

MechanicalAssessment_JamesT reviso_1503023

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Mechanical Engineering
Thermofluids and Applied Mechanics
Complex static and dynamic systems
PSDM6022

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1503023

1.1 Objective

Demonstrating knowledge of complex static and dynamic systems.

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2.1 Introduction

Critically evaluating complex and dynamic systems, discussing the importance of static and dynamic balancing in a non- coplanar system. Detailing elements of an epicyclical gear box and discussing the differences between thick and thin cylinder walls. Providing calculations for a lifting device and stresses in a shaft subjected to concentrated torques.

3.1 Element 1

a)

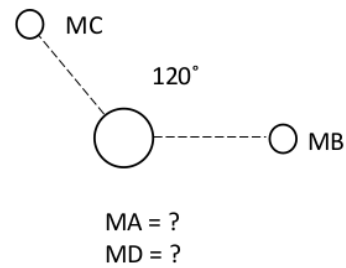
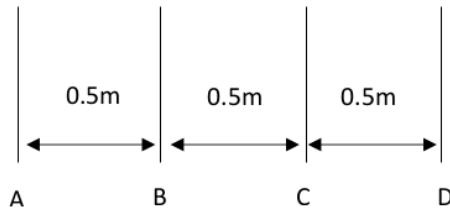
In rotating machinery, balancing of rotating bodies is important to eliminate vibrations, noise emissions and fatigue of components. Static balance refers to the ability of a number of masses attached to the same shaft without rotation MacCamhaoil 2016 explains, this occurs when the centre of gravity of the object is on the axis of rotation resulting inertia force ΣF_R . This is obtained by means of force polygon, these forces are considered to be in the same plane.

Dynamic balancing occurs when the rotating masses are at various planes along the shaft resulting in development of moments. For complete balance of inertia effect, for forces $\Sigma F = 0$. For moments $\Sigma M = 0$, this can be achieved once the force and moment polygons are closed thus balancing is a process of removing the effect of static and dynamic forces acting on the machine component.

When put into practice, non-coplanar balancing, a shaft with numerous bearings with difference masses at various distances apart, reactions of the bearings can be made which will provide a balanced system both statically and dynamically preventing unacceptable vibrations and premature failure thus for complete balance of inertia effect, achieving a closed force and moment polygon.



b)



$$MA = ?$$

$$MB = 0.6\text{kg}$$

$$MC = 0.3\text{kg}$$

$$MD = ?$$

$$RA = 100\text{mm}$$

$$RB = 120\text{mm}$$

$$RC = 120\text{mm}$$

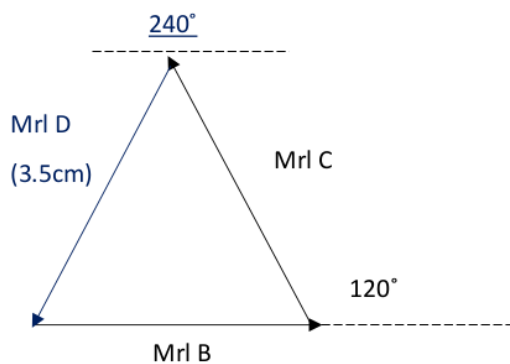
$$RD = 90\text{mm}$$

Firstly construct a Mrl table,

Plane	Mass (kg)	Radius (m)	Mr (kgm)	R (m)	Mrl (kgm ²)	Θ
A	MA	0.1	0.1MA	0	0	Θ_A
B	0.6	0.12	0.072	0.05	0.0036	0
C	0.3	0.12	0.036	0.1	0.0036	120
D	MD	0.09	0.09MD	0.15	0.0135	Θ_D

Now draw the Mrl polygon,

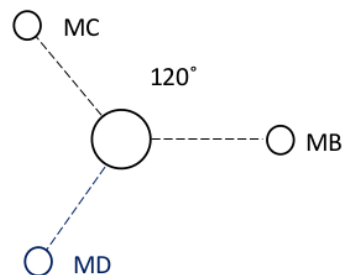
$$\text{Scale } 1\text{cm} = 0.001 \text{ kgm}^2$$



$$\text{Mrl D} = 0.0035\text{kgm}$$

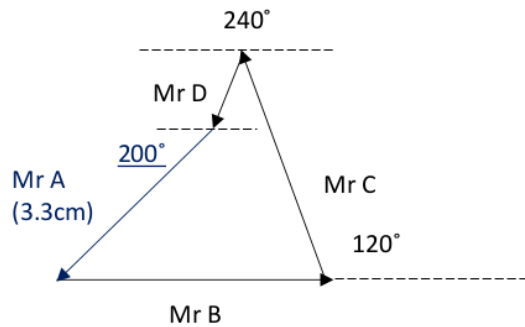
$$0.0135\text{MD} = 0.0035$$

$$\text{MD} = \frac{0.0035}{0.0135} = 0.259 \approx \text{mass} = \underline{0.26\text{kg}}$$



Now draw the forces polygon,

Scale 1cm = 0.015kgm

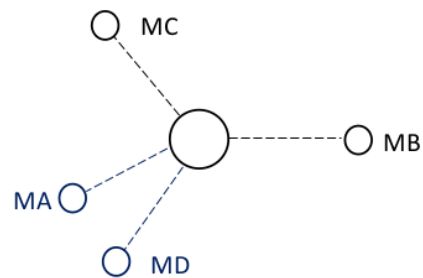


$$\text{Mr D} = 0.09 \times 0.26 \therefore \text{Mr D} = 0.0156\text{kgm (1cm)}$$

$$\text{Mr A} = 3.3 \times 0.015 = 0.0495\text{kgm}$$

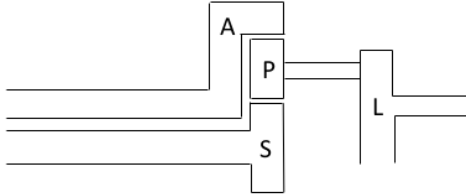
$$0.1 \text{ MA} = 0.0495$$

$$\text{MA} = \frac{0.0495}{0.1} = 0.495 \therefore \text{mass} = \underline{0.5\text{kg}}$$



4.1 Element 2

a)



Above shows a typical epicyclic gear box, where 'S' represents the sun, 'P' represents the planet. 'A' is the annulus, also known as the ring gear and 'L' represents the carrier.

b)

These complex gear systems is part of an automatic transition used vehicles, these provide high gear ratios. The input known as the sun rotates at a certain speed, with that the planet gear must rotate at the same speed otherwise the teeth in the gears will clash. The sun and the planet gear will rotate in the same direction, with the annulus being stationary the planet gears are able to spin and rotate producing velocity thus the carrier spinning with the planet gears. Yet when the sun gear is held stationary, the annulus rotates, this provides a higher speed ratio as the annulus gear radius is higher where $V = \text{Radius} \times \omega$.

c)

i)

Once the arm is stationary of the epicyclic gearbox, the sun and annulus rotate at opposite directions, in this case as the arm is fixed the planet gears can only spin not turn. The planet gear and the annulus both spin in the same direction, this provides a reverse gear. By constructing a table of elements if the arm is fixed and any other gears, a negative rotation is applied seen as -1 with other gears rotating in a positive revolution as +1. This has the effect of cancelling the rotation initially, then other elements can be compared with the rotation then, other gears can be considered by the number of teeth and ration of the fixed element.

ii)

As the sun gear is fixed, as the annulus radius is higher the planets rotate at a higher speed with the arm as the planet and annulus gears are rotating around the sun gear.

iii)

Whilst the annulus is fixed, the velocities are in opposite direction at the top of the planet gear to the annulus as it is stationary which is 0. Because the annulus is stationary, the carrier will rotate the planet gears around yet this provides a low speed ratio.

d)

In cases where of a epicycle gear box where all gears are free to rotate, by constructing a table whereby the fixed arm will be $-n$ rev and all other components will rotate $+m$ revs. By determining the rotation based on the number of teeth on each gear we can determine the speed of rotation.

e)

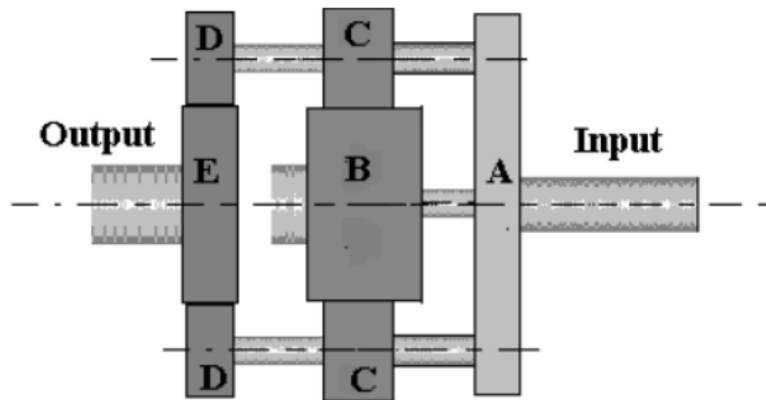
Gear trains are used to transmit rotary motion from one shaft to another, the speed is dependent on the numbers of teeth on each gear. A simple gear train will consist of a driver and driven and an idler to change direction of the output yet has no effect on gear ratio. Compound gears are also popular where there can be multiple driver and driven gears. Epicyclical gears offer greater radial speed and reliability. The transmission offers less vibration and fatigue than that of the gear train system which can be noisy and wear due to high bearing loads. The epicyclical gear system can achieve many gearing ratios with high velocity yet they may offer low efficiency compared to simple gear system, C. Chen, 2014 explains this is a result of sliding friction between meshing gear tooth surfaces.



f)

Epicyclical gearing train it can produce large gear ratios in a small space. The gearing system consists of many planet gears rotating around the central sun gear. The planet gear carrier is driven by the input torque whilst the sun gear provides the output torque while the annulus gear is fixed. As used in automation, Handschuh 2013 explains by that when epicycle gears are used in series, these provide a much higher gear ratio and act as bearings as well as transmissions. The gears are positioned whereby the output shaft of the initial stage becomes the input shaft of the next.

Task 2



$$TC = 200, TB = 50, TD = 50, TE = 200$$

	A	B	C/D	E
Rotate whole train + 1 Rev	+1	+1	+1	+1
Rotate gear -1 Rev & fix arm	0	-1	$+1 \cdot \left[\frac{50}{200} \right]$	$-1 \cdot \left[\frac{50 \times 50}{200 \times 200} \right]$
Add lines 1 & 2	1	0	$1.25 \left[\frac{5}{4} \right]$	$0.9375 \left[\frac{15}{16} \right]$

Thus the ratio of the gear box is 0.94: 1 or 15: 16. 

5.1 Element 3

Moment of inertia of the motor shaft as 0.04kgm^2

Moment of inertia of the drum and cables on the shaft as 25kgm^2

Gear reduction 30:1

Drum Diameter 800mm

Load 500kg

Motor torque 104Nm

Mechanical Efficiency 85%

$$T = I \times \alpha \quad a = \alpha \times r$$

$$T = 104\text{Nm} \quad I = 25\text{kgm}^2 \quad \alpha = ?$$

$$\frac{T}{I} = \alpha$$

$$85\% \text{ efficiency} = 88.4\text{Nm}$$

$$\frac{T}{I} \longrightarrow \frac{88.4}{25} = \alpha \quad \therefore \alpha = 3.5\text{rads/s}$$

$$a = \alpha \times r \longrightarrow 3.5 \times 0.4 \quad \therefore a = 1.4\text{m/s}^2$$



6.1 Element 4

A thin walled cylinder is regarded to have a wall thickness less than <20 of the diameter. It is assumed that the hoop stress- the latter is constant across the thickness of the cylinder wall. Longitudinal and circumferential stresses are produced as σ_l and σ_c . So long that the material withstands this is balanced by the stress in the material where $F = \sigma_c$. These cylinders are subject to internal and external pressures, they can be found in aerosol cans to boiler steam drums.

Unlike thin cylinders, thick walled cylinders are those with a wall thickness $20 >$ often used in hydraulic cylinder environments. According to Juskiewicz & Nowak 2015, along with longitudinal and circumferential stresses, a stress in the radial direction σ_r is also taking into consideration. Lamé's theorem gives the solution to a thick cylinder which incorporates principle stresses from in all three directions.

i)

$$d = 30\text{mm} \quad D = 120\text{mm} \quad P_1 = 12\text{MPa} \quad P_2 = 6\text{MPa}$$

Applying boundary conditions,

$$\text{When } r = 0.015\text{m } \sigma_r = -12\text{MN/m}^2$$

$$\text{When } r = 0.060\text{m } \sigma_r = -5\text{MN/m}^2$$

$$\sigma_r = a + \frac{b}{r^2}$$

$$-12 \times 10^6 = \frac{a + b}{(0.015)^2} \quad \text{————— 1}$$

$$-5 \times 10^6 = \frac{a + b}{(0.06)^2} \quad \text{————— 2}$$

$$\text{From equation 2, } -5 \times 10^6 = a + 277.77b$$

$$a = -5 \times 10^6 - 277.77b, \text{ sub into equation 1.}$$

$$-12 \times 10^6 = (-5 \times 10^6 - 277.77b) + 4444.44b$$

$$-12 \times 10^6 + 5 \times 10^6 = 4166.67b$$

$$-7 \times 10^6 = 4166.67b$$

$$\frac{-7 \times 10^6}{4166.67} = b \quad \therefore \quad b = -1679.99$$

$$a = -5 \times 10^6 - 277.77 \quad (-1679.99)$$

$$a = -4.53 \times 10^6$$

Now calculate the circumferential stress,

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

$$\sigma_c = a - \frac{b}{r^2}$$

$$\sigma_c = -4.53 \times 10^6 - \frac{(-1679.99)}{(0.015)^2} = 2.63 \text{ MN/m}^2$$

When $r = 0.06 \text{ m}$

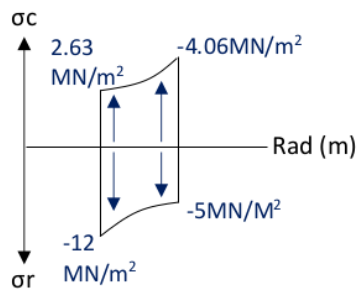
$$\sigma_c = -4.53 \times 10^6 - \frac{(-1679.99)}{(0.06)^2} = -4.06 \text{ MN/m}^2$$

Rad (m)	σ_r (N/m ²)	σ_c (N/m ²)
0.015	-12 MN	2.63 MN
0.06	-5 MN	-4.06 MN

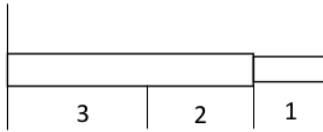


ii)

Stress profile,



7.1 Element 5



$$T_1 = -415 \text{ Nm}$$

$$T_2 = -415 + 1240 = 825 \text{ Nm}$$

$$T_3 = 825 - 1000 = -175 \text{ Nm}$$

Where general torsion equation $\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L}$

Using $\frac{T}{J} = \frac{\tau}{r} \longrightarrow \tau = \frac{Tr}{J}$

Polar second moment of area $J = \frac{\pi D^4}{32}$

i)

Calculating maximum shearing stress in the shaft.

Section 1,

$$J = \frac{\pi D^4}{32}, \quad \frac{\pi \times 0.03^4}{32} = 7.95 \times 10^{-8} \text{ m}^4$$

Shear stress,

$$\tau_1 = \frac{T_1 r_1}{J_1}, \quad \frac{415 \times 0.015}{7.95 \times 10^{-8}} = \tau \quad \therefore \tau = 78.30 \text{ MN/m}^2$$

Section 2,

$$J = \frac{\pi D^4}{32} - \frac{\pi d^4}{32}, \quad \frac{\pi \times 0.045^4}{32} - \frac{\pi \times 0.035^4}{32} = 2.55 \times 10^{-7} \text{ m}^4$$

Shear stress,

$$\tau_2 = \frac{825 \times 0.0225}{2.55 \times 10^{-7}} = \tau \quad \therefore \tau = 72.79 \text{ MN/m}^2$$

Section 3,

$$J = 2.55 \times 10^{-7} \text{ MN/m}^2$$

Shear stress,

$$\tau_3 = \frac{175 \times 0.0225}{2.55 \times 10^{-7}} = \tau \quad \therefore \tau = 15.44 \text{ MN/m}^2$$

∴ Maximum shear stress is section 1 of 78.30MN/m²

ii)

Calculating angle of rotation of each section.

Referring to the general torsion equation,

$$\text{Where } \frac{T}{J} = \frac{G\theta}{r} \longrightarrow \theta = \frac{TL}{JG}$$

Section 1,

$$\theta_1 = \frac{-415 \times 0.2}{(7.95 \times 10^{-8})(75 \times 10^9)} = -0.014 \therefore \theta = 0.014 \text{ Rads, } \frac{0.014}{2\pi} \times 360 = 0.80^\circ$$

Section 2,

$$\theta_2 = \frac{825 \times 0.4}{(2.55 \times 10^{-7})(75 \times 10^9)} = 0.017 \therefore \theta = 0.017 \text{ Rads, } \frac{0.017}{2\pi} \times 360 = 0.97^\circ$$

Section 3,

$$\theta_3 = \frac{-175 \times 0.9}{(2.55 \times 10^{-7})(75 \times 10^9)} = -0.0082 \therefore \theta = -0.0082 \text{ Rads, } \frac{0.0082}{2\pi} \times 360 = 0.47^\circ$$

iii)

Angle of rotation of D relative to A

$$\theta_{DA} = \theta_1 + \theta_2 + \theta_3$$

$$\theta_{DA} = -0.014 + 0.017 - 0.0082 = -5.2 \times 10^{-3} \therefore \theta_{DA} = -0.0052 \text{ Rads, } \frac{0.0052}{2\pi} \times 360 = 0.30^\circ$$



8.1 Bibliography

C. Chen, J. C. (2014). *Epicyclic Gears*. Lyon: Chandos Publishing.

Hands Schuh, R. F. (2013). *Epicyclic Gear Trains*. New York: Springer, Boston, MA.

Juszkiewicz, G., & Nowak, T. (2015). *Comparative study on thin and thick walled cylinder*. Cracow: Elsevier.

MacCamhaoil, M. (2016). *Static and Dynamic Balancing*. Denmark: Bruel & Kjaer.