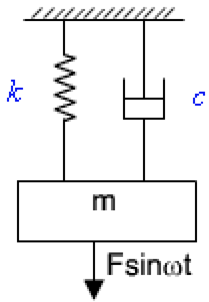
U-3, **Forced Harmonic Vibration**

- **Steady State Response due to Harmonic Oscillation :**
- Consider a spring-mass-damper system as shown in figure 4.1. The equation of motion of this system subjected to a harmonic force can be $F \sin \omega t$
- $$m\ddot{x} + kx + c\dot{x} = F \sin \omega t \quad (4.1)$$
- where, *m* , *k* and *c* are the mass, spring stiffness and damping coefficient of the system, *F* is the amplitude of the force, *w* is the excitation frequency or driving frequency.
- 

- Figure 4.1 Harmonically excited system

Forced Harmonic Vibration

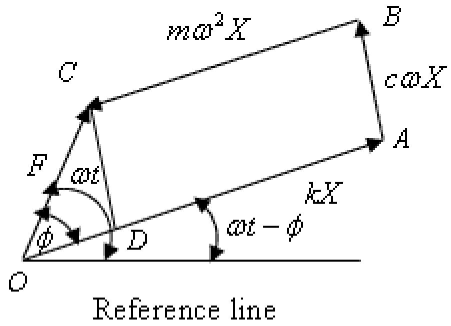


Figure 4.2: Force polygon

The steady state response of the system can be determined by solving equation(4.1) in many different ways. Here a simpler graphical method is used which will give physical understanding to this dynamic problem. From solution of differential equations it is known that the steady state solution (particular integral) will be of the form

$$x = X\sin(\omega t - \phi) \qquad (4.2)$$

As each term of equation (4.1) represents a forcing term viz., first, second and third terms, represent the inertia force, spring force, and the damping forces. The term in the right hand side of equation(4.1) is the applied force. One may draw a close polygon as shown in figure 4.2 considering the equilibrium of the system under the action of these forces. Considering a reference line these forces can be presented as follows.

- Spring force = $kx = kX\sin(\omega t - \phi)$ (This force will make an angle $\omega t - \phi$ with the reference line, represented by line OA).

Damping force = $c\dot{x} = c\omega X\cos(\omega t - \phi)$ (This force will be perpendicular to the spring force, represented by line AB).

Inertia force = $m\ddot{x} = -m\omega^2X\sin(\omega t - \phi)$ (this force is perpendicular to the damping force and is in opposite direction with the spring force and is represented by line BC) .

Applied force = $F\sin\omega t$ which can be drawn at an angle ωt with respect to the reference line and is represented by line OC.

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- From equation (1), the resultant of the spring force, damping force and the inertia force will be the applied force, which is clearly shown in figure 4.2.
- It may be noted that till now, we don't know about the magnitude of *X* and ϕ which can be easily computed from Figure 2. Drawing a line CD parallel to AB, from the triangle OCD of Figure 2,

$$F^2 = (c\omega X)^2 + (kX - m\omega^2 X)^2$$

$$X = \frac{F}{\sqrt{(k - m\omega^2)^2 + (c\omega)^2}}$$

$$\phi = \tan^{-1}\frac{c\omega}{k - m\omega^2}$$

$$X = \frac{F/k}{\sqrt{\left(1 - \frac{m}{k}\omega^2\right)^2 + \left(\frac{c\omega}{k}\right)^2}}$$

- Natural frequency $\omega_n = \sqrt{\frac{k}{m}}$
- Critical damping $c_c = 2m\omega_n$
- Damping factor or damping ratio

$$\zeta = \frac{c}{c_c}$$

Hence,

$$\frac{c\omega}{k} = \frac{c}{c_c}\frac{c_c\omega}{k} = \zeta\frac{2m\omega_n\omega}{k} = 2\zeta\frac{\omega}{\omega_n}$$

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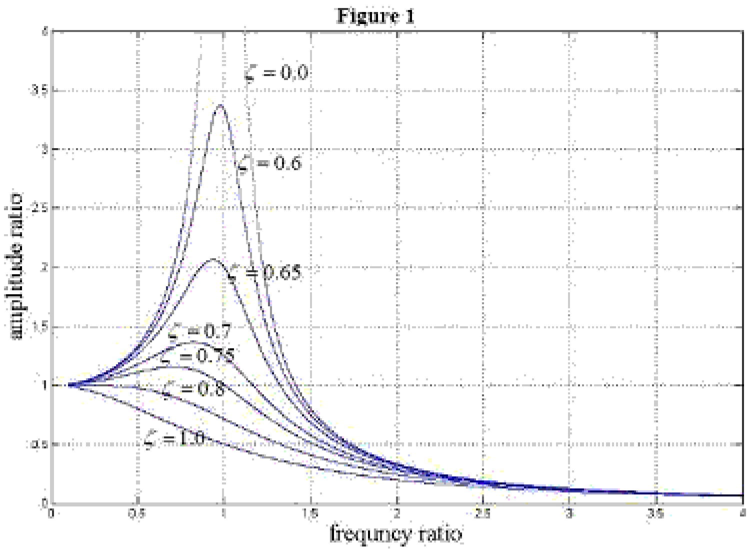
$$\Rightarrow \quad \frac{Xk}{F} = \frac{1}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right] + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]^2}}$$

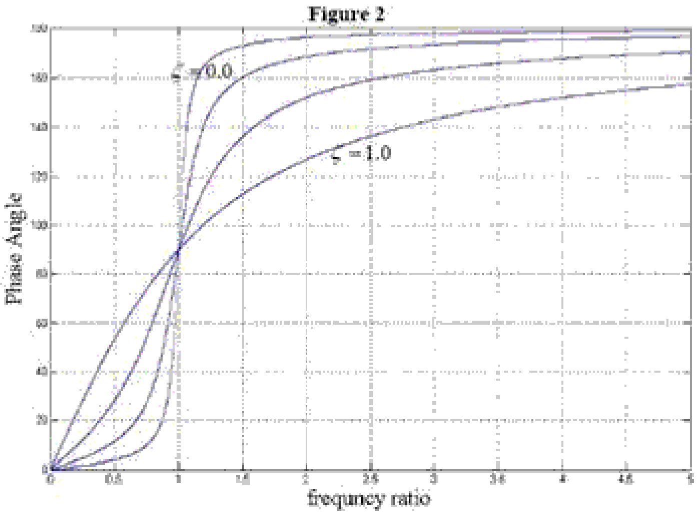
$$\tan\phi = \frac{2\zeta(\omega/\omega_n)}{1-(\omega/\omega_n)^2} \quad \text{or} \quad \phi = \tan^{-1}\left(\frac{2\zeta(\omega/\omega_n)}{1-(\omega/\omega_n)^2}\right)$$

As the ratio $\frac{F}{k}$ is the static deflection (X_0) of the spring, $\frac{Xk}{F} = \frac{X}{X_0}$

is known as the **magnification factor or amplitude ratio** of the system

Forced Harmonic Vibration





Forced Harmonic Vibration

- Following observation can be made from these plots.
- For undamped system (i.e. $\zeta = 0$) the magnification factor tends to infinity when the frequency of external excitation equals natural frequency of the system . $\left(\frac{\omega}{\omega_n} = 1\right)$
- But for underdamped systems the maximum amplitude of excitation has a definite value and it occurs at a frequency $\frac{\omega}{\omega_n} < 1.$
- For frequency of external excitation very less than the natural frequency of the system, with increase in frequency ratio, the dynamic deflection (X) dominates the static deflection (X_0), the magnification factor increases till it reaches a maximum value at resonant frequency (ω_r)
- For $\omega > \omega_r$, the magnification factor decreases and for very high value of frequency ratio (say) $\frac{\omega}{\omega_n} > 2$
- One may observe that with increase ın damping ratio, the resonant response amplitude decreases.
- Irrespective of value of ζ, at $\frac{\omega}{\omega_n} = 1$ the phase angle $\phi = 90^0$

Forced Harmonic Vibration

- For, $\frac{\omega}{\omega_n} < 1$. phase angle $\phi < 90^0$
- For $\frac{\omega}{\omega_n} > 1$, phase angle ϕ approaches 180^0 for very low value of ζ
- From phase angle and frequency ratio plot it is clear that, for very low value of frequency ratio, phase angle tends to zero and at resonant frequency, it is 90^0 and for very high value of frequency ratio it is 180^0.

For a underdamped system the total response of the system which is the combination of transient response and steady state response can be given by

$$x(t) = x_1 e^{-\zeta\omega_n t} \sin\left(\sqrt{1-\zeta^2}\,\omega_n t + \phi_1\right) + \frac{F_0}{k} \frac{\sin(\omega t - \phi)}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2}}$$

The parameter ϕ_1 will depend on the initial conditions.

It may be noted that as $t \to \infty$, the first part of equation (6) tends to zero and second part remains

Forced Harmonic Vibration

- Hence
- $$X = \frac{\left(\frac{F}{k}\right)\sin(\omega t - \phi)}{\sqrt{\left[\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_n}\right)\right)^2\right]}} \qquad (4.12)$$
- where $\tan\phi = \frac{2\zeta\omega_n\omega}{\left(-\omega^2 + \omega_n^2\right)}$ (from equation (4.10) and (4.11))
- or $$\phi = \tan^{-1}\left(\frac{2\zeta(\omega/\omega_n)}{1-(\omega/\omega_n)^2}\right) \qquad (4.13)$$

It may be noted that equation (4.12) and (4.13) are same as equation (4.3) and (4.5)

Rotating Unbalance

- One may find many rotating systems in industrial applications. The unbalanced force in such a system can be represented by a mass *m* with eccentricity *e* , which is rotating with angular velocity as shown in Figure 4.1.

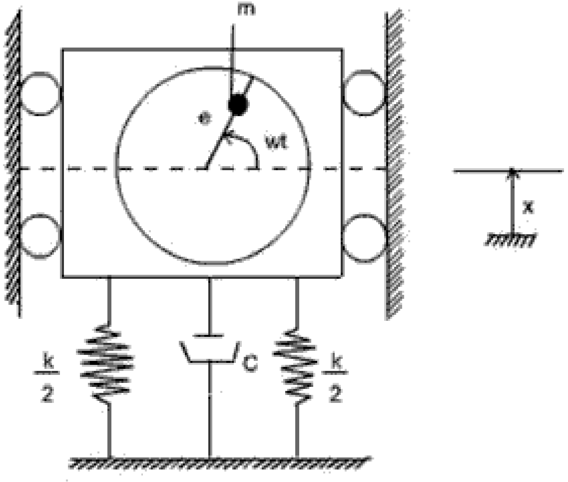


Figure 4.1 : Vibrating system with rotating unbalance

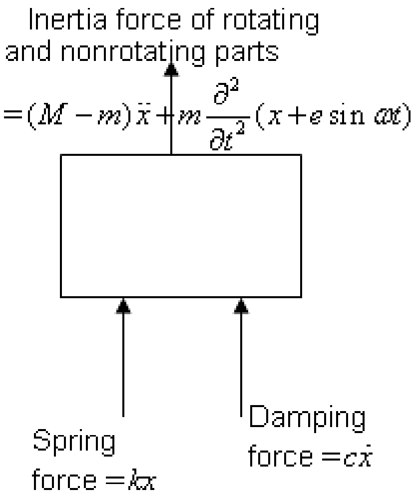


Figure 4.2. Freebody diagram of the system

Rotating Unbalance

- Let *x* be the displacement of the non rotating mass (*M*-*m*) from the static equilibrium position, then the displacement of the rotating mass *m* is
- From the freebody diagram of the system shown in figure 4.2, the equation of motion is $x + e\sin\omega t$
- $$(M-m)\ddot{x} + m\frac{\partial^2}{\partial t^2}(x + e\sin\omega t) + kx + c\dot{x} = 0$$ (4.1)
- or (4.2)
$$M\ddot{x} + k\dot{x} + cx = me\omega^2\sin\omega t$$
- This equation is same as equation (1) where *F* is replaced by . So from the force polygon as shown in figure 4.3
- (4.3) $me\omega^2$

$$me\omega^2 = \sqrt{\left\{(-M\omega^2+k)^2 + c\omega^2\right\}X^2}$$

or $$X = \frac{me\omega^2}{\sqrt{(k-M\omega^2)^2 + (c\omega)^2}}$$ (4.4)

Rotating Unbalance

$$\frac{X}{e} = \frac{\frac{m\omega}{M}}{\sqrt{\left(\frac{k}{M} - \omega^2\right)^2 + \left(\frac{c}{M}\omega\right)^2}} \qquad (4.5)$$

or

$$\frac{X}{e} = \frac{\frac{m\omega}{M}}{\sqrt{\left(\frac{k}{M} - \omega^2\right)^2 + \left(\frac{c}{M}\omega\right)^2}} \qquad (4.6)$$

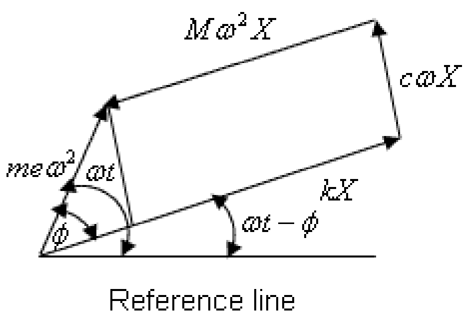


Figure 4.3: Force polygon

and

$$\tan\phi = \frac{2\zeta\frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \qquad (4.7)$$

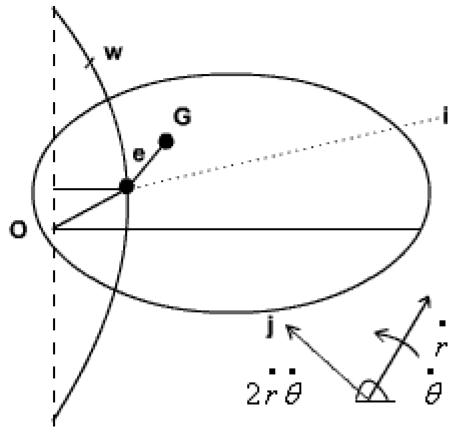
So the complete solution becomes

$$x(t) = x_1 e^{-\zeta\omega_n t}\sin\left(\sqrt{1-\zeta^2}\,\omega_n t + \phi_1\right) + \frac{me\omega^2}{\sqrt{(k - M\omega^2)^2 + (c\omega)^2}}\sin(\omega t - \phi) \qquad (4.8)$$

WHIRLING OF SHAFT

Whirling is defined as the rotation of the plane made by the bent shaft and the line of the centre of the bearing. It occurs due to a number of factors, some of which may include

(i) eccentricity,
(ii) unbalanced mass,
(iii) gyroscopic forces,
(iv) *fluid friction in bearing, viscous damping.*



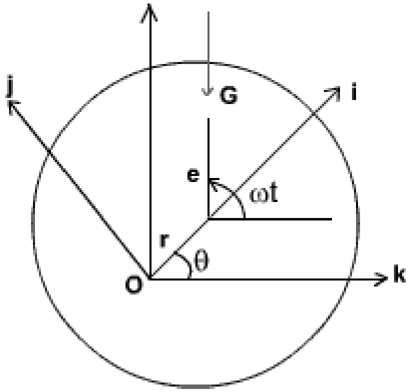


Figure 4.6: Whirling of shaft

WHIRLING OF SHAFT

- Consider a shaft AB on which a disc is mounted at ***S*** . ***G*** is the mass center of the disc, which is at a distance ***e*** from ***S*.** As the mass center of the disc is not on the shaft center, when the shaft rotates, it will be subjected to a centrifugal force. This force will try to bend the shaft. Now the neutral axis of the shaft, which is represented by line ASB, is different from the line joining the bearing centers AOB. The rotation of the plane containing the line joining bearing centers and the bend shaft (in this case it is AOBSA) is called the whirling of the shaft.
- Considering unit vectors ***i*** , ***j*** , ***k*** as shown in the figure 4.6(b), the acceleration of point *G* can be given by $a_G = a_S + a_{G/S}$

$$= \left[\ddot{r} - r\dot{\theta}^2 - e\omega^2\cos\left(\omega t - \theta\right)\right]\mathbf{i} + \left[r\ddot{\theta} - e\omega^2\sin\left(\omega t - \theta\right) + 2\dot{r}\dot{\theta}\right]\mathbf{j} \qquad (4.9)$$

- Assuming a viscous damping acting at *S*. The equation of motion in radial direction
- $$m\left[\ddot{r} - r\dot{\theta}^2 - e\omega^2\cos\left(\omega t - \theta\right)\right] + kr + C\dot{r} = 0 \qquad (4.10)$$
- $$m\left[r\theta + 2\dot{r}\dot{\theta} - e\ddot{\omega}^2\sin\left(\omega t - \theta\right)\right] + cr\dot{\theta} = 0 \qquad (4.11$$
-

WHIRLING OF SHAFT

- $$\ddot{r} + \frac{c}{m}\dot{r} + \left(\frac{k}{m} - \dot{\theta}^2\right) = e\omega^2 \cos(\omega t - \theta)$$
- (4.12)
- $$r\ddot{\theta} + \left(\frac{c}{m}r + 2\dot{r}\right)\dot{\theta} = e\omega^2 \sin(\omega t - \theta)$$ (4.13)
- Considering the synchronous whirl case, i.e. $\dot{\theta} = \omega$
- $$\theta = (\omega t - \phi)$$ (4.14)
- where ϕ is the phase angle between *e* and *r* .
- Taking , $\ddot{\theta} = \ddot{r} = \dot{r} = 0$ from equation (4.12)
- $$\left(\frac{k}{m} - \omega^2\right) = e\omega^2 \cos\phi$$ (4.14)
- and
- $$\frac{c}{m} r\omega = e\omega^2 \sin\phi$$ (4.15)

WHIRLING OF SHAFT

- Hence, $$\tan\phi = \frac{\frac{c}{m}\omega}{\left(\frac{k}{m}-\omega^2\right)} = \frac{2\zeta\frac{\omega}{\omega_n}}{1-\left(\frac{\omega}{\omega_n}\right)^2}$$ (4.16)
- as $\omega_n = \sqrt{\frac{k}{m}}$ and $\zeta = \frac{c}{c_c}$
- From equation (15), (4.18)
  $$\cos\phi = \frac{\frac{c}{m}\omega}{\sqrt{\left(\frac{k}{m}-\omega^2\right)^2+\left(\frac{c}{m}\omega\right)^2}}$$
- Substituting equation (4.18) in equation (4.15) yields
  $$\left(\frac{\mathrm{k}}{\mathrm{m}}-\omega^2\right)r = e\omega^2\frac{\left(\frac{k}{m}-\omega^2\right)}{\sqrt{\left(\frac{k}{m}-\omega^2\right)^2+\left(\frac{c}{m}\omega\right)^2}} = \frac{e\omega^2}{\sqrt{\left(\omega_n^2-\omega^2\right)+\left(2\zeta\frac{\omega}{\omega_n}\right)^2}}$$ (4.19)

or

- $$r = \frac{me\omega^2}{\sqrt{\left(k-m\omega^2\right)^2+\left(c\omega^2\right)^2}}$$ (4.20)

WHIRLING OF SHAFT

$$\frac{r}{e} = \frac{\omega^2 / \omega_n^2}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\zeta \frac{\omega}{\omega_n}\right]^2}} \qquad (4.21)$$

$$\tan\phi = \frac{2\zeta \frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \qquad (4.22)$$

- The eccentricity line *e* = *SG* leads the displacement line *r* = *OS* by phase angle ϕ which depends on the amount of damping and the rotation speed ratio $\frac{\omega}{\omega_n}$
- When the rotational speed equals to the natural frequency or critical speed, the amplitude is restrained by damping only.

From equation (22) at very high speed , $\omega >> \omega_n$ $\phi \to 180^0$

- and the center of mass *G* tends to approach the fixed point *O* and the shaft center S rotates about it in a circle of radius *e*.

SUPPORT MOTION

- Many machine components or instruments are subjected to forces from the support. For example while moving in a vehicle, the ground undulation will cause vibration, which will be transmitted, to the passenger. Such a system can be modelled by a spring-mass damper system as shown in figure 10.
- Here the support motion is considered in the form of $y = Y \sin \omega t$ which is transmitted to mass *m* , by spring (stiffness *k*) and damper (damping coefficient *c*).
- Let *x* be the vibration of mass about its equilibrium position.
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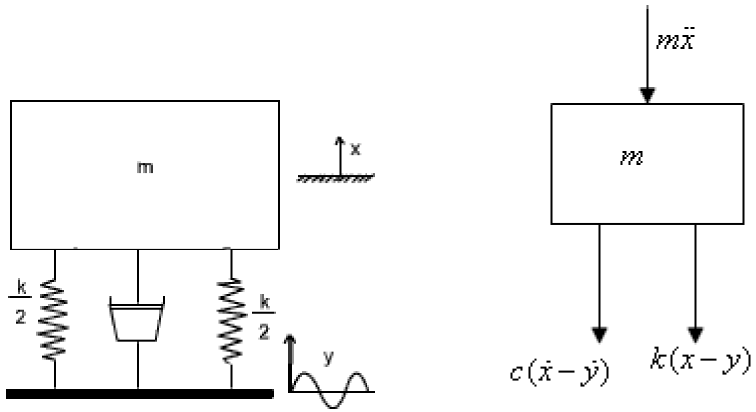

Figure 4.1: A system subjected to support motion   Figure 4.2: Freebody diagram

SUPPORT MOTION

- Figure 4.1: A system subjected to support motion and

  Figure 4.2: Freebody diagram
- Now to derive the equation of motion, from the freebody diagram of the mass as shown figure 2
- $$m\ddot{x} = -k(x-y) - c(\dot{x}-\dot{y}) \quad (4.1)$$
- let $z = x\text{-}y$ (4.2)
- $$m\ddot{z} + kz + c\dot{z} = -m\ddot{y} = m\omega^2 y \sin \omega t \quad (4.3)$$
- $$m\ddot{z} + kz + c\dot{z} = m\omega^2 y \sin \omega t \quad (4.4)$$
- As equation (4.4) is similar to equation (1) solution of equation (4.4) can be written as

$$z = Z \sin(\omega t - \phi) \quad (4.5)$$

$$Z = \frac{m\omega^2 y}{\sqrt{(k - m\omega^2)^2 + (c\omega)^2}}$$ and (6) $$\tan\phi = \frac{c\omega}{k - m\omega^2} \quad (4.6)$$

SUPPORT MOTION

- If the absolute motion x of the mass is required, we can solve for $x = z + y$.
- Using the exponential form of harmonic motion
- $$y = Ye^{i\omega t} \qquad (4.7)$$
- $$z = Ze^{i(\omega t-\varphi)} = \left(Ze^{-i\varphi}\right)e^{i\omega t} \qquad (4.8)$$
- $$x = Xe^{i(\omega t-\psi)} = \left(Xe^{-i\psi}\right)e^{i\omega t} \qquad (4.9)$$
- Substituting equation (4.9) in (4.1) one obtains
- $$\left\{m\left(Ze^{-i\phi}\right)\omega^2 + k\left(Ze^{-i\phi}\right) + ci\omega\left(Ze^{-i\phi}\right)\right\}e^{i\omega t} = m\omega^2 Ye^{i\omega t} \qquad (4.10)$$
- $$Ze^{-i\phi}\left(k - m\omega^2 + ic\omega\right) = m\omega^2 Y \qquad (4.11)$$
- $$Ze^{-i\phi} = \frac{m\omega^2 Y}{k - m\omega^2 + ic\omega} \qquad (4.12)$$
- $$x = \left(Ze^{-i\phi} + Y\right)e^{i\omega t} \qquad (4.13)$$
- $$x = \left(\frac{k - m\omega^2 + ic\omega + m\omega^2}{k - m\omega^2 + ic\omega}\right)Ye^{i\omega t} = X\left(\cos\psi - i\sin\psi\right)e^{i\omega t} \qquad (4.14)$$

SUPPORT MOTION

- The steady state amplitude and Phase from this equation are

$$\left|\frac{X}{Y}\right| = \sqrt{\frac{k^2 + (c\omega)^2}{(k - m\omega^2)^2 + (c\omega)^2}} \qquad (4.15)$$

$$\left|\frac{X}{Y}\right| = \sqrt{\frac{1 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2}{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2}} \qquad (4.16)$$

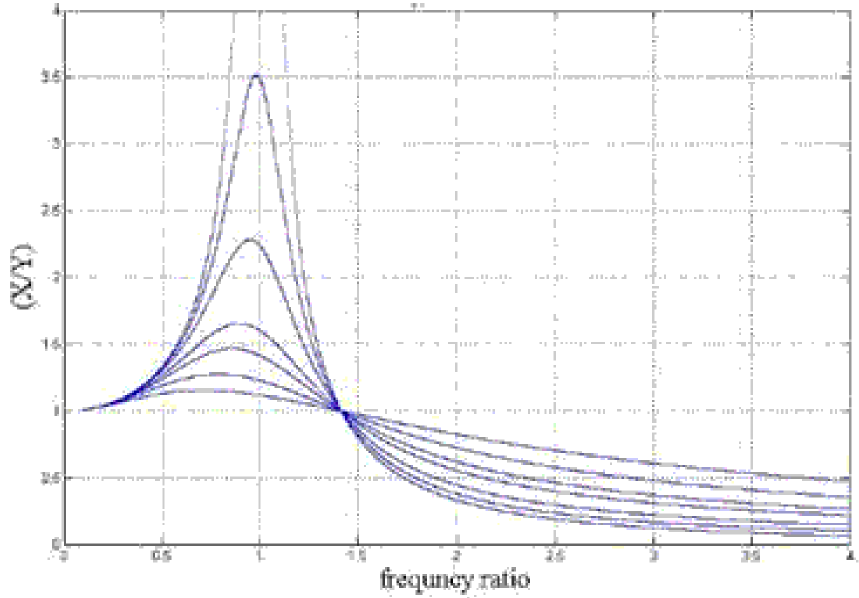

Figure 4.3: Amplitude ratio ~ frequency ratio plot for system with support motion

SUPPORT MOTION

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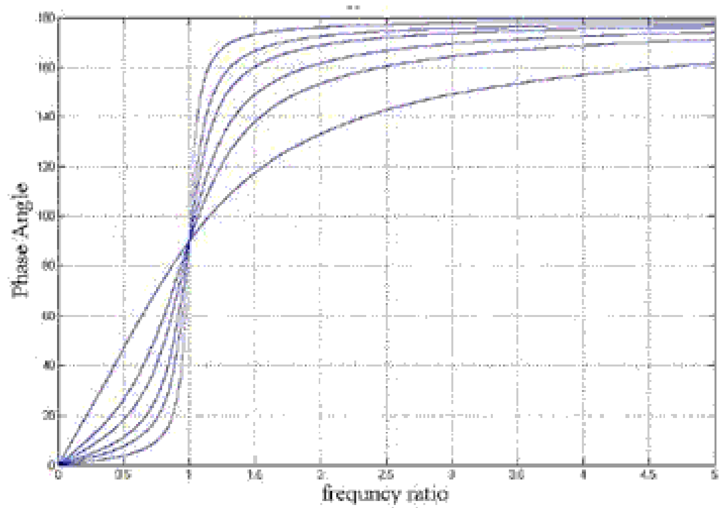

Figure 4.4: Phase angle ~ frequency ratio plot for system with support motion

From figure 4.3, it is clear that when the frequency of support motion nearly equal to the natural frequency of the system, resonance occurs in the system.
This resonant amplitude decreases with increase in damping ratio for $\frac{\omega}{\omega_n} < \sqrt{2}$. At $\frac{\omega}{\omega_n} = \sqrt{2}$, irrespective of damping factor, the mass vibrate with an amplitude equal to that of the support and for $\frac{\omega}{\omega_n} > \sqrt{2}$, amplitude ratio becomes less than 1, indicating that the mass will vibrate with an amplitude less than the support motion. But with increase in damping, in this case, the amplitude of vibration of the mass will increase. So in order to reduce the vibration of the mass, one should operate the system at a frequency very much greater than $\sqrt{2}$ times the natural frequency of the system. This is the principle of vibration isolation.

VIBRATION ISOLATION

- In many industrial applications, one may find the vibrating machine transmit forces to ground which in turn vibrate the neighbouring machines. So in that contest it is necessary to calculate how much force is transmitted to ground from the machine or from the ground to the machine.

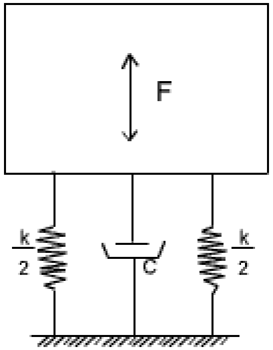

Figure 4.5 : A vibrating system

Figure 4.5 shows a system subjected to a force $F = F_0 \sin \omega t$

and vibrating with $x = X \sin(\omega t - \phi)$

.

This force will be transmitted to the ground only by the spring and damper.

Force transmitted to the ground

$$F_t = \sqrt{(KX)^2 + (c\omega X)^2} = KX\sqrt{1 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2} \quad (4.18)$$

It is known that for a disturbing force $F = F_0 \sin \omega t$

, the amplitude of resulting oscillation

$$X = \frac{\frac{F_0}{K}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2}} \quad (4.19)$$

VIBRATION ISOLATION

- Substituting equation (4.19) in (4.18) and defining the transmissibility TR as the ratio of the force transmitted Force to the disturbing force one obtains

$$\left|\frac{F_t}{F_0}\right| = \sqrt{\frac{1+\left(\frac{2\zeta\omega}{\omega_n}\right)^2}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left(\frac{2\zeta\omega}{\omega_n}\right)^2}} \qquad (4.20)$$

- Comparing equation (4.20) with equation (4.17) for support motion, it can be noted that

$$TR = \left|\frac{F_t}{F_0}\right| = \left|\frac{X}{Y}\right| \qquad (4.21)$$

- When damping is negligible

$$TR = \frac{1}{\left(\frac{\omega}{\omega_n}\right)^2 - 1} \qquad (4.22)$$

VIBRATION ISOLATION

- $\frac{\omega}{\omega_n}$ to be used always greater than $\sqrt{2}$
- Replacing $\omega_n^2 = \frac{g}{\Delta}$
- $$TR = \frac{1}{(2\pi f)^2 \frac{\Delta}{g} - 1} \qquad (4.23)$$
- To reduce the amplitude *X* of the isolated mass m without changing TR, *m* is often mounted on a large mass *M*. The stiffness *K* must then be increased to keep ratio *K/(m+M)* constant. The amplitude *X* is, however reduced, because *K* appears in the denominator of the expression

VIBRATION ISOLATION

$$X = \frac{\frac{F_0}{K}}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left(\frac{2\zeta\omega}{\omega_n}\right)^2}} \qquad (4.24)$$

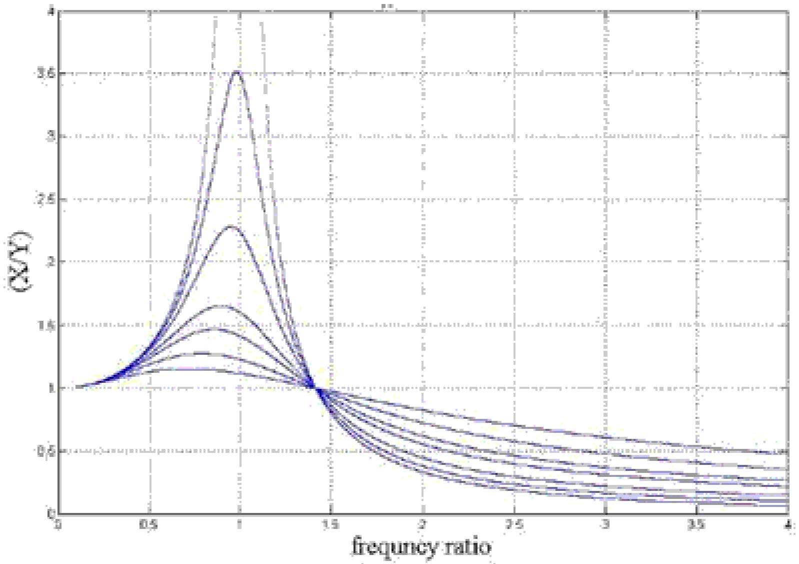

Figure 4.6: Transmissibility ~frequency ratio plot

VIBRATION ISOLATION

- Figure 4.6 shows the variation transmissibility with frequency ratio and it can be noted that vibration will be isolated when the system operates at a frequency ratio higher than $\sqrt{2}$

Equivalent Viscous Damping

- It is assumed that the energy dissipation takes place due to viscous type of damping where the damping force is proportional to velocity. But there are systems where the damping takes place in many other ways. For example, one may take surface to surface contact in vibrating systems and take Coulomb friction into account. In these cases one may still use the derived equations by considering an equivalent viscous damping. This can be achieved by equating the energy dissipated in the original and the equivalent system.
- The primary influence of damping on the oscillatory systems is that of limiting the amplitude at resonance. Damping has little influence on the response in the frequency regions away from resonance. In case of viscous damping, the amplitude at resonance is
- (4.25)

$$X = \frac{F_0}{c\,\omega_n} = \frac{F_0}{2\zeta k}$$

Where W_d must be evaluated from the particular type of damping

$$\pi C_{eq} \omega X^2 = W_d \qquad (4.26)$$

Structural Damping

- When materials are cyclically stressed, energy is dissipated internally within the material itself. Internal damping fitting this classification is called solid damping or structural damping. With the energy dissipation per cycle proportional to the square of the vibration amplitude, the loss coefficient is a constant and the shape of the hysteresis curve remains unchanged with amplitude and independent of the strain rate. Energy dissipated by structural damping can be written as
- (4.27)

$$W_d = \alpha X^2$$

- Where α is a constant with units of force displacement.
- By the concept of equivalent viscous damping
- or (4.28)

$$W_d = \alpha X^2 = \pi c_{eq} \omega X^2 \qquad c_{eq} = \frac{\alpha}{\pi \omega}$$

Coulomb Damping

- Coulomb damping is mechanical damping that absorbs energy by sliding friction, as opposed to viscous damping, which absorbs energy in fluid, or viscous, friction. Sliding friction is a constant value regardless of displacement or velocity. Damping of large complex structures with non-welded joints, such as airplane wings, exhibit coulomb damping.
- Work done per cycle by the Coulomb force F_d
- $$W_d = 4F_dX \qquad (4.29)$$
- For calculating equivalent viscous damping
- $$\pi C_{eq} \omega X^2 = 4F_dX \qquad (4.30)$$
- From the above equation equivalent viscous damping is found
- $$c_{eq} = \frac{4F_d}{\pi \omega X} \qquad (4.31)$$

Summary

- Some important features of steady state response for harmonically excited systems are as follows-
- The steady state response is always of the form $x(t) = X\sin(\omega t - \phi)$ Where it is having same frequency as of forcing. *X* is amplitude of the response, which is strongly dependent on the frequency of excitation, and on the properties of the spring-mass system.
- There is a phase lag ϕ between the forcing and the system response, which depends on the frequency of excitation and the properties of the spring-mass system.
- The steady state response of a forced, damped, spring mass system is independent of initial conditions.
- In this chapter response due to rotating unbalance, support motion, whirling of shaft and equivalent damping are also discussed.