = C\mathbf{x} + D\mathbf{u}$$

Draw schematic



Determine system equations

$$F - l(\theta_2 - \theta_1) = l\ddot{\theta}_2$$

$$l(\theta_2 - \theta_1) - l\theta_1 - l\dot{\theta}_1 = l\ddot{\theta}_1$$

Define states

$$x_1 = \theta_1$$
$$x_2 = \theta_2$$
$$x_3 = \dot{\theta}_1$$
$$x_4 = \dot{\theta}_2$$

Substitute state variables

$$\dot{x}_1 = x_3$$
$$\dot{x}_2 = x_4$$
$$\dot{x}_3 = x_2 - 2x_1 - x_3$$
$$\dot{x}_4 = F - x_2 + x_1$$

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Vibration Analysis – Modal analysis

Previously covered in Section 6

- Remember that;

$$\theta(t) = \mathrm{Re}\left\{\mathbf{u}_1 e^{\lambda_1 t} + \mathbf{u}_2 e^{\lambda_2 t} + \ldots + \mathbf{u}_n e^{\lambda_n t}\right\}$$

- Where each term is;

$$\mathbf{u}_1 e^{\lambda_1 t} = \mathbf{u}_1 e^{(\sigma + j\omega)t} = \mathbf{u}_1 e^{\sigma t} e^{j\omega t} = \mathbf{u}_1 e^{\sigma t}(\cos(\omega t) + i\sin(\omega t))$$

- For a single component;

$$\theta(t) = \mathbf{u}_1 e^{\lambda_1 t} \longrightarrow \dot{\theta}(t) = \lambda_1 \mathbf{u}_1 e^{\lambda_1 t}$$

- So that;

$\mathbf{x}(t) = \mathbf{v}_1 e^{\lambda_1 t}$ where; $\mathbf{v}_1 = \begin{bmatrix} \mathbf{u}_1 \\ \lambda_1 \mathbf{u}_1 \end{bmatrix}$

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Vibration Analysis – Modal analysis

Previously covered in Section 6

- For free vibration;

$$\lambda_1 \mathbf{v}_1 e^{\lambda_1 t} = A\mathbf{v}_1 e^{\lambda_1 t}$$

- So that;

$$\lambda_1 \mathbf{v}_1 = A\mathbf{v}_1$$

- Which means that λ_1 is a eigenvalue of A and v_1 is the corresponding eigenvector – from the definition of what an eigenvalue is.

- For the spring-mass-damper system previously (given parameter values);

$$\begin{aligned}
\lambda_1 &= -0.35 + 1.5\mathrm{j} \\
\lambda_2 &= -0.35 - 1.5\mathrm{j} \\
\lambda_3 &= -0.15 + 0.63\mathrm{j} \\
\lambda_4 &= -0.15 - 0.63\mathrm{j}
\end{aligned}$$

Matlab Example Output

[V,D] = eig(A)

```
D =

   1.0e+02 *

   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i  -5.4924 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.8696 + 3.6644i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.8696 - 3.6644i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.0332 + 0.2219i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.0332 - 0.2219i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.0000 + 0.0000i


V =

   0.0000 + 0.0000i   0.0000 + 0.0000i  -0.0000 - 0.0001i  -0.0000 + 0.0001i  -0.0048 - 0.0322i  -0.0048 + 0.0322i  -0.5000 + 0.0000i  -0.5000 + 0.0000i
   0.0000 + 0.0000i   0.0001 + 0.0000i   0.0006 + 0.0026i   0.0006 - 0.0026i  -0.0053 - 0.0278i  -0.0053 + 0.0278i  -0.5000 + 0.0000i  -0.5000 + 0.0000i
   0.0000 + 0.0000i  -0.0307 + 0.0000i  -0.9995 + 0.0000i  -0.9995 + 0.0000i   0.6340 - 0.0265i   0.6340 + 0.0265i  -0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i  -0.0001 + 0.0000i   0.0269 + 0.0007i   0.0269 - 0.0007i   0.7306 + 0.0000i   0.7306 + 0.0000i  -0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0018 + 0.0000i   0.0000 - 0.0000i   0.0000 + 0.0000i  -0.0096 + 0.0041i  -0.0096 - 0.0041i  -0.5000 + 0.0000i  -0.5000 + 0.0000i
   1.0000 + 0.0000i  -0.0000 + 0.0000i  -0.0000 - 0.0000i  -0.0000 + 0.0000i   0.0007 + 0.0037i   0.0007 - 0.0037i  -0.5000 + 0.0000i  -0.5000 + 0.0000i
   0.0000 + 0.0000i  -0.9994 + 0.0000i   0.0104 + 0.0139i   0.0104 - 0.0139i  -0.0592 - 0.2260i  -0.0592 + 0.2260i  -0.0000 + 0.0000i   0.0000 + 0.0000i
   0.0000 + 0.0000i   0.0151 + 0.0000i   0.0002 - 0.0003i   0.0002 + 0.0003i  -0.0839 + 0.0035i  -0.0839 - 0.0035i  -0.0000 + 0.0000i   0.0000 + 0.0000i
```

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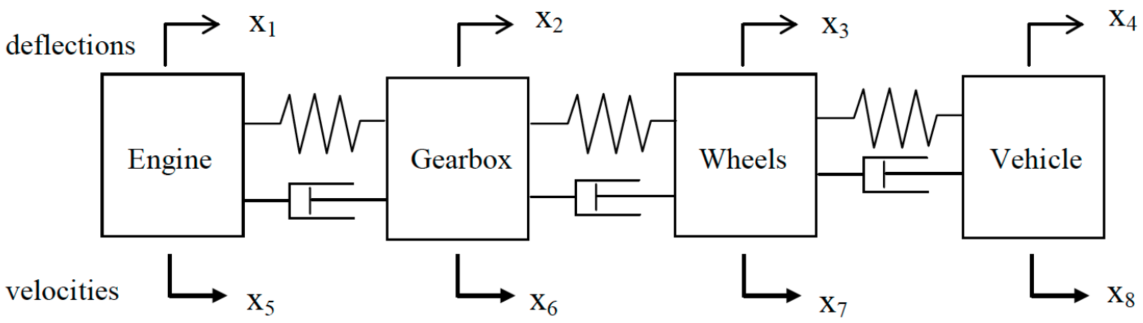
Eigenstructure – Four mass driveline model




```
[A,B,C,D] = linmod('drive4');
[V,D] = eig(A);
```

...and as if by magic!

	λ_1 = 0	λ_2 = -779.5	λ_3 = -86.9+366.4i	λ_4 = -86.9-366.4i
x1	0	0.0000	-0.0000 - 0.0001i	-0.0000 + 0.0001i
x2	0	-0.0000	0.0006 + 0.0026i	0.0006 - 0.0026i
x3	0	-0.0013	0.0000 - 0.0000i	0.0000 + 0.0000i
x4	1.000	0.0000	-0.0000 - 0.0001i	-0.0000 + 0.0001i
x5	0	-0.0000	0.0269 + 0.0007i	0.0269 - 0.0007i
x6	0	0.0129	-0.9996	-0.9996
x7	0	0.9998	0.0059 + 0.0116i	0.0059 - 0.0116i
x8	0	-0.0150	0.0003 - 0.0003i	0.0003 + 0.0003i

	λ_5 = -2.40 + 22.26i	λ_6 = -2.40- 22.26i	λ_7 = -1.4676e-006	λ_8 = 1.4676e-006
x1	-0.0035 - 0.0329i	-0.0035 - 0.0329i	0.5000	-0.5000
x2	-0.0039 - 0.0284i	-0.0039 - 0.0284i	0.5000	-0.5000
x3	-0.0068 + 0.0040i	-0.0068 + 0.0040i	0.5000	-0.5000
x4	0.0005 + 0.0038i	0.0005 + 0.0038i	0.5000	-0.5000
x5	0.7403	0.7403	-0.0000	-0.0000
x6	0.6412 - 0.0186i	0.6412 - 0.0186i	-0.0000	-0.0000
x7	-0.0720 - 0.1618i	-0.0720 - 0.1618i	-0.0000	-0.0000
x8	-0.0848 + 0.0025i	-0.0848 + 0.0025i	-0.0000	-0.0000

Eigenstructure – Four mass driveline model

- Normalising (as in previous example)
  - Shuffle;

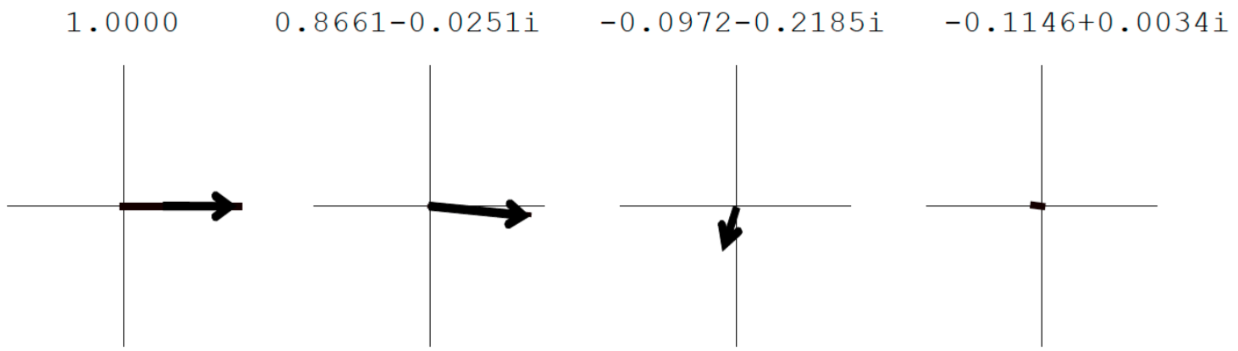
0.7403	/ 0.7403	=	1.0000
0.6412 - 0.0186i	/ 0.7403	=	0.8661 - 0.0251i
-0.0720 - 0.1618i	/ 0.7403	=	-0.0972 - 0.2185i
-0.0848 + 0.0025i	/ 0.7403	=	-0.1146 + 0.0034i

  - Rattle:

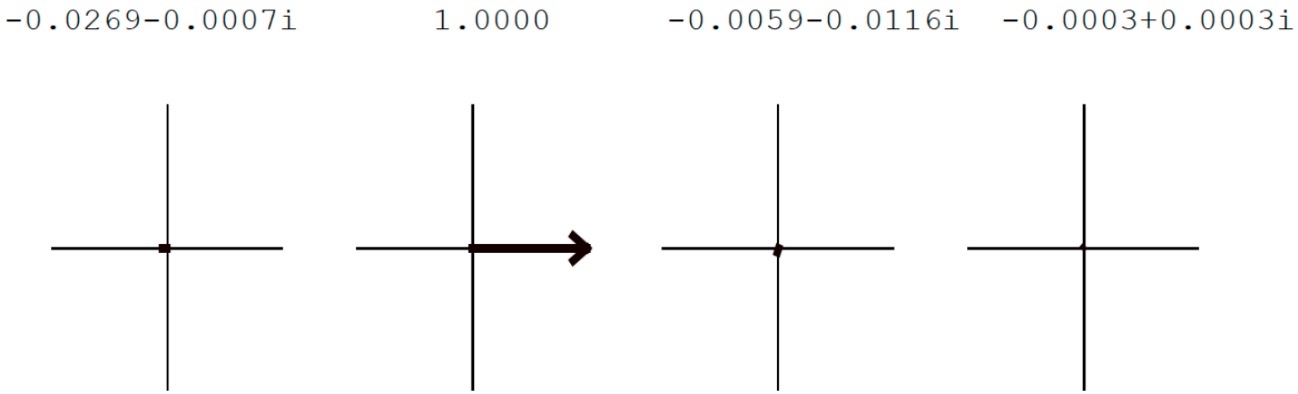
0.0269 + 0.0007i	/ -0.9996	=	-0.0269 - 0.0007i
-0.9996	/ -0.9996	=	1.0000
0.0059 + 0.0116i	/ -0.9996	=	-0.0059 - 0.0116i
0.0003 - 0.0003i	/ -0.9996	=	-0.0003 + 0.0003i

Eigenstructure – Four mass driveline model

- And plotting
  - Shuffle;



  - Rattle;



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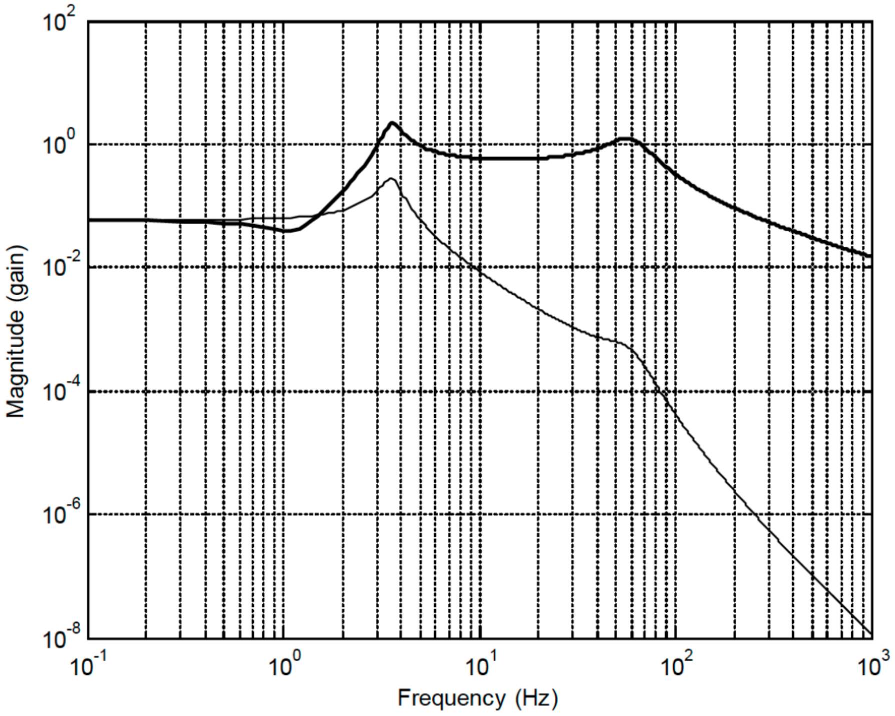
Eigenstructure – Four mass driveline model

- Bode plot to show frequency response between input torque, vehicle and transmission acceleration.

```
[A,B,C,D] = linmod('drive4');
sys = ss(A,B,C,D);
f = [0.1:0.1:1000]';
[mag,phase] = bode(sys,f*2*pi);
mag = squeeze(mag)';
loglog(f,mag);
grid on;
```



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