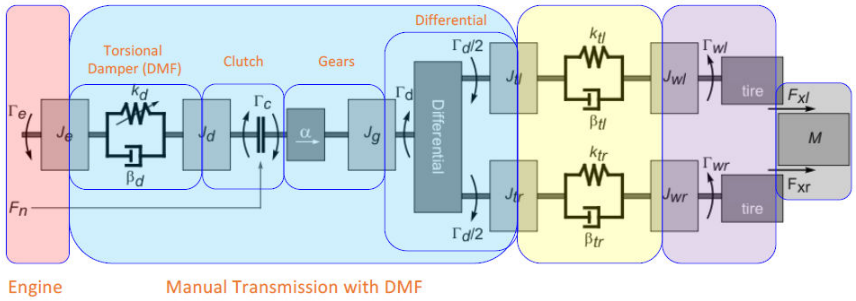
Torsional dynamics modelling

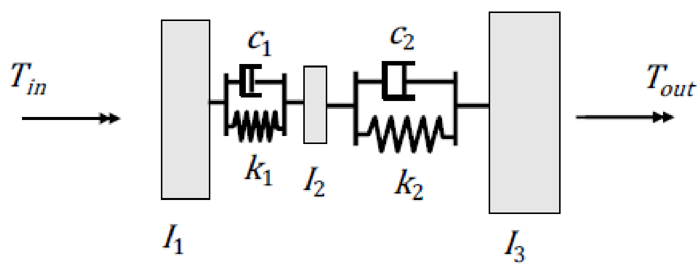
All the driveline system, which starts at the engine output shaft till the wheels is basically a torsional mechanical line. Since all the shafts can turn, this system is subjected to certian rotational DOF.



In particular the aim of this exercise is to concentrate on the torsional damper system, studying what is the effect of adopting a single flywheel respect to a dual mass one (DMF). However an initila analysis over a single mass flywheel is performed so to ahve a reference for the DMF.

SMF

The systems then presents like this:



A spring is connecting the two masses,which resambel the clutch in between. Therefore there is a total of **3DOF** (all torsional) system for the simplest model possible. Matrix writing comes in hand since permit to easily describe the system obtained, considering also that each inertia contribution is given as sum of other smaller interia components.

$$\begin{bmatrix} I_{1,SMF} & & \\ & I_{2,SMF} & \\ & & I_3 \end{bmatrix} \begin{Bmatrix} \ddot{\theta}_1 \\ \ddot{\theta}_2 \\ \ddot{\theta}_3 \end{Bmatrix} + \begin{bmatrix} c_1 & -c_1 & \\ -c_1 & c_1 + c_2 & -c_2 \\ & -c_2 & c_2 \end{bmatrix} \begin{Bmatrix} \dot{\theta}_1 \\ \dot{\theta}_2 \\ \dot{\theta}_3 \end{Bmatrix} + \begin{bmatrix} k_{1,SMF} & -k_{1,SMF} & \\ -k_{1,SMF} & k_{1,SMF} + k_2 & -k_2 \\ & -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \end{Bmatrix} = \begin{Bmatrix} T_{in} \\ 0 \\ T_{out} \end{Bmatrix}$$

$$I_{1,SMF} = I_{engine} + I_{fw}$$
$$I_{2,SMF} = I_{clutch} + I_{pressure\ plate} + I_{drive\ plate} + I_{propeller\ front\ shaft}\ \tau_G^2$$
$$I_{v,eq} = m_v R_{tyre}^2$$
$$I_3 = I_{propeller\ rear\ shaft}\ \tau_G^2 + (2I_{half\ shaft} + I_{v,eq})(\tau_G \tau_D)^2$$

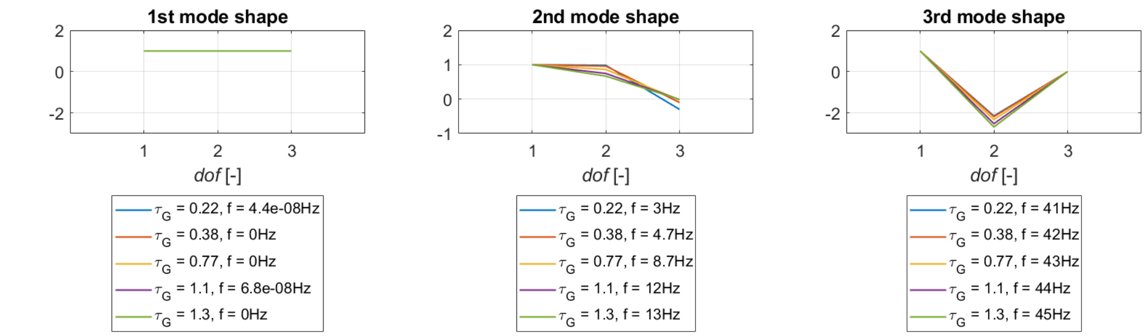
$$k_{1,SMF} = 5730\ [Nm/rad]$$
$$k_{prop} = \frac{k_{rear\ prop}\, k_{front\ prop}}{k_{rear\ prop} + k_{front\ prop}}$$
$$k_{half\ shaft,eq} = 2k_{halfshaft}\ \tau_D^2$$
$$k_2 = \frac{k_{prop}\, k_{half\ shaft,eq}}{k_{prop} + k_{half\ shaft,eq}}\ \tau_G^2$$

$$\boxed{\tau = \frac{\omega_{out}}{\omega_{in}}}$$

To study the dynamics of this system a modal analysis is performed putting attention to the frequency response plots of the **inertance** (in torsional dynamic: acceleration / force (torque in our case). Both must be performed for each gear and study the gear dependency over inertances, natural frequencies and which gears implies higher peaks.

Analysis

```
% SMF_gear_ratio_S -> reference script for this analysis
openfig('01mode_shape_SMF.fig','visible');
```

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openfig('02Ineritances_SMF.fig','visible');
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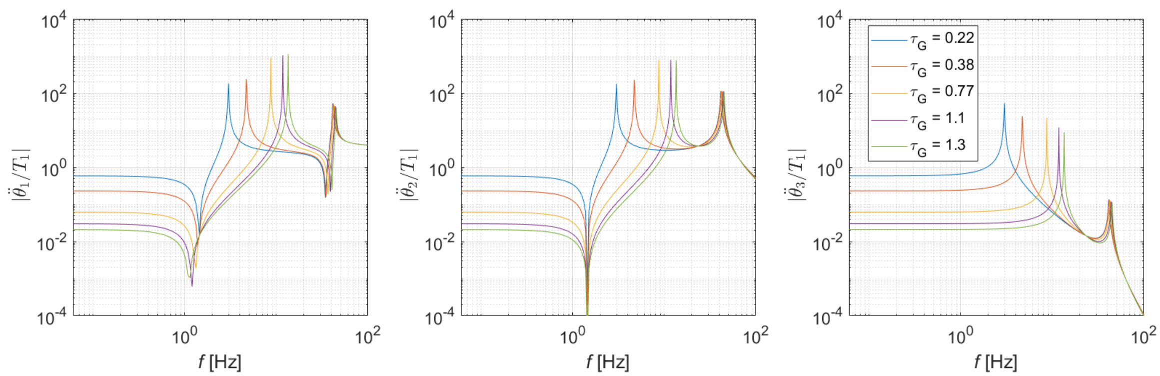


Remeber: The peaks are resonance an each one refers to a natural frequency. The nuemration follows the increasing frequency order. Instead, the lower peaks are anti-resonance points.

- What is the effect of gear shifting on mass and stiffness matrices?

Tau is defined as the rotational velocity out over input. Considering that tau multiplies both the inertia and the spring matrices, up to the third gear it will decrease both matrices (bigger effect on k since the dependcy is squared). For the fourth and fifth gear it increases the two matrices instead.

- Which is the natural frequency mostly influenced by the gear shifting?

By observing the Bode plot it seems to be the first natural frequency. This one in fact changes accordingly with higher gear, meaning that the peaks do not coincide. Menawhile the other natural frequency seems to be completely uneffected by the gear selection.

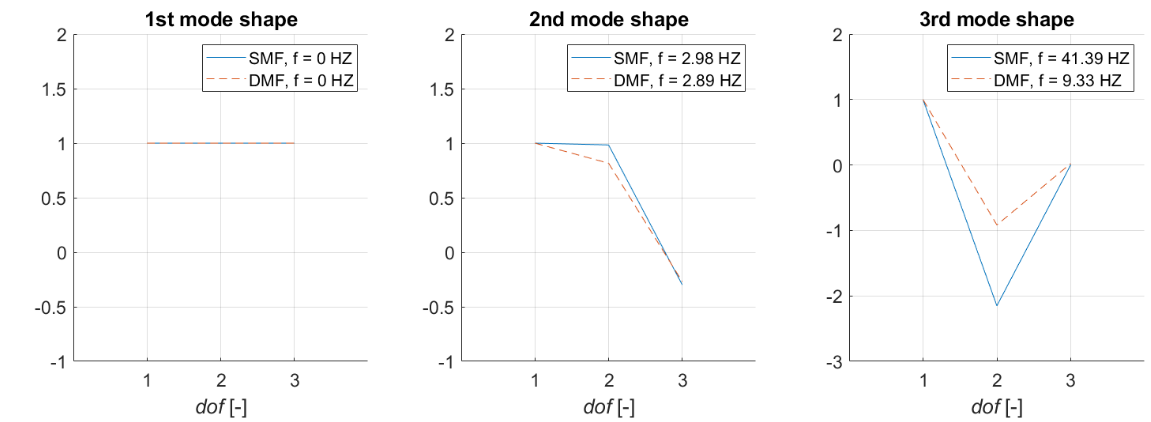
- Shifting from 1st to 5th gear ratio, do the natural frequencies move to lower or higher frequency levels?

Higher, since the peaks shift toward right whihc in the Bode plot represent the frequency range.

DMF

The DMF is chacaterised by a second mass attached at the end of the spring which implies soem slight difference respect to the SMF. In particualr in terms of formula they are actually the same, however since some terms now takeinto account the inertia also of the clutch disk, they change their value. These terms in particular are I_primay,fw (which decreases), I_propeller,shaft which increases and teh stiffness between I1 and I2 decreases considerably. Wiht these changes the modal analysis presents like:

```
openfig('03modeshapes_DMF.fig','visible');
```

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openfig('04Ineritances_DMF.fig','visible');
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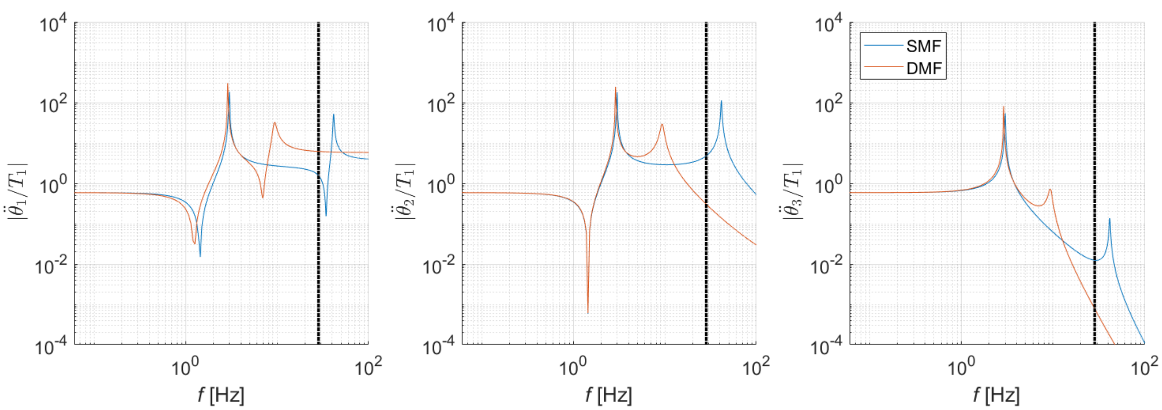


- Which is the desired effect of dual mass flywheel?

The DMF is introduced as system in order to damp the oscillations cmoing from the engine. Therefore it expected principle is that both natural frequencies are below the engine one, which is exaclty what is shown in the inertances Bodeplot. The black line is the engine frequency at a given rotational speed (in particular is the lower one) which has been calculated with the following formula considering a 4-stroke 4 cylider engine:

$$f_e = \frac{2\,n\,N_c}{60\,N_s}$$

n: revolutions per minute [rpm]
N_c: number of cylinders [-]
N_s: number of strokes during each cycle [-]

- Compute the critical damping in the DMF case using 1st gear ratio

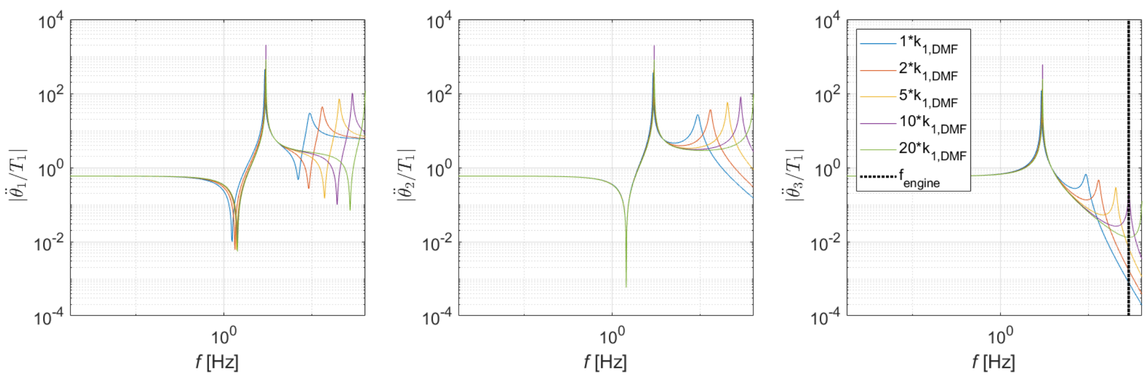
The critical damping can be calculated from its definition, where the first gear influence has been included into the I_eq_DMF term:

```
% c_cr_DMF = 2*(I_eq_DMF*k_1_DMF)^0.5;
disp(['Critical damping value for 1st gear: ',num2str(c_cr_DMF)])
```

```
Critical damping value for 1st gear: 10.9405
```

- Perform the sensitivity analyses on inertances with DMF stiffness k1 = [1 2 5 10 20]*k_1DMF with c1 = 0.05c_crDMF

```
openfig('06_Sensitivity_k1varying.fig','visible');
```



K has an effect only over the second natural frequency and higher is the gear higher is the transmission of vibrations.

- Perform the sensitivity analyses on inertances with DMF stiffness k1 = [0.5 0.2 0.1 0.05] with c1 = 0.05c_crDMF

```
openfig('07_Sensititivy_rVariation.fig','visible');
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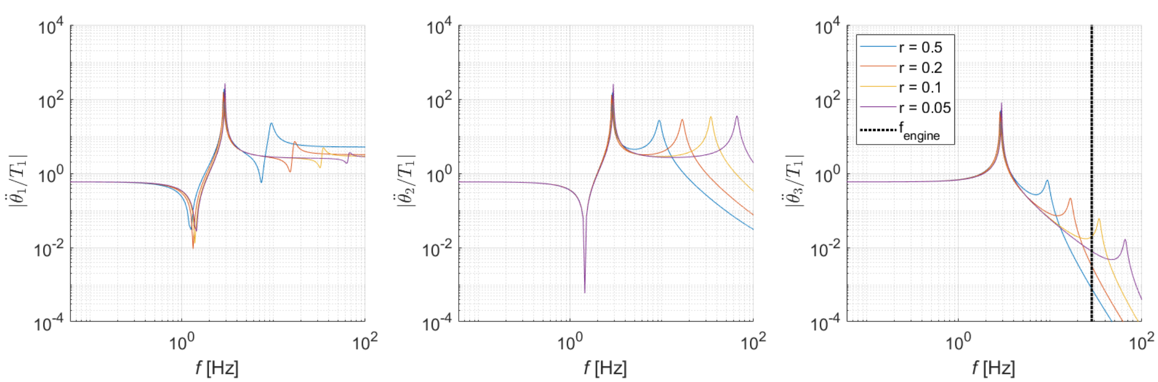
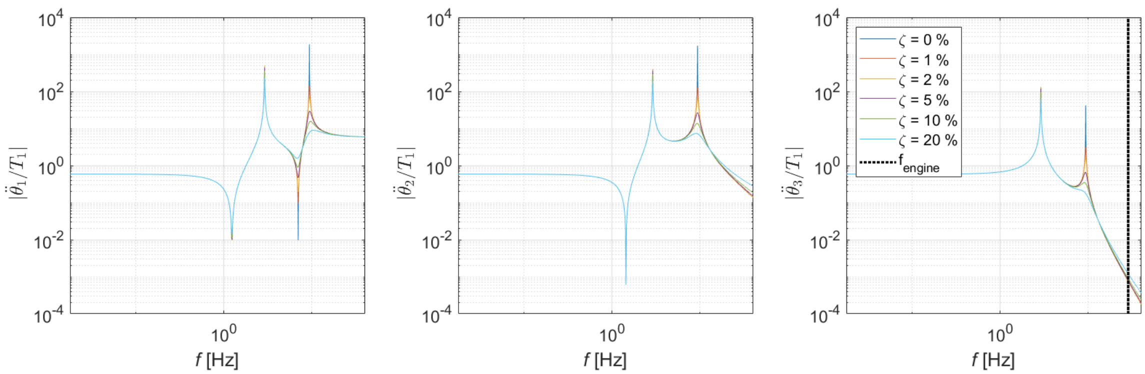


Also the r has an influence only on the second natural frequency but the effect is different respect to varying the stiffness. For the first harmonic the lower is r lower is the natural frequency (proportional depency as k), fot the second what changes is only the natural frequency with no modification over the trasmissibility. regarding the third mode shape, the pole at which is present the natural freqeuncy varies following a decreasing linear tendecy (both frequency and transmissibility increment). The slope of this tendecy is one of first order (-20 db/dec slope)

- Perform the sensitivity analyses on inertances with DMF stiffness c1 = [1% 2% 5% 10% 20%]*c_crDMF

```
openfig('08_Sensitivity_cVAriation.fig','visible');
```



Changing the damping value instead there are no modification over the natural frequencis. This is due to the fact that they changes only considering factor related to the free responce, the damping is taken into consdieration only during forced responces. Its influence is only over the amplification at the natural frequency. As someone can expect, higher is the damping higher the system will damp the oscillation resulting in a lower transmissibility and this is shown in the picture presented. However, higher damping translates into higher energy absortion, therefore the damping cannot be increased as much, a trade off has to be found.