# Dynamic vibration absorber



Iva Petríková

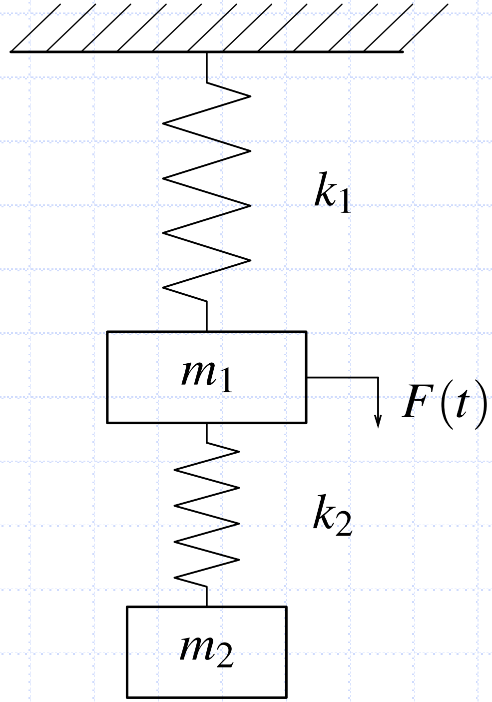
evropský sociální fond v ČR
EVROPSKÁ UNIE
MINISTERSTVO ŠKOLSTVÍ, MLÁDEŽE A TĚLOVÝCHOVY
OP Vzdělávání pro konkurenceschopnost
TECHNICKÁ UNIVERZITA V LIBERCI www.tul.cz
INVESTICE DO ROZVOJE VZDĚLÁVÁNÍ

# Dynamic vibration absorber

Figure represents a simple two mass system with harmonic force $F_0 \sin \omega t$ applied to $m_1$.

In order to avoid undesirable resonance conditions in many applications, the system stiffness and inertia characteristics must be changed. Another approach, to alleviate the resonant conditions, is to convert the single degree of freedom system (DOF) to a two DOF by adding an auxiliary spring and mass system.

The parameters of the added system can be selected in such a manner that the vibration of the main mass is eliminated.

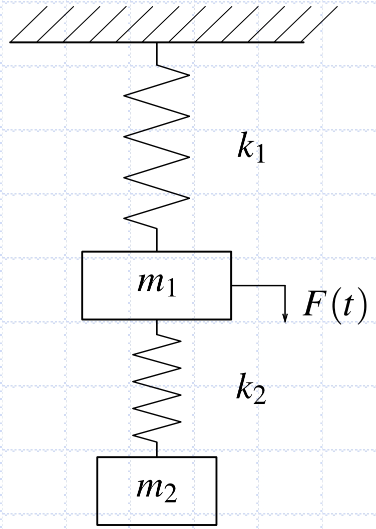


# Dynamic vibration absorber

- System with 2 DOF undamped

$$m_1\ddot{x}_1 = k_2\left(x_2 - x_1\right) - k_1 x_1 + F_0 \sin \omega t \qquad (1)$$

$$m_2\ddot{x}_2 = -k_2\left(x_2 - x_1\right) \qquad (2)$$




- Supposed solutions,

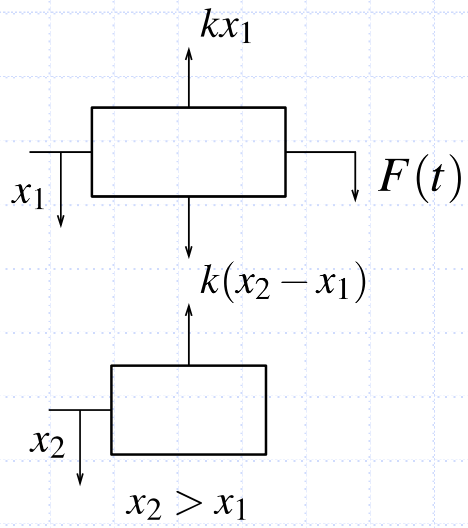
$$x_1 = X_1 \sin \omega t \qquad \ddot{x}_1 = -\omega^2 X_1 \sin \omega t$$

$$x_2 = X_2 \sin \omega t \qquad \ddot{x}_2 = -\omega^2 X_2 \sin \omega t$$

- Substitution to (1) a (2)

$$\left(k_1 + k_2 - \omega^2 m_1\right) X_1 - k_2 X_2 = F_0 \qquad (3)$$

$$-k_2 X_1 + \left(k_2 - \omega^2 m_1\right) X_2 = 0 \qquad (4)$$



# Dynamic vibration absorber

- Eq. (1) is divided by $k_1$ and eq. (2) $k_2$ denote relations

$$\sqrt{\frac{k_1}{m_1}} = \omega_{11}$$ natural frequency of single DOF with mass $m_1$

$$\sqrt{\frac{k_2}{m_2}} = \omega_{22}$$ natural frequency of single DOF with mass $m_2$

$$\frac{F_0}{k_1} = X_{ST}$$ static deflection of mass 1

$$\left[1 + \frac{k_2}{k_1} - \left(\frac{\omega}{\omega_{11}}\right)^2\right] X_1 - \frac{k_2}{k_1} X_2 = \frac{F_0}{k_1}$$

$$-X_1 + \left[1 - \left(\frac{\omega}{\omega_{22}}\right)^2\right] X_2 = 0$$

# Dynamic vibration absorber

- Amplitude of two masses are obtained from solution of equations (3) and (4)

$$\frac{X_1}{X_{ST}} = \frac{1-\left(\frac{\omega}{\omega_{22}}\right)^2}{\left[1+\frac{k_2}{k_1}-\left(\frac{\omega}{\omega_{11}}\right)^2\right]\left[1-\left(\frac{\omega}{\omega_{22}}\right)^2\right]-\frac{k_2}{k_1}} \quad (5) \qquad \frac{X_2}{X_{ST}} = \frac{1}{\left[1+\frac{k_2}{k_1}-\left(\frac{\omega}{\omega_{11}}\right)^2\right]\left[1-\left(\frac{\omega}{\omega_{22}}\right)^2\right]-\frac{k_2}{k_1}} \quad (6)$$

- It is evident from these equations that the amplitude $X_1$ of the main system becomes zero when the exciting frequency ω coincides with the natural frequency $\omega_{22}$ of mass 2. For this frequency amplitude $X_2$ is equal to:

$$X_2 = -\frac{k_1}{k_2} X_{ST} = -\frac{F_0}{k_2} \quad (7)$$

# Dynamic vibration absorber

- The negative sign idicates that amplitude $X_2$ is out of phase with exciting force. If $X_1=0$, the force $k_2 X_2$ exerted by spring 2 on mass $m_1$ is equal and opposite to the impressed force $F_0$. For exciting frequency of the system yields

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

$$k_2 X_2 = \omega^2 m_2 X_2 = -F_0$$

- Amplitude of main mass can reach two resonance frequency.

evropský sociální fond v ČR EVROPSKÁ UNIE MINISTERSTVO ŠKOLSTVÍ, MLÁDEŽE A TĚLOVÝCHOVY OP Vzdělávání pro konkurenceschopnost TECHNICKÁ UNIVERZITA V LIBERCI www.tul.cz
INVESTICE DO ROZVOJE VZDĚLÁVÁNÍ

# Dynamic vibration absorber

Resonant frequency can be obtained by setting the denominator of the amplitude equation (5) and (6) to zero.

$$\left[1+\frac{k_2}{k_1}-\left(\frac{\omega}{\omega_{11}}\right)^2\right]\left[1-\left(\frac{\omega}{\omega_{22}}\right)^2\right]-\frac{k_2}{k_1}=0$$