SOLUTIONS

Final Examination for MECH 364 Mechanical Vibrations

8th December 2011

Time: 2 hrs. 30 mts. Max. Available Mark: 60

READ THE FOLLOWING INSTRUCTIONS CAREFULLY

1. This exam consists of 5 pages including this page
2. Please write your name and student number on the answer sheets
3. ANSWER ALL QUESTIONS
4. One letter-sized formula sheet is allowed

The space below is intentionally left blank. Continue onto the next page for the exam questions.

MARKS ALLOCATED FOR EACH SEGMENT ARE IN THE MARGINS.

PAGE NUMBERS ARE ON TOP RIGHT.

2

Question 1

Concepts tested : *FBD, Initial Conditions, Equivalent Systems, Free Vibration Response.*

(a) A container of mass m is being lowered from a helicopter at a constant downward velocity of V when the helicopter is hovering above the target point of delivery. To avoid a mishap the weight is suddenly stopped by the operator at point O in Figure(1). Assuming that all cables have an area of cross-section A, formulate the equations of motion for the vertical oscillations of the mass. Note that the axial stiffness of a cable made from a material of modulus E, of length L and area of cross-section A is $k = \frac{AE}{L}$. **(12 marks)**

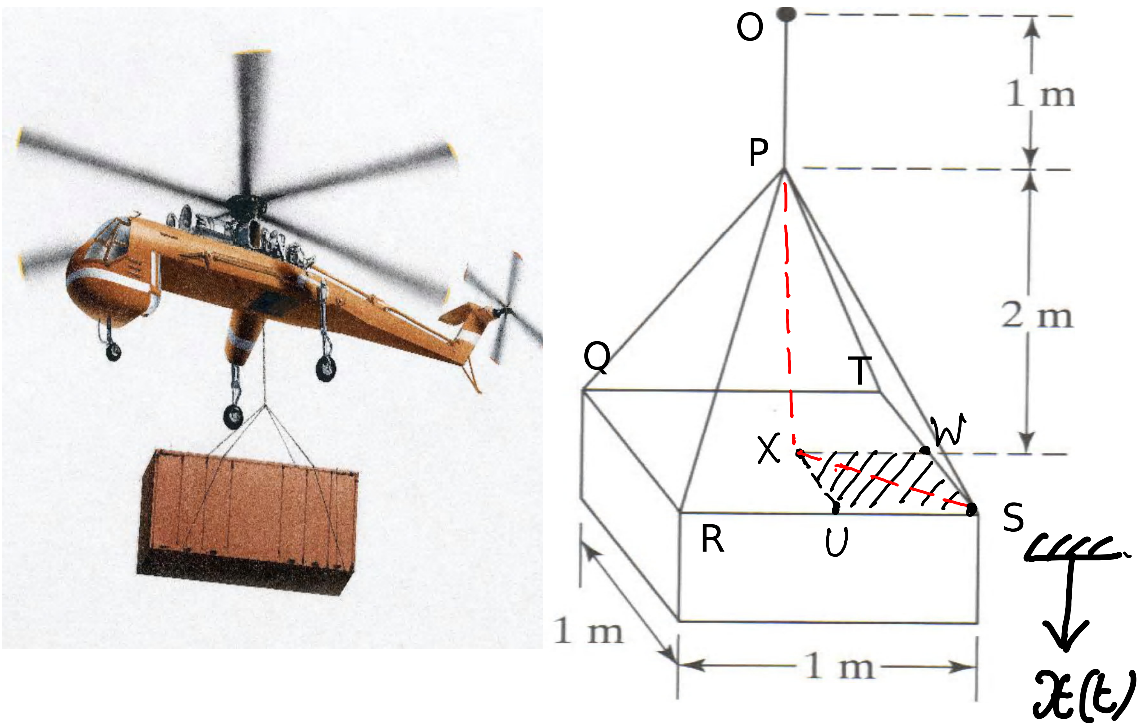


Figure 1: Figure for question 1. Point O originates at the helicopter.

(b) Find the maximum displacement amplitude of vertical oscillations of the container for the geometric parameters shown in the left side of Figure(1). Assume $E = 210$ GPa, $M = 200$ kg, $V = 1$ m/s and the diameter of wires is 5 cm. **(6 marks)**

(b) List at least two limitations of the above modelling approach. **(2 marks)**

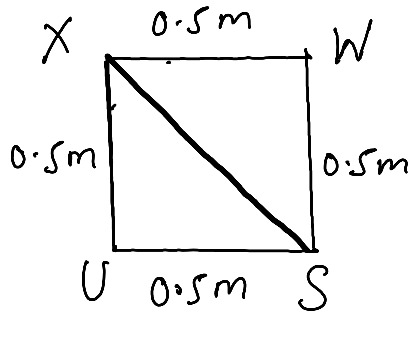
QUESTION 1

a) MODEL EACH CABLE AS A SPRING OF CONSTANT $K = \frac{AE}{L}$

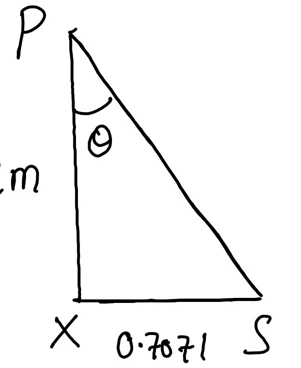
WE CAN SEE THAT THE SPRINGS ASSOCIATED WITH THE CABLES PQ, PR, PS AND PT ARE IN PARALLEL AS THEY ALL UNDERGO SAME RELATIVE DISPLACEMENT. THE RESULTANT EQUIVALENT SPRING IS CONNECTED IN SERIES WITH THE SPRING ASSOCIATED WITH THE CABLE OP. WE MAKE USE OF THESE FACTS.

WE SELECT VERTICALLY DOWNWARD DISPLACEMENT x (SEE FIG. 1 ON PAGE #2) AS OUR CO-ORDINATE.

THE ANGLE MADE BY SPRINGS PQ, PR, PS, PT WITH RESPECT TO THE DISPLACEMENT CO-ORDINATE, x, IS OBTAINED AS FOLLOWS. FROM THE SQUARE 'USWX'



$$SX = \sqrt{SW^2 + WX^2}$$
$$= \sqrt{0.5^2 + 0.5^2} = 0.7071\ m$$



FROM $\triangle$ PXS

$$\tan\theta = \frac{XS}{PX} = \frac{0.7071}{2}$$

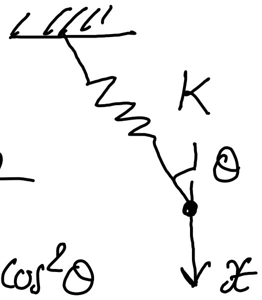
$\Rightarrow \theta = 19.471°$

$\Rightarrow \cos\theta = 0.9428$ & $PS = PX/\cos\theta = 2.1213\ m$

NOW, THE EQUIVALENT SPRING ASSOCIATED WITH THE 4 INCLINED CABLES CAN BE OBTAINED AS

$$K_{INCLINED} = 4K\cos^2\theta$$

USING THE FOLLOWING FIGURE



RESOLVE x ALONG SPRING

$$P.E. = \frac{1}{2}K_{eq}x^2 = \frac{1}{2}K(x\cos\theta)^2$$

$$= \frac{1}{2}K\cos^2\theta\, x \Rightarrow K_{eq} = K\cos^2\theta$$

FOR EACH INCLINED CABLE MODELLED AS A SPRING OF CONSTANT $K = AE/L = \frac{AE}{PS} = \frac{AE}{2.1213}$ N/m

NOW THE RESULTANT SPRING ASSOCIATED WITH THE 4 INCLINED CABLES IS IN SERIES WITH $K_{vertical}$ WHERE $K_{VERTICAL} = \frac{AE}{OP} = \frac{AE}{1} = AE$ N/m

THE EQUIVALENT SPRING OF 4 INCLINED SPRINGS IN SERIES WITH $K_{VERTICAL}$ IS

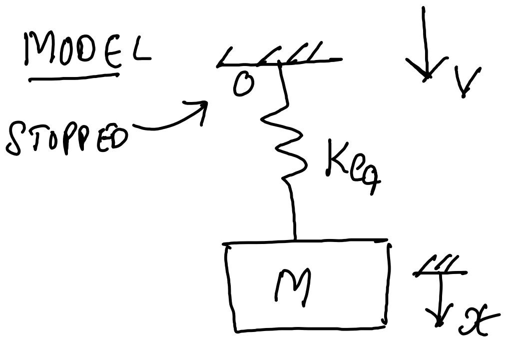
$$\frac{1}{K_{eq}} = \frac{1}{K_{INCLINED}} + \frac{1}{K_{VERTICAL}}$$

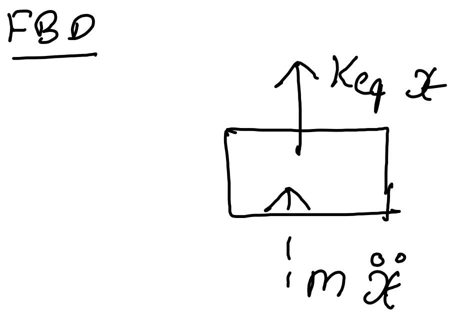
$$\Rightarrow \frac{1}{K_{eq}} = \frac{1}{4K\cos^2\theta} + \frac{1}{AE} = \frac{1}{4\frac{AE}{2.1213}(0.9428)^2} + \frac{1}{AE}$$

$$\Rightarrow \boxed{K_{eq} = 0.6263\, AE}$$

8 MARKS

WITH THE ABOVE WE CAN SET-UP THE MODEL.





x IS MEASURED WITH RESPECT TO EQUILIBRIUM WHERE GRAVITATIONAL WEIGHT mg CANCELS STATIC DEFLECTION IN "SPRING" K_{eq}.

EQUATION OF MOTION: $\downarrow \sum_{+ve} F_x = 0$ (D'ALEMBERT)

$$\Rightarrow -m\ddot{x} - K_{eq}\,x = 0$$

$$\Rightarrow \boxed{m\ddot{x} + K_{eq}\,x = 0, \quad K_{eq} = 0.6263\,AE} \quad \text{— (1)}$$

4 MARKS

b) WHEN 'O' IS STOPPED

$$x(0) = 0, \quad \dot{x}(0) = V \quad \text{— (2)}$$

2 MARKS

THE VIBRATION RESPONSE IS OBTAINED BY SOLVING (1)

$$x = x_h + \cancel{x_p}^{\,0} = x_h = A\cos\omega_n t + B\sin\omega_n t \quad \text{— (3)}$$

(2) IN (3) GIVE: $x(0) = 0 \Rightarrow A = 0$

$$\dot{x}(0) = 0 \Rightarrow B = \frac{V}{\omega_n}$$

2 MARKS

$\therefore \quad x = x_h = \dfrac{V}{\omega_n}\sin\omega_n t; \quad \omega_n = \sqrt{\dfrac{K_{eq}}{M}} = \sqrt{\dfrac{0.6263\,AE}{M}}$

b) NOW MAXIMUM DISPLACEMENT AMPLITUDE IS

$$x_{max} = \frac{V}{\omega_n}$$

GIVEN $V = 1\,m/s$, $E = 210 \times 10^9\ Pa$,

d = DIAMETER OF CABLE $= 5 \times 10^{-2}\ m$

A = AREA OF C/S OF CABLE $= \frac{\pi d^2}{4} = \frac{\pi \times (5\times10^{-2})^2}{4}$

$= 0.002\ m^2$

$K_{eq} = 0.6263\ AE = 0.6263 \times 0.002 \times 210 \times 10^9$

$= 2.5824 \times 10^8\ N/m$

M = MASS OF CONTANER = 200 kg

$$\omega_n = \sqrt{\frac{K_{eq}}{M}} = \sqrt{\frac{2.5824 \times 10^8}{200}} = 1.1363 \times 10^3\ rad/s$$

2 MARKS

$$\boxed{x_{max} = \frac{V}{\omega_n} = \frac{1}{1.1363 \times 10^3} = 8.8 \times 10^{-4}\ m = 0.88\ mm}$$

c) 2 MARKS

LIMITATIONS

(1) CONTAINER IS A POINT MASS i.e. NO MASS MOMENT OF INERTIA

(2) MOTION OF HELICOPTER IGNORED

(3) WIND RESISTANCE/EXCITATION IGNORED

(4) CABLES ARE RIGIDLY ATTACHED TO EACH OTHER AND TO THE MASS.

Question 2

Concepts tested : *Isolation System Design, Vibration Absorbers*

a) Explain in two sentences or less how isolation systems and absorber systems work. **(4 marks)**

b) An electronic control unit of mass $m = 2$ kg is located in the stores pod of a space mission shown below in Figure(2). It needs to be isolated from vibration inputs originating at the base in the form of acceleration $\ddot{x}$ as shown. The acceleration input $\ddot{x}$ is a *broad band* random vibration within the range 10 Hz and 1 kHz. It is required to limit the maximum displacement transmission ratio around *resonance* to 2, and the transmitted displacement ratio not to exceed 0.2 in the *entire* input frequency range. Design the isolator stiffness and damping. Which other design criterion would you consider in addition to the above? **(14 marks)**

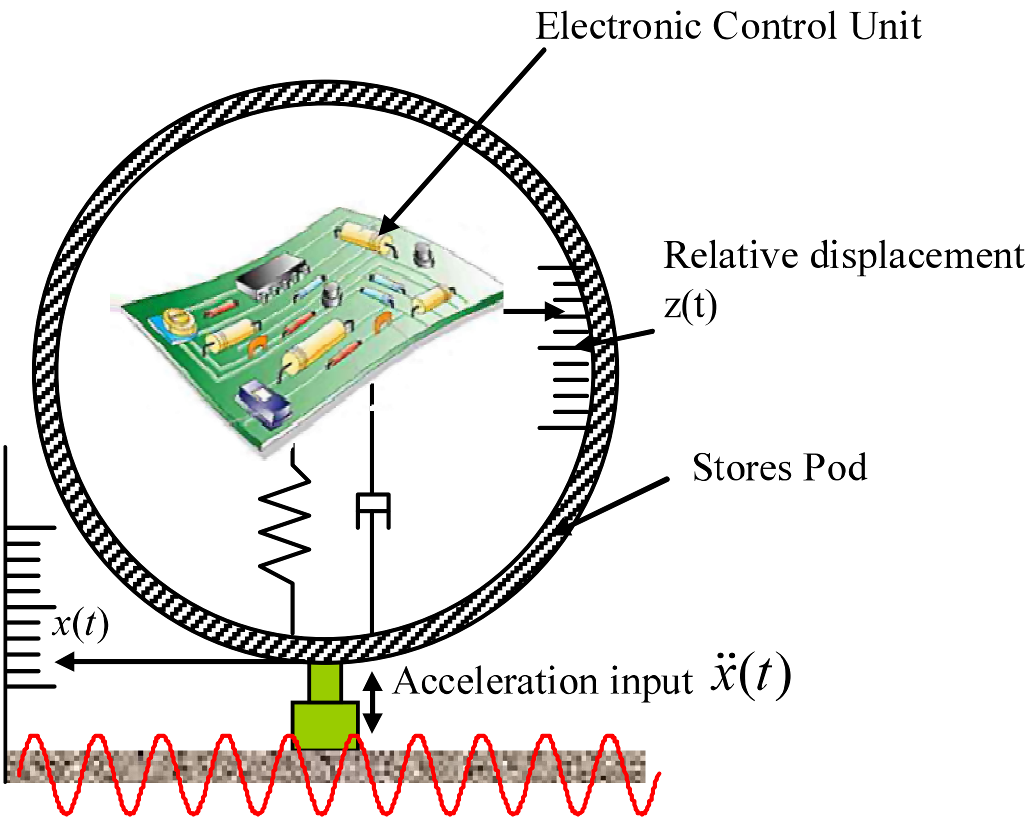


Figure 2: Figure for Question 2.

b) What is the relation between displacement, velocity, and acceleration transmission ratios at a *steady* operating frequency ω? **(2 marks)**

a) ISOLATION SYSTEMS COMPRISE A SPRING-DAMPER ARRANGEMENT (RESILIENT MATERIAL SUCH AS RUBBER) INSERTED IN THE VIBRATION TRANSMISSION PATH SUCH THAT THE NATURAL FREQUENCY OF THE SYSTEM WITH ISOLATOR IS WELL BELOW THE FORCING FREQUENCY. THIS LEADS TO REDUCED TRANSMITTED VIBRATION OR ISOLATION. THE DYNAMICS OF THE MAIN SYSTEM IS SLOWED DOWN IN RELATION TO THE EXCITATION FREQUENCY AT THE EXPENSE OF STIFFNESS REDUCTION.

2 MARKS

A VIBRATION ABSORBER IS AN AUXILARY SYSTEM ATTACHED TO THE MAIN SYSTEM SUCH THAT, AT TUNED FREQUENCIES, THE ABSORBER EXERTS A LARGE COUNTER VIBRATING FORCE AT THE POINT OF ATTACHMENT. VIBRATION REDUCTION IN THE MAIN SYSTEM IS ACHIEVED AT THE EXPENSE OF

(i) INTRODUCING TWO ADDITIONAL RESONANCES, AND

(ii) SUBJECTING THE ABSORBER SPRING-MASS UNIT TO UNDERGO SUSTAINED RESONANT OSCILLATIONS.

2 MARKS

b) GIVEN $m = 2\,kg$; $10\,Hz < \omega < 1000\,Hz \Rightarrow 20\pi < \omega < 2000\pi$ rad/s

$$TR = TR_d = \frac{\sqrt{1+(2\zeta r)^2}}{\sqrt{(1-r^2)^2+(2\zeta r)^2}} \qquad r = \frac{\omega}{\omega_n}$$

$$[TR]_{max} = [TR]_{r\approx 1} = \sqrt{\frac{1+4\zeta^2}{4\zeta^2}} = 2$$

2 MARKS

$$\Rightarrow 1+4\zeta^2 = 4\,(4\zeta^2) \Rightarrow \zeta = \frac{1}{\sqrt{12}}$$

TR = 0.2 IN THE RANGE $20\pi < \omega < 2000\pi$ rad/s

START WITH $\omega = 20\pi$ rad/s

$$TR = 0.2 = \sqrt{\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2}} \quad ; \text{ CALL } r^2 = x$$

SQUARING BOTH SIDES & RE-ARRANGING

$$\Rightarrow \left[(1-x)^2 + 4\zeta^2 x\right](TR)^2 = 1+4\zeta^2 x$$

$$\Rightarrow (TR)^2 x^2 + x\left[4\zeta^2 (TR)^2 - 2(TR)^2 - 4\zeta^2\right] + (TR)^2 - 1 = 0$$

WITH $\zeta = \frac{1}{\sqrt{12}}$ & $TR = 0.2$

$$\Rightarrow 0.04x^2 - 0.4x - 0.96 = 0$$

$$\Rightarrow x = \frac{0.4 \pm \sqrt{(0.4)^2 + 4 \times 0.04 \times 0.96}}{2 \times 0.04} = 12 \text{ or } -2$$

IGNORE

$$x = r^2 = 12 \Rightarrow r = \frac{\omega}{\omega_n} = \sqrt{12} \Rightarrow \omega_n = \frac{\omega}{\sqrt{12}}$$

$$\Rightarrow \omega_n = \sqrt{\frac{K}{m}} = \frac{\omega}{r} = \frac{20\pi}{\sqrt{12}} \text{ rad/s}$$

$$\Rightarrow K = m\left(\frac{20\pi}{\sqrt{12}}\right)^2 = 2\left(\frac{20\pi}{\sqrt{12}}\right)^2$$

$$\therefore K = 657.9736 \text{ N/m}$$

GIVEN $\zeta = \frac{C}{2M\omega_n} = \frac{1}{\sqrt{12}}$

8 MARKS

$$C = 2 \frac{1}{\sqrt{12}} M\omega_n = 2 \frac{1}{\sqrt{12}} \sqrt{KM}$$

$$= \frac{2}{\sqrt{12}} \sqrt{657.9736 \times 2} = 20.944 \ \frac{\text{N-s}}{\text{m}}$$

2 MARKS

CHECK: FOR $\omega = 2000\pi$ rad/s & $\zeta = \frac{1}{\sqrt{12}}$, $\omega_n = \frac{20\pi}{\sqrt{12}}$ rad/s

$$r = \frac{\omega}{\omega_n} = \frac{2000\pi}{20\pi}\sqrt{12} = 346.41$$

$$TR = \sqrt{\frac{1 + (2\zeta r)^2}{(1-r^2)^2 + (2\zeta r)^2}} = 0.0017 << 0.2 \text{ O.K.}$$

$$\boxed{\text{SELECT } K = 658 \text{ N/m AND } C = 21 \ \frac{\text{N-s}}{\text{m}}}$$

b) 2 MARKS

ANOTHER ADDITIONAL DESIGN CRITERION IS MAXIMUM PERMITTED RELATIVE DISPLACEMENT $Z(t)$.

c) IN A STEADY HARMONIC MOTION AT FREQUENCY ω rad/s

$$X_{VEL} = \omega X \quad ; \quad X_{accln} = -\omega^2 X$$

$$Y_{VEL} = \omega Y \quad ; \quad Y_{accln} = -\omega^2 Y$$

[RECALL "SHAKYTABLE" LAB]

$$TR_d = \left|\frac{Y}{X}\right|$$

2 MARKS

$$TR_{VEL} = \left|\frac{\omega Y}{\omega X}\right| = \left|\frac{Y}{X}\right| = TR_d$$

$$TR_{accln} = \left|\frac{-\omega^2 Y}{-\omega^2 X}\right| = \left|\frac{Y}{X}\right| = TR_d$$

$$\therefore \quad \boxed{TR_d = TR_{VEL} = TR_{accln}} \text{ IN STEADY STATE}$$

Question 3

Concepts tested : *Kinematics, FBD, Forced Vibration, DMF, Shaky Table Laboratory*

Helpful hint: If you are unable to find the accelerations within 15 minutes in part a) you may proceed to part b).

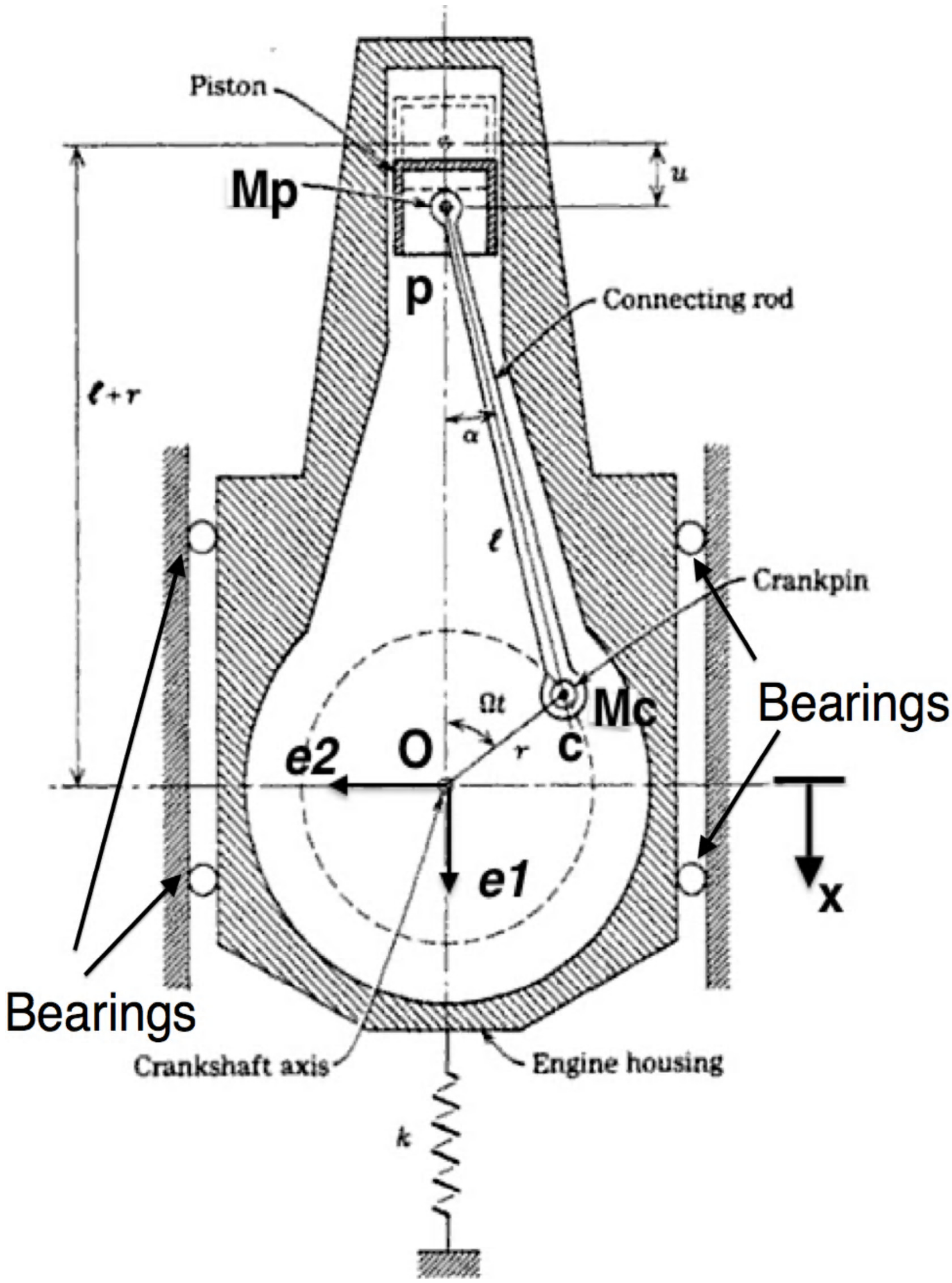


Figure 3: Figure for Question 3. Note the distances: $OC = r$, $CP = l$. All links are rigid and Ω is constant.

(a) A single cylinder engine with unbalanced masses lumped at the crank pin (M_c) and piston head (M_p) is shown in Figure(3). Using kinematics show that the *absolute inertial* acceleration of M_p is $\boldsymbol{a}_p \approx \ddot{x}\boldsymbol{e}_1 + \Omega^2 r \left[\cos\Omega t + \frac{r}{l}\cos 2\Omega t\right]\boldsymbol{e}_1$ and that of M_c is $\boldsymbol{a}_c = \ddot{x}\boldsymbol{e}_1 + r\Omega^2\cos\Omega t\boldsymbol{e}_1 + r\Omega^2\sin\Omega t\boldsymbol{e}_2$, Ω is **constant** and $+ve$ in clockwise direction. The unit vectors $\boldsymbol{e}_1$ and $\boldsymbol{e}_2$ are as shown in Figure(3). **You may find the identity $\sqrt{1-x} \approx 1 - \frac{1}{2}x$ useful in approximating the acceleration $\boldsymbol{a}_P$. You may also find the trigonometric identity $2\sin\theta\cos\theta = \sin 2\theta$ useful.** **(8 marks)**

(b) Using the accelerations from part a) construct the Free Body Diagram (FBD) for the engine housing (of mass M_e) and show that the governing equation of motion for vertical vibrations is $M\ddot{x} + kx = -(M_p + M_c)r\Omega^2\cos\Omega t - M_p\frac{r^2}{l}\Omega^2\cos 2\Omega t$ where $M = M_e + M_c + M_p$. Show that the steady state displacement amplitude of the forced vibration response, ignoring the homogeneous part, is $x(t) = -0.0029\cos\Omega t + 5.36\times 10^{-4}\cos 2\Omega t$ m for the parameters: $r = 0.2$ m, $l = 0.6$ m, $\Omega = 600$ rpm, $M_p = 3.2$ kg, $M_c = 0.9$ kg, $M = 227$ kg and $k = 2\times 10^6$ N/m. Can you explain the signs of displacement amplitudes in the above response? What is the influence of the ratio $\frac{r}{l}$ on the forces exerted by the unbalance masses. **(10 marks)**

(c) What is the **total** horizontal reaction force on the bearings between the engine housing and the rest of the body. **(2 marks)**

Parting thoughts ....

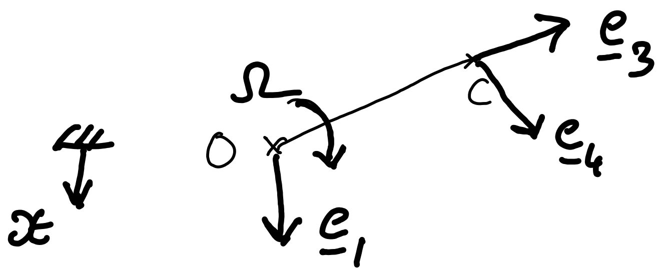
1. I enjoyed explaining this challenging but practically important material to you at 8AM every week!

2. Notice how difficult it is for an engineer to *intuitively* guess the forcing frequencies in Question 3 to avoid resonance for single cylinder engine, unless one knows how to *apply* kinematics and *basic* mathematics. The unexpected forcing at 2Ω is counter-intuitive! So are practical *engineering vibration* problems!!

ALL THE VERY BEST IN YOUR FUTURE ENDEAVOURS!

We shall not cease from exploration And the end of all our exploring Will be to arrive where we started And know the place for the first time. *–TS Eliot in Little Gidding.*

a) **ACCELERATION OF M_C**



$|\underline{e}_3| = |\underline{e}_4| = 1$

DISPLACEMENT WITH RESPECT TO A FIXED OBSERVER

$$\underline{r}_C = \underline{r}_O + \underline{r}_{C/O} = x\,\underline{e}_1 + r\,\underline{e}_3$$

$$\dot{\underline{r}}_C = \dot{x}\,\underline{e}_1 + r\Omega\,\underline{e}_4 \quad \text{(SUMMARY SHEET)}$$

r = FIXED

3 MARKS

$$\boxed{\begin{aligned} \underline{a}_C = \ddot{\underline{r}}_C &= \ddot{x}\,\underline{e}_1 - r\Omega^2\,\underline{e}_3 \\ &= \ddot{x}\,\underline{e}_1 + r\Omega^2\cos\Omega t\,\underline{e}_1 + r\Omega^2\sin\Omega t\,\underline{e}_2 \end{aligned}}$$

Note: $-\underline{e}_3 = +\underline{e}_1\cos\Omega t + \underline{e}_2\sin\Omega t$

ACCELERATION OF M_P:

DISPLACEMENT: $\underline{r}_P = \underline{r}_O + \underline{r}_{P/O}$

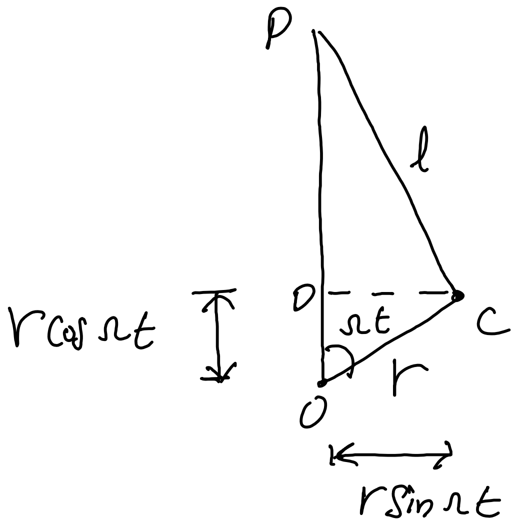
$$= x\,\underline{e}_1 - OP\,\underline{e}_1 \qquad \text{— (1)}$$

NOTICE THE −ve SIGN.

∵ P IS ABOVE 'O'

& $\underline{e}_1$ IS +ve DOWNWARDS!

FROM ΔOPC



$$OP = OD + DP = r\cos\Omega t + \sqrt{PC^2 - CD^2}$$

$$= r\cos\Omega t + \sqrt{l^2 - r^2\sin^2\Omega t}$$

$$= r\cos\Omega t + l\sqrt{1 - \left(\frac{r}{l}\right)^2 \sin^2\Omega t}$$

$$\simeq r\cos\Omega t + l - \frac{r^2}{2l}\sin^2\Omega t \quad - (2)$$

USING $\sqrt{1 - \left(\frac{r}{l}\right)^2\sin^2\Omega t} \simeq 1 - \frac{1}{2}\left(\frac{r}{l}\right)^2\sin^2\Omega t$

(2) IN (1) GIVES

$$\underline{r}_p \simeq x\,\underline{e}_1 - r\cos\Omega t\,\underline{e}_1 - l\,\underline{e} + \frac{r^2}{2l}\sin^2\Omega t$$

VELOCITY: $\dot{\underline{r}}_p \simeq \dot{x}\,\underline{e}_1 + r\Omega\sin\Omega t\,\underline{e}_1 + \frac{r^2}{2l}2\Omega\sin\Omega t\cos\Omega t\,\underline{e}_1$

$$= \dot{x}\,\underline{e}_1 + r\Omega\sin\Omega t\,\underline{e}_1 + \frac{r^2}{2l}\Omega\sin(2\Omega t)\,\underline{e}_1$$

NOTE: $2\sin\Omega t\cos\Omega t = \sin 2\Omega t$
(TRIG. IDENTITY)

ACCELERATION: $\underline{a}_p = \ddot{\underline{r}}_p$

$$= \ddot{x}\,\underline{e}_1 + r\Omega^2\cos\Omega t\,\underline{e}_1 + \frac{r^2}{2\ell}2\Omega^2\cos(2\Omega t)\,\underline{e}_1$$

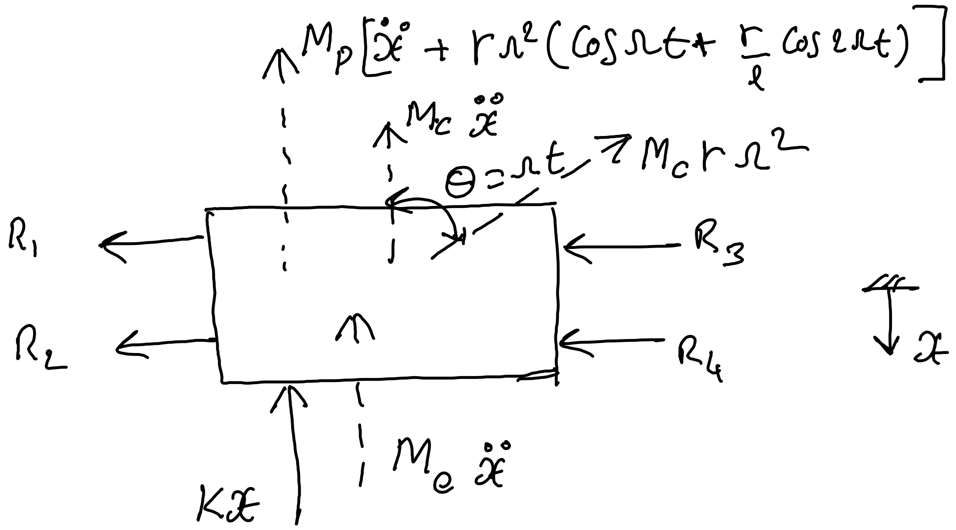
5 MARKS

$$\boxed{\underline{a}_p \approx \ddot{x}\,\underline{e}_1 + r\Omega^2\left[\cos\Omega t + \frac{r}{\ell}\cos(2\Omega t)\right]\underline{e}_1}$$

NOTE: THE APPROXIMATION ARISES FROM TRUNCATING THE SERIES EXPANSION OF $\sqrt{1-\left(\frac{r}{\ell}\right)^2\sin^2\Omega t}$ FOR SMALL $\left(\frac{r}{\ell}\right)$

b) FREE BODY DIAGRAM FOR M_e

5 MARKS



NOTE: GRAVITATIONAL WEIGHT CANCELS STATIC DEFLECTION IN THE SPRING AT STATIC EQUILIBRIUM.

$\downarrow \sum F_x = 0 \Rightarrow$
+ve

$$-M_e \ddot{x} - M_p\left(\ddot{x} + r\Omega^2 [\cos\Omega t + \frac{r}{l}\cos 2\Omega t]\right) - M_c \ddot{x}$$
$$- M_c r \Omega^2 \cos\Omega t - kx = 0$$

$$\Rightarrow (M_p + M_e + M_c)\ddot{x} + kx = -(M_p + M_c) r\Omega^2 \cos\Omega t$$
$$- M_p \frac{r^2}{l}\Omega^2 \cos 2\Omega t$$

STEADY STATE DISPLACEMENT:

$$x = x_p = -\frac{(M_p + M_c) r\Omega^2}{K - (M_p + M_e + M_c)\Omega^2}\cos\Omega t$$

$$- \frac{M_p r^2 \Omega^2}{l\,[K - (M_p + M_e + M_c)(2\Omega)^2]}\cos 2\Omega t$$

NOTE THIS!

WITH $M_p = 3.2$ kg ; $M_c = 0.9$ kg ; $M = M_p + M_e + M_c = 227$ kg

$\Omega = \frac{600 \times 2\pi}{60}$ rad/s ; $r = 0.2$ m, $l = 0.6$ m

3MARKS

$$\boxed{x = -0.0029\cos\Omega t + 5.36\times10^{-4}\cos 2\Omega t}$$

THE OPPOSITE SIGNS CAN BE EXPLAINED BASED ON THE FACT THAT $\omega_n = \sqrt{\frac{K}{M}} = \sqrt{\frac{2\times10^6}{227}} = 93.9$ rad/s

$\Omega = 600\,rpm = \frac{600}{60} \times 2\pi\ rad/s = 62.8\ rad/s$

(i) $\boxed{\omega_n > \Omega}$ SO RESPONSE DUE TO $-(M_p + M_c)\, r\,\Omega^2 \cos \Omega t$ IS IN-PHASE $\Rightarrow$ $-$VE SIGN FOR DISPLACEMENT, SAME AS FORCE

2 MARKS

(ii) $\boxed{\omega_n < 2\Omega}$ SO RESPONSE DUE TO $-M_p \frac{r^2}{l}\Omega^2 \cos 2\Omega t$ IS OUT-OF-PHASE $\Rightarrow$ $-(-) = +$VE SIGN FOR DISPLACEMENT, OPPOSITE SIGN AS FORCE!!

SMALLER THE $\frac{r}{l}$ RATIO SMALLER IS THE CONTRIBUTION OF HIGHER ORDER FORCING TERMS, UNLESS RESONANCE WITH HIGHER ORDER FORCING TERMS TAKES PLACE.

NOTE: DMF CURVE EXPLAINS WHY DISPLACEMENT AT $2\Omega t$ IS MUCH SMALLER COMPARED TO THAT AT Ω.

c) TOTAL HORIZONTAL REACTION $= R_1 + R_2 + R_3 + R_4 = M_c r \Omega^2 \sin \Omega t$

$= 710.6 \sin \Omega t$

2 MARKS

$\therefore$ MAX. HORIZONTAL REACTION $= 710.6\ N$

— THE END —

HAPPY HOLIDAYS!

Srikanth