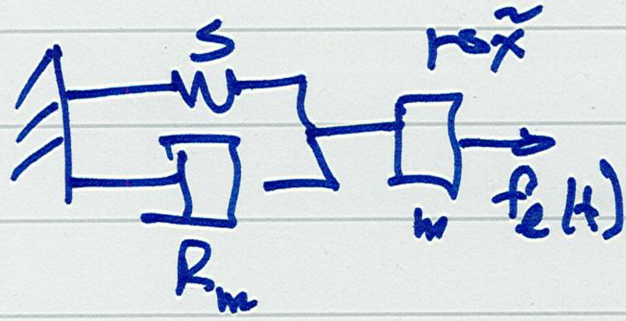
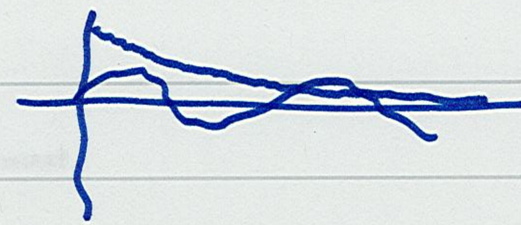
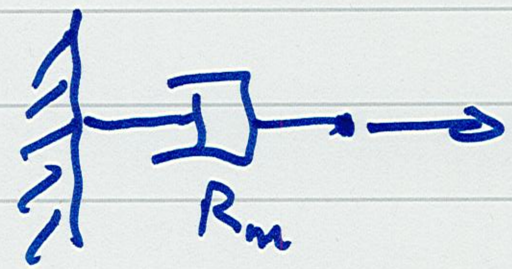




$$f_e(t) = F_e{}^{j\omega t}$$

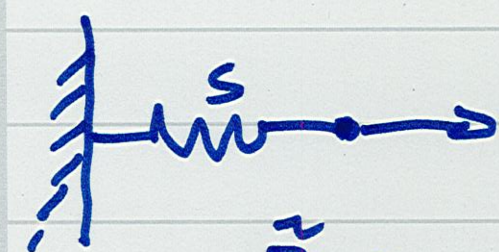
$$\tilde{x} = \frac{F_e{}^{j\omega t}}{j\omega\left(R_m + j\left(\omega m - \frac{s}{\omega}\right)\right)}$$

$$\tilde{Z}_m = \underbrace{R_m}_{\text{resistance}} + j\underbrace{\left(\omega m - \frac{s}{\omega}\right)}_{\text{reactance}}$$



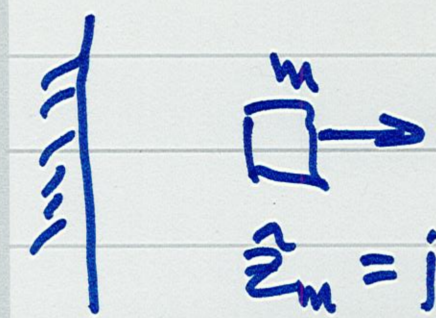
$\tilde{f}_r = Fe^{j\omega t}$

$\hat{z}_m = R_m$ $\underline{\text{real}}$ & $\underline{\text{positive}}$



$\tilde{f}_r = Fe^{j\omega t}$

$\tilde{z}_m = -j\frac{s}{\omega}$ imaginary, negative
inversely proportional to freq



$\tilde{f}_r = Fe^{j\omega t}$

$\hat{z}_m = j\omega m$ imaginary, positive
+ linearly prop. to freq

1.3.4 Mechanical ~~Response~~ Resonance

Defn: Occurs when the imaginary component of the mechanical impedance $\rightarrow 0$

$$\mathrm{Im}\{\tilde{z}_m\} = 0$$

$$\tilde{z}_m = R_m + i\left(\omega m - \frac{s}{\omega}\right) \qquad \tilde{x}_m = \mathrm{Im}\{\tilde{z}_m\} = \omega m - \frac{s}{\omega} = 0$$

$$\tilde{x}_m = 0 \text{ when } \omega = \omega_0 = \sqrt{\frac{s}{m}}$$

undamped natural freq.

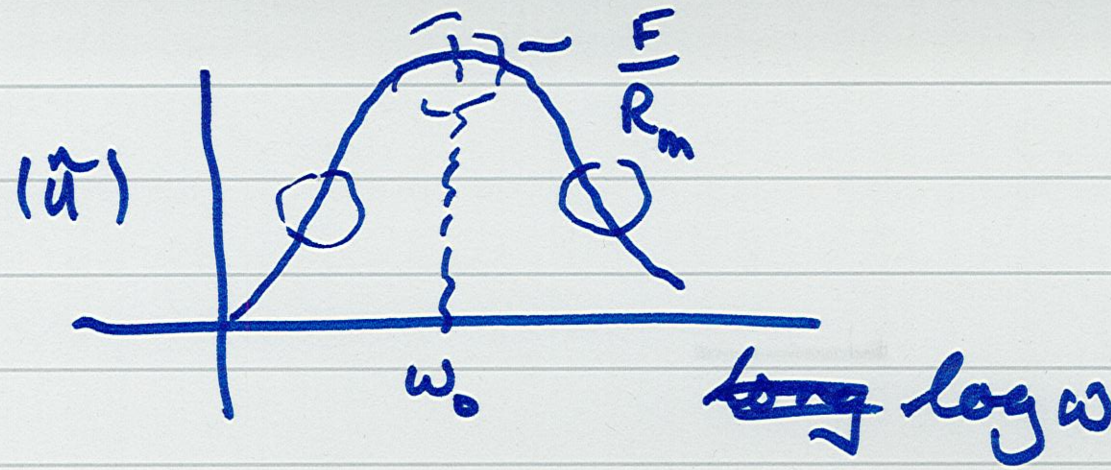
Resonance occurs when the system is driven at its undamped natural freq.

$$\hat{u} = \frac{F e^{i\omega t}}{R_m + j\underbrace{\left(\omega m - \frac{s}{\omega}\right)}_{=0 \text{ when } \omega = \omega_0}}$$

drive at resonance

velocity → $\underline{\underline{\text{max}}}$

and $\tilde{Z}_m$ is purely real



3 frequency ranges of interest

$\omega < \omega_0$

$\omega \simeq \omega_0$

$\omega > \omega_0$

(i) $\omega < \omega_0$ $\qquad$ $\tilde{Z}_m = R_m + j\left(\omega m - \frac{s}{\omega}\right)$

stiffness controlled region

$\tilde{Z}_m \simeq -j\frac{s}{\omega}$ $\qquad$ $\tilde{x} \simeq \frac{F}{s} e^{j\omega t}$

$|\tilde{x}|$ is independent of frequency.

(ii) near resonance $\omega \simeq \omega_0$

$\tilde{Z}_m \simeq R_m$ $\qquad$ $\tilde{x} = \frac{F}{j\omega R_m} e^{j\omega t}$

$\tilde{z}$

$\tilde{u} = j\omega\tilde{x} = \frac{F}{R_m} e^{j\omega t}$ $\qquad R_m + j\left(\omega m - \frac{s}{\omega}\right)$

$|\tilde{u}| \simeq \frac{F}{R_m}$ independent of frequency

"damping controlled region"

(iii) $\omega > \omega_0$ mass controlled region

$\tilde{z}_m \simeq j\omega m$

$\tilde{a} \simeq \frac{F}{m} e^{j\omega t}$

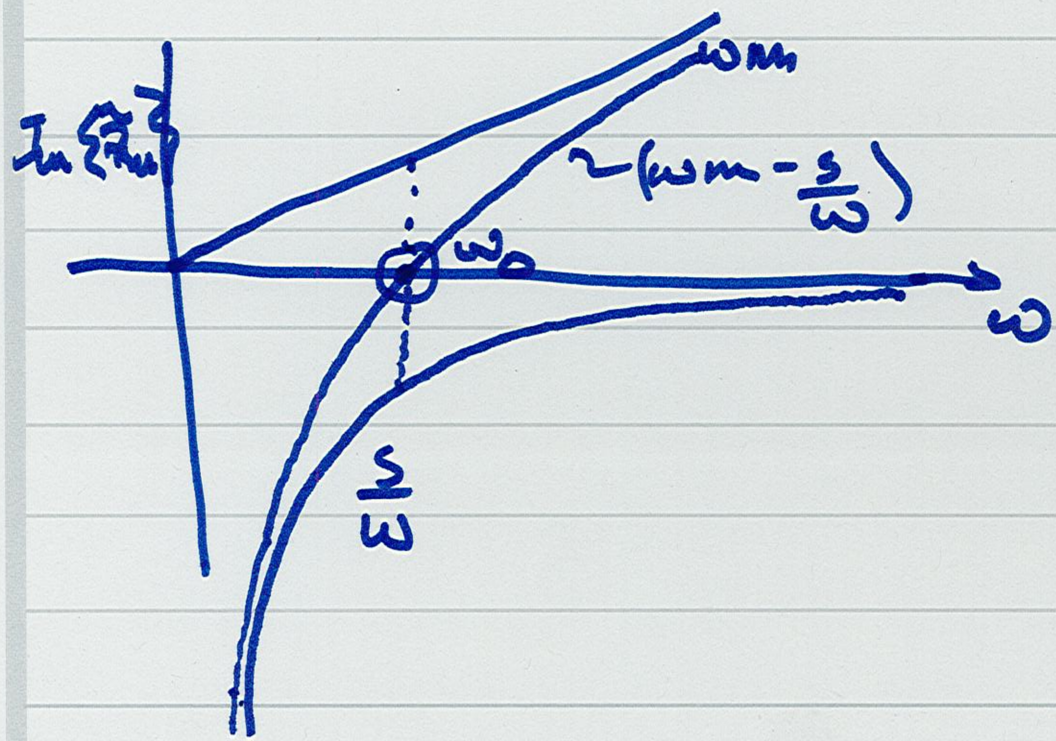
Transducers are designed to
operate in one of the ranges

$\tilde{Z}_m \simeq j\omega m$

- imag
- positive
- linearly prop to freq

$Z_m \simeq -j\frac{s}{\omega}$

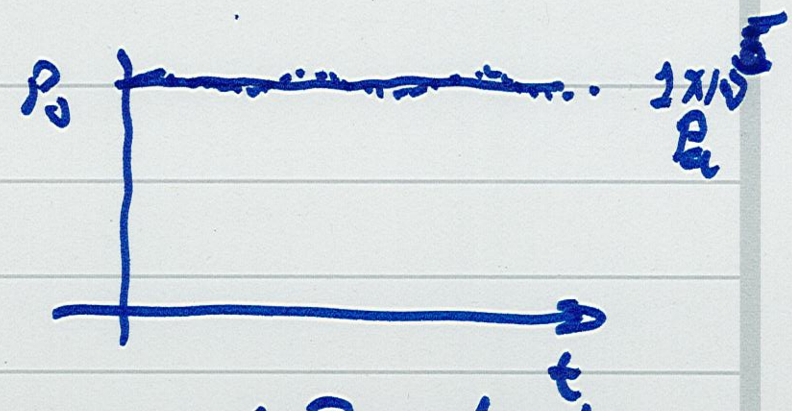
- imag
- negative
- inversely prop to freq.



1.4 superposition

"linear" acoustic

"linear" vibration



1 Pa-level sound.

small amplitude fluctuations in sound pressure or velocity

linear, small amplitude response

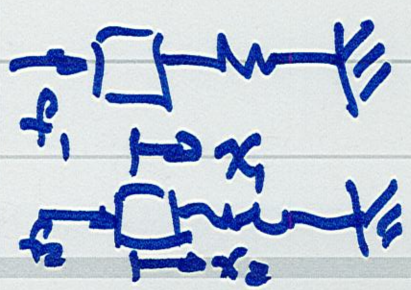
- output of a linear system is linearly proportional to the input

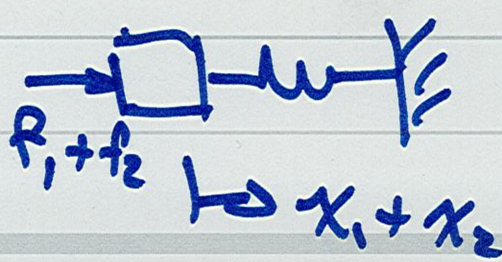
- double the force - response doubles

- linear system responds at the forcing frequency

$$F e^{i\omega t} \longrightarrow \tilde{A} e^{i\omega t}$$

- response to two or more inputs is linear sum of the individual responses





- convenient to break up input forces into frequency components (frequency analysis)
- find the response to each frequency component
- sum up the individual responses to give the total response.

Frequency domain analysis