

The Vibration Absorber

THE CENTRIFUGAL PENDULUM-TYPE ABSORBER

SCHAEFFLER



Modern internal combustion engines, which have very high torque in the low speed range, enable a particularly fuel-saving and environmentally-friendly driving style. However, the disruptive noise that is produced particularly in the low speed range in conventional vehicles prevents this potential from being fully exploited. High-performance torsional vibration dampers such as dual mass flywheels play an important role in unlocking the full potential of modern engines. The centrifugal pendulum-type absorber developed by LuK functions as a speed-adaptive mass damper and enables the full performance potential of modern drive systems to be exploited without sacrificing comfort.

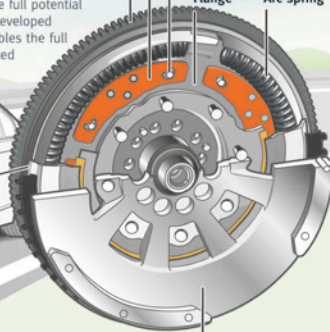
Primary flywheel with ring gear

Pendulum mass

Rollers

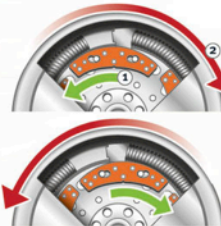
Flange

Arc spring



Secondary flywheel

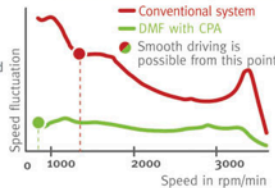
BALANCING FORCE



The pendulum masses (1) vibrate in the opposite direction to the torsional vibrations of the engine (2). They have a particularly large deflection at low speeds. In simple terms: The centrifugal pendulum-type absorber balances out undesirable torsional vibrations. Disruptive engine noise in the low speed range is almost completely eliminated.

SMOOTHER RUNNING

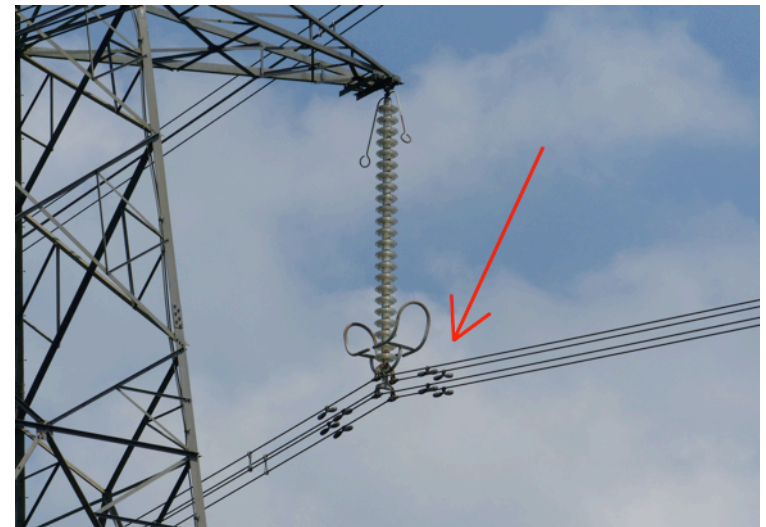
In contrast to conventional systems, the dual mass flywheel with CPA significantly reduces vibrations caused by the rotational irregularity of the crankshaft. The effect: In addition to reducing disruptive noise, the dual mass flywheel with CPA enables particularly comfortable driving at low speeds and therefore saves fuel.



LOWER FUEL CONSUMPTION

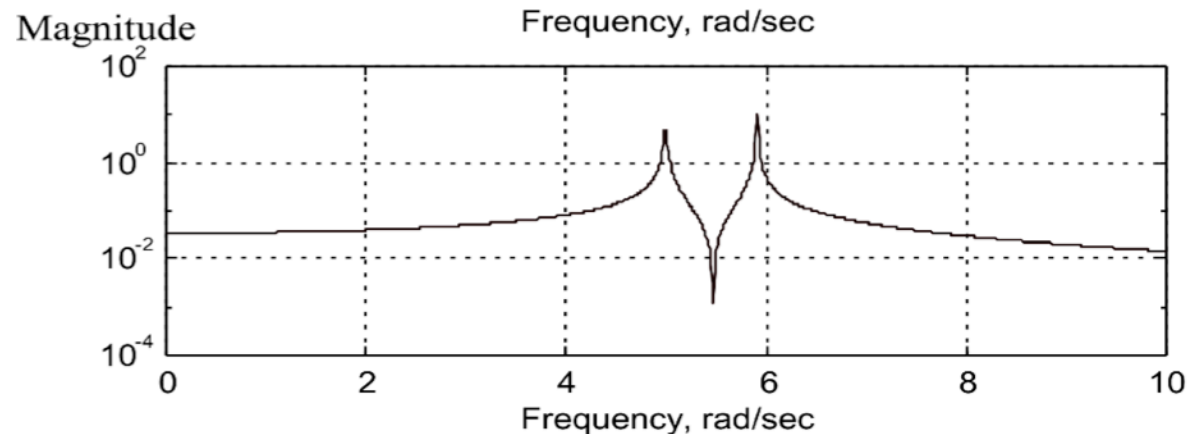
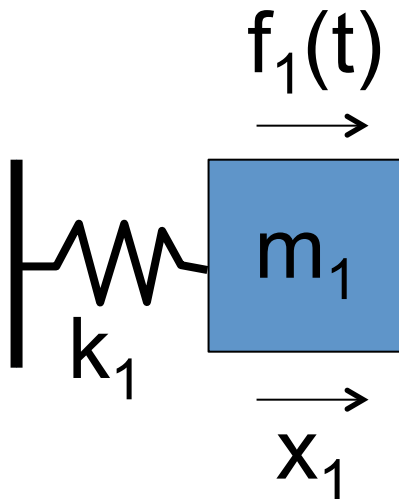
The capability of driving at particularly low speeds unlocks significant potential for reducing fuel consumption compared with conventional systems and even systems with dual mass flywheels.

FUEL CONSUMPTION IN L/100 KM

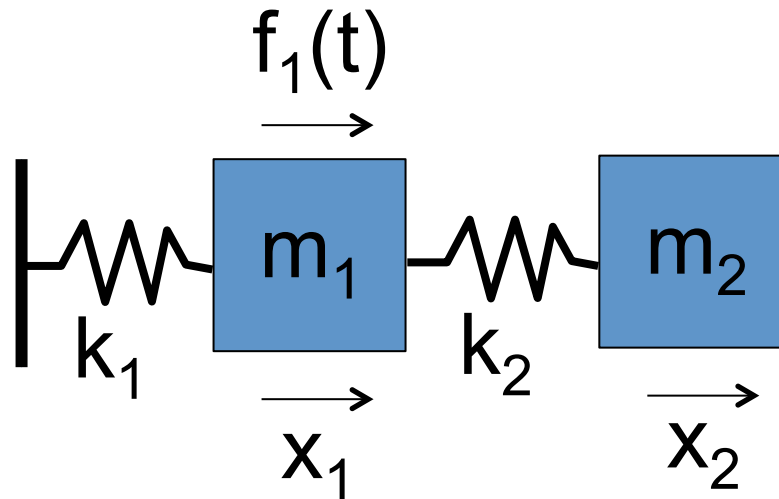


Starting with a single degree of freedom system...

- It is desired to cancel the oscillatory motion of an SDOF system subject to a harmonic forcing function, $f_1(t) = \sin \omega t$
- Recall that the driving point response of a 2-DOF system shows an anti-resonance between the resonant points



So we will add a second mass to the system



- The second mass dynamics will be tuned to the input excitation frequency:

$$\frac{k_2}{m_2} = \omega = \omega_{22}$$

Now to equations of motion give:

$$\begin{bmatrix} m_1 & \\ & m_2 \end{bmatrix} \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{bmatrix} (k_1 + k_2) & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{pmatrix} F_1 \sin \omega t \\ 0 \end{pmatrix}$$

Assuming harmonic motion results in:

$$\begin{bmatrix} (k_1 + k_2 - m_1 \omega^2) & -k_2 \\ -k_2 & k_2 - m_2 \omega^2 \end{bmatrix} \begin{pmatrix} X_1 \\ X_2 \end{pmatrix} = \begin{pmatrix} F_1 \\ 0 \end{pmatrix}$$

We can use Cramer's Rule to find the solution to X_1 :

- Recall Cramer's Rule:



$$\begin{aligned} [A](X) &= (F) \\ \begin{bmatrix} \alpha_1 \\ \alpha_2 \end{bmatrix} &= \begin{bmatrix} (F)(A_2)(A_3)\cdots \\ (A_1)(F)(A_3)\cdots \end{bmatrix} \end{aligned}$$

$$\text{Then: } X_1 = \frac{|\alpha_1|}{|A|}; \quad X_2 = \frac{|\alpha_2|}{|A|}$$

For our particular case:

$$\begin{aligned} X_1 &= \frac{\begin{vmatrix} F_1 & -k_2 \\ 0 & -m_2\omega^2 + k_2 \end{vmatrix}}{\begin{vmatrix} (-m_1\omega^2 + k_1 + k_2) & -k_2 \\ -k_2 & (-m_2\omega^2 + k_2) \end{vmatrix}} \\ &= \frac{F_1(k_2 - m_2\omega^2)}{(-m_1\omega^2 + k_1 + k_2)(-m_2\omega^2 + k_2) - k_2^2} \end{aligned}$$

This is simplified to:

$$X_1 = \frac{F_1 \left(1 - \frac{\omega^2}{\omega_{22}^2} \right)}{\left(k_1 + k_2 - m_1 \omega^2 \right) \left(1 - \frac{\omega^2}{\omega_{22}^2} \right) - k_2}$$

In dimensionless form this becomes:

$$\frac{X_1 k_1}{F_1} = \frac{F_1 \left(1 - \frac{\omega^2}{\omega_{22}^2} \right)}{\left(1 + \frac{k_1}{k_2} - \frac{\omega^2}{\omega_{11}^2} \right) \left(1 - \frac{\omega^2}{\omega_{22}^2} \right) - \frac{k_2}{k_1}}$$

Now, when $\omega = \omega_{22}$ we see that $X_1 = 0$

- Solving for X_2 :

$$\frac{X_2 k_1}{F_1} = \frac{1}{\left(1 + \frac{k_1}{k_2} - \frac{\omega^2}{\omega_{11}^2}\right) \left(1 - \frac{\omega^2}{\omega_{22}^2}\right) - \frac{k_2}{k_1}}$$

$$\text{Or: } X_2 = -\frac{F_1}{k_2}$$

We see that the motion of m_1 is zero, and the motion of m_2 is 180 degrees out of phase with the force, and the force transmitted through spring k_2 is $k_2 X_2 = -F_1$

Some things to consider in the design of the vibration absorber

- The size of k_2 is dependent on the allowable value of X_2 . We will have two resonant peaks on either side of the operating frequency. The spread in the frequency is controlled by both k_2 and m_2 .

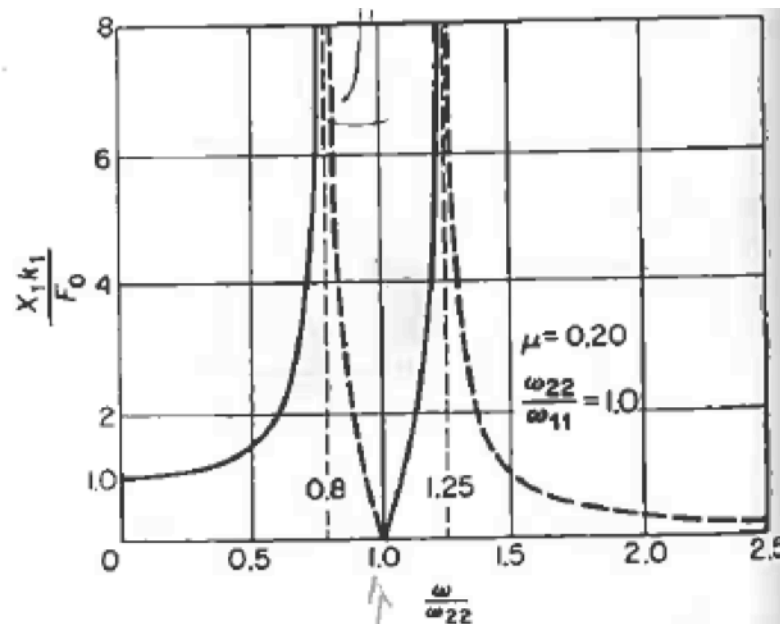


FIGURE 5.6.2. Response vs. frequency.

A nice metric to consider in designing a vibration absorber is the mass ratio m_2 / m_1 when determining the resonant frequency spread:

From this plot, the resonant peaks are determined relative to the center frequency where $X_2 = 0$.

A larger spread of resonant frequencies results in a greater range of operating frequencies that will not cause damaging resonant response

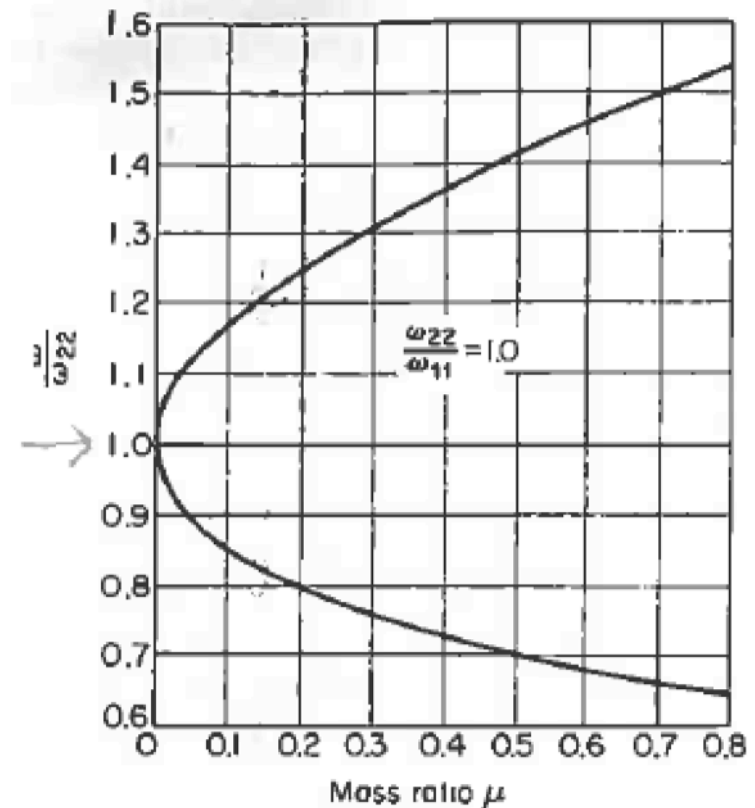
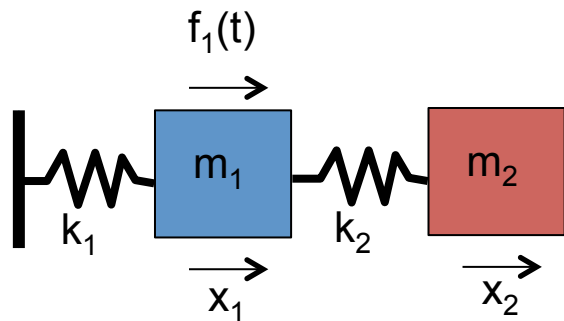


FIGURE 5.6.3 Natural frequencies vs. $\mu \times m_2/m_1$.

Example

- Design a vibration absorber for the system shown where we desire the system natural frequencies to be at least 30% away from the operating frequency of 400 rad/s



$$m_1 = .01295 \text{ lbs}^2 / \text{in}$$

$$k_1 = 2075 \text{ lb} / \text{in}$$

$$\omega_n = 400 \text{ rad} / \text{s} = \omega_{\text{operating}}$$

From the figure on the previous slide, $\frac{m_2}{m_1} \geq 0.5$

$$m_2 \geq .006475 \text{ lbs}^2 / \text{in}$$

Example cont.

$$\begin{aligned}k_2 &= \omega_{22}^2 m_2 \\ &= 400^2 (.006475) = 1036 \text{ lb / in}\end{aligned}$$

Checking this work shows that the two resonant frequencies are:

$$\omega_1 = 282 \text{ rad / s}$$

$$\omega_2 = 566 \text{ rad / s}$$

Which are greater than 30% away from the operating frequency

Now if the forcing function is determined to be $f(t) = \sin \omega t$

$$\text{Then: } X_2 = -\frac{F_1}{k_2} = -\frac{1}{1036} \approx -0.001 \text{ in}$$