Engingeering Mechanics Dynamics in SI Units 14th edition Hibbeler Solutions Manual

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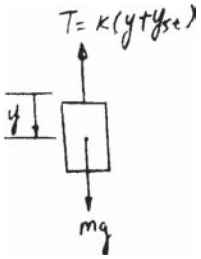
22–1.

A spring is stretched 175 mm by an 8-kg block. If the block is displaced 100 mm downward from its equilibrium position and given a downward velocity of 1.50 m/s, determine the differential equation which describes the motion. Assume that positive displacement is downward. Also, determine the position of the block when $t = 0.22$ s.

SOLUTION

$+\downarrow \Sigma F_y = ma_y; \quad mg - k(y + y_{st}) = m\ddot{y} \quad$ where $ky_{st} = mg$

$$\ddot{y} + \frac{k}{m}y = 0$$



Hence $\quad p = \sqrt{\frac{k}{m}} \quad$ Where $k = \frac{8(9.81)}{0.175} = 448.46$ N/m

$$= \sqrt{\frac{448.46}{8}} = 7.487$$

$\therefore \quad \ddot{y} + (7.487)^2 y = 0 \quad \ddot{y} + 56.1y = 0$ **Ans.**

The solution of the above differential equation is of the form:

$y = A \sin pt + B \cos pt$ **(1)**

$v = \dot{y} = Ap \cos pt - Bp \sin pt$ **(2)**

At $t = 0$, $y = 0.1$ m and $v = v_0 = 1.50$ m/s

From Eq. (1) $\quad 0.1 = A \sin 0 + B \cos 0 \quad B = 0.1$ m

From Eq. (2) $\quad v_0 = Ap \cos 0 - 0 \quad A = \frac{v_0}{p} = \frac{1.50}{7.487} = 0.2003$ m

Hence $\quad y = 0.2003 \sin 7.487t + 0.1 \cos 7.487t$

At $t = 0.22$ s, $\quad y = 0.2003 \sin [7.487(0.22)] + 0.1 \cos [7.487(0.22)]$

$= 0.192$ m **Ans.**

Ans:
$\ddot{y} + 56.1\, y = 0$
$y|_{t=0.22\text{ s}} = 0.192$ m

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22–2.

A spring has a stiffness of 800 N/m. If a 2-kg block is attached to the spring, pushed 50 mm above its equilibrium position, and released from rest, determine the equation that describes the block's motion. Assume that positive displacement is downward.

SOLUTION

$$p = \sqrt{\frac{k}{m}} = \sqrt{\frac{800}{2}} = 20$$

$$x = A \sin pt + B \cos pt$$

$x = -0.05$ m when $t = 0$,

$$-0.05 = 0 + B; \qquad B = -0.05$$

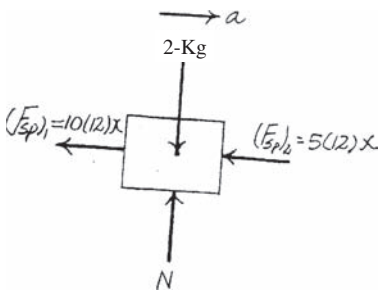
$$v = Ap \cos pt - Bp \sin pt$$

$v = 0$ when $t = 0$,

$$0 = A(20) - 0; \qquad A = 0$$

Thus,

$$x = -0.05 \cos (20t)$$ **Ans.**



Ans:
$x = -0.05 \cos (20t)$

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22–3.

A spring is stretched 200 mm by a 15-kg block. If the block is displaced 100 mm downward from its equilibrium position and given a downward velocity of 0.75 m/s, determine the equation which describes the motion. What is the phase angle? Assume that positive displacement is downward.

SOLUTION

$$k = \frac{F}{y} = \frac{15(9.81)}{0.2} = 735.75 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{735.75}{15}} = 7.00$$

$$y = A \sin \omega_n t + B \cos \omega_n t$$

$$y = 0.1 \text{ m when } t = 0,$$

$$0.1 = 0 + B; \qquad B = 0.1$$

$$v = A\omega_n \cos \omega_n t - B\omega_n \sin \omega_n t$$

$$v = 0.75 \text{ m/s when } t = 0,$$

$$0.75 = A(7.00)$$

$$A = 0.107$$

$$y = 0.107 \sin (7.00t) + 0.100 \cos (7.00t) \qquad \textbf{Ans.}$$

$$\phi = \tan^{-1}\left(\frac{B}{A}\right) = \tan^{-1}\left(\frac{0.100}{0.107}\right) = 43.0^\circ \qquad \textbf{Ans.}$$

Ans:
$y = 0.107 \sin (7.00t) + 0.100 \cos (7.00t)$
$\phi = 43.0^\circ$

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***22–4**

When a 2-kg block is suspended from a spring, the spring is stretched a distance of 40 mm. Determine the frequency and the period of vibration for a 0.5-kg block attached to the same spring.

SOLUTION

$$k = \frac{F}{y} = \frac{2(9.81)}{0.040} = 490.5 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{490.5}{0.5}} = 31.321$$

$$f = \frac{\omega_n}{2\pi} = \frac{31.321}{2\pi} = 4.985 = 4.98 \text{ Hz} \qquad \textbf{Ans.}$$

$$\tau = \frac{1}{f} = \frac{1}{4.985} = 0.201 \text{ s} \qquad \textbf{Ans.}$$

Ans:
$f = 4.98$ Hz
$\tau = 0.201$ s

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22–5.

When a 3-kg block is suspended from a spring, the spring is stretched a distance of 60 mm. Determine the natural frequency and the period of vibration for a 0.2-kg block attached to the same spring.

SOLUTION

$$k = \frac{F}{\Delta x} = \frac{3(9.81)}{0.060} = 490.5 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{490.5}{0.2}} = 49.52 = 49.5 \text{ rad/s} \qquad \textbf{Ans.}$$

$$f = \frac{\omega_n}{2\pi} = \frac{49.52}{2\pi} = 7.88 \text{ Hz}$$

$$\tau = \frac{1}{f} = \frac{1}{7.88} = 0.127 \text{ s} \qquad \textbf{Ans.}$$

Ans:
$\omega_n = 49.5$ rad/s
$\tau = 0.127$ s

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22–6.

An 8-kg block is suspended from a spring having a stiffness $k = 80$ N/m. If the block is given an upward velocity of 0.4 m/s when it is 90 mm above its equilibrium position, determine the equation which describes the motion and the maximum upward displacement of the block measured from the equilibrium position. Assume that positive displacement is measured downward.

SOLUTION

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{80}{8}} = 3.162 \text{ rad/s}$$

$$v = -0.4 \text{ m/s}, \qquad x = -0.09 \text{ m at } t = 0$$

$$x = A \sin \omega_n t + B \cos \omega_n t$$

$$-0.09 = 0 + B$$

$$B = -0.09$$

$$v = A\omega_n \cos \omega_n t - B\omega_n \sin \omega_n t$$

$$-0.4 = A(3.162) - 0$$

$$A = -0.126$$

Thus, $x = -0.126 \sin (3.16t) - 0.09 \cos (3.16t)$ m **Ans.**

$C = \sqrt{A^2 + B^2} = \sqrt{(-0.126)^2 + (-0.09)} = 0.155$ m **Ans.**

Ans:
$x = \{-0.126 \sin (3.16t) - 0.09 \cos (3.16t)\}$ m
$C = 0.155$ m

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22–7.

A pendulum has a cord of length L and is given a tangential velocity v toward the vertical from a position θ_0. Determine the equation which describes the angular motion.

SOLUTION

Given: $L = 0.4\ \text{m}$ $v = 0.2\ \text{m/s}$ $\theta_0 = 0.3\ \text{rad}$ $g = 9.81\ \text{m/s}^2$

Since the motion remains small

$$\omega_n = \sqrt{\frac{g}{L}} \qquad \theta = A\sin(\omega_n t) + B\cos(\omega_n t) \qquad A = \frac{-v}{\omega_n L} \qquad B = \theta_0$$

$$\theta = A\sin(\omega_n t) + B\cos(\omega_n t) \qquad A = -0.101\ \text{rad} \quad B = 0.30\ \text{rad} \quad \omega_n = 4.95\ \text{rad/s} \qquad \textbf{Ans.}$$

Ans:

$\theta = A\sin(\omega_n t) + B\cos(\omega_n t)$
$A = -0.101\ \text{rad}$
$B = 0.30\ \text{rad}$
$\omega_n = 4.95\ \text{rad/s}$

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***22–8.** A 2-kg block is suspended from a spring having a stiffness of 800 N/m. If the block is given an upward velocity of 2 m/s when it is displaced downward a distance of 150 mm from its equilibrium position, determine the equation which describes the motion. What is the amplitude of the motion? Assume that positive displacement is downward.

SOLUTION

$$p = \sqrt{\frac{k}{m}} = \sqrt{\frac{800}{2}} = 20$$

$$x = A \sin pt + B \cos pt$$

$x = 0.150$ m when $t = 0$,

$$0.150 = 0 + B; \qquad B = 0.150$$

$$v = Ap \cos pt - Bp \sin pt$$

$v = -2$ m/s when $t = 0$,

$$-2 = A(20) - 0; \qquad A = -0.1$$

Thus,

$$x = 0.1 \sin (20t) + 0.150 \cos (20t) \qquad \textbf{Ans.}$$

$$C = \sqrt{A^2 + B^2} = \sqrt{(0.1)^2 + (0.150)^2} = 0.180 \text{ m} \qquad \textbf{Ans.}$$

Ans:
$x = 0.1 \sin (20t) + 0.150 \cos (20t)$
$C = 0.180$ m

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22–9. A 3-kg block is suspended from a spring having a stiffness of $k = 200\,\text{N/m}$. If the block is pushed 50 mm upward from its equilibrium position and then released from rest, determine the equation that describes the motion. What are the amplitude and the natural frequency of the vibration? Assume that positive displacement is downward.

SOLUTION

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{200}{3}} = 8.16\ \text{rad/s} \qquad \textbf{Ans.}$$

$$x = A \sin \omega_n t + B \cos \omega_n t$$

$$x = -0.05\ \text{m when } t = 0,$$

$$-0.05 = 0 + B; \qquad B = -0.05$$

$$v = Ap \cos \omega_n t - B\omega_n \sin \omega_n t$$

$$v = 0 \text{ when } t = 0,$$

$$0 = A(8.165) - 0; \qquad A = 0$$

Hence,

$$x = -0.05 \cos (8.16t) \qquad \textbf{Ans.}$$

$$C = \sqrt{A^2 + B^2} = \sqrt{(0)^2 + (-0.05)} = 0.05\ \text{m} = 50\ \text{mm} \qquad \textbf{Ans.}$$

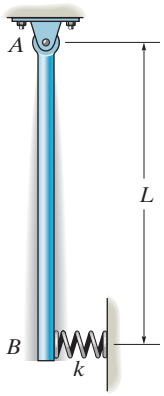
Ans:
$\omega_n = 8.16\ \text{rad/s}$
$x = -0.05 \cos (8.16t)$
$C = 50\ \text{mm}$

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22–10.

The uniform rod of mass m is supported by a pin at A and a spring at B. If B is given a small sideward displacement and released, determine the natural period of vibration.



SOLUTION

Equation of Motion. The mass moment of inertia of the rod about A is $I_A = \frac{1}{3}mL^2$. Referring to the FBD. of the rod, Fig. a,

$$\zeta + \Sigma M_A = I_A\alpha; \qquad -mg\left(\frac{L}{2}\sin\theta\right) - (kx\cos\theta)(L) = \left(\frac{1}{3}mL^2\right)\alpha$$

However; $x = L\sin\theta$. Then

$$\frac{-mgL}{2}\sin\theta - kL^2\sin\theta\cos\theta = \frac{1}{3}mL^2\alpha$$

Using the trigonometry identity $\sin 2\theta = 2\sin\theta\cos\theta$,

$$\frac{-mgL}{2}\sin\theta - \frac{KL^2}{2}\sin 2\theta = \frac{1}{3}mL^2\alpha$$

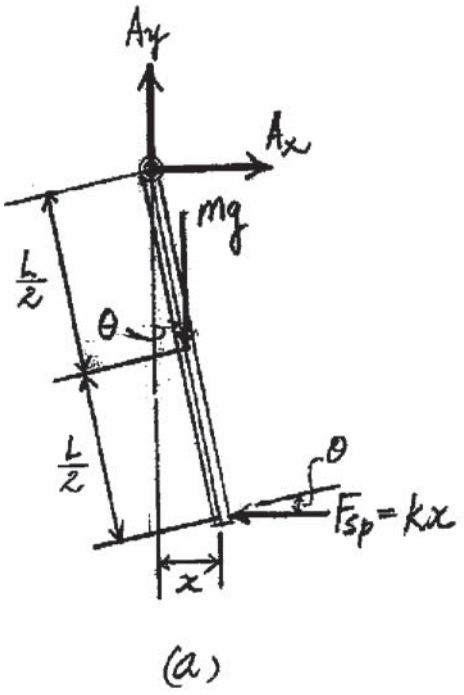
Here since θ is small $\sin\theta \simeq \theta$ and $\sin 2\theta \simeq 2\theta$. Also $\alpha = \ddot{\theta}$. Then the above equation becomes

$$\frac{1}{3}mL^2\ddot{\theta} + \left(\frac{mgL}{2} + kL^2\right)\theta = 0$$

$$\ddot{\theta} + \frac{3mg + 6kL}{2mL}\theta = 0$$

Comparing to that of the Standard form, $\omega_n = \sqrt{\frac{3mg + 6kL}{2mL}}$. Then

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{2mL}{3mg + 6kL}} \qquad \textbf{Ans.}$$





$KL^2 \sin\theta\cos\theta$

$\frac{KL^2}{2}$

Ans:

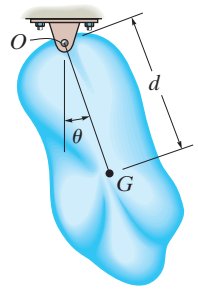
$$\tau = 2\pi\sqrt{\frac{2mL}{3mg + 6kL}}$$

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22–11.

The body of arbitrary shape has a mass m, mass center at G, and a radius of gyration about G of k_G. If it is displaced a slight amount θ from its equilibrium position and released, determine the natural period of vibration.



SOLUTION

$$\zeta + \Sigma M_O = I_O\,\alpha; \qquad -mgd\sin\theta = \left[mk_G^2 + md^2\right]\ddot{\theta}$$

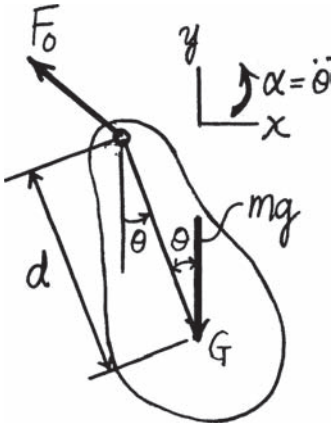
$$\ddot{\theta} + \frac{gd}{k_G^2 + d^2}\sin\theta = 0$$

However, for small rotation $\sin\theta \approx \theta$. Hence

$$\ddot{\theta} + \frac{gd}{k_G^2 + d^2}\theta = 0$$

From the above differential equation, $\omega_n = \sqrt{\dfrac{gd}{k_G^2 + d^2}}$.

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{\sqrt{\dfrac{gd}{k_G^2 + d^2}}} = 2\pi\sqrt{\frac{k_G^2 + d^2}{gd}}$$ **Ans.**



Ans:

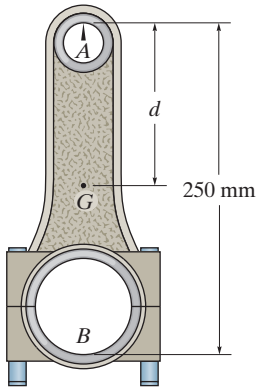
$$\tau = 2\pi\sqrt{\frac{k_G^2 + d^2}{gd}}$$

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***22–12.**

The connecting rod is supported by a knife edge at A and the period of vibration is measured as $\tau_A = 3.38$ s. It is then removed and rotated 180° so that it is supported by the knife edge at B. In this case the perod of vibration is measured as $\tau_B = 3.96$ s. Determine the location d of the center of gravity G, and compute the radius of gyration k_G.



SOLUTION

Free-body Diagram: In general, when an object of arbitary shape having a mass m is pinned at O and is displaced by an angular displacement of θ, the tangential component of its weight will create the *restoring moment* about point O.

Equation of Motion: Sum monent about point O to eliminate O_x and O_y.

$$\zeta + \Sigma M_O = I_O \alpha; \qquad -mg \sin\theta (l) = I_O \alpha \tag{1}$$

Kinematics: Since $\alpha = \dfrac{d^2\theta}{dt^2} = \ddot{\theta}$ and $\sin\theta = \theta$ if θ is small, then substitute these values into Eq. (1), we have

$$-mgl\theta = I_O \ddot{\theta} \qquad \text{or} \qquad \ddot{\theta} + \frac{mgl}{I_O}\theta = 0 \tag{2}$$

From Eq. (2), $\omega_n^2 = \dfrac{mgl}{I_O}$, thus, $\omega_n = \sqrt{\dfrac{mgl}{I_O}}$. Applying Eq. 22–12, we have

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{I_O}{mgl}} \tag{3}$$

When the rod is rotating about A, $\tau = \tau_A = 3.38$ s and $l = d$. Substitute these values into Eq. (3),we have

$$3.38 = 2\pi\sqrt{\frac{I_A}{mgd}} \qquad I_A = 0.2894mgd$$

When the rod is rotating about B, $\tau = \tau_B = 3.96$ s and $l = 0.25 - d$. Substitute these values into Eq. (3), we have

$$3.96 = 2\pi\sqrt{\frac{I_B}{mg\,(0.25 - d)}} \qquad I_B = 0.3972mg\,(0.25 - d)$$

However, the mass moment inertia of the rod about its mass center is

$$I_G = I_A - md^2 = I_B - m(0.25 - d)^2$$

Then,

$$0.2894mgd - md^2 = 0.3972mg\,(0.25 - d) - m\,(0.25 - d)^2$$

$$d = 0.1462 \text{ m} = 146 \text{ mm} \qquad \textbf{Ans.}$$

Thus, the mass moment inertia of the rod about its mass center is

$$I_G = I_A - md^2 = 0.2894m\,(9.81)(0.1462) - m\left(0.1462^2\right) = 0.3937\,m$$

The radius of gyration is

$$k_G = \sqrt{\frac{I_G}{m}} = \sqrt{\frac{0.3937m}{m}} = 0.627 \text{ m} \qquad \textbf{Ans.}$$

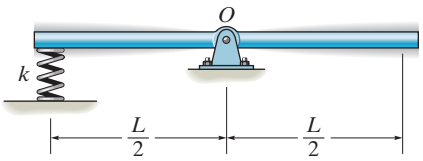
Ans:
$d = 146$ mm
$k_G = 0.627$ m

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22–13.

Determine the natural period of vibration of the uniform bar of mass m when it is displaced downward slightly and released.



SOLUTION

Equation of Motion. The mass moment of inertia of the bar about O is $I_0 = \frac{1}{12}mL^2$.

Referring to the FBD of the rod, Fig. a,

$$\circlearrowleft + \Sigma M_0 = I_0\alpha; \quad -ky\cos\theta\left(\frac{L}{2}\right) = \left(\frac{1}{12}mL^2\right)\alpha$$

However, $y = \frac{L}{2}\sin\theta$. Then

$$-k\left(\frac{L}{2}\sin\theta\right)\cos\theta\left(\frac{L}{2}\right) = \frac{1}{12}mL^2\alpha$$

Using the trigonometry identity $\sin 2\theta = 2\sin\theta\cos\theta$, we obtain

$$\frac{1}{12}mL^2\alpha + \frac{kL^2}{8}\sin 2\theta = 0$$

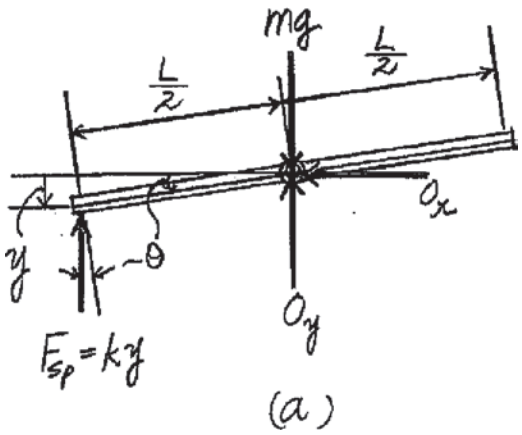
Here since θ is small, $\sin 2\theta \simeq 2\theta$. Also, $\alpha = \ddot{\theta}$. Then the above equation becomes

$$\frac{1}{12}mL^2\ddot{\theta} + \frac{kL^2}{4}\theta = 0$$

$$\ddot{\theta} + \frac{3k}{m}\theta = 0$$

Comparing to that of the Standard form, $\omega_n = \sqrt{\frac{3k}{m}}$. Then

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m}{3k}} \qquad \textbf{Ans.}$$



Ans:

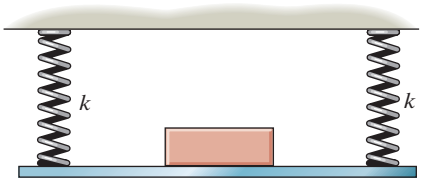
$$\tau = 2\pi\sqrt{\frac{m}{3k}}$$

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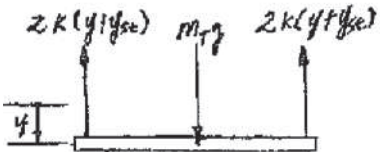
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22–14.

A platform, having an unknown mass, is supported by *four* springs, each having the same stiffness k. When nothing is on the platform, the period of vertical vibration is measured as 2.35 s; whereas if a 3-kg block is supported on the platform, the period of vertical vibration is 5.23 s. Determine the mass of a block placed on the (empty) platform which causes the platform to vibrate vertically with a period of 5.62 s. What is the stiffness k of each of the springs?



SOLUTION



$+\downarrow \Sigma F_y = ma_y; \qquad m\tau g - 4k(y + y_{\tau s}) = m\tau \ddot{y} \qquad$ Where $4k\, y_{\tau s} = m_\tau g$

$$\ddot{y} + \frac{4k}{m\tau} y = 0$$

Hence
$$P = \sqrt{\frac{4k}{m\tau}}$$

$$\tau = \frac{2\pi}{P} = 2\pi\sqrt{\frac{m\tau}{4k}}$$

For empty platform $m\tau = m_P$, where m_P is the mass of the platform.

$$2.35 = 2\pi\sqrt{\frac{m_P}{4k}} \qquad \textbf{(1)}$$

When 3-kg block is on the platform $m_\tau = m_P + 3$.

$$5.23 = 2\pi\sqrt{\frac{m_P + 3}{4k}} \qquad \textbf{(2)}$$

When an unknown mass is on the platform $m_\tau = m_P + m_B$.

$$5.62 = 2\pi\sqrt{\frac{m_P + m_B}{4k}} \qquad \textbf{(3)}$$

Solving Eqs. (1) to (3) yields :

$$k = 1.36\ \text{N/m} \qquad m_B = 3.58\ \text{kg} \qquad \textbf{Ans.}$$

$$m_P = 0.7589\ \text{kg}$$

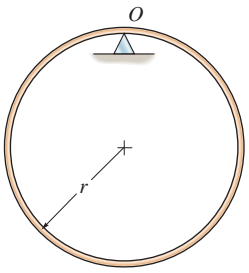
Ans:
$k = 1.36\ \text{N/m}$
$m_B = 3.58\ \text{kg}$

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22–15.

The thin hoop of mass m is supported by a knife-edge. Determine the natural period of vibration for small amplitudes of swing.



SOLUTION

$I_O = mr^2 + mr^2 = 2mr^2$

$\zeta + \sum M_O = I_O\alpha; \qquad -mgr\theta = (2mr^2)\ddot{\theta}$

$\ddot{\theta} + \left(\frac{g}{2r}\right)\theta = 0$

$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{2r}{g}}$ **Ans.**



Ans:

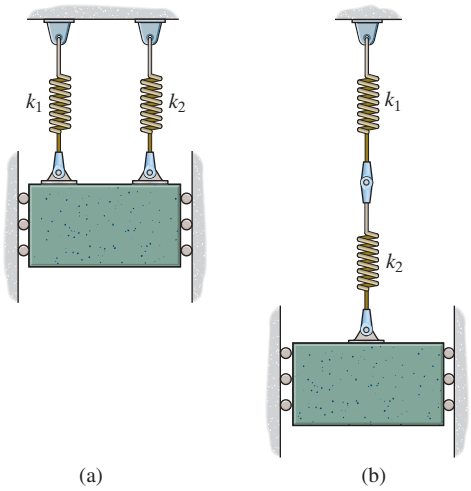
$\tau = 2\pi\sqrt{\frac{2r}{g}}$

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***22–16.**

A block of mass m is suspended from two springs having a stiffness of k_1 and k_2, arranged a) parallel to each other, and b) as a series. Determine the equivalent stiffness of a single spring with the same oscillation characteristics and the period of oscillation for each case.



SOLUTION

(a) When the springs are arranged in parallel, the equivalent spring stiffness is

$$k_{eq} = k_1 + k_2 \qquad \textbf{Ans.}$$

The natural frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} = \sqrt{\frac{k_1 + k_2}{m}}$$

Thus, the period of oscillation of the system is

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{\sqrt{\dfrac{k_1 + k_2}{m}}} = 2\pi\sqrt{\frac{m}{k_1 + k_2}} \qquad \textbf{Ans.}$$

(b) When the springs are arranged in a series, the equivalent stiffness of the system can be determined by equating the stretch of both spring systems subjected to the same load F.

$$\frac{F}{k_1} + \frac{F}{k_2} = \frac{F}{k_{eq}}$$

$$\frac{1}{k_1} + \frac{1}{k_2} = \frac{1}{k_{eq}}$$

$$\frac{k_2 + k_1}{k_1 k_2} = \frac{1}{k_{eq}}$$

$$k_{eq} = \frac{k_1 k_2}{k_1 + k_2} \qquad \textbf{Ans.}$$

The natural frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} = \sqrt{\frac{\left(\dfrac{k_1 k_2}{k_2 + k_1}\right)}{m}}$$

Thus, the period of oscillation of the system is

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{\sqrt{\dfrac{\left(\dfrac{k_1 k_2}{k_2 + k_1}\right)}{m}}} = 2\pi\sqrt{\frac{m(k_1 + k_2)}{k_1 k_2}} \qquad \textbf{Ans.}$$

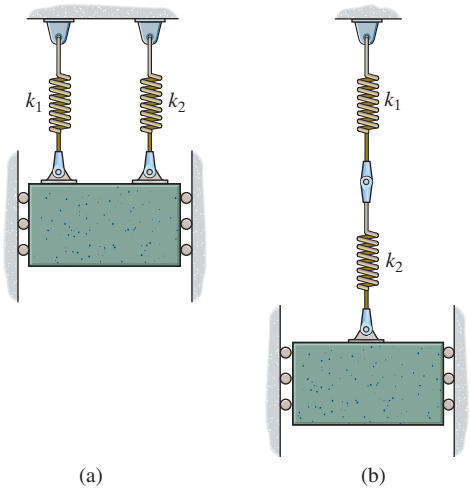
Ans:
$k_{eq} = k_1 + k_2$
$\tau = 2\pi\sqrt{\dfrac{m}{k_1 + k_2}}$
$k_{eq} = \dfrac{k_1 k_2}{k_1 + k_2}$
$\tau = 2\pi\sqrt{\dfrac{m(k_1 + k_2)}{k_1 k_2}}$

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22–17.

The 15-kg block is suspended from two springs having a different stiffness and arranged a) parallel to each other, and b) as a series. If the natural periods of oscillation of the parallel system and series system are observed to be 0.5 s and 1.5 s, respectively, determine the spring stiffnesses k_1 and k_2.



SOLUTION

The equivalent spring stiffness of the spring system arranged in parallel is $(k_{eq})_P = k_1 + k_2$ and the equivalent stiffness of the spring system arranged in a series can be determined by equating the stretch of the system to a single equivalent spring when they are subjected to the same load.

$$\frac{F}{k_1} + \frac{F}{k_2} = \frac{F}{(k_{eq})_S}$$

$$\frac{k_2 + k_1}{k_1 k_2} = \frac{1}{(k_{eq})_S}$$

$$(k_{eq})_S = \frac{k_1 k_2}{k_1 + k_2}$$

Thus the natural frequencies of the parallel and series spring system are

$$(\omega_n)_P = \sqrt{\frac{(k_{eq})_P}{m}} = \sqrt{\frac{k_1 + k_2}{15}}$$

$$(\omega_n)_S = \sqrt{\frac{(k_{eq})_S}{m}} = \sqrt{\frac{\left(\frac{k_1 k_2}{k_1 + k_2}\right)}{15}} = \sqrt{\frac{k_1 k_2}{15(k_1 + k_2)}}$$

Thus, the natural periods of oscillation are

$$\tau_P = \frac{2\pi}{(\omega_n)_P} = 2\pi\sqrt{\frac{15}{k_1 + k_2}} = 0.5 \qquad \textbf{(1)}$$

$$\tau_S = \frac{2\pi}{(\omega_n)_S} = 2\pi\sqrt{\frac{15(k_1 + k_2)}{k_1 k_2}} = 1.5 \qquad \textbf{(2)}$$

Solving Eqs. (1) and (2),

$k_1 = 2067$ N/m or 302 N/m **Ans.**

$k_2 = 302$ N/m or 2067 N/m **Ans.**

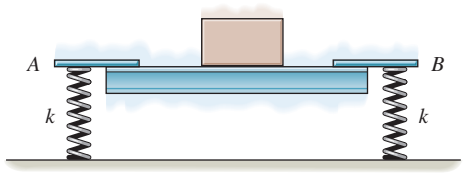
Ans:
$k_1 = 2067$ N/m
$k_2 = 302$ N/m
or vice versa

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22–18.

The uniform beam is supported at its ends by two springs A and B, each having the same stiffness k. When nothing is supported on the beam, it has a period of vertical vibration of 0.83 s. If a 50-kg mass is placed at its center, the period of vertical vibration is 1.52 s. Compute the stiffness of each spring and the mass of the beam.



SOLUTION

$$\tau = 2\pi\sqrt{\frac{m}{k}}$$

$$\frac{\tau^2}{(2\pi)^2} = \frac{m}{k}$$

$$\frac{(0.83)^2}{(2\pi)^2} = \frac{m_B}{2k} \qquad \textbf{(1)}$$

$$\frac{(1.52)^2}{(2\pi)^2} = \frac{m_B + 50}{2k} \qquad \textbf{(2)}$$

Eqs. (1) and (2) become

$$m_B = 0.03490k$$

$$m_B + 50 = 0.1170k$$

$$m_B = 21.2\ \text{kg} \qquad \textbf{Ans.}$$

$$k = 609\ \text{N/m} \qquad \textbf{Ans.}$$

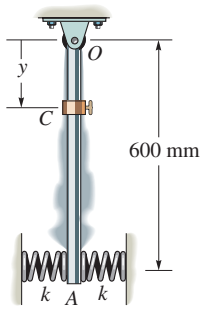
Ans:
$m_B = 21.2$ kg
$k = 609$ N/m

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22–19.

The slender rod has a mass of 0.2 kg and is supported at O by a pin and at its end A by two springs, each having a stiffness $k = 4$ N/m. The period of vibration of the rod can be set by fixing the 0.5-kg collar C to the rod at an appropriate location along its length. If the springs are originally unstretched when the rod is vertical, determine the position y of the collar so that the natural period of vibration becomes $\tau = 1$ s. Neglect the size of the collar.



SOLUTION

Moment of inertia about O:

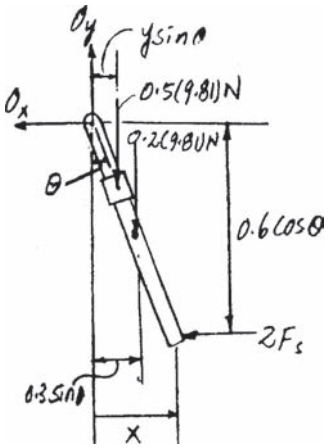
$$I_O = \frac{1}{3}(0.2)(0.6)^2 + 0.5y^2 = 0.024 + 0.5y^2$$

Each spring force $F_s = kx = 4x$.

$$\zeta + \Sigma M_O = I_O\,\alpha; \qquad -2(4x)(0.6\cos\theta) - 0.2(9.81)(0.3\sin\theta)$$

$$-0.5(9.81)(y\sin\theta) = (0.024 + 0.5y^2)\,\ddot{\theta}$$

$$-4.8x\cos\theta - (0.5886 + 4.905y)\sin\theta = (0.024 + 0.5y^2)\ddot{\theta}$$



However, for small displacement $x = 0.6\theta$, $\sin\theta \approx \theta$ and $\cos\theta = 1$. Hence

$$\ddot{\theta} + \frac{3.4686 + 4.905y}{0.024 + 0.5y^2}\theta = 0$$

From the above differential equation, $p = \sqrt{\dfrac{3.4686 + 4.905y}{0.024 + 0.5y^2}}$.

$$\tau = \frac{2\pi}{p}$$

$$1 = \frac{2\pi}{\sqrt{\dfrac{3.4686 + 4.905y}{0.024 + 0.5y^2}}}$$

$$19.74y^2 - 4.905y - 2.5211 = 0$$

$$y = 0.503\text{ m} = 503\text{ mm} \qquad \textbf{Ans.}$$

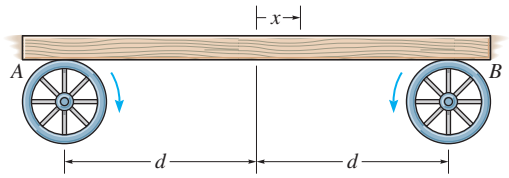
Ans:
$y = 503$ mm

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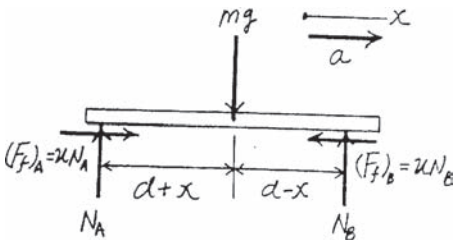
***22–20.**

A uniform board is supported on two wheels which rotate in opposite directions at a constant angular speed. If the coefficient of kinetic friction between the wheels and board is μ, determine the frequency of vibration of the board if it is displaced slightly, a distance x from the midpoint between the wheels, and released.



SOLUTION

***Freebody Diagram*:** When the board is being displaced x to the right, the *restoring force* is due to the unbalance friction force at A and B $\left[(F_f)_B > (F_f)_A\right]$.



***Equation of Motion*:**

$$\zeta + \Sigma M_A = \Sigma (M_A)_k\,; \qquad N_B\,(2d) - mg(d + x) = 0$$

$$N_B = \frac{mg(d + x)}{2d}$$

$$+\uparrow \Sigma F_y = m(a_G)_y\,; \qquad N_A + \frac{mg(d + x)}{2d} - mg = 0$$

$$N_A = \frac{mg(d - x)}{2d}$$

$$\pm\rightarrow \Sigma F_x = m(a_G)_x\,; \qquad \mu\left[\frac{mg(d - x)}{2d}\right] - \mu\left[\frac{mg(d + x)}{2d}\right] = ma$$

$$a + \frac{\mu g}{d}x = 0 \qquad \textbf{(1)}$$

***Kinematics*:** Since $a = \dfrac{d^2x}{dt^2} = \ddot{x}$, then substitute this value into Eq.(1), we have

$$\ddot{x} + \frac{\mu g}{d}x = 0 \qquad \textbf{(2)}$$

From Eq.(2), $\omega_n{}^2 = \dfrac{\mu g}{d}$, thus, $\omega_n = \sqrt{\dfrac{\mu g}{d}}$. Applying Eq. 22–4, we have

$$f = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{\mu g}{d}} \qquad \textbf{Ans.}$$

Ans:

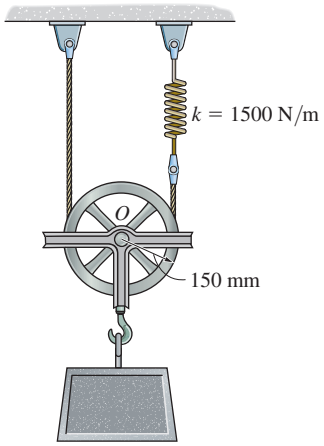
$$f = \frac{1}{2\pi}\sqrt{\frac{\mu g}{d}}$$

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22–21.

The 50-kg block is suspended from the 10-kg pulley that has a radius of gyration about its center of mass of 125 mm. If the block is given a small vertical displacement and then released, determine the natural frequency of oscillation.



SOLUTION

Equation of Motion: When the system is in the equilibrium position, the moment equation of equilibrium written about the *IC* using the free-body diagram of the system shown in Fig. *a* gives

$$\circlearrowleft + \Sigma M_{IC} = 0; \qquad \left(F_{sp}\right)_{st}(0.3) - 10(9.81)(0.15) - 50(9.81)(0.15) = 0$$

$$\left(F_{sp}\right)_{st} = 294.3\ \text{N}$$

Thus, the initial stretch of the spring is $s_0 = \dfrac{\left(F_{sp}\right)_{st}}{k} = \dfrac{294.3}{1500} = 0.1962$ m. Referring the pulley shown in Fig. *a*, the spring stretches further $s_A = r_{A/IC}\theta = 0.3\theta$ when the pulley rotates through a small angle θ. Thus, $F_{sp} = k(s_0 + s_1) = 1500(0.1962 + 0.3\theta) = 294.3 + 450\theta$. Also, $a_G = \ddot{\theta} r_{G/IC} = \ddot{\theta}(0.15)$. The mass moment of inertia of the pulley about its mass center is $I_G = mk_G^2 = 10(0.125^2) = 0.15625\ \text{kg}\cdot\text{m}^2$. Referring to the free-body and kinetic diagrams of the pulley shown in Fig. *b*,

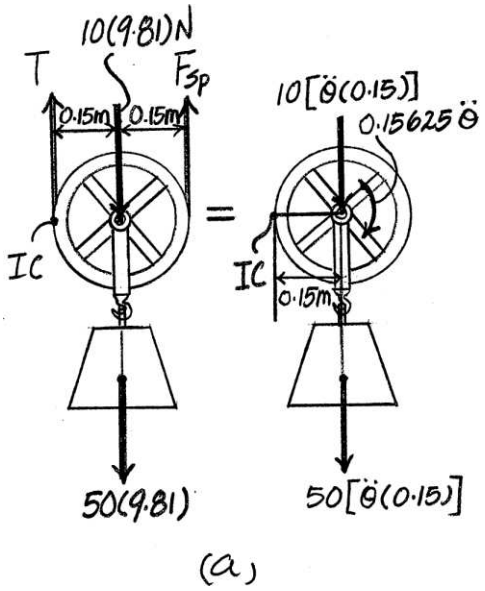
$$\Sigma M_{IC} = \Sigma(M_k)_{IC}; \qquad 10(9.81)(0.15) + 50(9.81)(0.15) - (294.3 + 450\theta)(0.3)$$

$$= 10\left[\ddot{\theta}(0.15)\right](0.15) + 50\left[\ddot{\theta}(0.15)\right](0.15) + 0.15625\ddot{\theta}$$

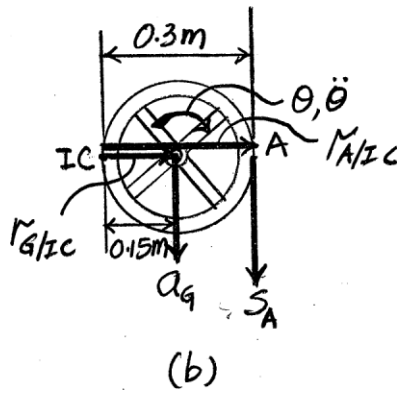
$$\ddot{\theta} + 89.63\theta = 0$$

Comparing this equation to that of the standard form, the natural frequency of the system is

$$\omega_n = \sqrt{89.63}\ \text{rad/s} = 9.47\ \text{rad/s} \qquad \textbf{Ans.}$$



(a)



(b)

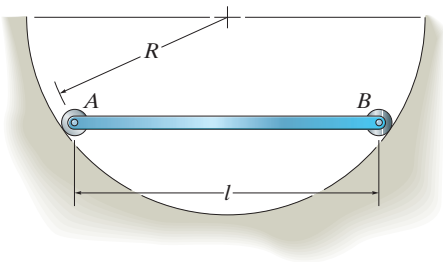
Ans:
$\omega_n = 9.47$ rad/s

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22–22.

The bar has a length l and mass m. It is supported at its ends by rollers of negligible mass. If it is given a small displacement and released, determine the natural frequency of vibration.



SOLUTION

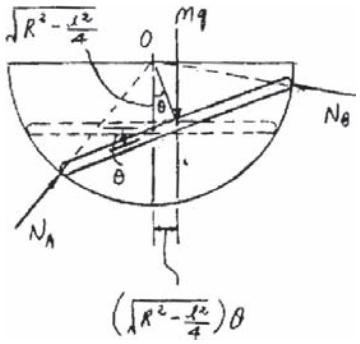
Moment of inertia about point O:

$$I_O = \frac{1}{12}ml^2 + m\left(\sqrt{R^2 - \frac{l^2}{4}}\right)^2 = m\left(R^2 - \frac{1}{6}l^2\right)$$

$$\zeta + \Sigma M_O = I_O\alpha; \qquad mg\left(\sqrt{R^2 - \frac{l^2}{4}}\right)\theta = -m\left(R^2 - \frac{1}{6}l^2\right)\ddot{\theta}$$

$$\ddot{\theta} + \frac{3g(4R^2 - l^2)^{\frac{1}{2}}}{6R^2 - l^2}\theta = 0$$

From the above differential equation, $\omega_n = \sqrt{\dfrac{3g(4R^2 - l^2)^{\frac{1}{2}}}{6R^2 - l^2}}$. **Ans.**



Ans:

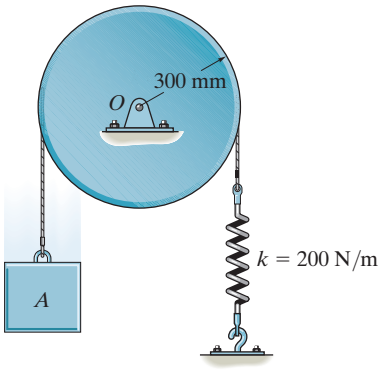
$$\omega_n = \sqrt{\frac{3g(4R^2 - l^2)^{1/2}}{6R^2 - l^2}}$$

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22–23.

The 20-kg disk, is pinned at its mass center O and supports the 4-kg block A. If the belt which passes over the disk is not allowed to slip at its contacting surface, determine the natural period of vibration of the system.



SOLUTION

Equation of Motion. The mass moment of inertia of the disk about its mass center O is $I_0 = \frac{1}{2}mr^2 = \frac{1}{2}(20)(0.3^2) = 0.9\ \text{kg}\cdot\text{m}^2$. When the disk undergoes a small angular displacement θ, the spring stretches further by $s = r\theta = 0.3\theta$. Thus, the total stretch is $y = y_{st} + 0.3\theta$. Then $F_{sp} = ky = 200(y_{st} + 0.3\theta)$. Referring to the FBD and kinetic diagram of the system, Fig. *a*,

$$\zeta + \Sigma M_0 = \Sigma(\mu_k)_0;\quad 4(9.81)(0.3) - 200(y_{st} + 0.3\theta)(0.3) = 0.90\alpha + 4[\alpha(0.3)](0.3)$$

$$11.772 - 60y_{st} - 18\theta = 1.26\alpha \qquad \textbf{(1)}$$

When the system is in equilibrium, $\theta = 0°$. Then

$$\zeta + \Sigma M_0 = 0;\qquad 4(9.81)(0.3) - 200(y_{st})(0.3) = 0$$

$$60y_{st} = 11.772$$

Substitute this result into Eq. (1), we obtain

$$-18\theta = 1.26\alpha$$

$$\alpha + 14.2857\theta = 0$$

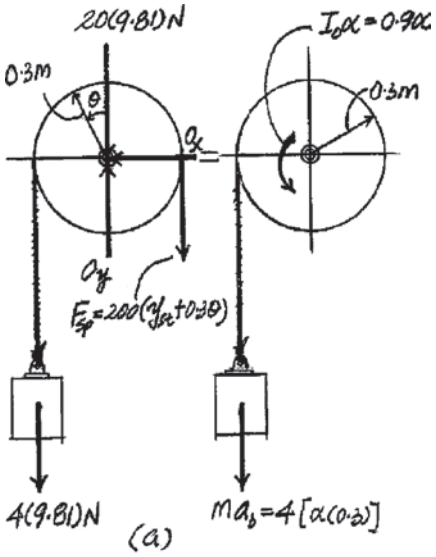
Since $\alpha = \ddot{\theta}$, the above equation becomes

$$\ddot{\theta} + 14.2857\theta = 0$$

Comparing to that of standard form, $\omega_n = \sqrt{14.2857} = 3.7796\ \text{rad/s}$.

Thus,

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{3.7796} = 1.6623\ \text{s} = 1.66\ \text{s} \qquad \textbf{Ans.}$$



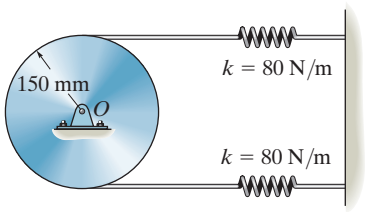
Ans:
$\tau = 1.66\ \text{s}$

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***22–24.**

The 10-kg disk is pin connected at its mass center. Determine the natural period of vibration of the disk if the springs have sufficient tension in them to prevent the cord from slipping on the disk as it oscillates. *Hint:* Assume that the initial stretch in each spring is δ_O.



SOLUTION

Equation of Motion. The mass moment of inertia of the disk about its mass center O is $I_0 = \frac{1}{2}Mr^2 = \frac{1}{2}(10)(0.15^2) = 0.1125\ \text{kg}\cdot\text{m}^2$. When the disk undergoes a small angular displacement θ, the top spring stretches further but the stretch of the spring is being reduced both by $s = r\theta = 0.15\theta$. Thus, $(F_{sp})_t = Kx_t = 80(\delta_0 - 0.15\theta)$ and $(F_{sp})_b = 80(\delta_0 - 0.15\theta)$. Referring to the FBD of the disk, Fig. *a*,

$$\zeta + \Sigma M_0 = I_0\alpha; \qquad -80(\delta_0 + 0.15\theta)(0.15) + 80(\delta_0 - 0.15\theta)(0.15) = 0.1125\alpha$$

$$-3.60\theta = 0.1125\alpha$$

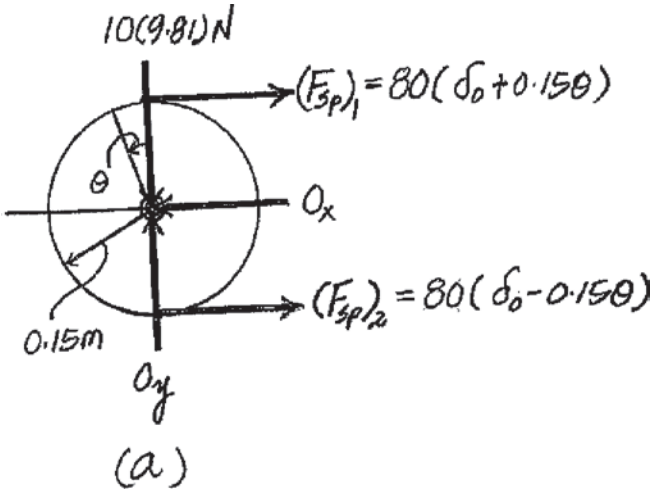
$$\alpha + 32\theta = 0$$

Since $\alpha = \ddot{\theta}$, this equation becomes

$$\ddot{\theta} + 32\theta = 0$$

Comparing to that of standard form, $\omega_n = \sqrt{32}\ \text{rad/s}$. Then

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{\sqrt{32}} = 1.1107\ \text{s} = 1.11\ \text{s} \qquad \textbf{Ans.}$$



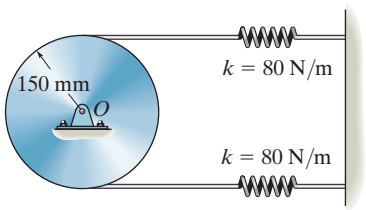
Ans:
$\tau = 1.11\ \text{s}$

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22–25.

If the disk in Prob. 22–24 has a mass of 10 kg, determine the natural frequency of vibration. *Hint:* Assume that the initial stretch in each spring is δ_O.



SOLUTION

Equation of Motion. The mass moment of inertia of the disk about its mass center O is $I_0 = \frac{1}{2}mr^2 = \frac{1}{2}(10)(0.15^2) = 0.1125\ \text{kg}\cdot\text{m}^2$ when the disk undergoes a small angular displacement θ, the top spring stretches but the bottom spring compresses, both by $s = r\theta = 0.15\theta$. Thus, $(F_{sp})_t = (F_{sp})_b = ks = 80(0.15\theta) = 12\theta$. Referring to the FBD of the disk, Fig. *a*,

$$\zeta + \Sigma M_0 = I_0\alpha; \qquad -12\theta(0.3) = 0.1125\alpha$$

$$-3.60\theta = 0.1125\alpha$$

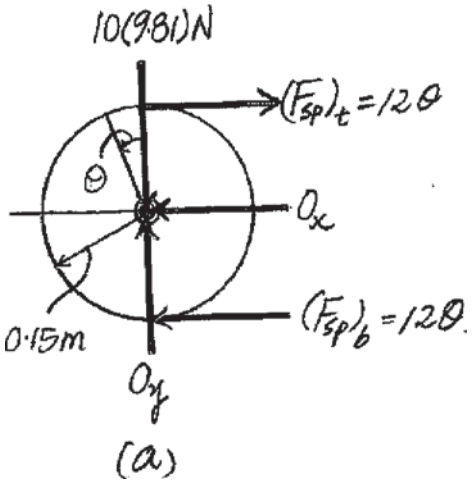
$$\alpha + 32\theta = 0$$

Since $\alpha = \ddot{\theta}$, this equation becomes

$$\ddot{\theta} + 32\theta = 0$$

Comparing to that of Standard form, $\omega_n = \sqrt{32}$ rad/s. Then

$$f = \frac{\omega_n}{2\pi} = \frac{\sqrt{32}}{2\pi} = 0.9003\ \text{Hz} = 0.900\ \text{Hz}$$



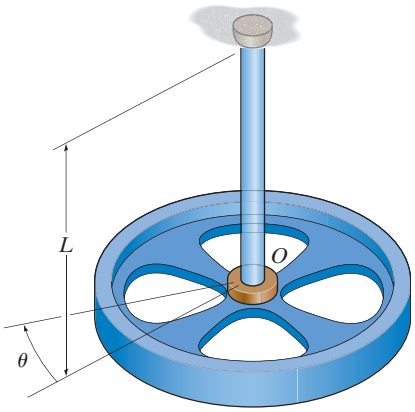
Ans:
$f = 0.900$ Hz

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22–26.

A flywheel of mass m, which has a radius of gyration about its center of mass of k_O, is suspended from a circular shaft that has a torsional resistance of $M = C\theta$. If the flywheel is given a small angular displacement of θ and released, determine the natural period of oscillation.

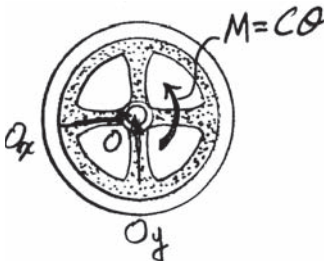


SOLUTION

Equation of Motion: The mass moment of inertia of the wheel about point O is $I_O = mk_O^2$. Referring to Fig. *a*,

$$\circlearrowleft + \qquad \Sigma M_O = I_O\,\alpha; \qquad -C\theta = mk_O^2\ddot{\theta}$$

$$\ddot{\theta} + \frac{C}{mk_O^2}\theta = 0$$



Comparing this equation to the standard equation, the natural circular frequency of the wheel is

$$\omega_n = \sqrt{\frac{C}{mk_O^2}} = \frac{1}{k_O}\sqrt{\frac{C}{m}}$$

Thus, the natural period of the oscillation is

$$\tau = \frac{2\pi}{\omega_n} = 2\pi k_O\sqrt{\frac{m}{C}} \qquad \textbf{Ans.}$$

Ans:

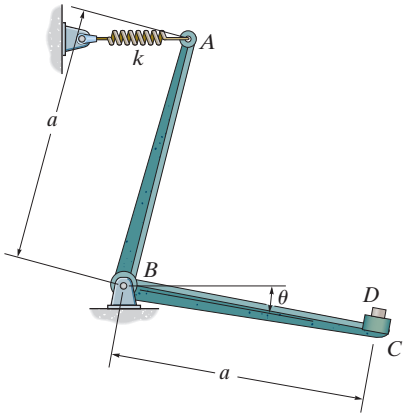
$$\tau = 2\pi k_O\sqrt{\frac{m}{C}}$$

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22–27.

If a block D of negligible size and of mass m is attached at C, and the bell crank of mass M is given a small angular displacement of θ, the natural period of oscillation is τ_1. When D is removed, the natural period of oscillation is τ_2. Determine the bell crank's radius of gyration about its center of mass, pin B, and the spring's stiffness k. The spring is unstrectched at $\theta = 0°$, and the motion occurs in the *horizontal plane*.



SOLUTION

Equation of Motion: When the bell crank rotates through a small angle θ, the spring stretches $s = a\theta$. Thus, the force in the spring is $F_{sp} = ks = k(a\theta)$. The mass moment of inertia of the bell crank about its mass center B is $I_B = Mk_B^2$. Referring to the free-body diagram of the bell crank shown in Fig. *a*,

$$\circlearrowleft + \Sigma M_B = I_B\alpha; \qquad -k(a\theta)\cos\theta(a) = Mk_B^2\ddot{\theta}_B$$

$$\ddot{\theta} + \frac{ka^2}{Mk_B^2}(\cos\theta)\theta = 0 \tag{1}$$

Since θ is very small, $\cos\theta \cong 1$. Then Eq.(1) becomes

$$\ddot{\theta} + \frac{ka^2}{Mk_B^2}\theta = 0$$

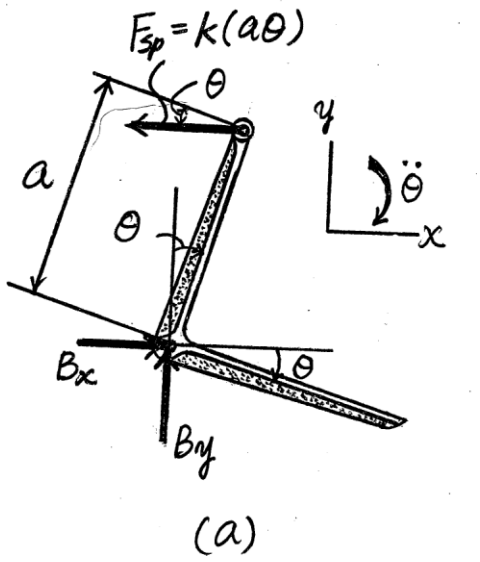
Since the bell crank rotates about point B, $a_C = \alpha r_{BC} = \ddot{\theta}(a)$. Referring to the free-body diagram shown in Fig. *b*,

$$\circlearrowleft + \Sigma M_B = \Sigma(M_k)_B; \qquad -k(a\theta)\cos\theta(a) = Mk_B^2\ddot{\theta} + m\left[\ddot{\theta}(a)\right](a)$$

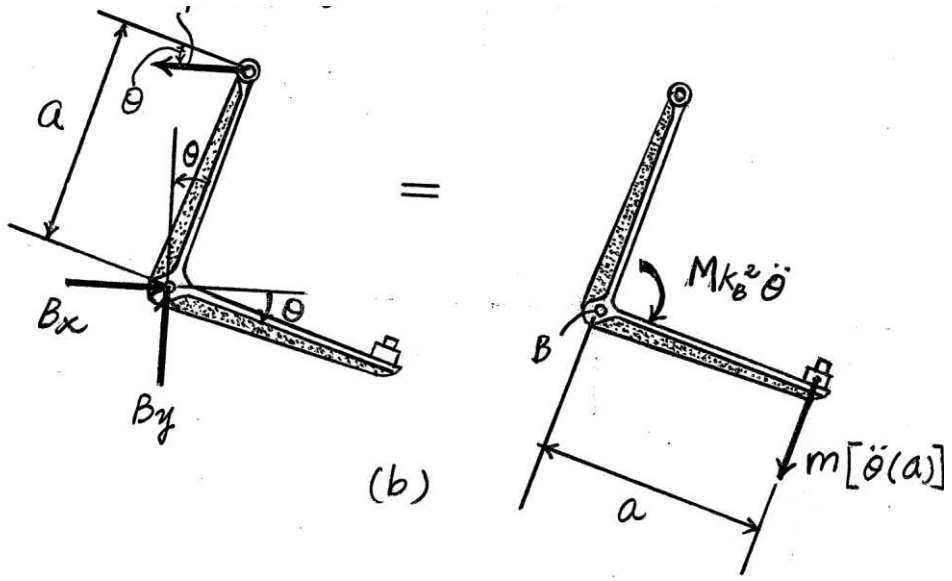
$$\ddot{\theta} + \frac{ka^2}{Mk_B^2 + ma^2}(\cos\theta)\theta = 0 \tag{2}$$

Again, $\cos\theta \cong 1$, since θ is very small. Thus, Eq. (2) becomes

$$\ddot{\theta} + \frac{ka^2}{Mk_B^2 + ma^2}\theta = 0$$



(a)



(b)

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22–27. continued

Thus, the natural frequencies of the two oscillations are

$$(\omega_n)_2 = \sqrt{\frac{ka^2}{Mk_B{}^2}}$$

$$(\omega_n)_1 = \sqrt{\frac{ka^2}{Mk_B{}^2 + ma^2}}$$

The natural periods of the two oscillations are

$$\tau_2 = \frac{2\pi}{(\omega_n)_2} = 2\pi\sqrt{\frac{Mk_B{}^2}{ka^2}}$$

$$\tau_1 = \frac{2\pi}{(\omega_n)_1} = 2\pi\sqrt{\frac{Mk_B{}^2 + ma^2}{ka^2}}$$

Solving,

$$k_B = a\sqrt{\frac{m}{M}\left(\frac{\tau_2{}^2}{\tau_1{}^2 - \tau_2{}^2}\right)}$$ **Ans.**

$$k = \frac{4\pi^2}{\tau_1{}^2 - \tau_2{}^2}\, m$$ **Ans.**

Ans:

$$k_B = a\sqrt{\frac{m}{M}\left(\frac{\tau_2^2}{\tau_1^2 - \tau_2^2}\right)}$$

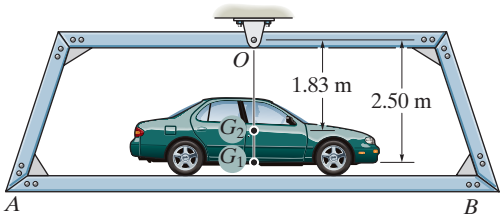
$$k = \frac{4\pi^2}{\tau_1^2 - \tau_2^2}\, m$$

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***22–28.**

The platform AB when empty has a mass of 400 kg, center of mass at G_1, and natural period of oscillation $\tau_1 = 2.38$ s. If a car, having a mass of 1.2 Mg and center of mass at G_2, is placed on the platform, the natural period of oscillation becomes $\tau_2 = 3.16$ s. Determine the moment of inertia of the car about an axis passing through G_2.



SOLUTION

Free-body Diagram: When an object arbitrary shape having a mass m is pinned at O and being displaced by an angular displacement of θ, the tangential component of its weight will create the *restoring moment* about point O.

Equation of Motion: Sum moment about point O to eliminate O_x and O_y.

$$\curvearrowleft + \Sigma M_O = I_O\alpha: \qquad -mg\sin\theta(l) = I_O\alpha \tag{1}$$

Kinematics: Since $\alpha = \dfrac{d^2\theta}{dt^2} = \ddot{\theta}$ and $\sin\theta = \theta$ if θ is small, then substituting these values into Eq. (1), we have

$$-mgl\theta = I_O\ddot{\theta} \qquad \text{or} \qquad \ddot{\theta} + \frac{mgl}{I_O}\theta = 0 \tag{2}$$

From Eq. (2), $\omega_n^2 = \dfrac{mgl}{I_O}$, thus, $\omega_n = \sqrt{\dfrac{mgl}{I_O}}$, Applying Eq. 22–12, we have

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{I_O}{mgl}} \tag{3}$$

When the platform is empty, $\tau = \tau_1 = 2.38$ s, $m = 400$ kg and $l = 2.50$ m. Substituting these values into Eq. (3), we have

$$2.38 = 2\pi\sqrt{\frac{(I_O)_p}{400(9.81)(2.50)}} \qquad (I_O)_p = 1407.55\ \text{kg}\cdot\text{m}^2$$

When the car is on the platform, $\tau = \tau_2 = 3.16$ s, $m = 400$ kg $+ 1200$ kg $= 1600$ kg.
$l = \dfrac{2.50(400) + 1.83(1200)}{1600} = 1.9975$ m and $I_O = (I_O)_C + (I_O)_p = (I_O)_C +$ 1407.55. Substituting these values into Eq. (3), we have

$$3.16 = 2\pi\sqrt{\frac{(I_O)_C + 1407.55}{1600(9.81)(1.9975)}} \qquad (I_O)_C = 6522.76\ \text{kg}\cdot\text{m}^2$$

Thus, the mass moment inertia of the car about its mass center is

$$(I_G)_C = (I_O)_C - m_C d^2$$
$$= 6522.76 - 1200(1.83^2) = 2.50(10^3)\ \text{kg}\cdot\text{m}^2 \qquad \textbf{Ans.}$$

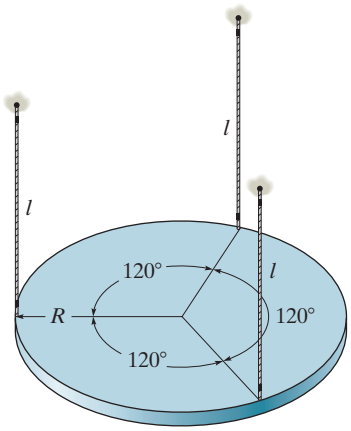
Ans:
$(I_G)_C = 2.50(10^3)\ \text{kg}\cdot\text{m}^2$

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22–29.

The plate of mass m is supported by three symmetrically placed cords of length l as shown. If the plate is given a slight rotation about a vertical axis through its center and released, determine the natural period of oscillation.



SOLUTION

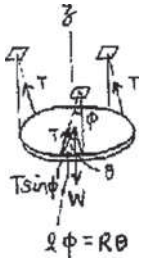
$$\Sigma M_z = I_z\alpha \qquad -3(T\sin\phi)R = \frac{1}{2}mR^2\ddot{\theta}$$

$$\sin\phi \equiv \phi$$

$$\ddot{\theta} + \frac{6T}{Rm}\phi = 0$$

$$\Sigma F_z = 0 \qquad 3T\cos\phi - mg = 0$$

$$\phi = 0, \qquad T = \frac{mg}{3}, \qquad \phi = \frac{R}{l}\theta$$



$$\ddot{\theta} + \frac{6}{Rm}\left(\frac{mg}{3}\right)\left(\frac{R}{l}\theta\right) = 0$$

$$\ddot{\theta} + \frac{2g}{l}\theta = 0$$

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{l}{2g}}$$

Ans.

Ans:

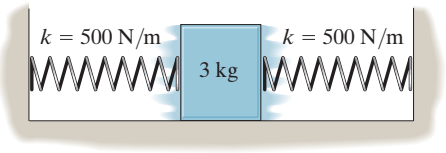
$$\tau = 2\pi\sqrt{\frac{l}{2g}}$$

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22–30.

Determine the differential equation of motion of the 3-kg block when it is displaced slightly and released. The surface is smooth and the springs are originally unstretched.



SOLUTION

$T + V = \text{const.}$

$T = \frac{1}{2}(3)\dot{x}^2$

$V = \frac{1}{2}(500)x^2 + \frac{1}{2}(500)x^2$

$T + V = 1.5\dot{x}^2 + 500x^2$

$1.5(2\dot{x})\,\ddot{x} + 1000x\dot{x} = 0$

$3\ddot{x} + 1000x = 0$

$\ddot{x} + 333x = 0$ **Ans.**

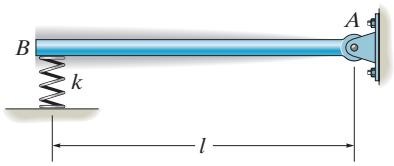
Ans:
$\ddot{x} + 333x = 0$

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22–31.

The uniform rod of mass m is supported by a pin at A and a spring at B. If the end B is given a small downward displacement and released, determine the natural period of vibration.



SOLUTION

$$T = \frac{1}{2}\left(\frac{1}{3}ml^2\right)\dot{\theta}^2$$

$$V = \frac{1}{2}k(y_{eq} + y_2)^2 - mgy_1$$

$$= \frac{1}{2}k(l\theta_{eq} + l\theta)^2 - mg\left(\frac{1}{2}\right)\theta$$

$$T + V = \frac{1}{6}ml^2\dot{\theta}^2 + \frac{1}{2}k(l\theta_{eq} + l\theta)^2 - mg\left(\frac{l\theta}{2}\right)$$

Time derivative

$$0 = \frac{1}{3}ml^2\dot{\theta}\ddot{\theta} + kl(\theta_{eq} + \theta)\dot{\theta} - mgl\frac{\dot{\theta}}{2}$$

For equilibrium

$$k(l\theta_{eq}) = mgl/2, \theta_{eq} = \frac{mg}{2k}$$

Thus,

$$0 = \frac{1}{3}ml\ddot{\theta} + k\theta$$

$$\ddot{\theta} + (3k/m)\theta = 0$$

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m}{3k}}$$ **Ans.**

Ans:

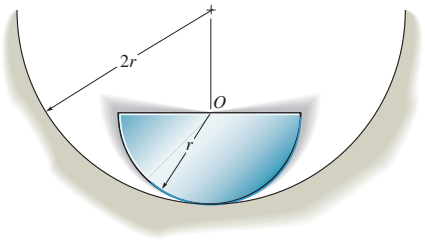
$$\tau = 2\pi\sqrt{\frac{m}{3k}}$$

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***22–32.**

The semicircular disk has a mass m and radius r, and it rolls without slipping in the semicircular trough. Determine the natural period of vibration of the disk if it is displaced slightly and released. *Hint:* $I_O = \frac{1}{2}mr^2$.



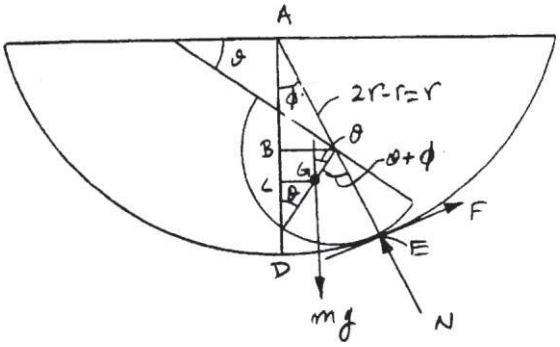
SOLUTION

$$AB = (2r - r)\cos\phi = r\cos\phi, \qquad BC = \frac{4r}{3\pi}\cos\theta$$

$$AC = r\cos\phi + \frac{4r}{3\pi}\cos\theta, \qquad DE = 2r\phi = r(\theta + \phi)$$

$$\phi = \theta$$

$$AC = r\left(1 + \frac{4}{3\pi}\right)\cos\theta$$



Thus, the change in elevation of G is

$$h = 2r - \left(r - \frac{4r}{3\pi}\right) - AC = r\left(1 + \frac{4}{3\pi}\right)(1 - \cos\theta)$$

Since no slipping occurs,

$$v_G = \dot{\theta}\left(r - \frac{4r}{3\pi}\right)$$

$$I_G = I_O - m\left(\frac{4r}{3\pi}\right)^2 = \left(\frac{1}{2} - \left(\frac{4}{3\pi}\right)^2\right)mr^2$$

$$T = \frac{1}{2}m\dot{\theta}^2r^2\left(1 - \frac{4}{3\pi}\right)^2 + \frac{1}{2}\left(\frac{1}{2} - \left(\frac{4}{3\pi}\right)^2\right)mr^2\dot{\theta}^2 = \frac{1}{2}mr^2\left(\frac{3}{2} - \frac{8}{3\pi}\right)\dot{\theta}^2$$

$$T + V = \frac{1}{2}mr^2\left(\frac{3}{2} - \frac{8}{3\pi}\right)\dot{\theta}^2 + mgr\left(1 + \frac{4}{3\pi}\right)(1 - \cos\theta)$$

$$0 = mr^2\left(\frac{3}{2} - \frac{8}{3\pi}\right)\dot{\theta}\,\ddot{\theta} + mgr\left(1 + \frac{4}{3\pi}\right)\sin\theta\dot{\theta}$$

$$\sin\theta \approx \theta$$

$$\ddot{\theta} + \frac{g\left(1 + \frac{4}{3\pi}\right)}{r\left(\frac{3}{2} - \frac{8}{3\pi}\right)}\theta = 0$$

$$\omega_n = 1.479\sqrt{\frac{g}{r}}$$

$$\tau = \frac{2\pi}{\omega_n} = 4.25\sqrt{\frac{r}{g}}$$

Ans.

Ans:

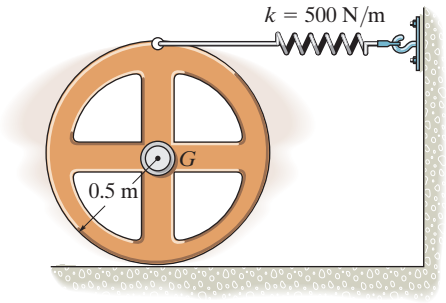
$$\tau = 4.25\sqrt{\frac{r}{g}}$$

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22–33.

If the 20-kg wheel is displaced a small amount and released, determine the natural period of vibration. The radius of gyration of the wheel is $k_G = 0.36$ m. The wheel rolls without slipping.



SOLUTION

Energy Equation. The mass moment of inertia of the wheel about its mass center is $I_G = mk_G = 20(0.361)^2 = 2.592\ \text{kg}\cdot\text{m}^2$. Since the wheel rolls without slipping, $v_G = \omega r = \omega(0.5)$. Thus,

$$T = \frac{1}{2}I_G\omega^2 + \frac{1}{2}mv_G^2$$

$$= \frac{1}{2}(2.592)\omega^2 + \frac{1}{2}(20)[\omega(0.5)]^2$$

$$= 3.796\,\omega^2 = 3.796\dot{\theta}^2$$

When the disk undergoes a small angular displacement θ, the spring stretches $s = \theta(1) = \theta$, Fig. *a*. Thus, the elastic potential energy is

$$V_e = \frac{1}{2}ks^2 = \frac{1}{2}(500)\theta^2 = 250\theta^2$$

Thus, the total energy is

$$E = T + V = 3.796\dot{\theta}^2 + 250\theta^2$$

Time Derivative. Taking the time derivative of the above equation,

$$7.592\dot{\theta}\ddot{\theta} + 500\theta\dot{\theta} = 0$$

$$\dot{\theta}(7.592\ddot{\theta} + 500\theta) = 0$$

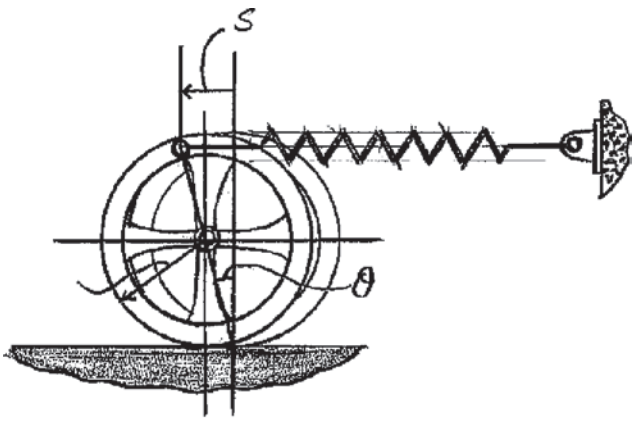
Since $\dot{\theta} \neq 0$, then

$$7.592\ddot{\theta} + 500\theta = 0$$

$$\ddot{\theta} + 65.8588\theta = 0$$

Comparing to that of standard form, $\omega_n = \sqrt{65.8588} = 8.1153$ rad/s. Thus,

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{8.1153} = 0.7742\ \text{s} = 0.774\ \text{s}$$ **Ans.**



(a)

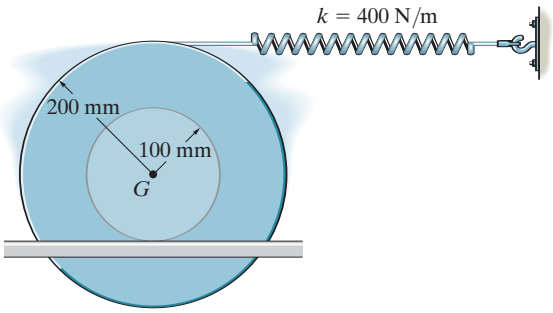
Ans:
$\tau = 0.774$ s

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22–34.

Determine the differential equation of motion of the 3-kg spool. Assume that it does not slip at the surface of contact as it oscillates. The radius of gyration of the spool about its center of mass is $k_G = 125$ mm.

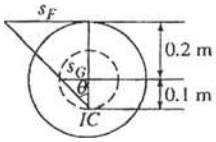


SOLUTION

Kinematics: Since no slipping occurs, $s_G = 0.1\theta$ hence $s_F = \frac{0.3}{0.1}S_G = 0.3\theta$. Also,

$$v_G = 0.1\dot{\theta}.$$

$$E = T + V$$



$$E = \frac{1}{2}[(3)(0.125)^2]\dot{\theta}^2 + \frac{1}{2}(3)(0.1\theta)^2 + \frac{1}{2}(400)(0.3\theta)^2 = \text{const.}$$

$$= 0.03844\dot{\theta}^2 + 18\theta^2$$

$$0.076875\ddot{\theta}\dot{\theta} + 36\theta\dot{\theta} = 0$$

$$0.076875\dot{\theta}(\ddot{\theta} + 468.29\theta) = 0 \text{ Since } 0.076875\theta \neq 0$$

$$\ddot{\theta} + 468\theta = 0 \qquad \textbf{Ans.}$$

Ans:

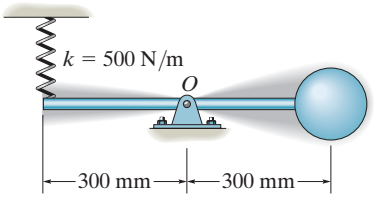
$$\ddot{\theta} + 468\theta = 0$$

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22–35.

Determine the natural period of vibration of the 3-kg sphere. Neglect the mass of the rod and the size of the sphere.

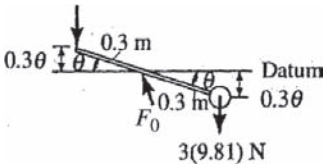


SOLUTION

$$E = T + V$$

$$= \frac{1}{2}(3)(0.3\dot{\theta})^2 + \frac{1}{2}(500)(\delta_{st} + 0.3\theta)^2 - 3(9.81)(0.3\theta)$$

$$E = \dot{\theta}[(3(0.3)^2\ddot{\theta} + 500(\delta_{st} + 0.3\theta)(0.3) - 3(9.81)(0.3)] = 0$$



By statics,

$$T(0.3) = 3(9.81)(0.3)$$

$$T = 3(9.81)\ \text{N}$$

$$\delta_{st} = \frac{3(9.81)}{500}$$

Thus,

$$3(0.3)^2\ddot{\theta} + 500(0.3)^2\theta = 0$$

$$\ddot{\theta} + 166.67\theta = 0$$

$$\omega_n = \sqrt{166.67} = 12.91\ \text{rad/s}$$

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{12.91} = 0.487\ \text{s}$$ **Ans.**

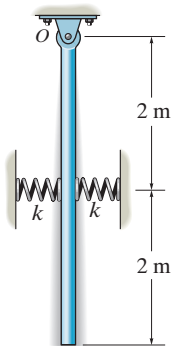
Ans:
$\tau = 0.487$ s

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***22–36.**

If the lower end of the 6-kg slender rod is displaced a small amount and released from rest, determine the natural frequency of vibration. Each spring has a stiffness of $k = 200\ \text{N/m}$ and is unstretched when the rod is hanging vertically.



SOLUTION

Energy Equation. The mass moment of inertia of the rod about O is $I_0 = \frac{1}{3}ml^2 = \frac{1}{3}(6)(4^2) = 32\ \text{kg}\cdot\text{m}^2$. Thus, the Kinetic energy is

$$T = \frac{1}{2}I_0\omega^2 = \frac{1}{2}(32)\dot{\theta}^2 = 16\dot{\theta}^2$$

with reference to the datum set in Fig. *a*, the gravitational potential energy is

$$V_g = mgy = 6(9.81)(-2\cos\theta) = -117.72\cos\theta$$

When the rod undergoes a small angular displacement θ the spring deform $x = 2\sin\Omega$. Thus the elastic potential energy is

$$V_e = 2\left(\frac{1}{2}kx^2\right) = 2\left[\frac{1}{2}(200)(2\sin\theta)^2\right] = 800\sin^2\theta$$

Thus, the total energy is

$$E = T + V = 16\dot{\theta}^2 + 800\sin^2\theta - 117.72\cos\theta$$

Time Derivative. Taking the first time derivative of the above equation

$$32\dot{\theta}\ddot{\theta} + 1600(\sin\theta\cos\theta)\dot{\theta} + 117.72(\sin\theta)\dot{\theta} = 0$$

Using the trigonometry identity $\sin 2\theta = 2\sin\theta\cos\theta$, we obtain

$$32\dot{\theta}\ddot{\theta} + 800(\sin 2\theta)\dot{\theta} + 117.72(\sin\theta)\dot{\theta} = 0$$

$$\dot{\theta}(32\ddot{\theta} + 800\sin 2\theta + 117.72\sin\theta) = 0$$

Since $\dot{\theta} \neq 0$,

$$32\ddot{\theta} + 800\sin 2\theta + 117.72\sin\theta) = 0$$

Since θ is small, $\sin 2\theta \simeq 2\theta$ and $\sin\theta = \theta$. The above equation becomes

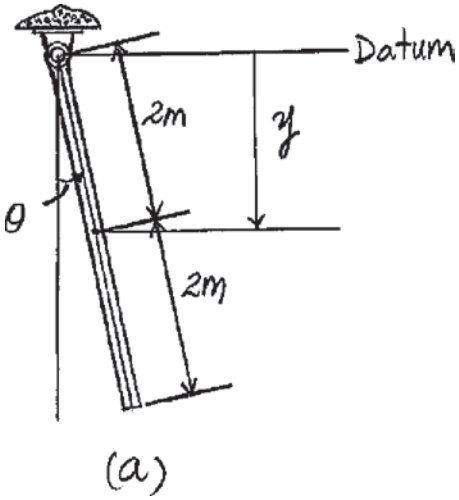
$$32\ddot{\theta} + 1717.72\theta = 0$$

$$\ddot{\theta} + 53.67875\theta = 0$$

Comparing to that of standard form, $\omega_n = \sqrt{53.67875} = 7.3266\ \text{rad/s}$.

Thus,

$$f = \frac{\omega_n}{2\pi} = \frac{7.3266}{2\pi} = 1.1661\ \text{Hz} = 1.17\ \text{Hz}$$ **Ans.**



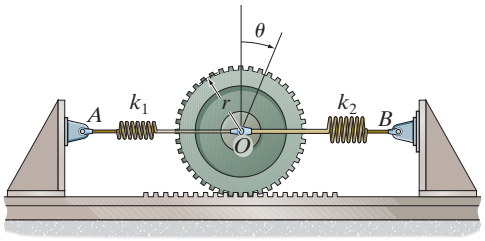
Ans:
$f = 1.17\ \text{Hz}$

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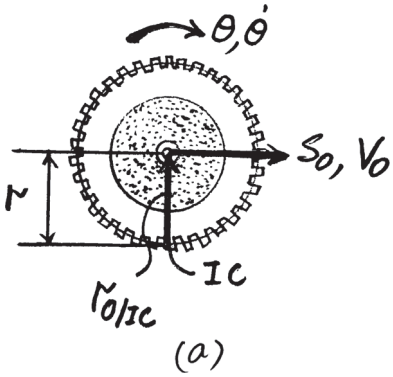
22–37.

The gear of mass m has a radius of gyration about its center of mass O of k_O. The springs have stiffnesses of k_1 and k_2, respectively, and both springs are unstretched when the gear is in an equilibrium position. If the gear is given a small angular displacement of θ and released, determine its natural period of oscillation.



SOLUTION

Potential and Kinetic Energy: Since the gear rolls on the gear rack, springs AO and BO stretch and compress $s_O = r_{O/IC}\theta = r\theta$. When the gear rotates a small angle θ, Fig. a, the elastic potential energy of the system is



$$V = V_e = \frac{1}{2}k_1 s_O{}^2 + \frac{1}{2}k_2 s_O{}^2$$

$$= \frac{1}{2}k_1 (r\theta)^2 + \frac{1}{2}k_2 (r\theta)^2$$

$$= \frac{1}{2}r^2(k_1 + k_2)\theta^2$$

Also, from Fig. a, $v_O = \dot{\theta} r_{O/IC} = \dot{\theta} r$. The mass moment of inertia of the gear about its mass center is $I_O = mk_O{}^2$.

Thus, the kinetic energy of the system is

$$T = \frac{1}{2}mv_O{}^2 + \frac{1}{2}I_O\,\omega^2$$

$$= \frac{1}{2}m(\dot{\theta} r)^2 + \frac{1}{2}\left(mk_O{}^2\right)\dot{\theta}^2$$

$$= \frac{1}{2}m(r^2 + k_O{}^2)\dot{\theta}^2$$

The energy function of the system is therefore

$$T + V = \text{constant}$$

$$\frac{1}{2}m\left(r^2 + k_O{}^2\right)\dot{\theta}^2 + \frac{1}{2}r^2(k_1 + k_2)\theta^2 = \text{constant}$$

Taking the time derivative of this equation,

$$m\left(r^2 + k_O{}^2\right)\dot{\theta}\,\ddot{\theta} + r^2(k_1 + k_2)\theta\dot{\theta} = 0$$

$$\dot{\theta}\left[m\left(r^2 + k_O{}^2\right)\ddot{\theta} + r^2(k_1 + k_2)\theta\right] = 0$$

Since $\dot{\theta}$ is not always equal to zero, then

$$m\left(r^2 + k_O{}^2\right)\ddot{\theta} + r^2(k_1 + k_2)\theta = 0$$

$$\ddot{\theta} + \frac{r^2(k_1 + k_2)}{m\left(r^2 + k_O{}^2\right)}\theta = 0$$

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22–37. continued

Comparing this equation to that of the standard form, the natural circular frequency of the system is

$$\omega_n = \sqrt{\frac{r^2(k_1 + k_2)}{m\left(r^2 + {k_O}^2\right)}}$$

Thus, the natural period of the oscillation is

$$\tau = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m\left(r^2 + {k_O}^2\right)}{r^2(k_1 + k_2)}}$$ **Ans.**

Ans:

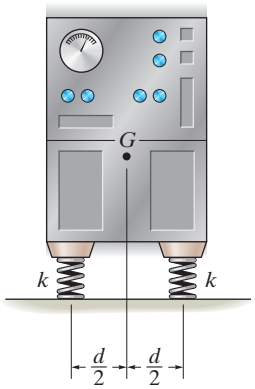
$$\tau = 2\pi\sqrt{\frac{m\left(r^2 + k_O^2\right)}{r^2(k_1 + k_2)}}$$

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22–38.

The machine has a mass m and is uniformly supported by *four* springs, each having a stiffness k. Determine the natural period of vertical vibration.



SOLUTION

$$T + V = \text{const.}$$

$$T = \frac{1}{2} m(\dot{y})^2$$

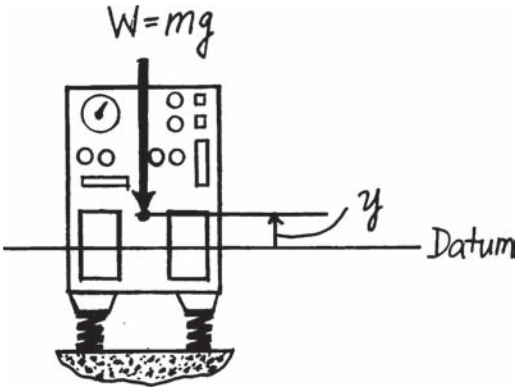
$$V = m\,g\,y + \frac{1}{2}(4k)(\Delta s - y)^2$$

$$T + V = \frac{1}{2} m(\dot{y})^2 + m\,g\,y + \frac{1}{2}(4k)(\Delta s - y)^2$$

$$m\,\dot{y}\,\ddot{y} + m\,g\,\dot{y} - 4k(\Delta s - y)\dot{y} = 0$$

$$m\,\ddot{y} + m\,g + 4ky - 4k\Delta s = 0$$

Since $\Delta s = \dfrac{mg}{4k}$



Then

$$m\ddot{y} + 4ky = 0$$

$$y + \frac{4k}{m} y = 0$$

$$\omega_n = \sqrt{\frac{4k}{m}}$$

$$\tau = \frac{2\pi}{\omega_n} = \pi\sqrt{\frac{m}{k}}$$

Ans.

Ans:

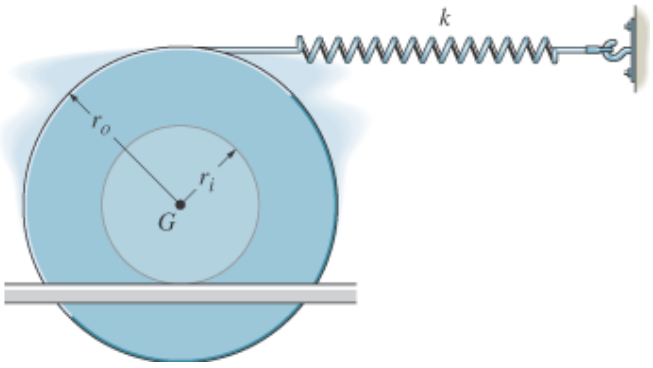
$$\tau = \pi\sqrt{\frac{m}{k}}$$

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22–39.

Determine the differential equation of motion of the spool of mass M. Assume that it does not slip at the surface of contact as it oscillates. The radius of gyration of the spool about its center of mass is k_G.



SOLUTION

$M = 3$ kg

$k_G = 125$ mm

$r_i = 100$ mm

$r_o = 200$ mm

$k = 400$ N/m

$$T + V = \frac{1}{2}M\left(k_G{}^2 + r_i{}^2\right)\theta'^2 + \frac{1}{2}k\left[\left(r_o + r_i\right)\theta\right]^2$$

$$M\left(k_G{}^2 + r_i{}^2\right)\theta' + k\left(r_o + r_i\right)^2\theta = 0 \qquad \omega_n = \sqrt{\frac{k\left(r_o + r_i\right)^2}{M\left(k_G{}^2 + r_i{}^2\right)}}$$

$\theta'' + \omega_n{}^2\theta = 0$ where $\omega_n{}^2 = 468$ rad /s^2 **Ans.**

Ans:

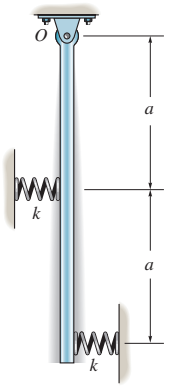
$\theta'' + \omega_n^2\theta = 0$
where $\omega_n^2 = 468$ rad/s^2

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***22–40.**

The slender rod has a mass m and is pinned at its end O. When it is vertical, the springs are unstretched. Determine the natural period of vibration.



SOLUTION

$$T + V = \frac{1}{2}\left[\frac{1}{3}m(2a)^2\right]\dot{\theta}^2 + \frac{1}{2}k(2\theta a)^2 + \frac{1}{2}k(\theta a)^2 + mga(1 - \cos\theta)$$

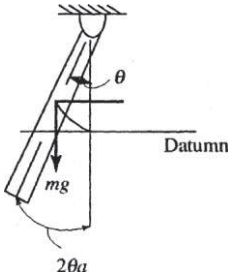
$$0 = \frac{4}{3}ma^2\dot{\theta}\,\ddot{\theta} + 4ka^2\theta\dot{\theta} + ka^2\theta\dot{\theta} + mga\sin\theta\dot{\theta}$$

$$\sin\theta = \theta$$

$$\frac{4}{3}ma^2\ddot{\theta} + 5ka^2\theta + mga\theta = 0$$

$$\ddot{\theta} + \left(\frac{15ka + 3mg}{4ma}\right)\theta = 0$$

$$\tau = \frac{2\pi}{\omega_n} = \frac{4\pi}{\sqrt{3}}\left(\frac{ma}{5ka + mg^2}\right)^{\frac{1}{2}}$$ **Ans.**



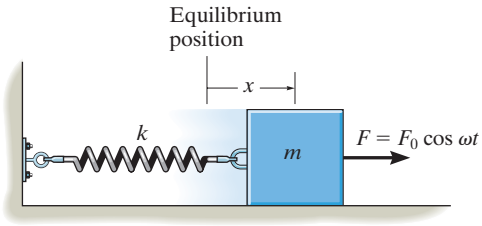
Ans:

$$\tau = \frac{2\pi}{\omega_n} = \frac{4\pi}{\sqrt{3}}\left(\frac{ma}{5ka + mg^2}\right)^{\frac{1}{2}}$$

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22–41. If the block-and-spring model is subjected to the periodic force $F = F_0 \cos \omega t$, show that the differential equation of motion is $\ddot{x} + (k/m)x = (F_0/m)\cos \omega t$, where x is measured from the equilibrium position of the block. What is the general solution of this equation?



SOLUTION

$$\stackrel{+}{\rightarrow} \Sigma F_x = ma_x; \qquad F_0 \cos \omega t - kx = m\ddot{x}$$

$$\ddot{x} + \frac{k}{m}x = \frac{F_0}{m}\cos \omega t \qquad \textbf{(Q.E.D.)}$$

$$\ddot{x} + p^2 x = \frac{F_0}{m}\cos \omega t \qquad \text{Where } p = \sqrt{\frac{k}{m}} \qquad \textbf{(1)}$$

The general solution of the above differential equation is of the form of $x = x_c + x_p$.

The complementary solution:

$$x_c = A \sin pt + B \cos pt$$

The particular solution:

$$s_p = .C \cos \omega t \qquad \textbf{(2)}$$

$$\ddot{x}_P = -C\omega^2 \cos \omega\, t \qquad \textbf{(3)}$$

Substitute Eqs. (2) and (3) into (1) yields:

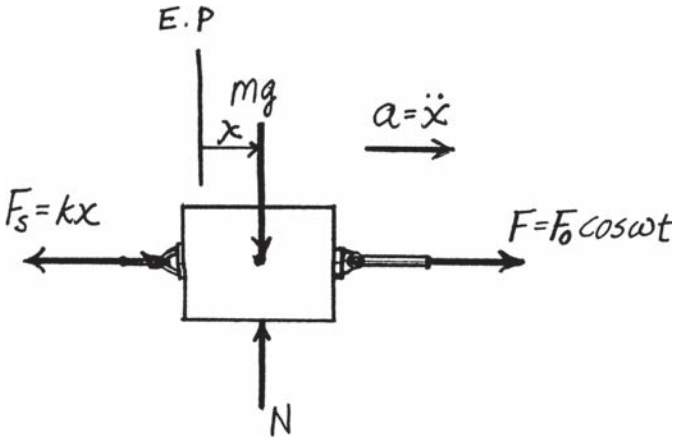
$$-C\omega^2 \cos \omega t + p^2 (C \cos \omega t) = \frac{F_0}{m}\cos \omega t$$

$$C = \frac{\dfrac{F_0}{m}}{p^2 - \omega^2} = \frac{F_0/k}{1 - \left(\frac{\omega}{p}\right)^2}$$

The general solution is therefore

$$s = A \sin pt + B \cos pt + \frac{F_0/k}{1 - \left(\dfrac{\omega}{p}\right)^2}\cos \omega t \qquad \textbf{Ans.}$$

The constants A and B can be found from the initial conditions.



Ans:

$$x = A \sin \omega_n t + B \cos \omega_n t + \frac{F_o/k}{1 - (\omega/p)^2}\cos \omega t$$

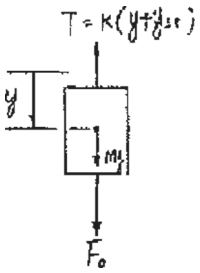
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22–42.

A block which has a mass m is suspended from a spring having a stiffness k. If an impressed downward vertical force $F = F_O$ acts on the weight, determine the equation which describes the position of the block as a function of time.

SOLUTION



$$+\uparrow \Sigma F_y = ma_y; \qquad k(y_{st} + y) - mg - F_0 = -m\ddot{y}$$

$$m\ddot{y} + ky + ky_{st} - mg = F_0$$

However, from equilbrium $ky_{st} - mg = 0$, therefore

$$m\ddot{y} + ky = F_0$$

$$\ddot{y} + \frac{k}{m}y = \frac{F_0}{m} \qquad \text{where } \omega_n = \sqrt{\frac{k}{m}}$$

$$\ddot{y} + \omega_n^2 y = \frac{F_0}{m} \qquad \textbf{[1]}$$

The general solution of the above differential equation is of the form of $y = y_c + y_p$.

$$y_c = A \sin \omega_n t + B \cos \omega_n t$$

$$y_P = C \qquad \textbf{[2]}$$

$$\ddot{y}_P = 0 \qquad \textbf{[3]}$$

Substitute Eqs. [2] and [3] into [1] yields :

$$0 + \omega_n^2 C = \frac{F_0}{m} \qquad C = \frac{F_0}{mp^2} = \frac{F_0}{k}$$

The general solution is therefore

$$y = A \sin \omega_n t + B \cos \omega_n t + \frac{F_0}{k} \qquad \textbf{Ans.}$$

The constants A and B can be found from the initial conditions.

Ans:

$$y = A \sin \omega_n t + B \cos \omega_n t + \frac{F_O}{k}$$

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22–43.

The block shown in Fig. 22–15 has a mass of 20 kg, and the spring has a stiffness $k = 600\ \text{N/m}$. When the block is displaced and released, two successive amplitudes are measured as $x_1 = 150\ \text{mm}$ and $x_2 = 87\ \text{mm}$. Determine the coefficient of viscous damping, c.

SOLUTION

Assuming that the system is underdamped.

$$x_1 = De^{-\left(\frac{c}{2m}\right)t_1} \tag{1}$$

$$x_2 = De^{-\left(\frac{c}{2m}\right)t_2} \tag{2}$$

Divide Eq. (1) by Eq. (2) $\dfrac{x_1}{x_2} = \dfrac{e^{-\left(\frac{c}{2m}\right)t_1}}{e^{-\left(\frac{c}{2m}\right)t_2}}$

$$\ln\left(\frac{x_1}{x_2}\right) = \left(\frac{c}{2m}\right)(t_2 - t_1) \tag{3}$$

However, $t_2 - t_1 = \tau_c = \dfrac{2\pi}{\omega_d} = \dfrac{2\pi}{\omega_n\sqrt{1 - \left(\frac{c}{c_c}\right)^2}}$ and $\omega_n = \dfrac{C_c}{2m}$

$$t_2 - t_1 = \frac{4m\pi}{C_c\sqrt{1 - \left(\frac{c}{C_c}\right)^2}} \tag{4}$$

Substitute Eq. (4) into Eq. (3) yields:

$$\ln\left(\frac{x_1}{x_2}\right) = \left(\frac{c}{2m}\right)\frac{4m\pi}{C_c\sqrt{1 - \left(\frac{C}{C_c}\right)^2}}$$

$$\ln\left(\frac{x_1}{x_2}\right) = \frac{2\pi\left(\frac{C}{C_c}\right)}{\sqrt{1 - \left(\frac{C}{C_c}\right)^2}} \tag{5}$$

From Eq. (5)

$$x_1 = 0.15\ \text{m}\ \ x_2 = 0.087\text{m}\ \ \omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{600}{20}} = 5.477\ \text{rad/s}$$

$$C_c = 2m\omega_n = 2(20)(5.477) = 219.09\ \text{N}\cdot\text{s/m}$$

$$\ln\left(\frac{0.15}{0.087}\right) = \frac{2\pi\left(\frac{c}{219.09}\right)}{\sqrt{1 - \left(\frac{c}{219.09}\right)^2}}$$

$$c = 18.9\ \text{N}\cdot\text{s/m} \qquad \textbf{Ans.}$$

Since $C < C_c$, the system is underdamped. Therefore, the assumption is OK!

Ans:

$c = 18.9\ \text{N}\cdot\text{s/m}$

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***22–44.**

A 4-kg block is suspended from a spring that has a stiffness of $k = 600$ N/m. The block is drawn downward 50 mm from the equilibrium position and released from rest when $t = 0$. If the support moves with an impressed displacement of $\delta = (10 \sin 4t)$ mm, where t is in seconds, determine the equation that describes the vertical motion of the block. Assume positive displacement is downward.

SOLUTION

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{600}{4}} = 12.25$$

The general solution is defined by Eq. 22–23 with $k\delta_0$ substituted for F_0.

$$y = A \sin \omega_n t + B \cos \omega_n t + \left(\frac{\delta_0}{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]} \right) \sin \omega t$$

$\delta = (0.01 \sin 4t)$m, hence $\delta_0 = 0.01, \omega = 4$, so that

$y = A \sin 12.25t + B \cos 12.25t + 0.0112 \sin 4t$

$y = 0.05$ when $t = 0$

$0.05 = 0 + B + 0; \qquad B = 0.05$ m

$\dot{y} = A(12.25) \cos 12.25t - B(12.25) \sin 12.25t + 0.0112(4) \cos 4t$

$v = y = 0$ when $t = 0$

$0 = A(12.25) - 0 + 0.0112(4); \qquad A = -0.00366$ m

Expressing the result in mm, we have

$y = (-3.66 \sin 12.25t + 50 \cos 12.25t + 11.2 \sin 4t)$ mm **Ans.**

Ans:
$y = (-3.66 \sin 12.25t + 50 \cos 12.25t + 11.2 \sin 4t)$ mm

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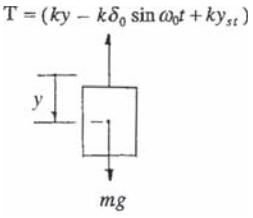
22–45.

Use a block-and-spring model like that shown in Fig. 22–13*a*, but suspended from a vertical position and subjected to a periodic support displacement $\delta = \delta_0 \sin \omega_0 t$, determine the equation of motion for the system, and obtain its general solution. Define the displacement y measured from the static equilibrium position of the block when $t = 0$.

SOLUTION

$$+\uparrow \Sigma F_x = ma_x; \qquad k(y - \delta_0 \sin \omega_0 t + y_{st}) - mg = -m\ddot{y}$$

$$m\ddot{y} + ky + ky_{st} - mg = k\delta_0 \sin \omega_0 t$$



However, from equilibrium

$$ky_{st} - mg = 0, \text{ therefore}$$

$$\ddot{y} + \frac{k}{m} y = \frac{k\delta_0}{m} \sin \omega t \qquad \text{where } \omega_n = \sqrt{\frac{k}{m}}$$

$$\ddot{y} + \omega_n^2 y = \frac{k\delta_0}{m} \sin \omega t \qquad \textbf{Ans. (1)}$$

The general solution of the above differential equation is of the form of $y = y_c + y_p$, where

$$y_c = A \sin \omega_n t + B \cos \omega_n t$$

$$y_p = C \sin \omega_0 t \qquad \textbf{(2)}$$

$$\ddot{y}_p = -C\omega_0^2 \sin \omega_0 t \qquad \textbf{(3)}$$

Substitute Eqs. (2) and (3) into (1) yields:

$$-C\omega^2 \sin \omega_0 t + \omega_n^2 (C \sin \omega_0 t) = \frac{k\delta_0}{m} \sin \omega_0 t$$

$$C = \frac{\frac{k\delta_0}{m}}{\omega_n^2 - \omega_0^2} = \frac{\delta_0}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2}$$

The general solution is therefore

$$y = A \sin \omega_n t + B \cos \omega_n t + \frac{\delta_0}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2} \sin \omega t \qquad \textbf{Ans.}$$

The constants A and B can be found from the initial conditions.

Ans:

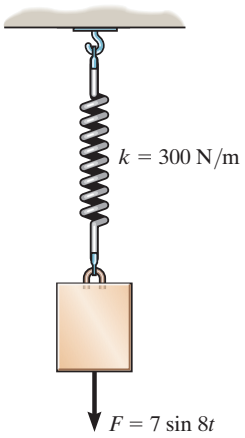
$$y = A \sin \omega_n t + B \cos \omega_n t + \frac{\delta_0}{1 - (\omega/\omega_n)^2} \sin \omega t$$

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22–46.

A 5-kg block is suspended from a spring having a stiffness of 300 N/m. If the block is acted upon by a vertical force $F = (7 \sin 8t)$ N, where t is in seconds, determine the equation which describes the motion of the block when it is pulled down 100 mm from the equilibrium position and released from rest at $t = 0$. Assume that positive displacement is downward.



SOLUTION

The general solution is defined by:

$$y = A \sin \omega_n t + B \cos \omega_n t + \left(\frac{\frac{F_0}{k}}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2} \right) \sin \omega_0 t$$

Since

$F = 7 \sin 8t, \quad F_0 = 7 \text{ N}, \quad \omega_0 = 8 \text{ rad/s}, \quad k = 300 \text{ N/m}$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{300}{5}} = 7.746 \text{ rad/s}$$

Thus,

$$y = A \sin 7.746t + B \cos 7.746t + \left(\frac{\frac{7}{300}}{1 - \left(\frac{8}{7.746}\right)^2} \right) \sin 8t$$

$y = 0.1$ m when $t = 0$,

$0.1 = 0 + B - 0; \quad B = 0.1$ m

$\dot{y} = A(7.746) \cos 7.746t - B(7.746) \sin 7.746t - (0.35)(8) \cos 8t$

$y = \dot{y} = 0$ when $t = 0$,

$\dot{y} = A(7.746) - 2.8 = 0; \quad A = 0.361$

Expressing the results in mm, we have

$y = (361 \sin 7.75t + 100 \cos 7.75t - 350 \sin 8t)$ mm **Ans.**

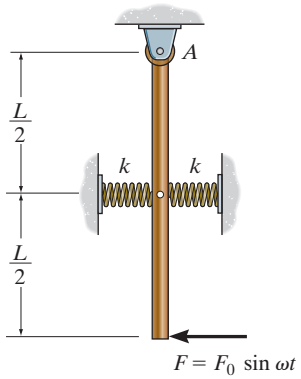
Ans:
$y = (361 \sin 7.75t + 100 \cos 7.75t - 350 \sin 8t)$ mm

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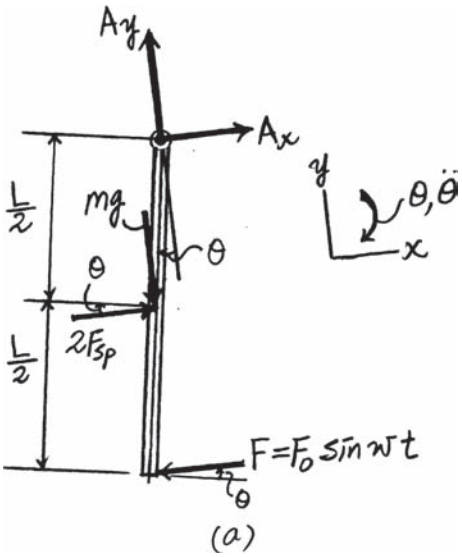
22–47.

The uniform rod has a mass of m. If it is acted upon by a periodic force of $F = F_0 \sin \omega t$, determine the amplitude of the steady-state vibration.



SOLUTION

Equation of Motion: When the rod rotates through a small angle θ, the springs compress and stretch $s = r_{AG}\theta = \frac{L}{2}\theta$. Thus, the force in each spring is $F_{sp} = ks = \frac{kL}{2}\theta$. The mass moment of inertia of the rod about point A is $I_A = \frac{1}{3}mL^2$. Referring to the free-body diagram of the rod shown in Fig. a,



$$+\Sigma M_A = I_A\alpha; \qquad F_O \sin \omega t \cos\theta(L) - mg \sin\theta\left(\frac{L}{2}\right) - 2\left(\frac{kL}{2}\theta\right)\cos\theta\left(\frac{L}{2}\right) = \frac{1}{3}mL^2\ddot{\theta}$$

Since θ is small, $\sin\theta \cong 0$ and $\cos\theta \cong 1$. Thus, this equation becomes

$$\frac{1}{3}mL\ddot{\theta} + \frac{1}{2}(mg + kL)\theta = F_O \sin \omega t$$

$$\ddot{\theta} + \frac{3}{2}\left(\frac{g}{L} + \frac{k}{m}\right)\theta = \frac{3F_O}{mL}\sin \omega t \qquad \textbf{(1)}$$

The particular solution of this differential equation is assumed to be in the form of

$$\theta_p = C \sin \omega t \qquad \textbf{(2)}$$

Taking the time derivative of Eq. (2) twice,

$$\ddot{\theta}_p = -C\omega^2 \sin \omega t \qquad \textbf{(3)}$$

Substituting Eqs. (2) and (3) into Eq. (1),

$$-C\omega^2 \sin \omega t + \frac{3}{2}\left(\frac{g}{L} + \frac{k}{m}\right)(C \sin \omega t) = \frac{3F_O}{mL}\sin \omega t$$

$$C\left[\frac{3}{2}\left(\frac{g}{L} + \frac{k}{m}\right) - \omega^2\right]\sin \omega t = \frac{3F_O}{mL}\sin \omega t$$

$$C = \frac{3F_O/mL}{\frac{3}{2}\left(\frac{g}{L} + \frac{k}{m}\right) - \omega^2}$$

$$C = \frac{3F_O}{\frac{3}{2}(mg + Lk) - mL\omega^2} \qquad \textbf{Ans.}$$

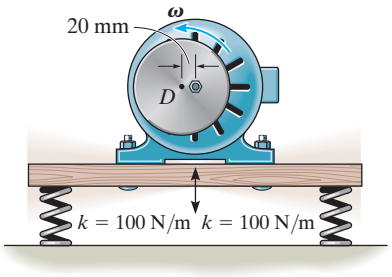
Ans:
$$C = \frac{3F_O}{\frac{3}{2}(mg + Lk) - mL\omega^2}$$

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***22–48.**

The electric motor has a mass of 50 kg and is supported by *four springs,* each spring having a stiffness of 100 N/m. If the motor turns a disk D which is mounted eccentrically, 20 mm from the disk's center, determine the angular velocity ω at which resonance occurs. Assume that the motor only vibrates in the vertical direction.



SOLUTION

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{4(100)}{50}} = 2.83 \text{ rad/s}$$

$$\omega_n = \omega = 2.83 \text{ rad/s} \qquad \textbf{Ans.}$$

Ans:

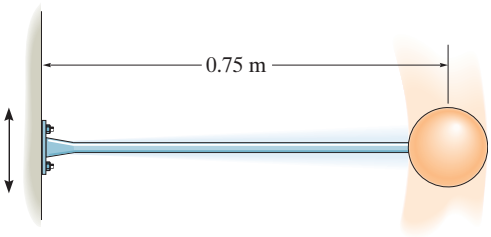
$\omega_n = \omega = 2.83$ rad/s

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22–49.

The light elastic rod supports a 4-kg sphere. When an 18-N vertical force is applied to the sphere, the rod deflects 14 mm. If the wall oscillates with harmonic frequency of 2 Hz and has an amplitude of 15 mm, determine the amplitude of vibration for the sphere.



SOLUTION

$$k = \frac{F}{\Delta y} = \frac{18}{0.014} = 1285.71 \text{ N/m}$$

$$\omega_0 = 2 \text{ Hz} = 2(2\pi) = 12.57 \text{ rad/s}$$

$$\delta_0 = 0.015 \text{ m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{1285.71}{4}} = 17.93$$

Using Eq. 22–22, the amplitude is

$$(x_p)_{max} = \left| \frac{\delta_0}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2} \right| = \left| \frac{0.015}{1 - \left(\frac{12.57}{17.93}\right)^2} \right|$$

$$(x_p)_{max} = 0.0295 \text{ m} = 29.5 \text{ mm} \qquad \textbf{Ans.}$$

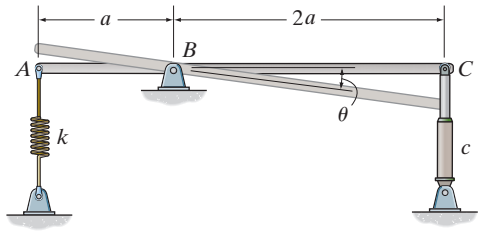
Ans:
$(x_p)_{\max} = 29.5$ mm

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22–50.

Find the differential equation for small oscillations in terms of θ for the uniform rod of mass m. Also show that if $c < \sqrt{mk}/2$, then the system remains underdamped. The rod is in a horizontal position when it is in equilibrium.



SOLUTION

Equation of Motion: When the rod is in equilibrium, $\theta = 0°$, $F_c = c\dot{y}_c = 0$ and $\ddot{\theta} = 0$. writing the moment equation of motion about point B by referring to the free-body diagram of the rod, Fig. a,

$$+\Sigma M_B = 0; \qquad -F_A\,(a) - mg\left(\frac{a}{2}\right) = 0 \qquad F_A = \frac{mg}{2}$$

Thus, the initial stretch of the spring is $s_O = \dfrac{F_A}{k} = \dfrac{mg}{2k}$. When the rod rotates about point B through a small angle θ, the spring stretches further by $s_1 = a\theta$. Thus, the force in the spring is $F_A = k(s_0 + s_1) = k\left(\dfrac{mg}{2k} + a\theta\right)$. Also, the velocity of end C of the rod is $v_c = \dot{y}_c = 2a\dot{\theta}$. Thus, $F_c = c\dot{y}_c = c(2a\dot{\theta})$. The mass moment of inertia of the rod about B is $I_B = \dfrac{1}{12}m(3a)^2 + m\left(\dfrac{a}{2}\right)^2 = ma^2$. Again, referring to Fig. a and writing the moment equation of motion about B,

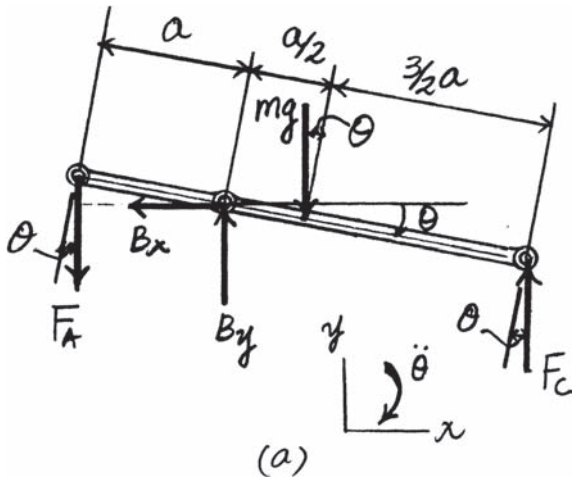
$$\Sigma M_B = I_B\,\alpha; \qquad k\left(\frac{mg}{2k} + a\theta\right)\cos\theta(a) + \left(2a\dot{\theta}\right)\cos\theta(2a) - mg\cos\theta\left(\frac{a}{2}\right)$$

$$= -ma^2\ddot{\theta}$$

$$\ddot{\theta} + \frac{4c}{m}\cos\theta\dot{\theta} + \frac{k}{m}(\cos\theta)\theta = 0$$

Since θ is small, $\cos\theta \cong 1$. Thus, this equation becomes

$$\ddot{\theta} + \frac{4c}{m}\dot{\theta} + \frac{k}{m}\theta = 0 \qquad \textbf{Ans.}$$



Comparing this equation to that of the standard form,

$$\omega_n = \sqrt{\frac{k}{m}} \qquad c_{eq} = 4c$$

Thus,

$$c_c = 2m\omega_n = 2m\sqrt{\frac{k}{m}} = 2\sqrt{mk}$$

For the system to be underdamped,

$$c_{eq} < c_c$$

$$4c < 2\sqrt{mk}$$

$$c < \frac{1}{2}\sqrt{mk} \qquad \textbf{Ans.}$$

Ans:

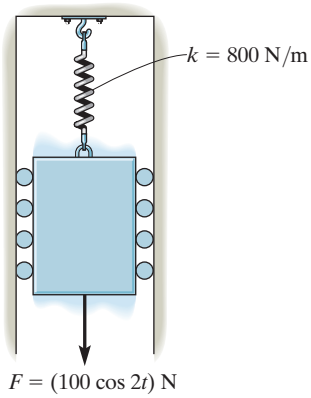
$$\ddot{\theta} + \frac{4c}{m}\dot{\theta} + \frac{k}{m}\dot{\theta} = 0$$

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22–51.

The 40-kg block is attached to a spring having a stiffness of 800 N/m. A force $F = (100 \cos 2t)$ N, where t is in seconds is applied to the block. Determine the maximum speed of the block for the steady-state vibration.



SOLUTION

For the steady-state vibration, the displacement is

$$y_p = \frac{F_0/k}{1 - (\omega_0/\omega_n)^2} \cos \omega t$$

Here $F_0 = 100$ N, $k = 800$ N/m, $\omega_0 = 2$ rad/s and

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{800}{40}} = \sqrt{20} \text{ rad/s}.$$

Thus

$$y_p = \frac{100/800}{1 - (2/\sqrt{20})^2} \cos 2t$$

$$y_P = 0.15625 \cos 2t$$

Taking the time derivative of this equation

$$v_p = \dot{y}_p = -0.3125 \sin 2t \qquad \textbf{(2)}$$

v_p is maximum when $\sin 2t = 1$. Thus

$$(v_p)_{\max} = 0.3125 \text{ m/s} \qquad \textbf{Ans.}$$

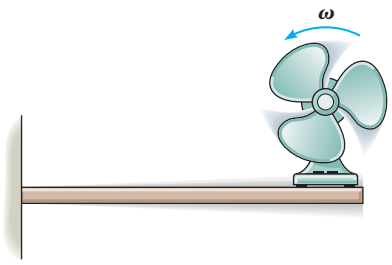
Ans:
$(v_p)_{\max} = 0.3125$ m/s

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***22–52.**

The fan has a mass of 25 kg and is fixed to the end of a horizontal beam that has a negligible mass. The fan blade is mounted eccentrically on the shaft such that it is equivalent to an unbalanced 3.5-kg mass located 100 mm from the axis of rotation. If the static deflection of the beam is 50 mm as a result of the weight of the fan, determine the angular velocity of the fan blade at which resonance will occur. *Hint:* See the first part of Example 22.8.



SOLUTION

$$k = \frac{F}{\Delta y} = \frac{25(9.81)}{0.05} = 4905 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{4905}{25}} = 14.01 \text{ rad/s}$$

Resonance occurs when

$$\omega = \omega_n = 14.0 \text{ rad/s} \qquad \textbf{Ans.}$$

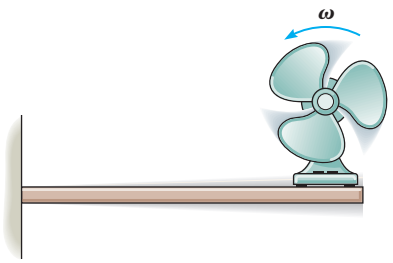
Ans:
$\omega = 14.0$ rad/s

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22–53.

In Prob. 22–52, determine the amplitude of steady-state vibration of the fan if its angular velocity is 10 rad/s.



SOLUTION

$$k = \frac{F}{\Delta y} = \frac{25(9.81)}{0.05} = 4905 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{4905}{25}} = 14.01 \text{ rad/s}$$

The force caused by the unbalanced rotor is

$$F_0 = mr\,\omega^2 = 3.5(0.1)(10)^2 = 35 \text{ N}$$

Using Eq. 22–22, the amplitude is

$$(x_p)_{\max} = \left| \frac{\frac{F_0}{k}}{1 - \left(\frac{\omega}{p}\right)^2} \right|$$

$$(x_p)_{\max} = \left| \frac{\frac{35}{4905}}{1 - \left(\frac{10}{14.01}\right)^2} \right| = 0.0146 \text{ m}$$

$$(x_p)_{\max} = 14.6 \text{ mm} \qquad \textbf{Ans.}$$

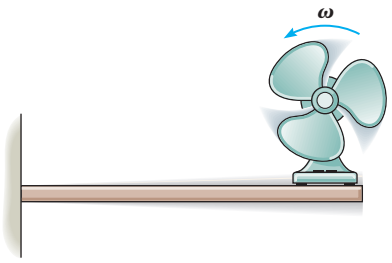
Ans:
$(x_p)_{\max} = 14.6$ mm

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22–54.

What will be the amplitude of steady-state vibration of the fan in Prob. 22–52 if the angular velocity of the fan blade is 18 rad/s? *Hint:* See the first part of Example 22.8.



SOLUTION

$$k = \frac{F}{\Delta y} = \frac{25(9.81)}{0.05} = 4905 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{4905}{25}} = 14.01 \text{ rad/s}$$

The force caused by the unbalanced rotor is

$$F_0 = mr\omega^2 = 3.5(0.1)(18)^2 = 113.4 \text{ N}$$

Using Eq. 22–22, the amplitude is

$$(x_p)_{\max} = \left| \frac{\frac{F_0}{k}}{1 - \left(\frac{\omega}{p}\right)^2} \right|$$

$$(x_p)_{\max} = \left| \frac{\frac{113.4}{4905}}{1 - \left(\frac{18}{14.01}\right)^2} \right| = 0.0355 \text{ m}$$

$$(x_p)_{\max} = 35.5 \text{ mm} \qquad \textbf{Ans.}$$

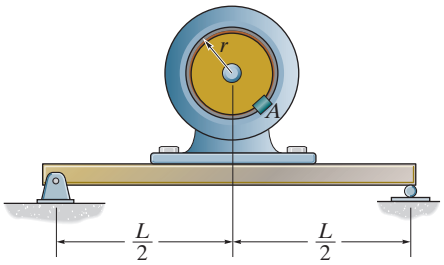
Ans:
$(x_p)_{\max} = 35.5$ mm

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22–55.

The motor of mass M is supported by a simply supported beam of negligible mass. If block A of mass m is clipped onto the rotor, which is turning at constant angular velocity of ω, determine the amplitude of the steady-state vibration. *Hint:* When the beam is subjected to a concentrated force of P at its mid-span, it deflects $\delta = PL^3/48EI$ at this point. Here E is Young's modulus of elasticity, a property of the material, and I is the moment of inertia of the beam's cross-sectional area.



SOLUTION

In this case, $P = k_{eq}\delta$. Then, $k_{eq} = \dfrac{P}{\delta} = \dfrac{P}{PL^3/48EI} = \dfrac{48EI}{L^3}$. Thus, the natural frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} = \sqrt{\frac{\frac{48EI}{L^3}}{M}} = \sqrt{\frac{48EI}{ML^3}}$$

Here, $F_O = ma_n = m(\omega^2 r)$. Thus,

$$Y = \frac{F_O/k_{eq}}{1 - \left(\dfrac{\omega}{\omega_n}\right)^2}$$

$$Y = \frac{\dfrac{m(\omega^2 r)}{48EI/L^3}}{1 - \dfrac{\omega^2}{48EI/ML^3}}$$

$$Y = \frac{mr\omega^2 L^3}{48EI - M\omega^2 L^3} \qquad \textbf{Ans.}$$

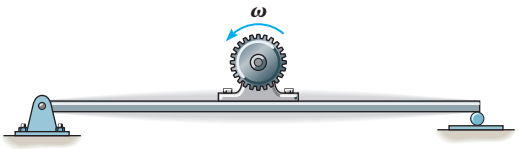
Ans:

$$Y = \frac{mr\omega^2 L^3}{48EI - M\omega^2 L^3}$$

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***22–56.** The electric motor turns an eccentric flywheel which is equivalent to an unbalanced 0.125-kg mass located 250 mm from the axis of rotation. If the static deflection of the beam is 25 mm because of the mass of the motor, determine the angular velocity of the flywheel at which resonance will occur. The motor has a mass of 75 kg. Neglect the mass of the beam.



SOLUTION

$$k = \frac{F}{\delta} = \frac{75(9.81)}{25} = 29.43 \text{ N/mm} \qquad \omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{29.43(10^3)}{75}} = 19.81$$

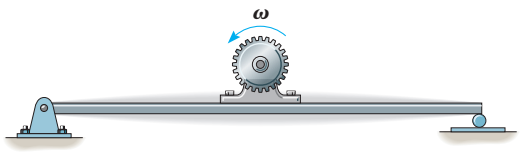
Resonance occurs when $\omega_0 = \omega_n = 19.81$ rad/s **Ans.**

Ans:
$\omega_0 = \omega_n = 19.81$ rad/s

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22–57. What will be the amplitude of steady-state vibration of the motor in Prob. 22–56 if the angular velocity of the flywheel is 20 rad / s?



SOLUTION

The constant value F_O of the periodic force is due to the centrifugal force of the unbalanced mass.

$$F_O = ma_n = mr\omega^2 = (0.125)\left(\frac{250}{1000}\right)(20)^2 = 12.5 \text{ N}$$

Hence $F = 12.5 \sin 20t$

$$k = \frac{F}{\delta} = \frac{75(9.81)}{25} = 29.43 \text{ N/mm} \qquad \omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{29.43(10^3)}{75}} = 19.809$$

From Eq. 22–21, the amplitude of the steady state motion is

$$C = \left|\frac{F_0/k}{1-\left(\frac{\omega_0}{\omega_n}\right)^2}\right| = \left|\frac{12.5/29.43}{1-\left(\frac{20}{19.809}\right)^2}\right| = 21.92 \text{ mm}$$ **Ans.**

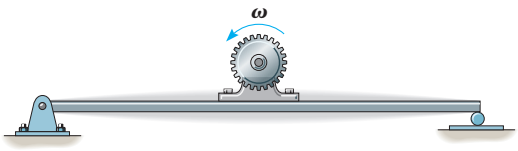
Ans:

$C = 21.92$ mm

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22–58. Determine the angular velocity of the flywheel in Prob. 22–56 which will produce an amplitude of vibration of 6 mm.



SOLUTION

The constant value F_O of the periodic force is due to the centrifugal force of the unbalanced mass.

$$F_O = ma_n = mr\omega^2 = (0.125)\left(\frac{250}{1000}\right)\omega^2 = 0.03125\omega^2$$

$$F = 0.03125\omega^2 \sin \omega t$$

$$k = \frac{F}{\delta} = \frac{75(9.81)}{25} = 29.43 \text{ N/mm} \qquad \omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{29.43(10^3)}{75}} = 19.809$$

From Eq. 22.21, the amplitude of the steady-state motion is

$$C = \left| \frac{F_0/k}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2} \right|$$

$$6 = \left| \frac{0.03125\left(\frac{\omega^2}{29.43}\right)}{1 - \left(\frac{\omega}{19.809}\right)^2} \right|$$

$\omega = 19.2$ rad/s **Ans.**

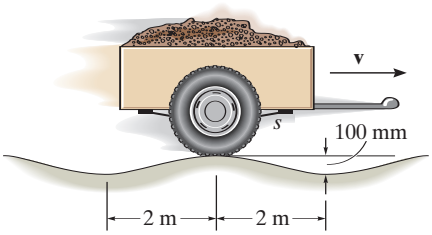
Ans:
$\omega = 19.2$ rad/s

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22–59.

The 450-kg trailer is pulled with a constant speed over the surface of a bumpy road, which may be approximated by a cosine curve having an amplitude of 50 mm and wave length of 4 m. If the two springs s which support the trailer each have a stiffness of 800 N/m, determine the speed v which will cause the greatest vibration (resonance) of the trailer. Neglect the weight of the wheels.



SOLUTION

The amplitude is $\delta_0 = 50 \text{ mm} = 0.05 \text{ m}$

The wave length is $\lambda = 4 \text{ m}$

$$k = 2(800) = 1600 \text{ N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{1600}{450}} = 1.89 \text{ rad/s}$$

$$\tau = \frac{2\pi}{\omega_n} = \frac{2\pi}{1.89} = 3.33 \text{ s}$$

For maximum vibration of the trailer, resonance must occur, i.e.,

$$\omega_0 = \omega_n$$

Thus, the trailer must travel $\lambda = 4 \text{ m}$, in $\tau = 3.33 \text{ s}$, so that

$$v_R = \frac{\lambda}{\tau} = \frac{4}{3.33} = 1.20 \text{ m.s} \qquad \textbf{Ans.}$$

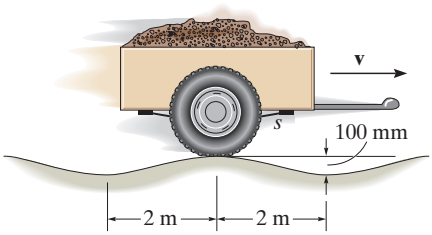
Ans:
$v_R = 1.20 \text{ m.s}$

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***22–60.**

Determine the amplitude of vibration of the trailer in Prob. 22–60 if the speed $v = 15$ km/h.



SOLUTION

$$v = 15 \text{ km/h} = \frac{15(1000)}{3600} \text{ m/s} = 4.17 \text{ m/s}$$

$$\delta_0 = 0.05 \text{ m}$$

As shown in Prob. 22–50, the velocity is inversely proportional to the period. Since $\frac{1}{\tau} = f$ the the velocity is proportional of f, ω_n and ω_0

Hence, the amplitude of motion is

$$(x_p)_{max} = \left| \frac{\delta_0}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2} \right| = \left| \frac{\delta_0}{1 - \left(\frac{v}{v_R}\right)^2} \right|$$

$$(x_p)_{max} = \left| \frac{0.05}{1 - \left(\frac{4.17}{1.20}\right)^2} \right| = 0.00453 \text{ m}$$

$$(x_p)_{max} = 4.53 \text{ mm} \qquad \textbf{Ans.}$$

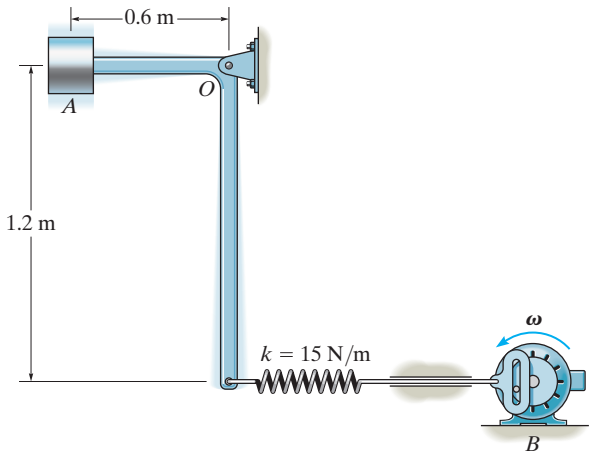
Ans:
$(x_p)_{\max} = 4.53$ mm

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22–61.

The small block at A has a mass of 4 kg and is mounted on the bent rod having negligible mass. If the rotor at B causes a harmonic movement $\delta_B = (0.1 \cos 15t)$ m, where t is in seconds, determine the steady-state amplitude of vibration of the block.



SOLUTION

$$+\Sigma M_O = I_O\,\alpha; \qquad 4(9.81)(0.6) - F_s(1.2) = 4(0.6)^2\ddot{\theta}$$

$$F_s = kx = 15(x + x_{st} - 0.1\cos 15t)$$

$$x_{st} = \frac{4(9.81)(0.6)}{1.2(15)}$$

Thus,

$$-15(x - 0.1\cos 15t)(1.2) = 4(0.6)^2\ddot{\theta}$$

$$x = 1.2\theta$$

$$\theta + 15\theta = 1.25\cos 15t$$

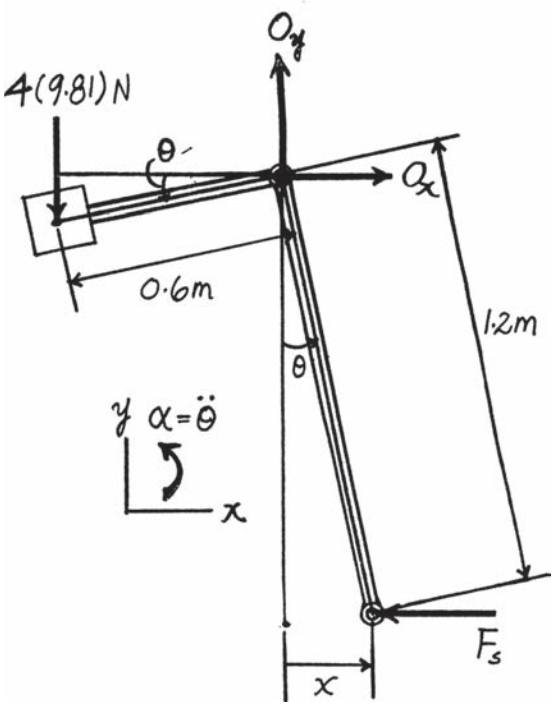
Set $x_p = C\cos 15t$

$$-C(15)^2\cos 15t + 15(C\cos 15t) = 1.25\cos 15t$$

$$C = \frac{1.25}{15 - (15)^2} = -0.00595 \text{ m}$$

$$\theta_{\max} = C = 0.00595 \text{ rad}$$

$$y_{\max} = (0.6 \text{ m})(0.00595 \text{ rad}) = 0.00357 \text{ rad}$$ **Ans.**



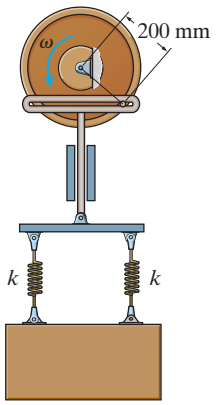
Ans:
$y_{\max} = 0.00357$ rad

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22–62.

The spring system is connected to a crosshead that oscillates vertically when the wheel rotates with a constant angular velocity of $\boldsymbol{\omega}$. If the amplitude of the steady-state vibration is observed to be 400 mm, and the springs each have a stiffness of $k = 2500$ N/m, determine the two possible values of $\boldsymbol{\omega}$ at which the wheel must rotate. The block has a mass of 50 kg.



SOLUTION

In this case, $k_{eq} = 2k = 2(2500) = 5000$ N/m Thus, the natural circular frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} = \sqrt{\frac{5000}{50}} = 10 \text{ rad/s}$$

Here, $\delta_O = 0.2$ m and $(Y_P)_{\max} = \pm 0.4$ m, so that

$$(Y_P)_{\max} = \frac{\delta_O}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

$$\pm 0.4 = \frac{0.2}{1 - \left(\frac{\omega}{10}\right)^2}$$

$$\frac{\omega^2}{100} = 1 \pm 0.5$$

Thus,

$$\frac{\omega^2}{100} = 1.5 \qquad \omega = 12.2 \text{ rad/s} \qquad \textbf{Ans.}$$

or

$$\frac{\omega^2}{100} = 0.5 \qquad \omega = 7.07 \text{ rad/s} \qquad \textbf{Ans.}$$

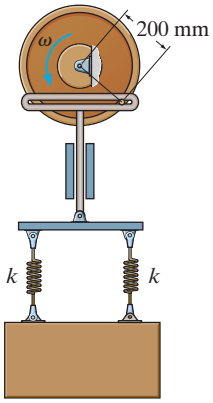
Ans:
$\omega = 12.2$ rad/s
$\omega = 7.07$ rad/s

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22–63.

The spring system is connected to a crosshead that oscillates vertically when the wheel rotates with a constant angular velocity of $\omega = 5$ rad/s. If the amplitude of the steady-state vibration is observed to be 400 mm, determine the two possible values of the stiffness k of the springs. The block has a mass of 50 kg.



SOLUTION

In this case, $k_{eq} = 2k$ Thus, the natural circular frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} = \sqrt{\frac{2k}{50}} = \sqrt{0.04k}$$

Here, $\delta_O = 0.2$ m and $(Y_P)_{\max} = \pm 0.4$ m, so that

$$(Y_P)_{\max} = \frac{\delta_O}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

$$\pm 0.4 = \frac{0.2}{1 - \left(\frac{5}{\sqrt{0.04k}}\right)^2}$$

$$\frac{625}{k} = 1 \pm 0.5$$

Thus,

$$\frac{625}{k} = 1.5 \qquad k = 417 \text{ N/m} \qquad \textbf{Ans.}$$

or

$$\frac{625}{k} = 0.5 \qquad k = 1250 \text{ N/m} \qquad \textbf{Ans.}$$

Ans:
$k = 417$ N/m
$k = 1250$ N/m

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***22–64.** A 3.5-kg block is suspended from a spring having a stiffness of $k = 1250$ N/m. The support to which the spring is attached is given simple harmonic motion which can be expressed as $\delta = (0.045 \sin 2t)$ m, where t is in seconds. If the damping factor is $c/c_c = 0.8$, determine the phase angle ϕ of forced vibration.

SOLUTION

$$p = \sqrt{\frac{k}{m}} = \sqrt{\frac{1250}{3.5}} = 18.898$$

$$\delta = 0.045 \sin 2t$$

$$\delta_0 = 0.045, \omega = 2$$

$$\phi' = \tan^{-1}\left(\frac{2\left(\frac{c}{c_c}\right)\left(\frac{\omega}{p}\right)}{1 - \left(\frac{\omega}{p}\right)^2}\right) = \tan^{-1}\left(\frac{2(0.8)\left(\frac{2}{18.898}\right)}{1 - \left(\frac{2}{18.898}\right)^2}\right)$$

$$\phi' = 9.72^\circ \qquad \textbf{Ans.}$$

Ans:
$\phi' = 9.72^\circ$

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22–65. Determine the magnification factor of the block, spring, and dashpot combination in Prob. 22–64.

SOLUTION

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{75}{\left(\frac{7}{32.2}\right)}} = 18.57$$

$$\delta = 0.15 \sin 2t$$

$$\delta_0 = 0.15, \quad \omega = 2$$

$$\text{MF} = \frac{1}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\left(\frac{c}{c_n}\right)\left(\frac{\omega}{\omega_n}\right)\right]^2}} = \frac{1}{\sqrt{\left[1 - \left(\frac{2}{18.57}\right)^2\right]^2 + \left[2(0.8)\left(\frac{2}{18.57}\right)\right]^2}}$$

$$\text{MF} = 0.997 \qquad \textbf{Ans.}$$

Ans:
MF = 0.997

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22–66.

A block having a mass of 7 kg is suspended from a spring that has a stiffness $k = 600\ \text{N/m}$. If the block is given an upward velocity of 0.6 m/s from its equilibrium position at $t = 0$, determine its position as a function of time. Assume that positive displacement of the block is downward and that motion takes place in a medium which furnishes a damping force $F = (50|v|)\ \text{N}$, where v is in m/s.

SOLUTION

$$c = 50\ \text{N s/m} \quad k = 600\ \text{N/m} \quad m = 7\ \text{kg}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{600}{7}} = 9.258\ \text{rad/s}$$

$$c_c = 2m\omega_n = 2(7)(9.258) = 129.6\ \text{N}\cdot\text{s/m}$$

Since $c < c_z$, the system is underdamped,

$$\omega_d = \omega_n\sqrt{1 - \left(\frac{c}{c_c}\right)^2} = 9.258\sqrt{1 - \left(\frac{50}{129.6}\right)^2} = 8.542\ \text{rad/s}$$

$$\frac{c}{2m} = \frac{50}{2(7)} = 3.751$$

From Eq. 22-32

$$y = D\left[e^{-\left(\frac{c}{2m}\right)t}\sin(\omega_d t + \phi)\right]$$

$$v = \dot{y} = D\left[e^{-\left(\frac{c}{2m}\right)t}\omega_d\cos(\omega_d t + \phi) + \left(-\frac{c}{2m}\right)e^{-\left(\frac{c}{2m}\right)t}\sin(\omega_d t + \phi)\right]$$

$$v = De^{-\left(\frac{c}{2m}\right)t}\left[\omega_d\cos(\omega_d t + \phi) - \frac{c}{2m}\sin(\omega_d t + \phi)\right]$$

Applying the initial condition at $t = 0$, $y = 0$ and $v = -0.6$ m/s.

$$0 = D[e^{-0}\sin(0 + \phi)] \qquad \text{since} \quad D \neq 0$$

$$\sin\phi = 0 \qquad \phi = 0°$$

$$-0.6 = De^{-0}[8.542\cos 0° - 0]$$

$$D = -0.0702\ \text{m}$$

$$y = [-0.0702\,e^{-3.57t}\sin(8.540)]\ \text{m} \qquad \textbf{Ans.}$$

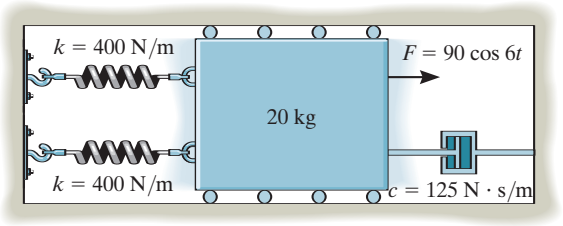
Ans:
$y = \{-0.0702\,e^{-3.57t}\sin(8.540)\}\ \text{m}$

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22–67.

The 20-kg block is subjected to the action of the harmonic force $F = (90 \cos 6t)$ N, where t is in seconds. Write the equation which describes the steady-state motion.



SOLUTION

$$F = 90 \cos 6t$$

$$F_0 = 90 \text{ N}, \qquad \omega_0 = 6 \text{ rad/s}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{800}{20}} = 6.32 \text{ rad/s}$$

From Eq. 22–29,

$$c_c = 2m\omega_n = 2(20)(6.32) = 253.0$$

Using Eqs. 22–39,

$$x = C' \cos(\omega_0 t - \phi)$$

$$C' = \frac{\frac{F_n}{k}}{\sqrt{\left[1 - \left(\frac{\omega_0}{\omega_r}\right)^2\right]^2 + \left[2\left(\frac{C}{C_r}\right)\left(\frac{\omega_0}{\omega_n}\right)\right]^2}}$$

$$= \frac{\frac{90}{800}}{\sqrt{[1 - (\frac{6}{6.32})^2]^2 + [2(\frac{125}{253.0})(\frac{6}{6.32})]^2}}$$

$$= 0.119$$

$$\phi = \tan^{-1}\left[\frac{\frac{c\omega_0}{k}}{1 - \left(\frac{\omega_0}{\omega_n}\right)^2}\right]$$

$$= \tan^{-1}\left[\frac{\frac{125(6)}{800}}{1 - (\frac{6}{6.32})^2}\right]$$

$$\phi = 83.9°$$

Thus,

$$x = 0.119 \cos(6t - 83.9°) \text{ m} \qquad \textbf{Ans.}$$

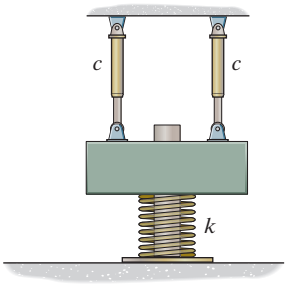
Ans:
$x = 0.119 \cos(6t - 83.9°)$ m

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***22–68.**

Two identical dashpots are arranged parallel to each other, as shown. Show that if the damping coefficient $c < \sqrt{mk}$, then the block of mass m will vibrate as an underdamped system.



SOLUTION

When the two dash pots are arranged in parallel, the piston of the dashpots have the same velocity. Thus, the force produced is

$$F = c\dot{y} + c\dot{y} = 2c\dot{y}$$

The equivalent damping coefficient c_{eq} of a single dashpot is

$$c_{eq} = \frac{F}{\dot{y}} = \frac{2c\dot{y}}{\dot{y}} = 2c$$

For the vibration to occur (underdamped system), $c_{eq} < c_c$. However, $c_c = 2m\omega_n = 2m\sqrt{\frac{k}{m}}$. Thus,

$$c_{eq} < c_c$$

$$2c < 2m\sqrt{\frac{k}{m}}$$

$$c < \sqrt{mk} \qquad \textbf{Ans.}$$

Ans:
$F = 2c\dot{y}$
$c_c = 2m\sqrt{\frac{k}{m}}$
$c < \sqrt{mk}$

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22–69. The damping factor, c/c_c, may be determined experimentally by measuring the successive amplitudes of vibrating motion of a system. If two of these maximum displacements can be approximated by x_1 and x_2, as shown in Fig. 22–16, show that $\ln(x_1/x_2) = 2\pi(c/c_c)/\sqrt{1-(c/c_c)^2}$. The quantity $\ln(x_1/x_2)$ is called the *logarithmic decrement*.

SOLUTION

Using Eq. 22–32,

$$x = D\left[e^{-\left(\frac{c}{2m}\right)t}\sin(\omega_d t + \phi)\right]$$

The maximum displacement is

$$x_{max} = De^{-\left(\frac{c}{2m}\right)t}$$

At $t = t_1$, and $t = t_2$

$$x_1 = De^{-\left(\frac{c}{2m}\right)t_1}$$

$$x_2 = De^{-\left(\frac{c}{2m}\right)t2}$$

Hence,

$$\frac{x_1}{x_2} = \frac{De^{-\left(\frac{c}{2m}\right)1}}{De^{-\left(\frac{c}{2m}\right)t_2} = e^{-\left(\frac{c}{2m}\right)(t_1-t_2)}}$$

Since $\omega_d t_2 - \omega_d t_1 = 2\pi$

then $t_2 - t_1 = \dfrac{2\pi}{\omega_d}$

so that $\ln\left(\dfrac{x_1}{x_2}\right) = \dfrac{c\pi}{m\omega_d}$

Using Eq. 22–33, $c_c = 2m\omega_n$

$$\omega_d = \omega_n\sqrt{1-\left(\frac{c}{c_c}\right)^2} = \frac{c_c}{2m}\sqrt{1-\left(\frac{c}{c_r}\right)^2}$$

So that,

$$\ln\left(\frac{x_1}{x_2}\right) = \frac{2\pi\left(\frac{c}{c_c}\right)}{\sqrt{1-\left(\frac{c}{c_c}\right)^2}}$$

Q.E.D.

Ans:

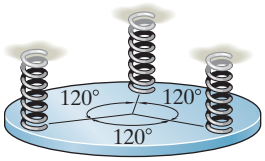
$$\ln\left(\frac{x_1}{x_2}\right) = \frac{2\pi\left(\frac{c}{c_c}\right)}{\sqrt{1-\left(\frac{c}{c_c}\right)^2}}$$

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22–70.

The 4-kg circular disk is attached to three springs, each spring having a stiffness $k = 180\ \text{N/m}$. If the disk is immersed in a fluid and given a downward velocity of 0.3 m/s at the equilibrium position, determine the equation which describes the motion. Consider positive displacement to be measured downward, and that fluid resistance acting on the disk furnishes a damping force having a magnitude $F = (60|v|)\ \text{N}$, where v is the velocity of the block in m/s.



SOLUTION

$$k = 540\ \text{N/m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{540}{4}} = 11.62\ \text{rad/s}$$

$$c_c = 2m\omega_n = 2(4)(11.62) = 92.95$$

$$F = 60v, \text{so that } c = 60$$

Since $c < c_c$, system is underdamped.

$$\omega_d = \omega_n\sqrt{1 - \left(\frac{c}{c_c}\right)^2}$$

$$= 11.62\sqrt{1 - \left(\frac{60}{92.95}\right)^2}$$

$$= 8.87\ \text{rad/s}$$

$$y = A[e^{-(\frac{c}{2m})t}\sin(\omega_d t + \phi)] \qquad \textbf{(1)}$$

$$y = 0, v = 0.3 \text{ at } t = 0$$

$$0 = A\sin\phi$$

$$A \neq 0 \text{ (trivial solution) so that } \phi = 0$$

$$v = y = A[-\frac{c}{2m}e^{-(\frac{c}{2m})t}\sin(\omega_d t + \phi) + e^{-(\frac{c}{2m})t}\cos(\omega_d t + \phi)(\omega_d)]$$

Since $\phi = 0$

$$0.3 = A[0 + 1(8.87)]$$

$$A = 0.0338$$

Substituting into Eq. (1)

$$y = 0.0338[e^{-(\frac{60}{2(4)})t}\sin(8.87)t]$$

Expressing the result in mm

$$y = [33.8e^{-7.5t}\sin(8.87t)]\ \text{mm} \qquad \textbf{Ans.}$$

Ans:

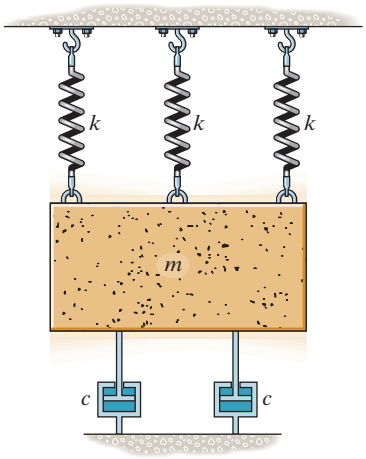
$y = [33.8e^{-7.5t}\sin(8.87t)]$ mm

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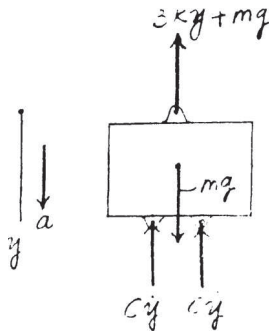
22–71.

Determine the differential equation of motion for the damped vibratory system shown. What type of motion occurs? Take $k = 100$ N/m, $c = 200$ N·s/m, $m = 25$ kg.



SOLUTION

Free-body Diagram: When the block is being displaced by an amount y vertically downward, the *restoring force* is developed by the three springs attached the block.



Equation of Motion:

$$+\uparrow \Sigma F_x = 0; \qquad 3ky + mg + 2c\dot{y} - mg = -m\ddot{y}$$

$$m\ddot{y} + 2c\dot{y} + 3ky = 0 \qquad \textbf{(1)}$$

Here, $m = 25$ kg, $c = 200$ N·s/m and $k = 100$ N/m. Substituting these values into Eq. (1) yields

$$25\ddot{y} + 400\dot{y} + 300y = 0$$

$$\ddot{y} + 16\dot{y} + 12y = 0 \qquad \textbf{Ans.}$$

Comparing the above differential equation with Eq. 22–27, we have $m = 1$kg, $c = 16$ N·s/m and $k = 12$ N/m. Thus, $\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{12}{1}} = 3.464$ rad/s

$$c_c = 2m\omega_n = 2(1)(3.464) = 6.928 \text{ N·s/m}$$

Since $c > c_c$, the system will not vibrate. Therefore it is **overdamped.** **Ans.**

Ans:

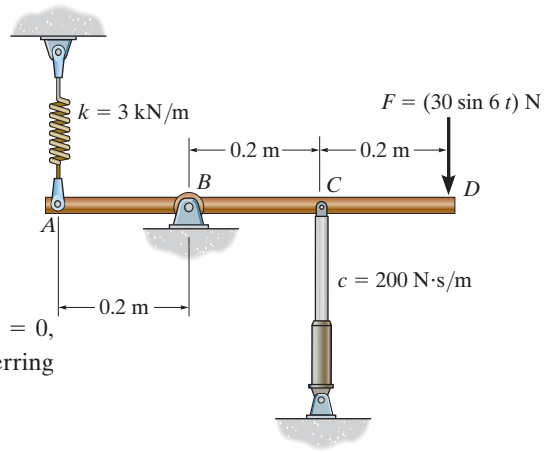
$\ddot{y} + 16\dot{y} + 12y = 0$
Since $c > c_c$, the system will not vibrate.
Therefore it is **overdamped.**

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***22–72.**

If the 12-kg rod is subjected to a periodic force of $F = (30 \sin 6t)$ N, where t is in seconds, determine the steady-state vibration amplitude θ_{max} of the rod about the pin B. Assume θ is small.

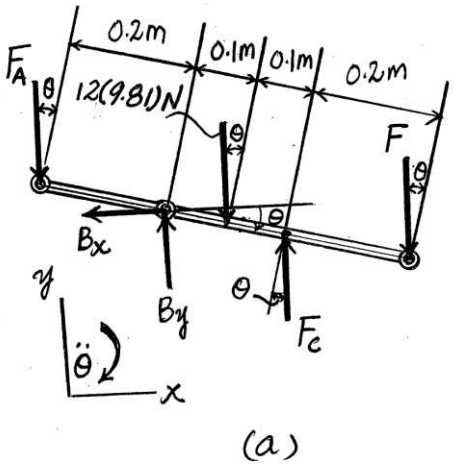


SOLUTION

Equation of Motion: When the rod is in equilibrium, $\theta = 0°$, $F = 0$, $F_c = c\dot{y}_c = 0$, and $\ddot{\theta} = 0$. Writing the moment equation of equilibrium about point B by referring to the free-body diagram of the rod, Fig. a,

$$+\sum M_B = 0; \qquad F_A(0.2) - 12(9.81)(0.1) = 0 \qquad F_A = 58.86 \text{ N}$$

Thus, the initial compression of the spring is $s_O = \frac{F_A}{k} = \frac{58.86}{3000} = 0.01962$ m. When the rod rotates about point B through a small angle θ, the spring compresses further by $s_1 = 0.2\theta$. Thus, the force in the spring is $F_A = k(s_0 + s_1) = 3000(0.01962 + 0.2\theta) = 58.85 + 600\theta$. Also, the velocity of point C on the rod is $v_c = \dot{y}_c = 0.2\dot{\theta}$. Thus, $F_c = c\dot{y}_c = 200(0.2\dot{\theta}) = 40\dot{\theta}$. The mass moment of inertia of the rod about B is $I_B = \frac{1}{12}(12)(0.6)^2 + 12(0.1)^2 = 0.48 \text{ kg} \cdot \text{m}^2$. Again, referring to Fig. a,



(a)

$$+\sum M_B = I_B\alpha; \quad (58.86 + 600\theta)\cos\theta(0.2) + 40\dot{\theta}\cos\theta(0.2) - (30\sin 6t)\cos\theta(0.4) - 12(9.81)\cos\theta(0.1) = -0.48\ddot{\theta}$$

$$\ddot{\theta} + 16.67\cos\theta\dot{\theta} + 250(\cos\theta)\theta = 25\sin 6t\cos\theta$$

Since θ is small, $\cos\theta \cong 1$. Thus, this equation becomes

$$\ddot{\theta} + 16.67\dot{\theta} + 250\theta = 25\sin 6t$$

Comparing this equation to that of the standard form,

$$\frac{k_{eq}}{m} = 250 \qquad k_{eq} = 250(12) = 3000 \text{ N/m}$$

$$\frac{c_{eq}}{m} = 16.667 \qquad c_{eq} = 16.667(12) = 200 \text{ N} \cdot \text{s/m}$$

$$\frac{F_O}{m} = 25 \qquad F_O = 25(12) = 300 \text{ N}$$

$$\omega_n = \sqrt{3000/12} = \sqrt{250}$$

Thus,

$$c_c = 2m\omega_n = 2(12)\sqrt{250} = 379.47 \text{ N} \cdot \text{s/m}$$

Then,

$$\frac{c_{eq}}{c_c} = \frac{200}{379.47} = 0.5270$$

$$\theta_{max} = \frac{300/3000}{\sqrt{\left[1 - \left(\frac{6}{\sqrt{250}}\right)^2\right]^2 + \left[\frac{2(0.5270)(6)}{\sqrt{250}}\right]^2}}$$

$$= 0.106 \text{ rad} \qquad \textbf{Ans.}$$

Ans:

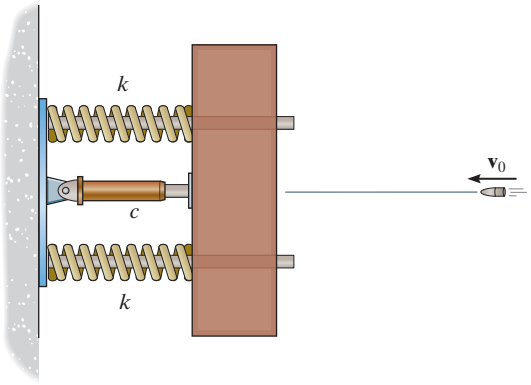
$\theta_{max} = 0.106$ rad

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22–73.

A bullet of mass m has a velocity of $\mathbf{v}_0$ just before it strikes the target of mass M. If the bullet embeds in the target, and the vibration is to be critically damped, determine the dashpot's critical damping coefficient, and the springs' maximum compression. The target is free to move along the two horizontal guides that are "nested" in the springs.



SOLUTION

Since the springs are arranged in parallel, the equivalent stiffness of the single spring system is $k_{eq} = 2k$. Also, when the bullet becomes embedded in the target, $m_T = m + M$. Thus, the natural frequency of the system is

$$\omega_n = \sqrt{\frac{k_{eq}}{m_T}} = \sqrt{\frac{2k}{m + M}}$$

When the system is critically damped

$$c = c_c = 2m_T\omega_n = 2(m + M)\sqrt{\frac{2k}{m + M}} = \sqrt{8(m + M)k} \qquad \textbf{Ans.}$$

The equation that describes the critically dampened system is

$$x = (A + Bt)e^{-\omega_n t}$$

When $t = 0, x = 0$. Thus,

$$A = 0$$

Then,

$$x = Bte^{-\omega_n t} \qquad \textbf{(1)}$$

Taking the time derivative,

$$v = \dot{x} = Be^{-\omega_n t} - B\omega_n te^{-\omega_n t}$$

$$v = Be^{-\omega_n t}(1 - \omega_n t) \qquad \textbf{(2)}$$

Since linear momentum is conserved along the horizontal during the impact, then

$$(\pm) \qquad mv_0 = (m + M)v$$

$$v = \left(\frac{m}{m + M}\right)v_0$$

Here, when $t = 0, v = \left(\frac{m}{m + M}\right)v_0$. Thus, Eq. (2) gives

$$B = \left(\frac{m}{m + M}\right)v_0$$

And Eqs. (1) and (2) become

$$x = \left[\left(\frac{m}{m + M}\right)v_0\right]te^{-\omega_n t} \qquad \textbf{(3)}$$

$$v = \left[\left(\frac{m}{m + M}\right)v_0\right]e^{-\omega_n t}(1 - \omega_n t) \qquad \textbf{(4)}$$

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22–73. Continued

The maximum compression of the spring occurs when the block stops. Thus, Eq. (4) gives

$$0 = \left[\left(\frac{m}{m + M}\right)v_0\right](1 - \omega_n t)$$

Since $\left(\frac{m}{m + M}\right)v_0 \neq 0$, then

$$1 - \omega_n t = 0$$

$$t = \frac{1}{\omega_n} = \sqrt{\frac{m + M}{2k}}$$

Substituting this result into Eq. (3)

$$x_{\max} = \left[\left(\frac{m}{m + M}\right)v_0\right]\left(\sqrt{\frac{m + M}{2k}}\right)e^{-1}$$

$$= \left[\frac{m}{e}\sqrt{\frac{1}{2k(m + M)}}\right]v_0 \qquad \textbf{Ans.}$$

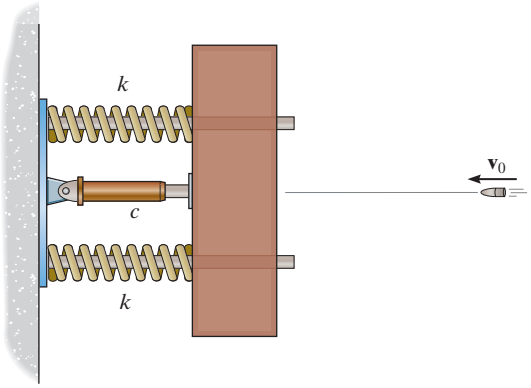
Ans:
$c_c = \sqrt{8(m + M)k}$
$x_{\max} = \left[\frac{m}{e}\sqrt{\frac{1}{2k(m + M)}}\right]v_0$

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22–74.

A bullet of mass m has a velocity $\mathbf{v}_0$ just before it strikes the target of mass M. If the bullet embeds in the target, and the dashpot's damping coefficient is $0 < c \ll c_c$, determine the springs' maximum compression. The target is free to move along the two horizontal guides that are"nested" in the springs.



SOLUTION

Since the springs are arranged in parallel, the equivalent stiffness of the single spring system is $k_{eq} = 2k$. Also, when the bullet becomes embedded in the target, $m_T = m + M$. Thus, the natural circular frequency of the system

$$\omega_n = \sqrt{\frac{k_{eq}}{m_T}} = \sqrt{\frac{2k}{m + M}}$$

The equation that describes the underdamped system is

$$x = Ce^{-(c/2m_T)t} \sin(\omega_d t + \phi) \tag{1}$$

When $t = 0$, $x = 0$. Thus, Eq. (1) gives

$$0 = C \sin \phi$$

Since $C \neq 0$, $\sin \phi = 0$. Then $\phi = 0$. Thus, Eq. (1) becomes

$$x = Ce^{-(c/2m_T)t} \sin \omega_d t \tag{2}$$

Taking the time derivative of Eq. (2),

$$v = \dot{x} = C\left[\omega_d e^{-(c/2m_T)t} \cos \omega_d t - \frac{c}{2m_T} e^{-(c/2m_T)t} \sin \omega_d t\right]$$

$$v = Ce^{-(c/2m_T)t}\left[\omega_d \cos \omega_d t - \frac{c}{2m_T} \sin \omega_d t\right] \tag{3}$$

Since linear momentum is conserved along the horizontal during the impact, then

$(\stackrel{+}{\leftarrow})$ $$mv_0 = (m + M)v$$

$$v = \left(\frac{m}{m + M}\right)v_0$$

When $t = 0$, $v = \left(\frac{m}{m + M}\right)v_0$. Thus, Eq. (3) gives

$$\left(\frac{m}{m + M}\right)v_0 = C\omega_d \qquad C = \left(\frac{m}{m + M}\right)\frac{v_0}{\omega_d}$$

And Eqs. (2) becomes

$$x = \left[\left(\frac{m}{m + M}\right)\frac{v_0}{\omega_d}\right]e^{-(c/2m_T)t} \sin \omega_d t \tag{4}$$

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22–74. Continued

The maximum compression of the spring occurs when

$$\sin \omega_d t = 1$$

$$\omega_d t = \frac{\pi}{2}$$

$$t = \frac{\pi}{2\omega_d}$$

Substituting this result into Eq. (4),

$$x_{\max} = \left[\left(\frac{m}{m + M}\right)\frac{v_0}{\omega_d}\right] e^{-[c/2(m + M)]\left(\frac{\pi}{2\omega_d}\right)}$$

However, $\omega_d = \sqrt{\frac{k_{eq}}{m_T} - \left(\frac{c}{2m_T}\right)^2} = \sqrt{\frac{2k}{m + M} - \frac{c^2}{4(m + M)^2}} = \frac{1}{2(m + M)}\sqrt{8k(m + M) - c^2}$. Substituting this result into Eq. (5),

$$x_{\max} = \frac{2mv_0}{\sqrt{8k(m + M) - c^2}} e^{-\left[\frac{\pi c}{2\sqrt{8k(m + M) - c^2}}\right]} \qquad \textbf{Ans.}$$

Ans:

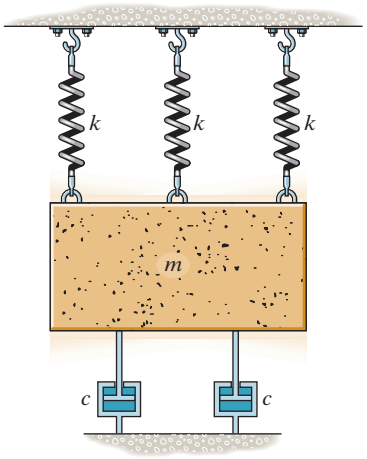
$$x_{\max} = \frac{2mv_0}{\sqrt{8k(m + M) - c^2}} e^{-\pi c/\left(2\sqrt{8k(m+M)-c^2}\right)}$$

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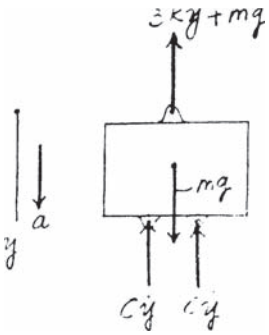
22–75.

Determine the differential equation of motion for the damped vibratory system shown. What type of motion occurs? Take $k = 100$ N/m, $c = 200$ N·s/m, $m = 25$ kg.



SOLUTION

Free-body Diagram: When the block is being displaced by an amount y vertically downward, the *restoring force* is developed by the three springs attached the block.



Equation of Motion:

$$+\uparrow \Sigma F_x = 0; \qquad 3ky + mg + 2c\dot{y} - mg = -m\ddot{y}$$

$$m\ddot{y} + 2c\dot{y} + 3ky = 0 \qquad \textbf{(1)}$$

Here, $m = 25$ kg, $c = 200$ N·s/m and $k = 100$ N/m. Substituting these values into Eq. (1) yields

$$25\ddot{y} + 400\dot{y} + 300y = 0$$

$$\ddot{y} + 16\dot{y} + 12y = 0 \qquad \textbf{Ans.}$$

Comparing the above differential equation with Eq. 22–27, we have $m = 1$kg, $c = 16$ N·s/m and $k = 12$ N/m. Thus, $\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{12}{1}} = 3.464$ rad/s

$$c_c = 2m\omega_n = 2(1)(3.464) = 6.928 \text{ N·s/m}$$

Since $c > c_c$, the system will not vibrate. Therefore it is **overdamped.** **Ans.**

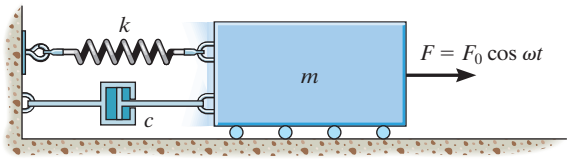
Ans:
$\ddot{y} + 16\dot{y} + 12y = 0$
Since $c > c_c$, the system will not vibrate. Therefore it is **overdamped.**

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***22–76.**

Draw the electrical circuit that is equivalent to the mechanical system shown. Determine the differential equation which describes the charge q in the circuit.



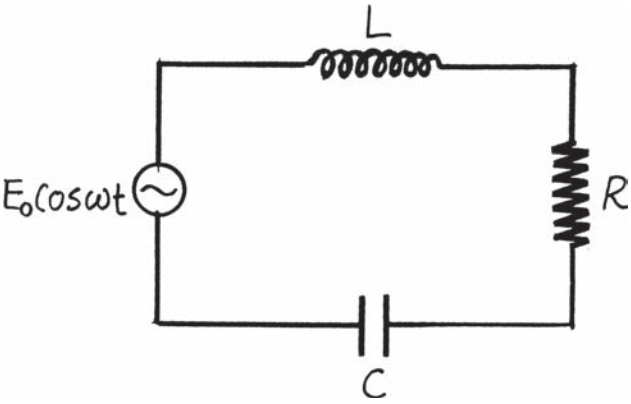
SOLUTION

For the block,

$$mx + cx + kx = F_0 \cos \omega t$$

Using Table 22–1,

$$Lq + Rq + (\frac{1}{C})q = E_0 \cos \omega t$$ **Ans.**



Ans:

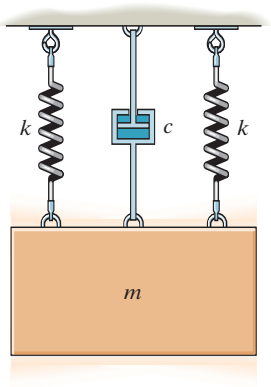
$$Lq + Rq + \left(\frac{1}{C}\right)q = E_0 \cos \omega t$$

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22–77.

Draw the electrical circuit that is equivalent to the mechanical system shown. What is the differential equation which describes the charge q in the circuit?



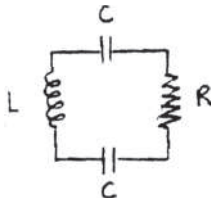
SOLUTION

For the block,

$$m\ddot{x} + c\dot{x} + 2k = 0$$

Using Table 22–1,

$$L\ddot{q} + R\dot{q} + \left(\frac{2}{C}\right)q = 0 \qquad \textbf{Ans.}$$



Ans:

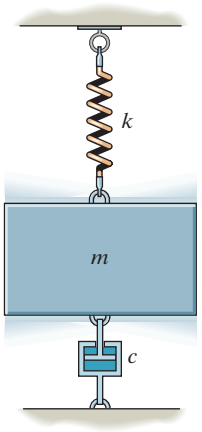
$$L\ddot{q} + R\dot{q} + \left(\frac{2}{C}\right)q = 0$$

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22–78.

Draw the electrical circuit that is equivalent to the mechanical system shown. Determine the differential equation which describes the charge q in the circuit.



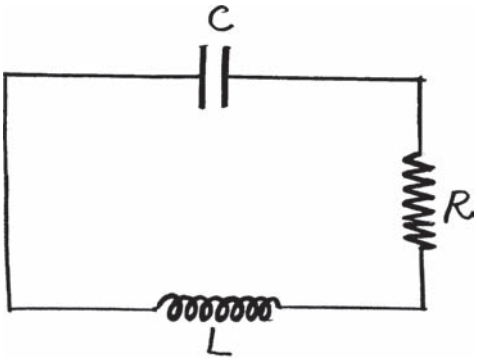
SOLUTION

For the block

$$m\ddot{y} + c\dot{y} + ky = 0$$

Using Table 22–1

$$L\ddot{q} + R\dot{q} + \frac{1}{C}q = 0 \qquad \textbf{Ans.}$$



Ans:

$$L\ddot{q} + R\dot{q} + \frac{1}{C}q = 0$$

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