Problems and Solutions Section 5.5 (5.53 through 5.66)

5.53 Design a Houdaille damper for an engine modeled as having an inertia of 1.5 kg·m^2 and a natural frequency of 33 Hz. Choose a design such that the maximum dynamic magnification is less than 6:

$$\left|\frac{Xk}{M_0}\right| < 6$$

The design consists of choosing J_2 and c_a, the required optimal damping.

Solution:

From Equation (5.50),

$$\left(\frac{Xk}{M_0}\right)_{\max} = 1 + \frac{2}{\mu}$$

Since $\left|\frac{Xk}{M_0}\right| < 6,$ then

$$6 > 1 + \frac{2}{\mu}$$

$$\mu > 0.4$$

Choose μ = 0.4. From Equation (5.49), the optimal damping is

$$\zeta_{op} = \frac{1}{\sqrt{2(\mu+1)(\mu+2)}} = 0.3858$$

The values of J_2 and c_a are

$$J_2 = \mu J_1 = (0.4)(1.5 \text{ kg}\cdot\text{m}^2/\text{rad}) = 0.6 \text{ kg}\cdot\text{m}^2/\mathit{rad}$$
$$c_a = 2\zeta_{\text{op}} J_2 \omega_p = 2(0.3858)(0.6)(33)(2\pi) = 95.98 \text{ N}\cdot\text{m}\cdot\text{s/rad}$$

5.54 Consider the damped vibration absorber of equation (5.37) with β fixed at β = 1/2 and μ fixed at μ = 0.25. Calculate the value of ζ that minimizes $|X/\Delta|$. Plot this function for several values of 0 < ζ < 1 to check your design. If you cannot solve this analytically, consider using a three-dimensional plot of $|X/\Delta|$ versus r and ζ to determine your design.

Solution:

From equation (5.37), with β = 0.5 and μ = 0.25, let

$$f(r,\zeta) = \frac{X}{\Delta}\sqrt{\frac{4\zeta^2 r^2 + \left(r^2 - 0.25\right)^2}{4\zeta^2 r^2\left(1.25r^2 - 1\right)^2 + \left[0.065r^2 - \left(r^2 - 1\right)\left(r^2 - 0.25\right)\right]^2}}$$

From equataions (5.44) and (5.45), with $f = \dfrac{A^{1/2}}{B^{1/2}}$,

$$\frac{\partial f}{\partial \zeta} = 0$$

becomes

$$BdA - AdB$$

Since $B = 4\zeta^2 r^2\left(1.25r - 1\right)^2 + \left[0.0625r^2 - \left(r^2 - 1\right)\left(r^2 - 0.25\right)\right]^2$ and $A = 4\zeta^2 r^2 + \left(r^2 - 0.25\right)^2$, then

$$dA = \frac{\partial A}{\partial \zeta} = 8\zeta r^2$$

$$dB = \frac{\partial B}{\partial \zeta} = 8\zeta r^2\left(1.25r^2 - 1\right)^2$$

So,

$$\left\{4\zeta^2 r^2\left(1.25r^2 - 1\right)^2 + \left[0.0625r^2 - \left(4^2 - 1\right)\left(r^2 - 0.25\right)\right]^2\right\}\left(8\zeta r^2\right)$$

$$= \left\{4\zeta^2 r^2 + \left(r^2 - 0.25\right)^2\right\}\left(8\zeta r^2\right)\left(1.25r^2 - 1\right)^2$$

$$\left[0.0625r^2 - \left(r^2 - 1\right)\left(r^2 - 0.25\right)^2\right] = \left(r^2 - 0.25\right)^2\left(1.25r^2 - 1\right)^2$$

Taking the square root yields

$$0.625r^2 - \left(r^2 - 1\right)\left(r^2 - 0.25\right) = \pm\left(r^2 - 0.25\right)\left(1.25r^2 - 1\right)$$

Solving for r yields

$$r = 0.4896,\ 0.9628$$

Now take the derivative

$$\frac{\partial f}{\partial r} = 0$$

becomes

$$BdA = AdB$$

Since $B = 4\zeta^2 r^2\left(1.25r^2 - 1\right)^2 + \left[0.0625r^2 - \left(r^2 - 1\right)\left(r^2 - 0.25\right)\right]^2$ and $A = 4\zeta^2 r^2 + \left(r^2 - 0.25\right)^2$, then

$$dA \equiv \frac{\partial A}{\partial \zeta} = 8\zeta^2 r + 2\left(r^2 - 0.25\right)\left(2r\right)$$

$$dB \equiv \frac{\partial B}{\partial \zeta} = 8\zeta^2 r\left(1.25r^2 - 1\right)^2 + 8\zeta^2 r^2\left(1.25r^2 - \left(2r\right)\right)\left(2.5r\right)$$

$$+2\left[0.0625r^2 - \left(4^2 - 1\right)\left(r^2 - 0.25\right)\right]\left[0.125r - \left(2r\right)\left(r^2 - 0.25\right) - \left(r^2 - 1\right)\left(2r\right)\right]$$

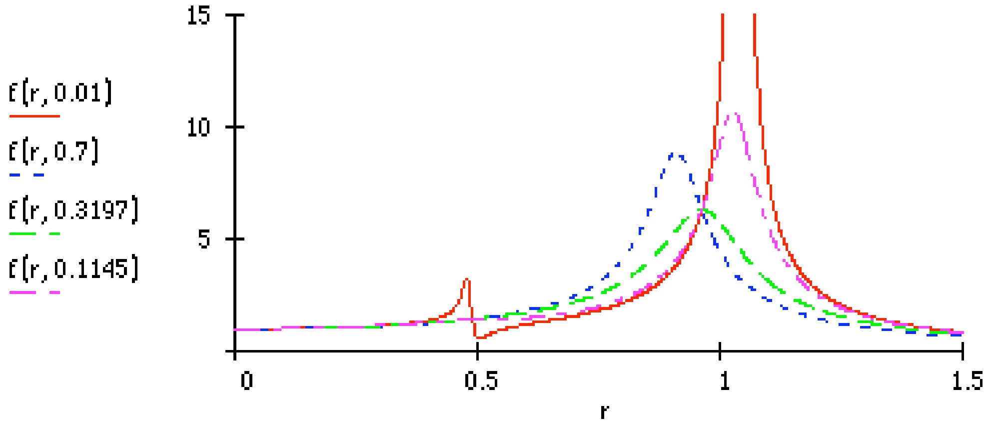
Solving $B\ dA = A\ dB$ for ζ yields

$$r = 0.4896 \rightarrow \zeta = 0.1145 \rightarrow \frac{X}{\delta_{st}} = 1.4279$$

$$r = 0.9628 \rightarrow \zeta = 0.3197 \rightarrow \frac{X}{\delta_{st}} = 6.3029$$

To determine the optimal damping ratio, make a plot of $\left|X / \Delta\right|$ versus r for ζ = 0.01, 0.1145, 0.3197, and 0.7.

$$f(r,\zeta) := \sqrt{\frac{(2\cdot\zeta\cdot r)^2 + (r^2 - 0.25)^2}{(2\cdot r\cdot\zeta)^2\cdot(1.25\cdot r^2 - 1^2)^2 + \left[0.0625\cdot r^2 - (r^2 - 1)\cdot(r^2 - 0.25)\right]^2}}$$



The value of $\zeta = 0.3197$ yields the best overall response (i.e., the lowest maximum).

5.55 For a Houdaille damper with mass ratio μ = 0.25, calculate the optimum damping ratio and the frequency at which the damper is most effective at reducing the amplitude of vibration of the primary system.

Solution:

From equation (5.49), with μ = 0.25,

$$\zeta_{op} = \frac{1}{\sqrt{2(\mu+1)(\mu+2)}} = 0.422$$

From equation (5.48),

$$r = \sqrt{\frac{2}{2+\mu}} = 0.943$$

The damper would be most effective at $\omega = r\omega_n = 0.943\omega_n$, i.e., where the amplitude is greatest:

5.56 Consider again the system of Problem 5.53. If the damping ratio is changed to $\zeta = 0.1$, what happens to $\left| Xk / M_0 \right|$?

Solution:

If $\zeta_{op} = 0.1$, the value of μ becomes

$$0.1 = \frac{1}{\sqrt{2(\mu+1)(\mu+2)}}$$
$$0.02\mu^2 + 0.06\mu = 0.96 = 0$$
$$\mu = -8.589,\ 5.589$$

Clearly μ = 5.589 is the physical solution. The maximum value of $\left| \frac{Xk}{M_0} \right|$ would be

$$\left(\frac{Xk}{M_0} \right)_{\max} = 1 + \frac{2}{\mu} = 1.358$$

which is less than 6 (the requirement of Problem 5.53). Note that the value of μ is extremely large.

5.57 Derive Equation (5.42) from Equation (5.35) and derive Equation (5.49) for the optimal damping ratio.

Solution:

Equation (5.37) is derived from Equation (5.35) in Problem 5.39.

Start with Equation (5.37):

$$\frac{Xk}{F_0} = \sqrt{\frac{(2\zeta r)^2 + (r^2 - \beta^2)^2}{(2\zeta r)^2(r^2 - 1 + \mu r^2)^2 + \left[\mu r^2\beta^2 - (r^2 - 1)(r^2 - \beta^2)\right]^2}}$$

To derive Equation (5.42), which is the same as Equation (5.39), note that $c = k_a = \omega_a = 0,$ which also means $\beta = 0$. Since this is a moment equation, F_0 is replaced by M_0. Therefore,

$$\frac{Xk}{F_0} = \sqrt{\frac{(2\zeta r)^2 + r^4}{(2\zeta r)^2(r^2 - 1 + \mu r^2)^2 + (r^2 - 1)^2 r^4}} = \sqrt{\frac{4\zeta^2 + r^4}{4\zeta^2(r^2 + \mu r^2 - 1)^2 + (r^2 - 1)^2 r^2}}$$

which is Equation (5.42) after canceling r^2.

To derive Equation (5.49), first let Equation (5.42) be $f(r,\zeta)$. Since $f = \dfrac{A^{1/2}}{B^{1/2}}$, where $A = 4\zeta^2 + r^2$ and $B = 4\zeta^2(r^2 + \mu r^2 - 1)^2 + (r^2 - 1)^2 r^2,$ then

$$\frac{\partial f}{\partial \zeta} = 0$$

becomes

$$BdA = AdB$$

where

$$dA \equiv \frac{\partial A}{\partial \zeta} = 8\zeta$$

$$dB \equiv \frac{\partial B}{\partial \zeta} = 8\zeta(r^2 + \mu r^2 - 1)^2$$

So,

$$\left\{4\zeta^2\left(r^2+\mu r^2-1\right)^2+\left(r^2-1\right)^2 r^2\right\}\left(8\zeta\right)=\left\{4\zeta^2+r^2\right\}\left(8\zeta\right)\left(r^2+\mu r^2-1\right)^2$$
$$\left(r^2-1\right)^2=\left(r^2+\mu r^2-1\right)^2$$
$$\left(r^2-1\right)=\pm\left(r^2+\mu r^2-1\right)$$

Taking the minus sign (the plus sign yields $r = 0$).

$$\left(2+\mu\right)r^2-2=0$$
$$r=\sqrt{\frac{2}{\left(2+\mu\right)}}$$

Now take the other partial derivative $\frac{\partial f}{\partial r}=0$, which becomes

$$BdA = AdB$$
$$dA\equiv\frac{\partial A}{\partial r}=2r$$
$$dB\equiv\frac{\partial B}{\partial r}=16\zeta^2 r\left(1+\mu\right)\left(r^2+\mu r^2-1\right)+4r^3\left(r^2-1\right)+2r\left(r^2-1\right)^2$$

So,

$$\left\{4\zeta^2\left(r^2+\mu r^2-1\right)^2+\left(r^2-1\right)^2 r^2\right\}\left(2r\right)$$
$$=\left\{4\zeta^2+r^2\right\}\left[16\zeta^2 r\left(1+\mu\right)\left(r^2+\mu r^2-1\right)+4r^3(r^2-1)+2r\left(r^2-1\right)^2\right]$$

Substituting $r=\sqrt{\frac{2}{\left(2+\mu\right)}}$ yields, after rearranging

$$4\zeta^2\left[\frac{2}{2+\mu}+\frac{2\mu}{2+\mu}-1\right]^2+\left[\frac{2}{2+\mu}-1\right]^2\left(\frac{2}{2+\mu}\right)$$
$$=\left[4\zeta^2+\frac{2}{2+\mu}\right]\left[8\zeta^2\left(1-\mu\right)\left(\frac{2}{2+\mu}+\frac{2\mu}{2+\mu}-1\right)+2\left(\frac{2}{2+\mu}\right)\left(\frac{2}{2+\mu}-1\right)+\left(\frac{2}{2+\mu}-1\right)^2\right]$$

Expanding and canceling terms yields

$$4\zeta^4(1+\mu)(2+\mu)+2\zeta^2\mu-\frac{2}{2+\mu}=0$$

The physical solution for ζ is

$$\zeta=\frac{1}{\sqrt{2(1+\mu)(2+\mu)}}$$

which is Equation (5.49).

5.58 Consider the design suggested in Example 5.5.1. Calculate the percent change in the maximum deflection if the damping constant changes 10% from an optimal value. If the optimal damping is fixed but the mass of the absorber changes by 10%, what percent change in $\left|Xk / M_0\right|_{\max}$ results? Is the optimal absorber design more sensitive to changes in c_a or m_a?

Solution:

From Problems 5.51 and 5.46, F_0 = 100 N, $k = 4.144 \times 10^6$ N/m, and μ = 0.133. The optimal damping is

$$\zeta_{op} = \frac{1}{\sqrt{2(1+\mu)(2+\mu)}} = 0.4549$$

The deflection is given by Equation (5.42), and M_0 replaced by F_0,

$$X = \frac{F_0}{k}\sqrt{\frac{4\zeta^2 + r^2}{4\zeta^2\left(r^2 + \mu r^2 - 1\right)^2 + \left(r^2 - 1\right)^2 r^2}}$$

Also, the maximum displacement will occur at $r = \sqrt{\dfrac{2}{(2+\mu)}}$ = 0.9683. If the damping constant changes by 10%, ζ will also change by 10% since $\zeta = \dfrac{c_a}{2m\omega_p}$. The value of X for 0.9 ζ_{op}, ζ_{op}, and 1.1 ζ_{op} is

$$\begin{aligned}
\zeta &= 0.9\zeta_{op} & &\rightarrow & X &= 3.870 \times 10^{-4}\ \text{m}\\
\zeta &= \zeta_{\text{op}} & &\rightarrow & X &= 3.870 \times 10^{-4}\ \text{m}\\
\zeta &= 1.1\zeta_{\text{op}} & &\rightarrow & X &= 3.870 \times 10^{-4}\ \text{m}
\end{aligned}$$

There is no change in X with a 10% change in ζ_{op}.

If m_a changes by 10%, μ will also change by 10% since $\mu = \dfrac{m_a}{m}$. The value of $\left(\dfrac{Xk}{F_0}\right)_{\max}$ for 0.9μ, μ, and 1.1μ is

$$0.9\mu \quad \rightarrow \quad r = 0.9714 \quad \rightarrow \quad \left(\frac{Xk}{F_0}\right)_{\max} = 17.708(+10.4\%)$$

$$\mu \quad \rightarrow \quad r = 0.9683 \quad \rightarrow \quad \left(\frac{Xk}{F_0}\right)_{\max} = 16.038$$

$$1.1\mu \quad \rightarrow \quad r = 0.9318 \quad \rightarrow \quad \left(\frac{Xk}{F_0}\right)_{\max} = 14.671(-8.5\%)$$

The displacement is more sensitive to changes in m_a than c_a.

5.59 Consider the elastic isolation problem described in Figure 5.26. Derive equations (5.57) and (5.58) from equation (5.53).

Solution:

Rewrite equation (5.53) in matrix form as

$$\begin{bmatrix} k_1 - m\omega^2 + jc\omega & -jc\omega \\ -jc\omega & -\left(k_2 + jc\omega\right) \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix}$$

The inverse of the matrix on the left is

$$\frac{1}{-k_2\left(k_1 - m\omega^2\right) - jc\omega\left(k_1 + k_2 m\omega^2\right)} \begin{bmatrix} -\left(k_2 + jc\omega\right) & jc\omega \\ jc\omega & k_1 m\omega^2 + jc\omega \end{bmatrix}$$

Solving for X_1 and X_2 yields

$$X_1 = \frac{\left(k_2 + jc\omega\right)F_0}{k_2\left(k_1 - m\omega^2\right) + jc\omega\left(k_1 + k_2 - m\omega_{dr}^2\right)}$$

$$X_2 = \frac{c\omega_{dr} F_0 j}{k_2\left(k_1 - m\omega^2\right) + jc\omega\left(k_1 + k_2 - m\omega_{dr}^2\right)}$$

which are equations (5.54) and (5.55).

5.60 Use the derivative calculation for finding maximum and minimum to derive equations (5.57) and (5.58) for the elastic damper system.

Solution:

From equation (5.56)

$$T.R. = \frac{\sqrt{1+4(1+\gamma)^2\zeta^2 r^2}}{\sqrt{(1-r^2)^2 + 4\zeta^2 r^2(1+\gamma - r^2\gamma)^2}}$$

Equation (5.45) is applicable here, so that

$$BdA = AdB$$

where $A = 1+4(1+\gamma)^2\zeta^2 r^2$ and $B = (1-r^2)^2 + 4\zeta^2 r^2(1+\gamma - r^2\gamma)^2$ differentiating with respect to ζ yields

$$dA \equiv \frac{\partial A}{\partial \zeta} = 8(1+\gamma)^2 \zeta r^2$$
$$dB \equiv \frac{\partial B}{\partial \zeta} = 8\zeta r^2(1+\gamma - r^2\gamma)^2$$

So,

$$\left\{(1-r^2)^2 + 4\zeta^2 r^2(1+\gamma - r^2\gamma)^2\right\}(8)(1+\gamma)^2\zeta r^2$$
$$= \left\{1+4(1+\gamma)^2\zeta^2 r^2\right\}(8\zeta r^2)(1+\gamma - r^2\gamma)^2$$
$$(1-r^2)^2(1+\gamma)^2 = (1+\gamma - r^2\gamma)^2$$
$$(1-r^2)(1+\gamma) = \pm(1+\gamma - r^2\gamma)$$

The minus sign yields the physical result

$$r^2(2\gamma+1) = 2(1+\gamma)$$
$$r = \sqrt{\frac{2(1+\gamma)}{1+2\gamma}}$$

which is equation (5.57)

Differentiating with respect to *r* yields

$$dA \equiv \frac{\partial A}{\partial r} = 8\left(1+\gamma\right)^2 \zeta^2 r$$
$$dB \equiv \frac{\partial B}{\partial r} = 2\left(1-r^2\right)\left(-2r\right) + 8\zeta^2 r\left(1+\gamma-r^2\gamma\right)^2 + 8\zeta^2 r^2\left(1+\gamma-r^2\gamma\right)\left(-2r\gamma\right)$$

So,

$$\left\{\left(1-r^2\right)^2 + 4\zeta^2 r^2\left(1+\gamma-r^2\gamma\right)^2\right\}\left(8\zeta^2 r\right)\left(1+\gamma\right)^2$$
$$= \left\{1+4\left(1+\gamma\right)^2\zeta^2 r^2\right\}\left[-4r\left(1-r^2\right)+8\zeta^2 r\left(1+\gamma-r^2\gamma\right)^2 - 16\gamma\zeta^2 r^3\left(1+\gamma-r^2\gamma\right)\right]$$

Substituting for *r* and manipulating yields

$$\left[64\gamma\left(1+\gamma\right)^5\left(\frac{1}{1+2\gamma}\right)\right]\zeta^4 + 8\left[\gamma\left(1+\gamma\right)^2 + \left(1+\gamma\right)^3\left(1+2\gamma\right) - 2\left(1+\gamma\right)^4\right]\zeta^2 - \left(1+2\gamma\right) = 0$$

Solving for ζ yields the physical result

$$\zeta = \frac{\sqrt{2\left(1+2\gamma\right)/\gamma}}{4\left(1+\gamma\right)}$$

which is Equation (5.58).

5.61 A 1000-kg mass is suspended from ground by a 40,000-N/m spring. A viscoelastic damper is added, as indicated in Figure 5.26. Design the isolator (choose k_2 and c) such that when a 70-N sinusoidal force is applied to the mass, no more than 100 N is transmitted to ground.

Solution:

From equation (5.59),

$$(T.R.)_{\max} = 1 + 2\gamma$$
$$\frac{F_T}{F_0} = \frac{100}{70} = 1.429 = 1 + 2\gamma$$
$$\gamma = 0.2143$$

The isolator stiffness should be

$$k_2 = \gamma k_1 = (0.2143)(40{,}000) = 8571\,\text{N/m}$$

From equation (5.58),

$$\zeta_{op} = \frac{\sqrt{2(1+2\gamma)/\gamma}}{4(1+\gamma)} = 0.7518$$

The isolator damping should be

$$c = 2\zeta_{op}\sqrt{\frac{k_1}{m}} = 2(0.7518)\sqrt{\frac{40{,}000}{1000}} = 9.51\,\text{kg/s}$$

5.62 Consider the isolation design of Example 5.5.2. If the value of the damping coefficient changes 10% from the optimal value (of 188.56 kg/s), what percent change occurs in $(\text{T.R.})_{max}$? If c remains at its optimal value and k_2 changes by 10%, what percent change occurs in $(\text{T.R.})_{max}$? Is the design of this type of isolation more sensitive to changes in damping or stiffness?

Solution:

From Example 5.5.2, c = 188.56 kg/s and k_2 = 200 N/m. If the value of c changes by 10%, the value of T.R. becomes (with r = 5 and γ = 0.5),

$$\begin{array}{lllll} 0.9c & \rightarrow & \zeta_{op} = 0.4243 & \rightarrow & T.R. = 0.1228(-1.78\%) \\ c & \rightarrow & \zeta_{op} = 0.4714 & \rightarrow & T.R. = 0.1250 \\ 1.1c & \rightarrow & \zeta_{op} = 0.5185 & \rightarrow & T.R. = 0.1267\left(+1.39\%\right) \end{array}$$

If the value of k_2 changes by 10%, the value of T.R. becomes (with r = 5 and ζ = 0.4714),

$$\begin{array}{lllll} 0.9k_2 & \rightarrow & \gamma = 0.45 & \rightarrow & T.R. = 0.1327\left(+6.17\%\right) \\ k_2 & \rightarrow & \gamma = 0.5 & \rightarrow & T.R. = 0.1250 \\ 1.1k_2 & \rightarrow & \gamma = 0.55 & \rightarrow & T.R. = 0.1183\left(-5.31\%\right) \end{array}$$

This design is more sensitive to changes in stiffness.

5.63 A 3000-kg machine is mounted on an isolator with an elastically coupled viscous damper such as indicated in Figure 5.26. The machine stiffness (k_1) is 2.943×10^6 N/m, $\gamma = 0.5$, and $c = 56.4 \times 10^3$ N·s/m. The machine, a large compressor, develops a harmonic force of 1000 N at 7 Hz. Determine the amplitude of vibration of the machine.

Solution:

The amplitude of vibration is given by Equation (5.54) as

$$X_1 = \frac{\left(k_2 + jc\omega\right)F_0}{k_2\left(k_1 - m\omega^2\right) + jc\omega\left(k_1 + k_2 - m\omega_{dr}^2\right)}$$

Since F_0 = 1000 N, $\omega = 7(2\pi) = 43.98$ rad/s, m = 3000 kg, $c = 56.4 \times 10^3$ N·s/m, $k_1 = 2.943 \times 10^6$ N/m, and $k_2 = \gamma k_1 = 1.4715 \times 10^6$ N/m, then

$$X_1 = -4.982 \times 10^{-4} - 1.816 \times 10^{-4}\, j$$

The magnitude is

$$\left|X_1\right| = 5.303 \times 10^{-4} \text{ m}$$

5.64 Again consider the compressor isolation design given in Problem 5.63. If the isolation material is changed so that the damping in the isolator is changed to $\zeta = 0.15$, what is the force transmitted? Next determine the optimal value for the damping ratio and calculate the resulting transmitted force.

Solution:

From Problem 5.63, $\gamma = 0.5$, $F_0 = 1000$ N, and $r = \dfrac{\omega}{\sqrt{k_1 / m}} = \dfrac{7(2\pi)}{\sqrt{2.943 \times 10^6 / 3000}} =$ 1.404. Since $\zeta = 0.15$, the transmitted force is [from Equation (5.56)],

$$F_T = F_0 \sqrt{\frac{1 + 4(1+\gamma)^2 \zeta^2 r^2}{(1 - r^2)^2 + 4\zeta^2 r^2 (1 + \gamma - r^2 \gamma)^2}} = 1188 \text{ N}$$

The optimal value for the damping ratio is found from equation (5.58):

$$\zeta_{op} = \frac{\sqrt{2(1+2\gamma)/\gamma}}{4(1+\gamma)} = 0.4714$$

The transmitted force is then

$$F_T = 1874 \text{ N}$$

5.65 Consider the optimal vibration isolation design of Problem 5.64. Calculate the optimal design if the compressor's steady-state driving frequency changes to 24.7 Hz. If the wrong optimal point is used (i.e., if the optimal damping for the 7-Hz driving frequency is used), what happens to the transmissibility ratio?

Solution:

From Problems 5.63 and 5.64, $\gamma = 0.5$, $F_0 = 1000$ N, $k_1 = 2.943 \times 10^6$ N, and $m = 3000$ kg.

The optimal damping is

$$\zeta_{op} = \frac{\sqrt{2(1+2\gamma)/\gamma}}{4(1+\gamma)} = 0.4714$$

The value of c and k_2 would be

$$c = 2\zeta_{op}\sqrt{k_1 m} = 88.589 \text{ kg/s}$$
$$k_2 = \gamma k_1 = 1.472 \times 10^6 \text{ N/m}$$

The isolation design is independent of the driving frequency in this problem, so the transmissibility ratio would not change.

5.66 Recall the optimal vibration absorber of Problem 5.53. This design is based on a steady-state response. Calculate the response of the primary system to an impulse of magnitude M_0 applied to the primary inertia J_1. How does the maximum amplitude of the transient compare to that in steady state?

Solution:

The response of the system given in Problem 5.53 cannot be solved by the means of modal analysis given in Chapter 4 because the system is not proportionally damped. However, the steady-state response of a damped system to an impulse is simply zero. Therefore, the maximum amplitude of the transient will be of interest. For a sinusoidal input, a numerical simulation might be necessary to determine the effects of the transient response.