وزارة التعليم العالي والبحث العلمي
الجامعة التقنية الشمالية
المعهد التقني الدور

الحقيبة التعليمية

القسم العلمي: قسم التقنيات الميكانيكية

اسم المقرر: اجزاء مكائن متقدم

المرحلة / المستوى: الثاني

الفصل الدراسي: الثاني

السنة الدراسية:2025-2026

معلومات عامة

اسم المقرر:	تقنية اجزاء المكائن
القسم:	قسم التقنيات الميكانيكة
الكلية:	المعهد التقني الدور
المرحلة / المستوى	الثاني
الفصل الدراسي:	الثاني
عدد الساعات الاسبوعية:	نظري 2 عملي 0
عدد الوحدات الدراسية:	2
الرمز:	MTP205
نوع المادة	نظري * عملي كلهما
هل يتوفر نظير للمقرر في الاقسام الاخرى	لا يوجد
اسم المقرر النظير	-
القسم	-
رمز المقرر النظير	-
معلومات تدريسي المادة	
اسم مدرس (مدرسي) المقرر:	نور عبداللطيف عزت
اللقب العلمي:	مدرس مساعد
سنة الحصول على اللقب	2023
الشهادة :	ماجستير
سنة الحصول على الشهادة	2020
عدد سنوات الخبرة (تدريس)	3 سنوات

الوصف العام للمقرر

تصميم المكائن هي فرع من فروع الهندسة الميكانيكية تركز على تطوير وتصميم الأجهزة والأنظمة الميكانيكية التي تستخدم في مختلف الصناعات. وتهدف إلى إنشاء حلول هندسية تلبي الاحتياجات الفنية والوظيفية بأقل تكلفة ممكنة وبأعلى كفاءة ممكنة.

الاهداف العامة

- يتعلم الطلاب كيفية تحليل المكونات الميكانيكية المختلفة باستخدام مبادئ الفيزياء والرياضيات، مثل تحميلات القوى والضغوط والعزم.
- يتعلم الطلاب كيفية اختيار المواد المناسبة لكل تطبيق، مع توفير التوازن المثلى بين الموثوقية والتكلفة والأداء.
- يتدرب الطلاب على تصميم الآلات والأجزاء باستخدام برامج الكمبيوتر وأدوات النمذجة الهندسية، مع التركيز على تحقيق الأداء المطلوب والكفاءة.
- يتدرب الطلاب على التعامل مع التحليلات الإجهادية لضمان متانة واستدامة التصاميم، بالإضافة إلى تحسين تصميماتهم الهيكلية.
- يتعلم الطلاب كيفية التفكير الإبداعي والابتكار في تصميم المكائن، وتحسين الأداء والموثوقية وتقليل التكاليف.

الأهداف الخاصة

- تحسين أداء الأجزاء بما يتناسب مع متطلبات التصميم، مثل زيادة الكفاءة والدقة والمتانة والقدرة على تحمل الأحمال.
- تحقيق تكاليف أقل للإنتاج والصيانة، من خلال اختيار المواد المناسبة وتحسين التصميمات لتقليل النفايات وتحسين عمليات التصنيع.
- تطوير تصاميم تتميز بالموثوقية العالية، مما يضمن استمرارية عمل المكائن بكفاءة ودون توقفات غير مخطط لها.
- تطوير تصاميم فعّالة تلبي الاحتياجات الفنية والوظيفية بأقل تكلفة ممكنة وبأعلى كفاءة ممكنة.

أمثلة الأهداف التعليمية .

إكساب المتعلم مهارات الحسابات التصميمية.

تشجيع الطلاب على استخدام التفكير الإبداعي لتطوير حلول تصميمية مبتكرة تلبي الاحتياجات المحددة للتطبيق.

تعلم الطلاب كيفية تقييم تأثير الأحمال الميكانيكية والظروف البيئية على أداء المكائن، وتحسين التصاميم بناءً على هذه الاستجابات.

تدريب الطلاب على قراءة وتحليل وتقييم التصاميم الميكانيكية بناءً على المعايير الهندسية والمواصفات المحددة.

الأهداف السلوكية او نواتج التعلم

- ان يتعلم الطالب التصاميم الميكانيكية وفق المعايير الهندسية والمواصفات المحددة.
- ان يتعلم الطالب استخدام الادوات والبرمجيات الهندسية المختلفة لتصميم وتحليل المكائن.
- ان يميز الطالب المواد الهندسية المختلفة والتي تلبي الاحتياجات الفعلية باقل تكلفة وبأعلى كفاءة ممكنة.
- **أمثلة أهداف تدريسية:**

بعد الانتهاء من المحاضرة سيكون الطالب قادرا على ان:

- كيفية تقييم تأثير الأحمال الميكانيكية والظروف البيئية على أداء المكائن، وتحسين التصاميم بناءً على هذه الاستجابات.
- قراءة وتحليل وتقييم التصاميم الميكانيكية بناءً على المعايير الهندسية والمواصفات المحددة.
- استخدام الأدوات والبرمجيات الهندسية المختلفة لتصميم وتحليل المكائن، مثل النمذجة الهندسية بالحاسوب وبرامج CAD/CAM .

المتطلبات السابقة

- لا يحتاج الطالب الى أي مواد دراسية سابقة او مهارات معينة.

الأهداف السلوكية او مخرجات التعليم الأساسية		
ت	تفصيل الهدف السلوكي او مخرج التعليم	آلية التقييم
1	. فهم الأساسيات الميكانيكية	• **المحاضرات .** • **الواجبات الفردية والمشاركة .** • **الاختبارات.**
2	تصميم الآلات والمكائن	• **المحاضرات .** • **الواجبات الفردية والمشاركة .** • **الاختبارات.**
3	استخدام البرمجيات الهندسية	• **المحاضرات .** • **الواجبات الفردية والمشاركة .** • **الاختبارات.**

4	التحليل والتقييم	• **الاختبارات** • **المشاريع الفردية والجماعية.**

أساليب التدريس (حدد مجموعة متنوعة من أساليب التدريس لتناسب احتياجات الطلاب ومحتوى المقرر)

الاسلوب او الطريقة	مبررات الاختيار
1. **المحاضرات النظرية والعملية**	تهدف إلى تعليم الطالب كيفية التحليل والتعامل مع المسائل
2. **العصف الذهني**	اسلوب يهدف إلى توليد أفكار جديدة وحلول مبتكرة للمشاكل أو التحديات المطروحة
3. **التفكير الناقد**	تهدف إلى تقييم وتحليل الأفكار والمعلومات بشكل منطقي ومنظم
4. **التواصل اللغوي**	للتأثير على الآخرين وإقناعهم بالآراء أو الفكر الذي يتم تبادله، مما يساهم في بناء القناعات والاتفاقات.
5. **المناقشة والحوار**	تهدف كلاهما إلى تبادل الآراء والأفكار بين الأفراد ومناقشة المواضيع المختلفة

الفصل الاول من المحتوى العلمي							
عنوان الفصل	الوقت						
التوزيع الزمني	النظري	العملي	العنوان الرئيسي	العنوان الفرعي	طريقة التدريس	التقنيات	طرق القياس
الأسبوع الاول والثاني	2		Belts	Types of Belts	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
				Design of Belts	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الاسبوع الثالث	2		استخدام الذكاء الاصطناعي في تصميم الاحزمة	**تعليم الطلاب كيفية استخدام أدوات تعتمد على الذكاء الاصطناعي لتحسين تصميم الأحزمة، مثل اختيار المواد المناسبة أو التنبؤ بالأداء في ظروف مختلفة.**	محاضرة وعرض تقديمي	عرض تقديمي، تفاعل الطلبة مع الذكاء الاصطناعي لتصميم الاحزمة	

الفصل الثاني							
عنوان الفصل	الوقت						
التوزيع الزمني	النظري	العملي	العنوان الرئيسي	العنوان الفرعي	طريقة التدريس	التقنيات	طرق القياس
الأسبوع الرابع	2		Design of Shafts	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الاسبوع الخامس	2		استخدام الذكاء الاصطناعي في تصميم الاعمدة	**استخدام خوارزميات الذكاء الاصطناعي لتحليل أنماط الاهتزازات الناتجة عن دوران الأعمدة واقتراح تعديلات في التصميم لتقليل عدم الاستقرار.**	عرض تقديمي	تعليم الطلاب كيفية استخدام الذكاء الاصطناعي في تصميم الاعمدة	
الأسبوع السادس والسابع	2		Design of Journal Bearings	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة

التوزيع الزمني	نظري	عملي	العنوان الرئيسي	العنوان الفرعي	طريقة التدريس	التقنيات	طرق القياس
الفصل الثالث							
عنوان الفصل	الوقت						
الأسبوع الثامن والتاسع	2		Selection of Ball Bearings	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الأسبوع العاشر	2		Design of Gears by Lewis Equation	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الاسبوع الحادي عشر			Cams	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة

الفصل الرابع (من المحتوى العلمي)							
عنوان الفصل	الوقت						
التوزيع الزمني	نظري	عملي	العناوين الرئيسية	العناوين الفرعية	طريقة التدريس	التقنيات	طرق القياس
الأسبوع الثاني عشر	2		Gears Trains	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الأسبوع الثالث عشروالرابع عشر	2		Design of Simple Gears Box	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة
الأسبوع الخامس عشر	2		Worm Gears	-	محاضرة	عرض تقديمي، شرح، أسئلة وأجوبة	عرض الامثلة والمناقشة وملاحظة مستوى الطلبة

المحتوى العلمي

Belt drives

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. the amount of power transmitted depends upon the following factors:

1.the velocity of the belt

2.the tension under which the belt is placed on the pulleys.

3.the arc of contact between the belt and the smaller pulley.

4.the conditions under which the belt is used.

Types of belt drives

The belt drives are usually classified into the following three groups:

1.light drives: these are used to transmit small powers at belt speeds up to 10 m / s as in agricultural machines.

2.medium drives: these are used to transmit medium powers at belt speeds over 10 m / s to 22 m/ s as in machine tools.

3. heavy drives: these are used to transmit large powers at belt speeds over 22 m / s as in compressors and generators.

Types of belts

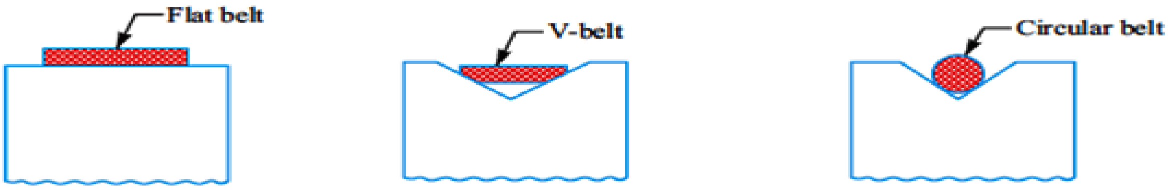
1.flat belt 2.V-belt 3.circular belt

Material used for belts

1.lather belts 2. Cotton or fabric belts 3.rubber belts 4.balata belts

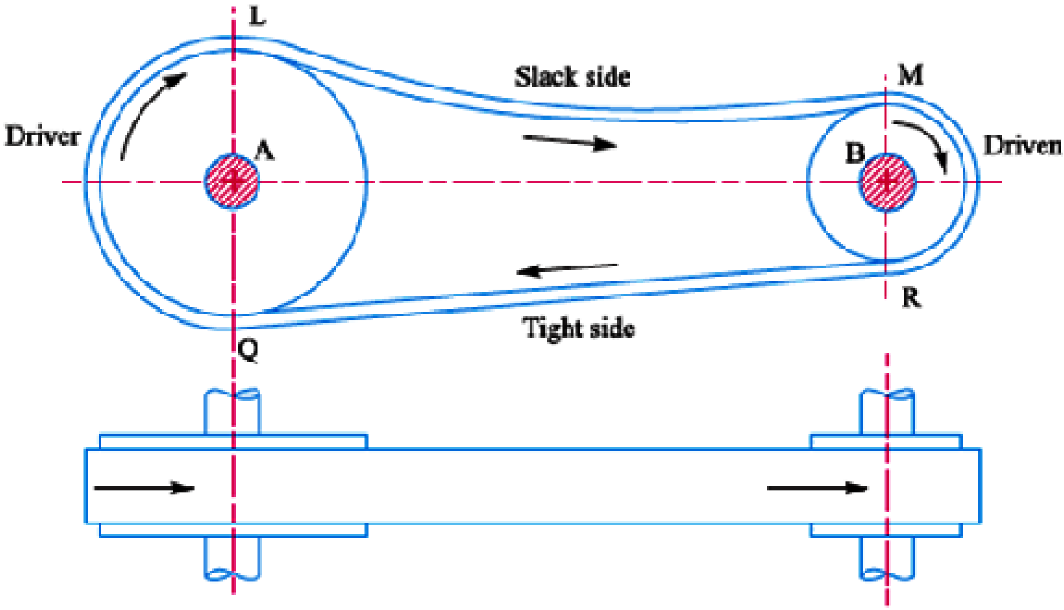
Types of flat belt drives

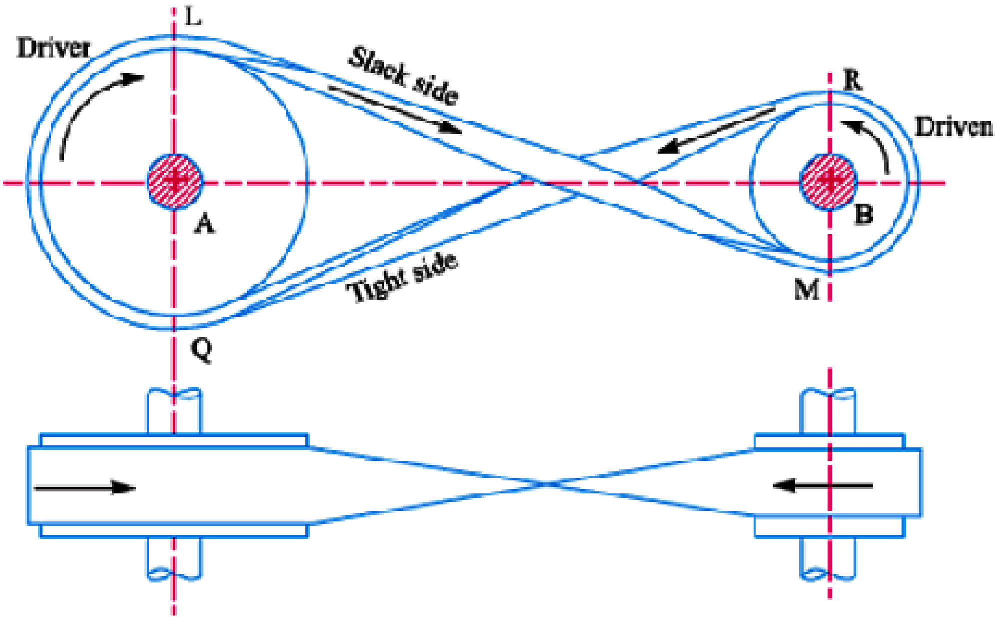
1.open belt drive 2.crossed or twist belt drive



(*a*) Flat belt. (*b*) V-belt. (*c*) Circular belt.

Types of belts





Velocity ratio of belt drive

$$v_1 = \frac{\pi d_1 N_1}{60} \quad m/s \qquad v_2 = \frac{\pi d_2 N_2}{60} \quad m/s$$

$$v_1 = v_2$$

$$\pi d_1 N_1 = \pi d_2 N_2$$

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

d_1 – diameter of the driver

d_2 – diameter of the follower

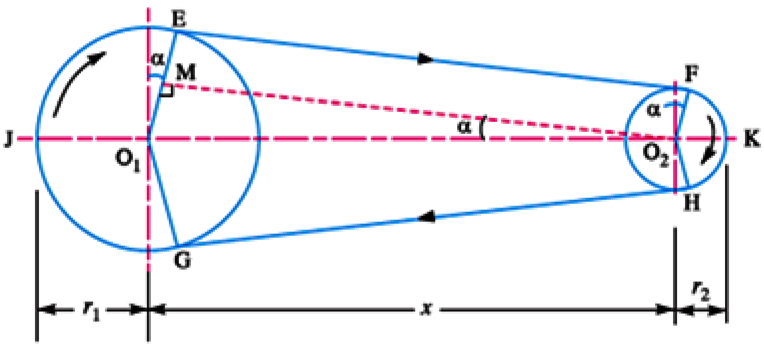
N_1 – speed of the driver in R.P.M

N_2 – speed of the follower in R.P.M

v_1 – velocity of the belt on the drive pulley

v_2 – velocity of the belt on the driven pulley

Length of an open belt drive



open belt drive

Let the belt leaves the larger pulley at E and G and the smaller pulley at F and H

Through O_2 draw O_2M parallel to FE.

From the geometry of the figure, we find that O_2M will be perpendicular to O_1E.

Let the angle $MO_2O_1 = \alpha$ radians.

We know that the length of the belt,

$$L = \text{Arc } GJE + EF + \text{Arc } FKH + HG$$

$$= 2\,(\text{Arc } JE + EF + \text{Arc } FK) \qquad ...(i)$$

From the geometry of the figure, we also find that

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E - EM}{O_1O_2} = \frac{r_1 - r_2}{x}$$

Since the angle α is very small, therefore putting

$$\sin\alpha = \alpha \text{ (in radians)} = \frac{r_1 - r_2}{x} \qquad ...(ii)$$

$$\therefore \qquad \text{Arc } JE = r_1\left(\frac{\pi}{2} + \alpha\right) \qquad ...(iii)$$

Similarly,

$$\text{arc } FK = r_2\left(\frac{\pi}{2} - \alpha\right) \qquad ...(iv)$$

and

$$EF = MO_2 = \sqrt{(O_1O_2)^2 - (O_1M)^2} = \sqrt{x^2 - (r_1 - r_2)^2}$$

$$= x\sqrt{1 - \left(\frac{r_1 - r_2}{x}\right)^2}$$

Expanding this equation by binomial theorem, we have

$$EF = x\left[1 - \frac{1}{2}\left(\frac{r_1 - r_2}{x}\right)^2 + \ldots\right] = x - \frac{(r_1 - r_2)^2}{2x} \qquad ...(v)$$

Substituting the values of arc JE from equation *(iii)*, arc FK from equation *(iv)* and EF from equation *(v)* in equation *(i)*, we get

$$L = 2\left[r_1\left(\frac{\pi}{2} + \alpha\right) + x - \frac{(r_1 - r_2)^2}{2x} + r_2\left(\frac{\pi}{2} - \alpha\right)\right]$$

$$= 2\left[r_1 \times \frac{\pi}{2} + r_1\alpha + x - \frac{(r_1 - r_2)^2}{2x} + r_2 \times \frac{\pi}{2} - r_2\alpha\right]$$

$$= 2\left[\frac{\pi}{2}(r_1 + r_2) + \alpha\,(r_1 - r_2) + x - \frac{(r_1 - r_2)^2}{2x}\right]$$

$$= \pi\,(r_1 + r_2) + 2\alpha\,(r_1 - r_2) + 2x - \frac{(r_1 - r_2)^2}{x}$$

Substituting the value of $\alpha = \dfrac{(r_1 - r_2)}{x}$ from equation *(ii)*, we get

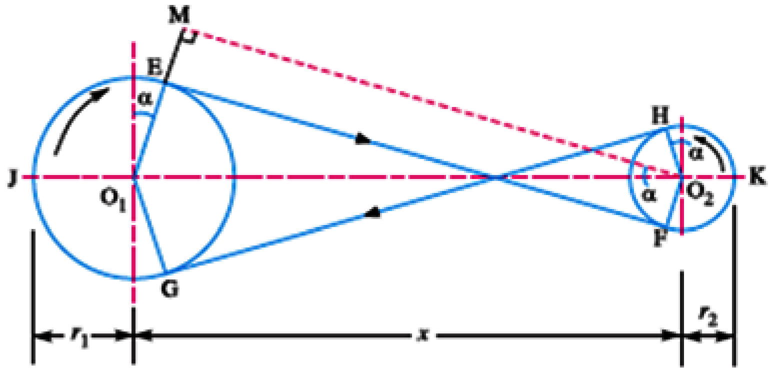
$$L = \pi\,(r_1 + r_2) + 2 \times \frac{(r_1 - r_2)}{x}\,(r_1 - r_2) + 2x - \frac{(r_1 - r_2)^2}{x}$$

$$= \pi\,(r_1 + r_2) + \frac{2\,(r_1 - r_2)^2}{x} + 2x - \frac{(r_1 - r_2)^2}{x}$$

$$= \pi\,(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \qquad \text{... (in terms of pulley radii)}$$

$$= \frac{\pi}{2}\,(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x} \qquad \text{... (in terms of pulley diameters)}$$

Length of a cross belt drive



crossed belt drive

Let r_1 and r_2 = Radii of the larger and smaller pulleys,

x = Distance between the centres of two pulleys (*i.e.* O_1O_2), and

L = Total length of the belt.

Let the belt leaves the larger pulley at E and G and the smaller pulley at F and H

Through O_2 draw O_2M parallel to FE.

From the geometry of the figure, we find that O_2M will be perpendicular to O_1E.

Let the angle $MO_2O_1 = \alpha$ radians.

We know that the length of the belt,

$$L = \text{Arc } GJE + EF + \text{Arc } FKH + HG$$

$$= 2\,(\text{Arc } JE + FE + \text{Arc } FK) \qquad ...(i)$$

From the geometry of the figure, we find that

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{O_1E + EM}{O_1O_2} = \frac{r_1 + r_2}{x}$$

Since the angle α is very small, therefore putting

$$\sin\alpha = \alpha \text{ (in radians)} = \frac{r_1 + r_2}{x} \qquad ...(ii)$$

$\therefore$

$$\text{Arc } JE = r_1\left(\frac{\pi}{2} + \alpha\right) \qquad ...(iii)$$

Similarly,

$$\text{arc } FK = r_2\left(\frac{\pi}{2} + \alpha\right) \qquad ...(iv)$$

and

$$EF = MO_2 = \sqrt{(O_1O_2)^2 - (O_1M)^2} = \sqrt{x^2 - (r_1 + r_2)^2}$$

$$= x\sqrt{1 - \left(\frac{r_1 + r_2}{x}\right)^2}$$

Expanding this equation by binomial theorem, we have

$$EF = x\left[1 - \frac{1}{2}\left(\frac{r_1 + r_2}{x}\right)^2 + ...\right] = x - \frac{(r_1 + r_2)^2}{2x} \qquad ...(v)$$

Substituting the values of arc JE from equation (*iii*), arc FK from equation (*iv*) and EF from equation (*v*) in equation (*i*), we get,

$$L = 2\left[r_1\left(\frac{\pi}{2}+\alpha\right)+x-\frac{(r_1+r_2)^2}{2x}+r_2\left(\frac{\pi}{2}+\alpha\right)\right]$$

$$= 2\left[r_1\times\frac{\pi}{2}+r_1.\alpha+x-\frac{(r_1+r_2)^2}{2x}+r_2\times\frac{\pi}{2}+r_2.\alpha\right]$$

$$= 2\left[\frac{\pi}{2}(r_1+r_2)+\alpha(r_1+r_2)+x-\frac{(r_1+r_2)^2}{2x}\right]$$

$$= \pi(r_1+r_2)+2\alpha(r_1+r_2)+2x-\frac{(r_1+r_2)^2}{2x}$$

Substituting the value of $\alpha = \dfrac{(r_1+r_2)}{x}$ from equation (*ii*), we get

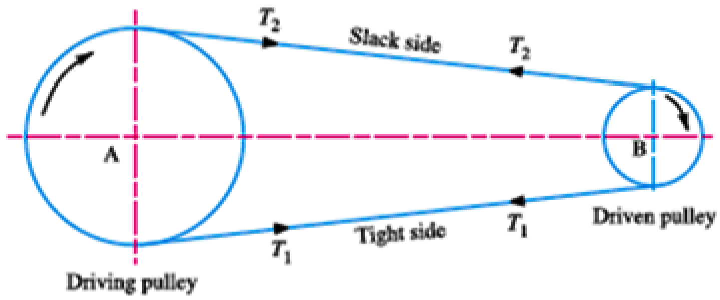
$$L = \pi(r_1+r_2)+2\times\frac{(r_1+r_2)}{x}(r_1+r_2)+2x-\frac{(r_1+r_2)^2}{x}$$

$$= \pi(r_1+r_2)+\frac{2(r_1+r_2)^2}{x}+2x-\frac{(r_1+r_2)^2}{x}$$

$$= \pi(r_1+r_2)+2x+\frac{(r_1+r_2)^2}{x} \qquad \text{... (in terms of pulley radii)}$$

$$= \frac{\pi}{2}(d_1+d_2)+2x+\frac{(d_1+d_2)^2}{4x} \qquad \text{... (in terms of pulley diameters)}$$

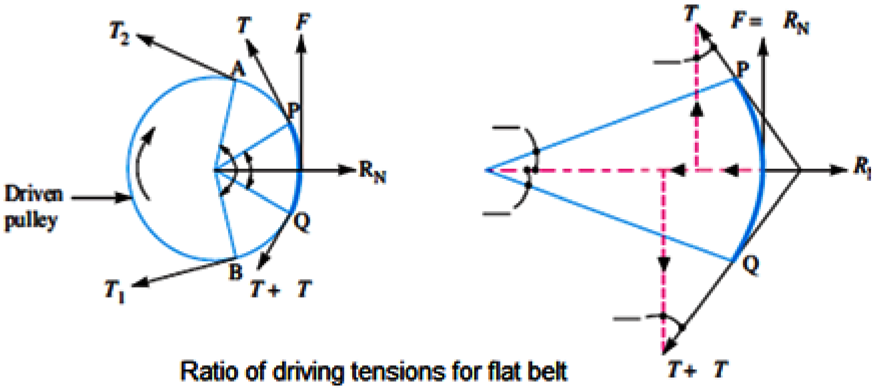
Power transmitted by a belt



power transmitted by a belt

$$P=(T_1-T_2)\times v$$
$$v-\text{velocity of the belt } m/s$$
$$T_1-\text{tension in the tight side}$$
$$T_2-\text{tension in the slack side}$$

Ratio of driving tensions for flat belt drive



Ratio of driving tensions for flat belt

$$F=\mu\times R_N$$

$F - frictional\ force$

$\mu - coeficient\ of\ friction$

$R_N - normal reaction\ force$

$$\log_e \left(\frac{T_1}{T_2} \right) = \mu \times \theta$$

$$\left(\frac{T_1}{T_2} \right) = e^{\mu \times \theta}$$

$$2.3 \log \left(\frac{T_1}{T_2} \right) = \mu \times \theta$$

$\theta - angle of\ contact$

$$\theta = (180^0 - 2\alpha) \frac{\pi}{180} \quad rad \qquad (for\ open belt drive)$$

$$\theta = (180^0 + 2\alpha) \frac{\pi}{180} \quad rad \qquad (for\ cross - belt drive)$$

Ex1:

Find the length of belt which is necessary to drive a pulley of 80 cm dia. is running parallel at a distance of 12 meters from the driving pulley of 480 cm dia.

Sol:

1.length of open belt drive

$$L=\frac{\pi}{2}(d_1+d_2)+2X+\frac{(d_1-d_2)^2}{4X}$$

$$L=\frac{\pi}{2}(480+80)+2\times1200+\frac{(480-80)^2}{4\times1200}$$

$$L=33.13\ m$$

2.length of cross-belt drive

$$L=\frac{\pi}{2}(d_1+d_2)+2X+\frac{(d_1+d_2)^2}{4X}$$

$$L=\frac{\pi}{2}(480+80)+2\times1200+\frac{(480+80)^2}{4\times1200}$$

$$L=33.45\ m$$

Ex2:

Find the length of the flat belt under the following conditions; diameter of the small pulley 630 mm, rotate at 740 R.P.M , and diameter of large pulley 2000 mm, center distance is 6 meters. Also find the power , if the friction factor is 0.3 and the tension in the slack side is 300 N.

Sol:

$$v=\frac{\pi d N}{60}=\frac{\pi\times0.63\times740}{60}=24.5\ m/s$$

$$L=\frac{\pi}{2}(d_1+d_2)+2X+\frac{(d_1-d_2)^2}{4X}$$

$$L=\frac{\pi}{2}(2000+630)+2\times6000+\frac{(2000-630)^2}{4\times6000}$$

$$L=16209\,mm\ =16.21m$$

$$\sin\alpha=\frac{r_1-r_2}{X}=\frac{1000-315}{6000}$$

$$\alpha=6.5^0$$

$$\theta=180-2\alpha=180-2\times6.5=166.8\times\frac{\pi}{180}=2.9\ rad$$

$$2.3\log(\frac{T_1}{T_2})=\mu\times\theta=0.3\times2.9=0.87$$

$$\log(\frac{T_1}{T_2})=\frac{0.87}{2.3}=0.378$$

$$\frac{T_1}{T_2}=2.387$$

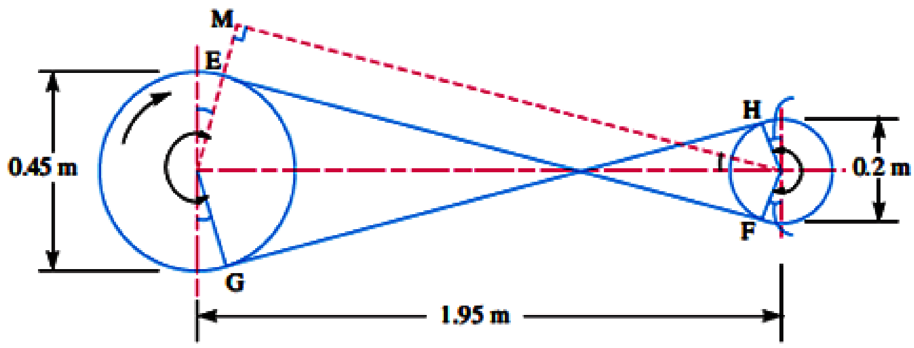
$$T_1=2.387\times T_2=2.387\times300=716\ N$$

$$power\ P=(T_1-T_2)\times v=(716-300)\times24.5=10192W$$

Ex3:

Tow pulleys one 450 mm diameter and the other 200 mm diameter, on parallel shafts 1.95 m apart are connected by a crossed belt. Find the length of belt required and the angle of contact between the belt and each pulley.

What power can be transmitted by the belt when the larger pulley rotates at 200 rev / min, if the maximum permissible tension in the belt



is 1 KN, and the coefficient of friction between the belt and pulley is 0.25 ?

$$L = \pi\,(r_1 + r_2) + 2X + \frac{(r_1 - r_2)^2}{X}$$

$$= \pi \times (0.225 + 0.1) + 2 \times 1.95 + \frac{(0.225 + 0.1)^2}{1.95}$$

$$= 1.02 + 3.9 + 0.054 = 4.974\ m$$

$$\sin\alpha = \frac{r_1 + r_2}{X} = \frac{0.225 + 0.1}{1.95} = 0.1667$$

$$\alpha = 9.6^\circ$$

$$\theta = 180^\circ + 2\alpha = 180^\circ + 2 \times 9.6^\circ = 199.2^\circ = 199.2^\circ \times \frac{\pi}{180} = 3.477\ rad$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu \times \theta = 0.25 \times 3.477 = 0.8693$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.8693}{2.3} = 0.378$$

$$\frac{T_1}{T_2} = 2.387$$

$$T_2 = \frac{T_1}{2.387} = \frac{1000}{2.387} = 419N$$

$$v = \frac{\pi\ d_1\ N_1}{60} = \frac{\pi \times 0.45 \times 200}{60} = 4.713 m/s$$

power transitted

$$P = (T1 - T2) \times v = (1000 - 419) \times 4.713 = 2738\ W = 2.7\ KW$$

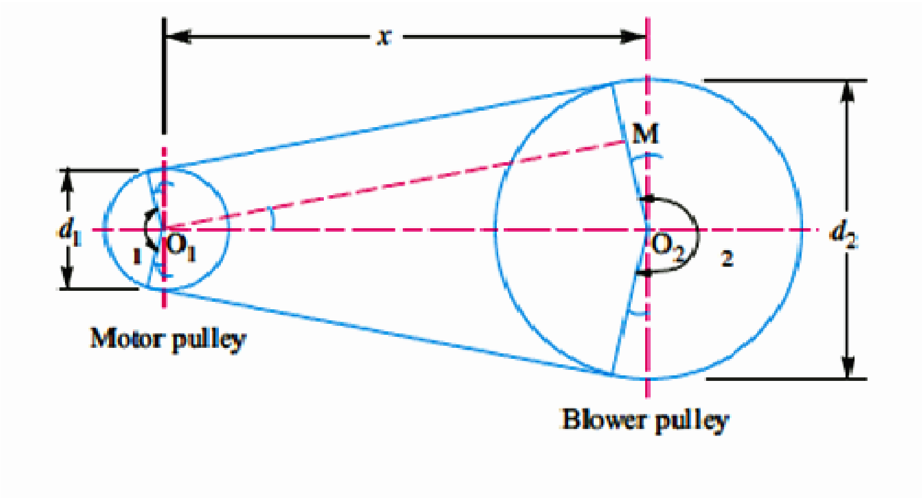
Ex4:

In a horizontal belt drive for a centrifugal blower , the blower is belt driven at 600 R.P.M by a 15 KW , 1750 R.P.M . electric motor . the center distance is twice the diameter of the large pulley , the density of the belt material = 1500 kg / m^3 ; maximum allowable stress = 4 MPa;

μ_1 = 0.5(motor pulley); μ_2 = 0.4 (blower pulley);

peripheral velocity of the belt = 20 m/s. determine the following :

1.Pulley diameter 2.Belt length



Sol:

$$v_1 = \frac{\pi d_1 \; N_1}{60}$$

$$20 = \frac{\pi \times d_1 \times 1750}{60} = 91.64 d_1$$

$$d_1 = \frac{20}{91.64} = 218 \; mm$$

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

$$d_2 = \frac{N_1 \times d_1}{N_2} = \frac{218 \times 1750}{600} = 636 \; mm$$

$$X = 2 d_2 = 2 \times 636 = 1272 \; mm$$

$$L = \frac{\pi}{2}(d_1 + d_2) + 2X + \frac{(d_1 - d_2)^2}{4X}$$

$$= \frac{\pi}{2}(218 + 636) + 2 \times 1272 + \frac{(218 - 636)^2}{4 \times 1272}$$

$$= 1342 + 2544 + 34 = 3920 mm = 3.92 \; m$$

استخدام الذكاء الاصطناعي في تصميم الاحزمة

دمج الذكاء الاصطناعي في مادة الأحزمة (Belt) ضمن تقنية أجزاء المكائن يمكن أن يعزز العملية التعليمية ويوفر تطبيقات عملية لفهم واستخدام هذه الأنظمة بشكل أفضل. فيما يلي بعض الأفكار حول كيفية دمج الذكاء الاصطناعي

1. محاكاة عمل الأحزمة باستخدام تقنياتAI

- يمكن استخدام نماذج محاكاة تعتمد على الذكاء الاصطناعي لتوضيح كيفية عمل الأحزمة في نقل الحركة والقوة.
- استخدام برامج مثل MATLAB أو Simulink مع تقنيات الذكاء الاصطناعي لتحليل وتصميم أنظمة نقل الحركة بواسطة الأحزمة.

2 . التنبؤ بالعمر الافتراضي للأحزمة

- إدخال مفاهيم التنبؤ بمدة صلاحية الأحزمة أو توقيت صيانتها باستخدام نماذج تعلم الآلة.

3 . تطوير نظام مراقبة ذكي

- استخدام مستشعرات مدمجة ونماذج الذكاء الاصطناعي لرصد أداء الأحزمة في الوقت الفعلي.
- ربط النظام بشاشة عرض أو تطبيق يمكن للطلاب من خلاله متابعة البيانات.

4 . استخدام الذكاء الاصطناعي في تصميم الأحزمة

- تعليم الطلاب كيفية استخدام أدوات تعتمد على الذكاء الاصطناعي لتحسين تصميم الأحزمة، مثل اختيار المواد المناسبة أو التنبؤ بالأداء في ظروف مختلفة.

فوائد الدمج:

- تعزيز الفهم العملي والنظري للمادة.
- تطوير مهارات التفكير النقدي وحل المشكلات لدى الطلاب.
- توفير بيئة تعليمية مبتكرة تربط بين التكنولوجيا الحديثة والتطبيقات الصناعية.

Shafts

A shaft is a rotating machine element which is used to transmit power from one place to another. In order to transfer the power from one shaft to another, the various members such as pulleys, gears, etc...are mounted on it by means of keys or splines.

An axle is a similar in shape to the shaft, it is a stationary machine element and is used for transmission of bending moment only, and it acts as a support for some rotating body such as car wheel.

A spindle is a short shaft that imparts motion either to a cutting tool (drill press spindles) or to a work piece (lathe spindles)

Standard sizes of transmission shafts

25 mm to 60 mm with 5 mm steps

60 mm to 110 mm with 10 mm steps

110 mm to 140 mm with 15 mm steps

140 mm to 500 mm with 20 mm steps

The standard lengths of shafts are 5 m, 6 m and 7 m.

Stresses in shafts

1. Shear stresses due to the transmission of torque (torsional load)

2. Bending stresses (tensile or compressive) due to the forces acting upon machine elements like gears, pulleys as well as due to the weight of the shaft itself.

3. Stresses due to combined torsional and bending loads.

Design of shafts:

(a) Shafts subjected to twisting moment only

(b) Shafts subjected to bending moment only

(c) Shafts subjected to combined twisting and bending moments.

1. Shafts subjected to twisting moment (torque):

The diameter of the shaft may be obtained by using the torsion equation:

$$\frac{T}{J} = \frac{\tau_s}{r}$$

$$T - twisting\,moment\,(torque)$$

$$J - polar\,moment\,of\,inertia$$

$$\tau_s - torsional\,shear\,stress$$

$$r - radius\,of\,the\,shaft$$

$$J = \frac{\pi}{32} \times d^4 \qquad\qquad r = \frac{d}{2}$$

$$\frac{T}{\frac{\pi}{32} \times d^4} = \frac{\tau_s}{\frac{d}{2}}$$

$$T = \frac{\pi}{16} \times \tau_s \times d^3 \qquad for\,solid\,shafts$$

And for hollow shafts the polar moment of inertia

$$J = \frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right] \qquad\qquad r = \frac{d_o}{2}$$

$$\frac{T}{\frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right]} = \frac{\tau_s}{\frac{d_o}{2}} \qquad or$$

$$T = \frac{\pi}{16} \times \tau_s \times \left[\frac{(d_o)^4 - (d_i)^4}{d_o}\right]$$

$$k = \frac{d_i}{d_o} \qquad ratio\,of\,inside\,and\,outside\,diameter\,of\,the\,shaft$$

$$T = \frac{\pi}{16} \times \tau_s \times (d_o)^3 \times (1 - k^4)$$

The power transmitted by the shaft (in watts)

$$P = T \times w \qquad w - angular\ velocity\ in\ rad/s$$

$$w = \frac{2\pi N}{60}$$

$$P = \frac{2\pi N \times T}{60} \qquad or \qquad T = \frac{P \times 60}{2\pi N}$$

$T - twisting\ moment$

$N - speed\ of\ the\ shaft\ in\ R.P.M$

In case of belt drives, the twisting moment is given by

$$T = (T_1 - T_2) \times R$$

$T_1\ ,T_2\ - tensions\ in\ tight\ and\ slack\ side\ of\ the\ belt$

$R - radius\ of\ the\ pulley$

2. Shafts subjected to bending moment:

When the shaft is subjected to a bending moment only then the maximum stress (tensile or compressive) is given by the bending equation:

$$\frac{M}{I} = \frac{S_b}{y}$$

$M - bending\ moment$

$I - moment\ of\ inertia\ of\ the\ shaft\ about\ the\ axis\ of\ rotation$

$S_b - bending\ stress$

$y - distance\ from\ nutral\ axis\ to\ the\ outer\ surface$

Moment of inertia for the solid shaft is

$$I = \frac{\pi}{64} \times d^4 \qquad y = \frac{d}{2}$$

$$\therefore \frac{M}{\frac{\pi}{64} \times d^4} = \frac{S_b}{\frac{d}{2}} \qquad or$$

$$M = \frac{\pi}{32} \times S_b \times d^3$$

And moment of inertia for hollow shaft is

$$I = \frac{\pi}{64}\left[(d_o)^4 - (d_i)^4\right] = \frac{\pi}{64} \times (d_o)^4 \times (1 - k^4) \qquad where \; k = \frac{d_i}{d_o}$$

$$y = \frac{d_o}{2}$$

$$\frac{M}{\frac{\pi}{64} \times (d_o)^4 \times (1 - k^4)} = \frac{S_b}{\frac{d_o}{2}} \qquad or$$

$$M = \frac{\pi}{32} \times S_b \times (d_o)^3 \times (1 - k^4)$$

3. Shafts subjected to combined twisting and bending moments:

(A) Maximum shear stress theory or Guest's theory; it is used for ductile materials such as mild steel

$$\tau_{\max} = \frac{1}{2}\sqrt{(S_b)^2 + 4\tau^2}$$

$$\tau_{\max} = \frac{1}{2}\sqrt{\left(\frac{32M}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2} = \frac{16}{\pi d^3} \times \sqrt{M^2 + T^2} \qquad or$$

$$\frac{\pi}{16} \times \tau_{\max} \times d^3 = \sqrt{M^2 + T^2}$$

$$\sqrt{M^2 + T^2} = T_e$$

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

T_e -Equivalent twisting moment: it is the twisting moment which when acting alone, produces the same shear stress as the actual twisting moment.

(b) Maximum normal stress theory or Rankin's theory it is used for brittle materials such as cast iron.

$$S_{b\max} = \frac{1}{2}(S_b) + \frac{1}{2}\sqrt{(S_b)^2 + 4\tau^2}$$

$$= \frac{1}{2} \times \frac{32\ M}{\pi\ d^3} + \frac{1}{2}\sqrt{\left(\frac{32\ M}{\pi\ d^3}\right)^2 + 4\left(\frac{16\ T}{\pi\ d^3}\right)^2}$$

$$= \frac{32}{\pi\ d^3}\left[\frac{1}{2}(M + \sqrt{M^2 + T^2})\right] \qquad or$$

$$\frac{\pi}{32} \times S_{b\max} \times d^3 = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$$

$$\frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = M_e$$

$$\therefore\ M_e = \frac{\pi}{32} \times S_b \times d^3$$

M_e - Equivalent bending moment: it is the moment which when acting alone produces the same tensile or compressive stress as the actual bending moment.

Note: in case of hollow shafts the equivalent equations are written as follows:

$$T_e = \frac{\pi}{16} \times \tau \times (d_o)^3 \times (1 - k^4)$$

$$M_e = \frac{\pi}{32} \times S_b \times (d_o)^3 \times (1 - k^4)$$

Ex 1: A line shaft rotating at 200 R.P.M. is to transmit 20 KW. The shaft may be assumed to be made of mild steel with an allowable shear stress of 42 MPs. determine the diameter of the shaft, neglecting the bending moment.

Sol:

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 200} = 955\ N.m = 955 \times 10^3\ N.mm$$

$$T = \frac{\pi}{16} \times \tau \times d^3$$

$$955 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3 = 8.25 d^3$$

$$d^3 = \frac{955 \times 10^3}{8.25} = 115733 \qquad or \quad d = 48.7\,mm$$

$$or\ d \cong 50\,mm$$

Ex 2: A solid shaft is transmitting 1 MW at 240 R.P.M. determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20 %. Take the maximum shear stress as 60 MPs.

Sol:

$$T_{mean}=\frac{P\times 60}{2\pi N}=\frac{1\times 10^{6}\times 60}{2\pi\times 240}=39784N.m=39784\times 10^{3}\ N.mm$$

$$T_{\max}=1.2T_{mean}=1.2\times 39784\times 10^{3}=47741\times 10^{3}\ N.mm$$

$$47741\times 10^{3}=\frac{\pi}{16}\times\tau\times d^{3}=\frac{\pi}{16}\times 60\times d^{3}=11.78d^{3}$$

$$d^{3}=\frac{47741\times 10^{3}}{11.78}=4053\times 10^{3}\qquad or\ \ d=159.4\,mm$$

$$or\quad d\cong 160\ mm$$

Ex 3: Find the diameter of a solid steel shaft to transmit 20 KW at 200 R.P.M. the ultimate shear stress for the steel may be taken as 360 MPs and a factor of safety as 8.

If a hollow shaft

Is to be used in place of the solid shaft, find the inside and outside diameter when the ratio of inside to outside diameters is 0.5 .

Sol:

$$F.S = \frac{\tau_{ul}}{\tau_{working}}$$

$$\tau = \frac{\tau_{ul}}{F.S} = \frac{360}{8} = 45\ N/mm^2$$

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 200} = 955 N.m = 955 \times 10^3\ N.mm$$

1. *diameter of the solid shaft*

$$T = \frac{\pi}{16} \times \tau \times d^3$$

$$955 \times 10^3 = \frac{\pi}{16} \times 45 \times d^3 = 8.84 d^3$$

$$d^3 = \frac{955 \times 10^3}{8.84} = 108032 \qquad or \qquad d = 47.6\ mm \cong 50 mm$$

2. *diameter of hollow shaft*

$$955 \times 10^3 = \frac{\pi}{16} \times \tau \times (d_o)^3 \times (1 - k^4) = \frac{\pi}{16} \times 45 \times (d_o)^3 \times \{1 - (0.5)^4\}$$

$$= 8.3 \times (d_o)^3$$

$$(d_o)^3 = \frac{955 \times 10^3}{8.3} = 115060 \qquad or \qquad d_o = 48.6\ mm \ \cong 50\ mm$$

$$d_i = 0.5 \times d_o \ = 0.5 \times 50 = 25\ mm$$

Ex 4: A solid circular shaft is subjected to a bending moment of 3000 N.m, and a torque of 10000 N.m , the shaft is made of 45 C 8 steel having ultimate tensile stress of 700 MPs, and an ultimate shear stress of 500 MPs. Assuming the factor of safety as 6, determine the diameter of the shaft.

Sol:

$$S_t \ or \ S_b = \frac{S_{ul}}{F.S} = \frac{700}{6} = 116.7 \ N.mm^2$$

$$\tau = \frac{\tau_{ul}}{F.S} = \frac{500}{6} = 83.3 \ N.mm^2$$

$$S_t - tensilestress \qquad S_b - bendingstress$$

$$S_{ul} \ , \tau_{ul} \ - ultimatetensile \ and \ ultimateshear \ stress$$

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3\times10^6)^2 + (10\times10^6)^2} = 10.44\times10^6 \ N.mm$$

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

$$10.44\times10^6 = \frac{\pi}{16} \times 83.3 \times d^3 = 16.36 \ d^3$$

$$d^3 = \frac{10.44\times10^6}{16.36} = 0.636\times10^6 \qquad or \qquad d = 86 mm$$

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = \frac{1}{2}[M + T_e]$$

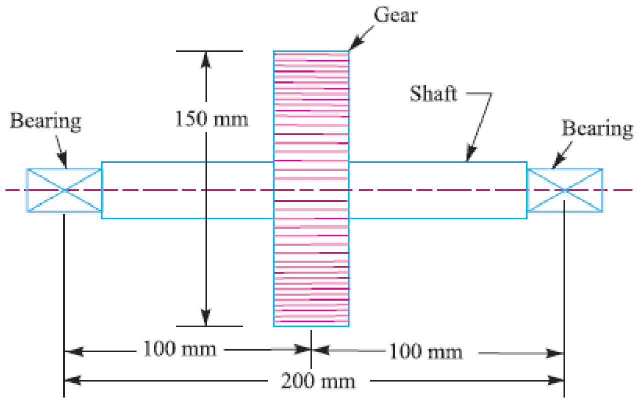
$$= \frac{1}{2}(3\times10^6 + 10.44\times10^6) = 6.72\times10^6 \ N.mm$$

$$M_e = \frac{\pi}{32} \times S_b \times d^3$$

$$6.72\times10^6 = \frac{\pi}{32} \times 116.7 \times d^3 = 11.46 \ d^3$$

$$d^3 = \frac{6.72\times10^6}{11.46} = 0.586\times10^6 \qquad or \qquad d = 83.7 \ mm$$

Ex 5: A shaft is supported at the ends in ball bearing carries a spur gear at its mid span and is to transmit 7.5 KW at 300 R.P.M, the diameter of the gear is 150 mm, the distances between the centre line of bearing and gear is 100 mm each. If the shaft is made of steel and the allowable shear stress is 45 MPs, determine the diameter of the shaft. The pressure angle of the teeth of the gear may take as 20^0 .



Sol:

$$T = \frac{P \times 60}{2\pi N} = \frac{7500 \times 60}{2\pi \times 300} = 238.7 \ N.m$$

the ta gential force on the gear

$$T = F_t \times \frac{D}{2}$$

$$F_t = \frac{2 \times T}{D} = \frac{2 \times 238.7}{0.15} = 3182.7 \ N$$

the normal force on the tooth of gear

$$F_n = \frac{F_t}{\cos 20^0} = \frac{3182.7}{0.9397} = 3387 \ N$$

the bending moment at the centre of gear is

$$M = \frac{F_n \times l}{4} = \frac{3387 \times 0.2}{4} = 169.4 \ N.m$$

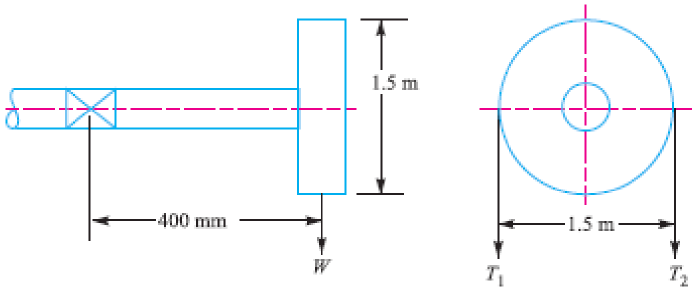
$$T_e = \sqrt{M^2 + T^2} = \sqrt{(169.4)^2 + (238.7)^2} = 292.7 \ N.m = 292.7 \times 10^3 \ N.mm$$

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

$$292.7 \times 10^3 = \frac{\pi}{16} \times 45 \times d^3 = 8.84 \ d^3$$

$$d^3 = \frac{292.7 \times 10^3}{8.84} = 33 \times 10^3 \qquad or \qquad d = 32 \ mm \cong 35 \ mm$$

Ex 6: A line shaft is driven by means of a motor placed vertically below it. The pulley on the line shaft is 1.5 meter in diameter and has belt tensions 5.4 KN on the tight side and 1.8 KN on the slack side of the belt. Both these tensions may assume to be vertical, as in fig. find the diameter of the shaft if the maximum allowable shear stress is 42 MPs.



Sol:

$$T=(T_1-T_2)\times R=(5400-1800)\times 0.75=2700\,N.m=2700\times 10^3\ N.mm$$

the total vertical load on the pulley is

$$W=T_1+T_2=5400+1800=7200\ N$$

$\therefore$ *bending moment*

$$M=W\times L=7200\times 400=2880\times 10^3\ N.mm$$

$$T_e=\sqrt{M^2+T^2}=\sqrt{(2880\times 10^3)^2+(2700\times 10^3)^2}=3950\times 10^3\ N.mm$$

$$T_e=\frac{\pi}{16}\times\tau\times d^3$$

$$3950\times 10^3=\frac{\pi}{16}\times 42\times d^3=8.25d^3$$

$$d^3=\frac{3950\times 10^3}{8.25}=479\times 10^3 \qquad or \qquad d=78\ mm\cong\ 80\ mm$$

استخدام الذكاء الاصطناعي في تصميم الاعمدة

استخدام الذكاء الاصطناعي في تصميم الأعمدة (Shafts) في الهندسة الميكانيكية يمكن أن يحسّن الكفاءة والدقة في التصميم، ويُسهّل من التنبؤ بالأداء والظروف التشغيلية. فيما يلي طرق ومجالات لاستخدام الذكاء الاصطناعي في تصميم الأعمدة:

1. تحليل الإجهاد والإجهادات الديناميكية

- **تعليم الآلة:(Machine Learning)**
  - استخدام نماذج التعلم الآلي لتحليل الإجهادات الثابتة والديناميكية على الأعمدة بناءً على معايير مثل الحمولة، السرعة، والمواد.
  - يمكن تدريب النماذج باستخدام بيانات تحليل العناصر المحدودة (FEA) لتقديم توقعات دقيقة للأداء.

2. اختيار المواد الأمثل

- **الشبكات العصبية الاصطناعية:(ANNs)**
  - تدريب الشبكات العصبية على اختيار المواد المثالية بناءً على المتطلبات، مثل القوة، الكثافة، التكلفة، ومقاومة التآكل.
  - يمكن أن يقترح النظام مواد بديلة لتحسين الأداء أو تقليل التكاليف.

3. تصميم أعمدة خفيفة الوزن باستخدام التحسين التوليدي

- **الذكاء الاصطناعي والتحسين التوليدي:(Generative Design)**
  - توليد تصاميم مبتكرة وخفيفة الوزن للأعمدة باستخدام برامج الذكاء الاصطناعي، مثل **Autodesk Generative Design**، مع مراعاة معايير الأداء والقيود الهندسية.
  - تساعد هذه التقنية في تقليل الوزن دون التضحية بالمتانة.

4. التنبؤ بالعمر الافتراضي والصيانة الوقائية

- **التعلم العميق:(Deep Learning)**
  - تدريب أنظمة AI لتحليل البيانات التشغيلية (مثل الاهتزازات ودرجات الحرارة) لتقدير العمر الافتراضي للأعمدة والتنبؤ بأعطالها قبل حدوثها.
  - يمكن دمج ذلك مع إنترنت الأشياء (IoT) لتوفير مراقبة حية.

5. تقليل الاهتزازات وتحسين الاستقرار

- استخدام خوارزميات الذكاء الاصطناعي لتحليل أنماط الاهتزازات الناتجة عن دوران الأعمدة واقتراح تعديلات في التصميم لتقليل عدم الاستقرار.
- تطبيق أنظمة تحكم ذكية تعمل بالذكاء الاصطناعي لضبط الأعمدة أثناء التشغيل.

6. دمج الذكاء الاصطناعي مع أدوات التصميم الهندسي

- دمج تقنيات الذكاء الاصطناعي مع برامج التصميم مثل **SolidWorks**، **ANSYS**، و **CATIA**لتسريع عمليات التحليل والتصميم.
- تطوير وحدات إضافية للبرامج تتيح التوصيات الذكية بناءً على الظروف الهندسية المختلفة.

الفوائد:

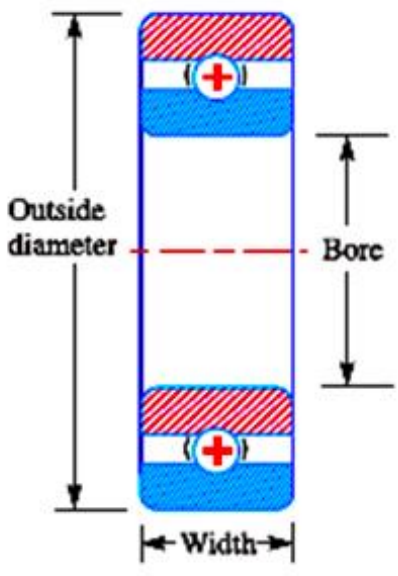
- تحسين الكفاءة والدقة في التصميم.
- تقليل وقت وتكلفة التطوير.
- تقديم حلول مبتكرة لتحديات التصميم.
- تعزيز الفهم العملي والنظري للطلاب.

Rolling contact bearing

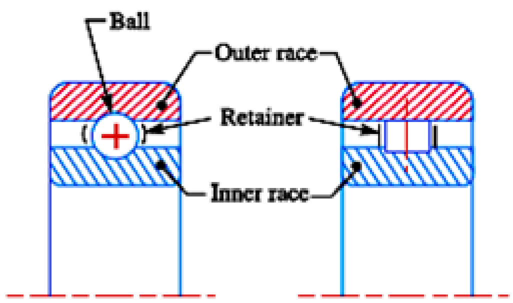
A machine element which supports another moving element(known as ِ rolling) , it permits a relative motion between the contact surface of the members while carrying the load.

Types of bearings

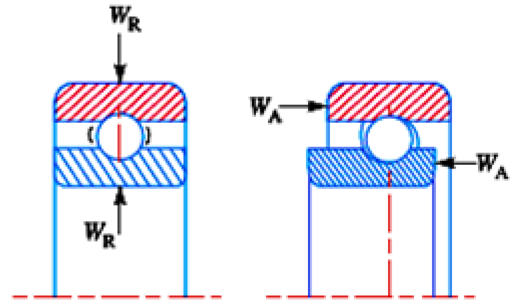
1.ball bearings 2.roller bearings



The ball bearings are usually used for light loads and the roller bearings are used for heavier loads.



(a) Ball bearing. (b) Roller bearing.

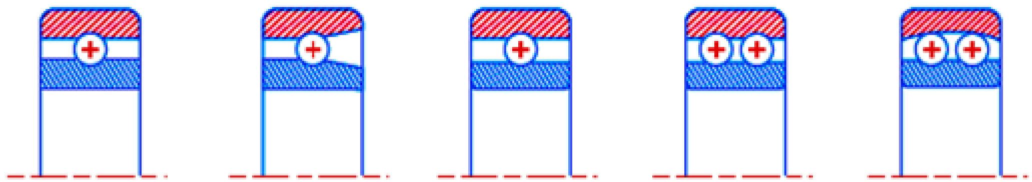


(a) Radial ball bearing. (b) Thrust ball bearing.

the rolling contact bearing, depending upon the load to be carried, are classified as :

a. radial bearings b. thrust bearings

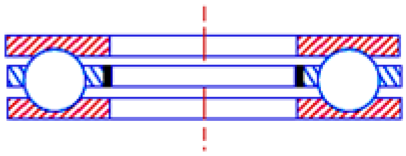
types of radial ball bearings



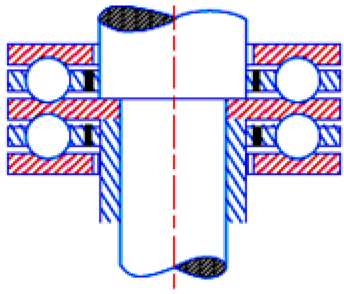
(a) Single row deep groove. (b) Filling notch. (c) Angular contact. (d) Double row. (e) Self-aligning.

Thrust ball bearings

The thrust ball bearings are used for carrying loads at speeds below 2000 R.P.M



(a) Single direction thrust ball bearing.



(b) Double direction thrust ball bearing.

Basic static load rating (C_0) :

it is the static radial load(in case of radial ball or roller bearings) or axial load(in case of thrust ball or roller bearings)which acts on non rotating bearings.

1.for radial ball bearings

$$C_0 = f_0 \times i \times Z \times D^2 \times \cos\alpha$$

$$f_0 - factor\,depending\,on\,the\,type\,of\;bearing$$

$$i - No.of\;rows\,of\;balls\;in\;any\,bearing$$

$$Z - No.of\;balls\;per\,row$$

$$D - diameter\,of\;ball\,in\;mm$$

$$\alpha - Nomi\;al\,angle\,of\;contact$$

2.for radial roller bearings

$$C_0 = f_0 \times i \times Z \times l_e \times D \times \cos\alpha$$

$$l_e - length\,of\;roller$$

3.for thrust ball bearings

$$C_0 = f_0 \times Z \times D^2 \times \sin\alpha$$

4.for thrust roller bearings

$$C_0 = f_0 \times Z \times l_e \times D \times \sin\alpha$$

Note:

$f_0 -$ May takes the following values: 3.33 , 12.3 , 21.6 , 49 , 98.1

Static equivalent load(W_{0R}):

$$W_{0R} = X_0 W_R + Y_0 W_A$$
$$X_0 - radialload\ factor$$
$$Y_0 - axialor\, thrustload\ factor$$
$$W_R - radial\ load$$
$$W_A - axial\ load$$

Values of X_0 and Y_0 for radial bearings.

S.No.	Type of bearing	Single row bearing		Double row bearing	
		X_0	Y_0	X_0	Y_0
1.	Radial contact groove ball bearings	0.60	0.50	0.60	0.50
2.	Self aligning ball or roller bearings and tapered roller bearing	0.50	0.22 cot θ	1	0.44 cot θ
3.	Angular contact groove bearings :				
	$\alpha = 15°$	0.50	0.46	1	0.92
	$\alpha = 20°$	0.50	0.42	1	0.84
	$\alpha = 25°$	0.50	0.38	1	0.76
	$\alpha = 30°$	0.50	0.33	1	0.66
	$\alpha = 35°$	0.50	0.29	1	0.58
	$\alpha = 40°$	0.50	0.26	1	0.52
	$\alpha = 45°$	0.50	0.22	1	0.44

Basic dynamic load rating (C):

$$C = f_c\,(i\,\cos\alpha)^{0.7} \times Z^{\frac{2}{3}} \times D^{1.8}$$

$$f_c - dynamic\ factor\ depending\ on\ the\ geomerty\ of\ bearing\ component$$

Dynamic equivalent load (W):

Values of *X* and *Y* for dynamically loaded bearings.

Type of bearing	*Specifications*	$\frac{W_A}{W_R} \le e$		$\frac{W_A}{W_R} > e$		*e*
		X	*Y*	*X*	*Y*	
Deep groove ball bearing	$\frac{W_A}{C_0} = 0.025$				2.0	0.22
	= 0.04				1.8	0.24
	= 0.07				1.6	0.27
	= 0.13	1	0	0.56	1.4	0.31
	= 0.25				1.2	0.37
	= 0.50				1.0	0.44
Angular contact ball bearings	Single row		0	0.35	0.57	1.14
	Two rows in tandem		0	0.35	0.57	1.14
	Two rows back to back	1	0.55	0.57	0.93	1.14
	Double row		0.73	0.62	1.17	0.86
Self-aligning bearings	Light series : for bores					
	10 – 20 mm		1.3		2.0	0.50
	25 – 35	1	1.7	6.5	2.6	0.37
	40 – 45		2.0		3.1	0.31
	50 – 65		2.3		3.5	0.28
	70 – 100		2.4		3.8	0.26
	105 – 110		2.3		3.5	0.28
	Medium series : for bores					
	12 mm		1.0	0.65	1.6	0.63
	15 – 20		1.2		1.9	0.52
	25 – 50		1.5		2.3	0.43
	55 – 90		1.6		2.5	0.39
Spherical roller bearings	For bores :					
	25 – 35 mm		2.1		3.1	0.32
	40 – 45	1	2.5	0.67	3.7	0.27
	50 – 100		2.9		4.4	0.23
	100 – 200		2.6		3.9	0.26
Taper roller bearings	For bores :					
	30 – 40 mm				1.60	0.37
	45 – 110	1	0	0.4	1.45	0.44
	120 – 150				1.35	0.41

Values of service factor (Ks)

S.No.	Type of service	Service factor (K_S) for radial ball bearings
1.	Uniform and steady load	1.0
2.	Light shock load	1.5
3.	Moderate shock load	2.0
4.	Heavy shock load	2.5
5.	Extreme shock load	3.0

the design dynamic load = basic dynamic load × service factor

Life of a bearing(L):

It is the number of revolutions or hours which the bearing runs before the first evidence of fatigue develops in the material.

$$L = \left(\frac{C}{W}\right)^k \times 10^6 \ revolutions$$

$$k = 3 \ for\, ball\, bearings$$

$$k = \frac{10}{3} \ for\, roller\, bearings$$

The relationship between the life in revolutions(L) and the life in working hours ( )is given:

$$L = 60\, N \times L_H$$

$$N - speed\ in\ R.P.M$$

The rating life of a group of bearings is defined as the number of revolutions or hours that 90 percent of a group of bearings will exceed before the first evidence of fatigue develops

The average life of bearing is 5 times the rating life.

The longest life of a single bearing is about 30 –50 times the rating life.

Basic static and dynamic capacities of various types of radial ball bearings

Bearing No.	Basic capacities in kN							
	Single row deep groove ball bearing		Single row angular contact ball bearing		Double row angular contact ball bearing		Self-aligning ball bearing	
	Static (C_0)	Dynamic (C)	Stctic (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)	Static (C_0)	Dynamic (C)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
200	2.24	4	—	—	4.55	7.35	1.80	5.70
300	3.60	6.3	—	—	—	—	—	—
201	3	5.4	—	—	5.6	8.3	2.0	5.85
301	4.3	7.65	—	—	—	—	3.0	9.15
202	3.55	6.10	3.75	6.30	5.6	8.3	2.16	6
302	5.20	8.80	—	—	9.3	14	3.35	9.3
203	4.4	7.5	4.75	7.8	8.15	11.6	2.8	7.65
303	6.3	10.6	7.2	11.6	12.9	19.3	4.15	11.2
403	11	18	—	—	—	—	—	—
204	6.55	10	6.55	10.4	11	16	3.9	9.8
304	7.65	12.5	8.3	13.7	14	19.3	5.5	14
404	15.6	24	—	—	—	—	—	—
205	7.1	11	7.8	11.6	13.7	17.3	4.25	9.8
305	10.4	16.6	12.5	19.3	20	26.5	7.65	19
405	19	28	—	—	—	—	—	—
206	10	15.3	11.2	16	20.4	25	5.6	12
306	14.6	22	17	24.5	27.5	35.5	10.2	24.5
406	23.2	33.5	—	—	—	—	—	—
207	13.7	20	15.3	21.2	28	34	8	17
307	17.6	26	20.4	28.5	36	45	13.2	30.5
407	30.5	43	—	—	—	—	—	—
208	16	22.8	19	25	32.5	39	9.15	17.6
308	22	32	25.5	35.5	45.5	55	16	35.5
408	37.5	50	—	—	—	—	—	—
209	18.3	25.5	21.6	28	37.5	41.5	10.2	18
309	30	41.5	34	45.5	56	67	19.6	42.5
409	44	60	—	—	—	—	—	—
210	21.2	27.5	23.6	29	43	47.5	10.8	18
310	35.5	48	40.5	53	73.5	81.5	24	50
410	50	68	—	—	—	—	—	—

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
211	26	34	30	36.5	49	53	12.7	20.8
311	42.5	56	47.5	62	80	88	28.5	58.5
411	60	78	—	—	—	—	—	—
212	32	40.5	36.5	44	63	65.5	16	26.5
312	48	64	55	71	96.5	102	33.5	68
412	67	85	—	—	—	—	—	—
213	35.5	44	43	50	69.5	69.5	20.4	34
313	55	72	63	80	112	118	39	75
413	76.5	93	—	—	—	—	—	—
214	39	48	47.5	54	71	69.5	21.6	34.5
314	63	81.5	73.5	90	129	137	45	85
414	102	112	—	—	—	—	—	—
215	42.5	52	50	56	80	76.5	22.4	34.5
315	72	90	81.5	98	140	143	52	95
415	110	120	—	—	—	—	—	—
216	45.5	57	57	63	96.5	93	25	38
316	80	96.5	91.5	106	160	163	58.5	106
416	120	127	—	—	—	—	—	—
217	55	65.5	65.5	71	100	106	30	45.5
317	88	104	102	114	180	180	62	110
417	132	134	—	—	—	—	—	—
218	63	75	76.5	83	127	118	36	55
318	98	112	114	122	—	—	69.5	118
418	146	146	—	—	—	—	—	—
219	72	85	88	95	150	137	43	65.5
319	112	120	125	132	—	—	—	—
220	81.5	96.5	93	102	160	146	51	76.5
320	132	137	153	150	—	—	—	—
221	93	104	104	110	—	—	56	85
321	143	143	166	160	—	—	—	—
222	104	112	116	120	—	—	64	98
322	166	160	193	176	—	—	—	—

Examples for bearings

Ex1:

Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 R.P.M , for an average life of 5 years at 10 hours per day .assume uniform and steady load.

Sol:

$$_H = 5\times300\times10 = 15000 \ hours \qquad (300\, working\, day\ per\ year)$$

$$= 60\,N \times L_H = 60\times1600\times15000 = 1440\times10^6 \ revolutions$$

From the table for dynamically loaded bearings let us take the value

$$\frac{W_A}{C_0} = 0.5$$

$$\frac{W_A}{W_R} = \frac{5000}{4000} = 1.25 \qquad which\, is\ greater\, than \quad e = 0.44$$

$$then \quad X = 0.56 \qquad Y = 1$$

$$W = X\,V\,W_R + Y\,W_A \qquad V = 1$$

$$= 0.56\times1\times4000 + 1\times5000 = 7240\ N$$

$$L = \left(\frac{C}{W}\right)^k \times10^6 \quad rev.$$

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{k}} = 7240\times\left(\frac{1440\times10^6}{10^6}\right)^{\frac{1}{3}} = 81760N = 81.76\ KN$$

$$from\ the\ table\ we\ select\, the\ bearing\, No.315$$

Ex2:

Design a self aligning ball bearing for a radial load of 7000 N and a thrust load of 2100 N the desired life of bearing is 160 millions of revolutions at 300 R.P.M . Assume uniform and steady load.

Sol:

$$\frac{W_A}{W_R}=\frac{2100}{7000}=0.3$$

$$From\,table\,we\;find\,that\;for\,a\,self\;aligning\,ball\,bearing$$

$$X=0.65 \qquad\qquad Y=3.5$$

$$W=X\,V\,W_R+Y\,W_A=0.65\times1\times7000+3.5\times2100=11900\;N$$

$$from\,table\,we\;find\,that\,the\,service\;factor\;K_s=1$$

$$\therefore the\,bearing\,should\,be\,selected\;for\;W=11900\;N$$

$$L=\left(\frac{C}{W}\right)^k\times10^6\quad rev.$$

$$C=W\left(\frac{L}{10^6}\right)^{1/k}=11900\times\left(\frac{160\times10^6}{10^6}\right)^{1/3}=64600\;N=64.6\;KN$$

$$from\;table\,we\;select\;bearing\,number\,219\;having\,C=65.5\,KN$$

Ex3:

A single row angular contact ball bearing number 310 is used for an axial flow compressor. The bearing is to carry a radial load of 2500 N, and an axial load or thrust load of 1500 N. assuming light shock load, determine the rating life of bearing.

Sol:

$$\frac{W_A}{W_R}=\frac{1500}{2500}=0.6$$

From table we find the values X and Y

$$X = 1 \qquad\qquad Y = 0$$

$$W = XVW_R + YW_A$$

$$V = 1 \qquad \textit{for most bearings}$$

$$W = 1 \times 1 \times 2500 + 0 \times 1500 = 2500N$$

From table we find the service factor for light shock load

$$K_S = 1.5$$

the design dynamic load = basic dynamic load × service factor

$$W = W \times K_S = 2500 \times 1.5 = 3750N$$

From table we find the basic dynamic load for the bearing number 310 is:

C = 53 KN = 53000 N

$$L = \left(\frac{C}{W}\right)^k \times 10^6 = \left(\frac{53000}{3750}\right)^3 \times 10^6 = 2823 \times 10^6 \ \textit{revolutions}$$

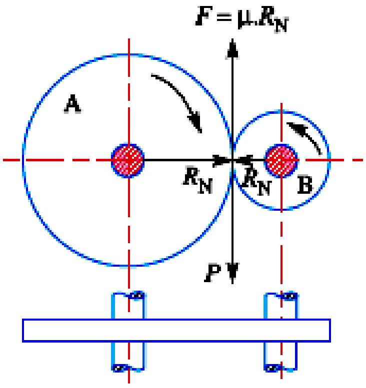
Gears

Gears are an adaptation of rolling cylinders and cones, designed to ensure positive motion.

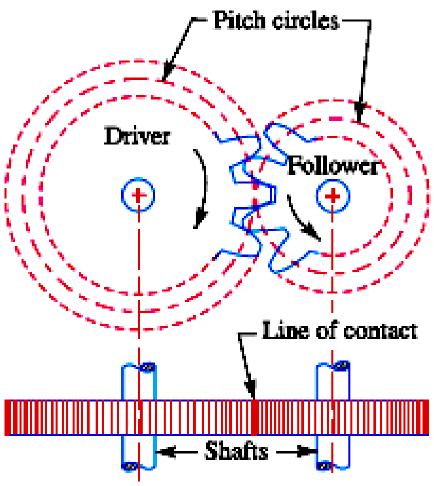
Gears are used to transmit power between shafts rotating usually at different speeds.

Types of Gears

Spur gears, have teeth parallel to the axis of rotation and are used to transmit motion
from one shaft to another, parallel, shaft. Of all types, the spur gear is the simplest .

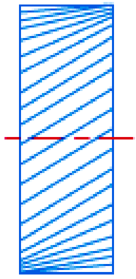


Friction wheels.

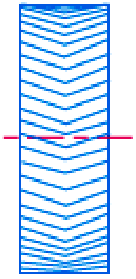


Gear or toothed wheel.

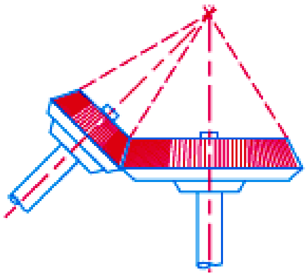
Helical gears have teeth inclined to the axis of rotation. Helical gears can be used for
the same applications as spur gears and, when so used, are not as noisy, because of the
more gradual engagement of the teeth during meshing. The inclined tooth also develops thrust loads and bending couples, which are not present with spur gearing.
Sometimes helical gears are used to transmit motion between nonparallel shafts.



Single helical gear.



Double helical gear.

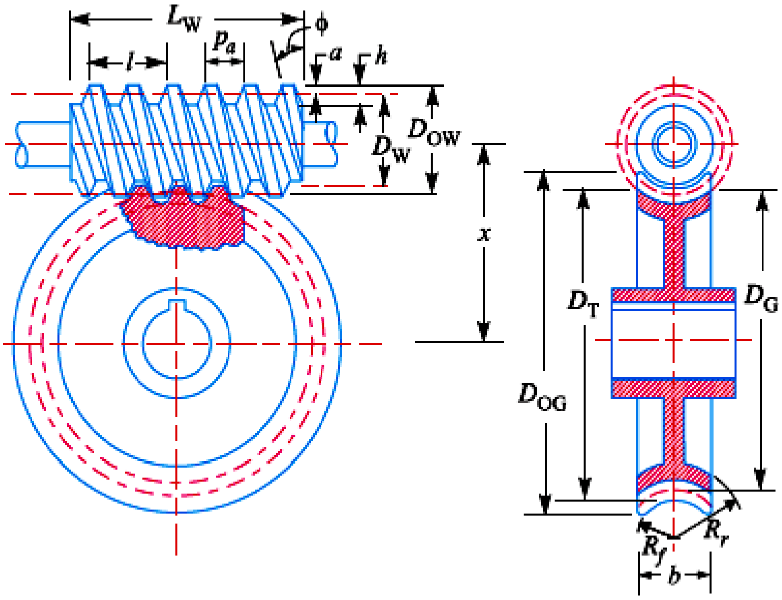


Bevel gear.

Bevel gears, have teeth formed on conical surfaces and are used mostly for transmitting motion between intersecting shafts.

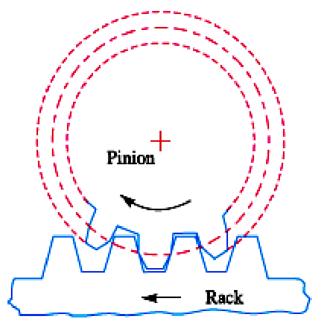
Worms and *worm gears*, represent the fourth basic gear type. the worm resembles a screw.
The direction of rotation of the worm gear, also called the worm wheel, depends upon
the direction of rotation of the worm and upon whether the worm teeth are cut right-hand or left-hand. Worm-gear sets are also made so that the teeth of one or both wrap
partly around the other. Worm-gear sets are mostly used when the speed ratios of the two

shafts are quite high, say, 3 or more.

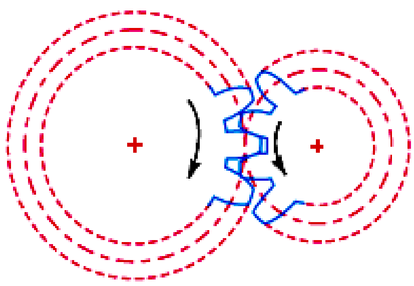


The gears are classified according to the type of gearing as :

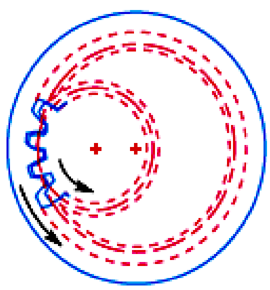
a)external gearing b)internal gearing c)rack and pinion



rack and pinion



External gearing

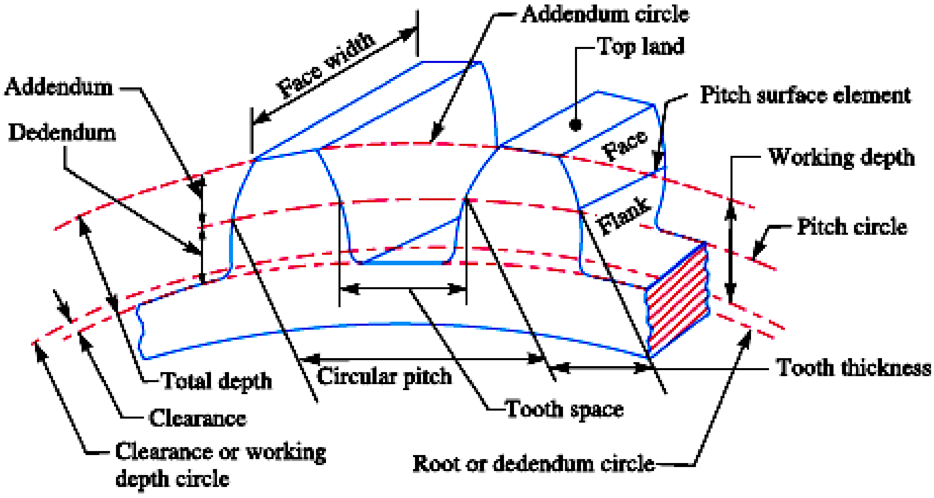


Internal gearing

Terms used in gears

1.pitch circle: it is an imaginary circle which by pure rolling action , would give the same motion as the actual gear.

2.pitch circle diameter: it is the diameter of pitch circle.



3.pressure angle: it is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. It is usually denoted by Φ. The standard pressure angles are 14 $1/2^0$ and 20^0.

4.addendum: it is the radial distance of a tooth from the pitch circle to the top of the tooth.

5.dedendum: it is the radial distance of a tooth from the pitch circle to the bottom of the tooth.

6.circular pitch: it is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth .

$$P_c = \frac{\pi D}{T}$$

D – diameter of pitch circle

T – number of teeth on the wheel.

Note: if D_1 and D_2 are the diameters of two meshing gears having the teeth T_1 and T_2 . then for them to mesh correctly:

$$P_c = \frac{\pi D_1}{T_1} = \frac{\pi D_2}{T_2} \qquad or \qquad \frac{D_1}{D_2} = \frac{T_1}{T_2}$$

7.diametral pitch: it is the ratio of number of teeth of the pitch circle diameter in millimeters.

$$P_d = \frac{T}{D} = \frac{\pi}{P_c}$$

8.module: it is the ratio of the pitch circle diameter in millimeters to the number of teeth.

$$m = \frac{D}{T}$$

Note1: the standard modules are 1 , 1.25 , 1.5 , 2 , 2.5 , 3 , 4 , 5, 6 , 8 , 10 , 12 , 16 , 20 , 25 , 32 , 40 and 50

The modules 1.125,1.375, 1.75 , 2.25, 2.75, 3.5 , 4.5 ,5.5 , 7, 9, 11 , 14 , 18 , 22 , 28 , 36 and 45 are of second choice.

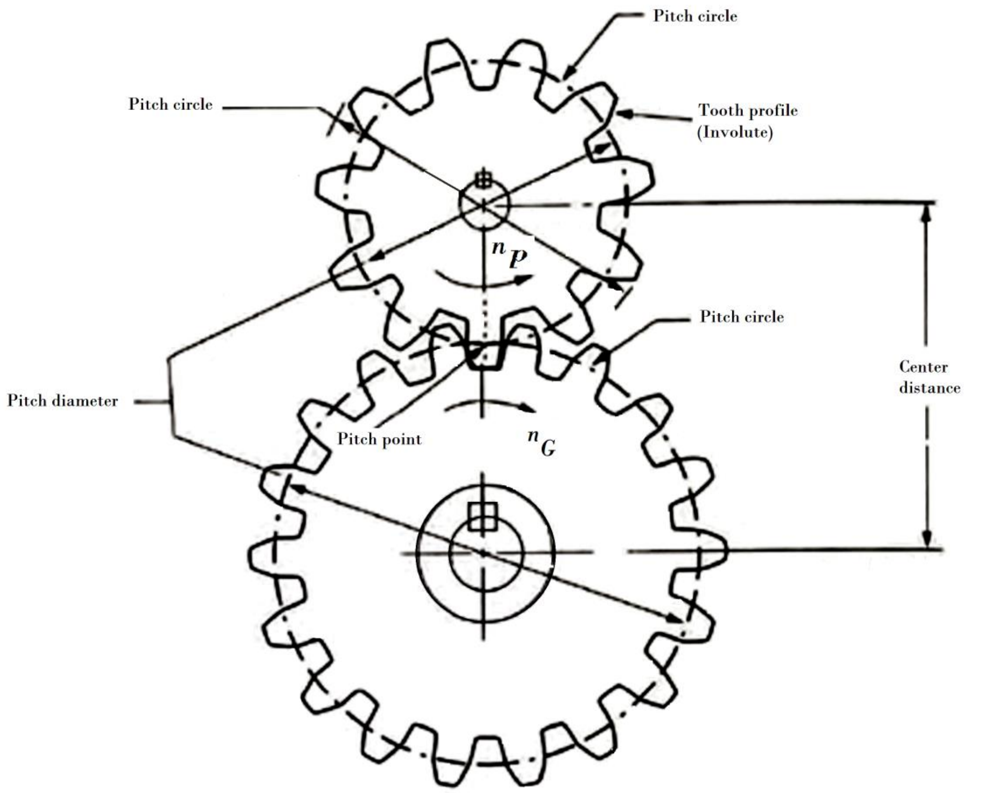
Note2:if D_1 and D_2 are pitch circle diameters of gears 1 and 2 then the velocity ratio is

$$\frac{w1}{w1} = \frac{D_2}{D_1} = \frac{T_2}{T_1}$$

Design of spur gears:

Sometimes, the spur gears (driver and driven) are to be designed for the given velocity ratio and the distance between the centers of their shafts.

Let x – distance between the centers of two shafts



N_1, N_2 - speed of the driver and driven

T_1, T_2 – number of teeth of the driver and driven

d_1 , d_2 – pitch circle diameter of the driver and the driven

$$x = \frac{d_1 + d_2}{2}$$

And speed ratio or velocity ratio

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{T_2}{T_1}$$

Systems of gear teeth:

1. 14 $1/2^0$ composite system

2. 14 $1/2^0$ full depth involute system

3. 20^0 full depth involute system

4. 20^0 stub involute system (has a strong tooth to take a heavy loads)

Standard proportions of gear systems:

The following table shows the standard proportions in module (m) for the four gear systems

Standard proportions of gear systems.

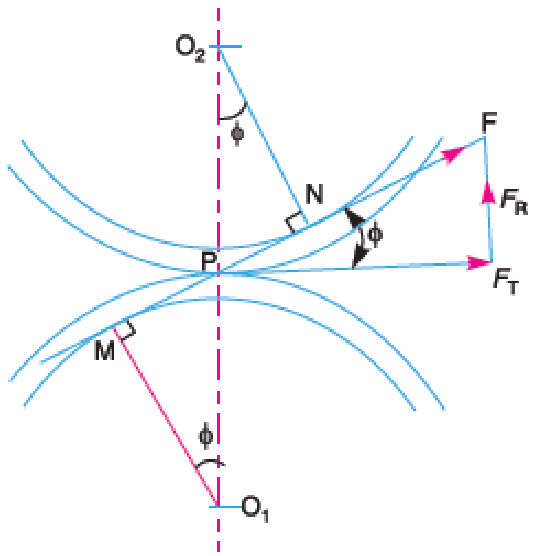
S. No.	Particulars	$14\frac{1}{2}°$ composite or full depth involute system	20° full depth involute system	20° stub involute system
1.	Addendum	1*m*	1*m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*

If *F* is the maximum tooth pressure, then tangential force $F_T = F\cos\phi$

and radial or normal force $F_N = F\sin\phi$

Torque exerted on the gear shaft $T = F_T \times r$

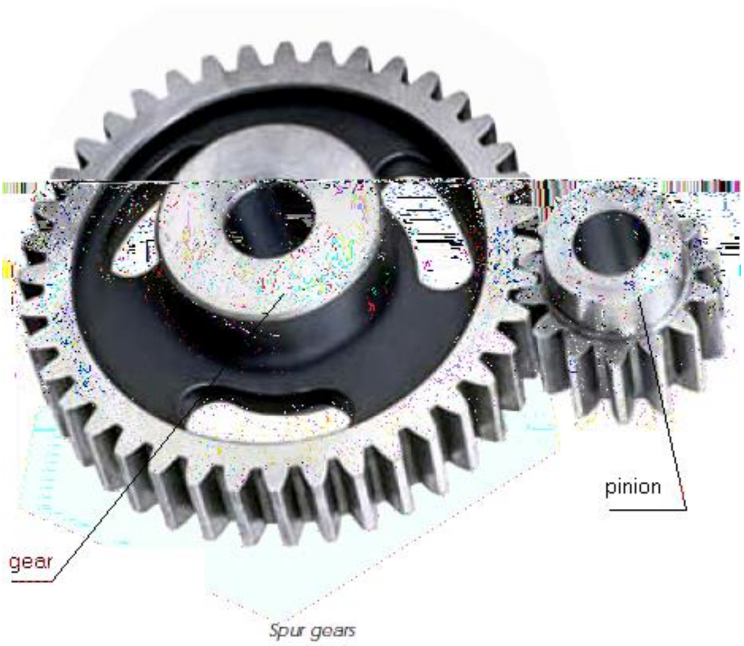
r – radius of the pitch circle of the gear



Minimum number of teeth on the pinion

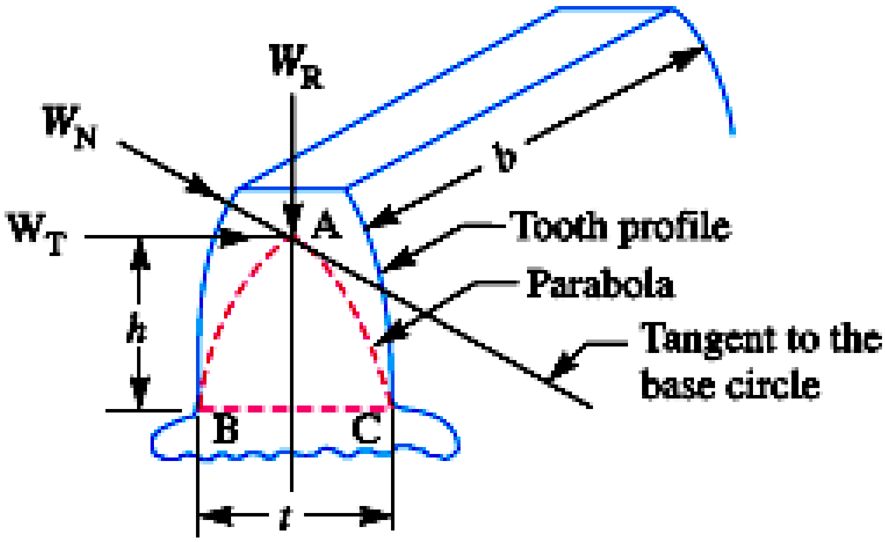
Minimum number of teeth on the pinion

S. No.	Systems of gear teeth	Minimum number of teeth on the pinion
1.	$14\frac{1}{2}^\circ$ Composite	12
2.	$14\frac{1}{2}^\circ$ Full depth involute	32
3.	20° Full depth involute	18
4.	20° Stub involute	14



Strength of gear teeth –Lewis Equation

The maximum bending stress(working stress) at the section BC is :


W_R
W_N
W_T
A
b
Tooth profile
Parabola
Tangent to the
base circle
h
B
C
t

$$S_W = \frac{M.y}{I}$$

$$\therefore M - maximum\ bending\ moment\ at\ the\ critical\ se\ tion$$

$$M = W_T \times h$$

$$W_T - ta\ ngential\ load\ acting\ at\ the\ tooth$$

$$h - length\ of\ the\ tooth$$

$$y - half\ the\ thickness\ of\ the\ tooth$$

$$y = \frac{t}{2}$$

$$I - moment\ of\ inertia\ about\ the\ centre\ line\ of\ the\ tooth$$

$$I = \frac{b.t^3}{12}$$

$$b - width\ of\ gear\ face$$

$$\therefore \quad S_W = \frac{(W_T \times h) \times \frac{t}{2}}{\frac{b \times t^3}{12}} = \frac{(W_T \times h) \times 6}{b \times t^2}$$

$$W_T = \frac{S_W \times b \times t^2}{6\ h} \qquad let \qquad t = x \times p_c \qquad h = k \times p_c$$

$$W_T = S_W \times b \times \frac{x^2 \times p_c{}^2}{6 \times k \times p_c} = S_W \times b \times p_c \times \frac{x^2}{6 \times k}$$

$$let \qquad y = \frac{x^2}{6 \times k}$$

$$W_T = S_W \times b \times p_c \times y = S_W \times b \times \pi m \times y$$

$S_W = S_0 \times C_v$

$S_0 - allowable\ static\ stress$

$C_v\ -\ velocity\ factor$

$C_v = \frac{3}{3+v}$ $for\ ordinary\ cut\ gears operating\ at\ velocities\ upto\ 12.5\ m/s$

$= \frac{4.5}{4.5+v}$ $for\ carefully\ cut\ gears operating at\ velocities\ upto\ 12.5\ m/s$

$= \frac{6}{6+v}$ $for\ very\ accurately\ cut\ and\ ground\ metallic\ gears\ operating$

$at\ velocities\ upto\ 20\ m/s$

$= \left(\frac{0.75}{1+v}\right) + 0.25 \quad for\ non-metallic\ gears$

$v - pitch\ line\ velocity\ in\ \ m/s$

$v = \frac{\pi\ D\ N}{60}$

Design procedure for spur gears:

1.tangential tooth load is obtained from the power transmitted and the pitch line velocity:

$$W_T = \frac{P}{v} \times C_S$$

$W_T - tangential\ tooth\ load\ in\ \ newtons$

$P - power\ transmitted\ \ in\ watts$

$C_S - service\ \ factor$

Values of service factor.

Type of load	Type of service		
	Intermittent or 3 hours per day	8-10 hours per day	Continuous 24 hours per day
Steady	0.8	1.00	1.25
Light shock	1.00	1.25	1.54
Medium shock	1.25	1.54	1.80
Heavy shock	1.54	1.80	2.00

2.apply the Lewis equation as :

$$W_T = S_W \times b \times p_c \times y = S_W \times b \times \pi m \times y = (S_0 \times C_v) \times b \times \pi m \times y$$

$$y = 0124 - \frac{0.684}{T} \quad for \ 14\tfrac{1}{2}^0 \quad coposite\ and\ \ full\ depth\ involute\ system$$

$$= 0.154 - \frac{0.912}{T} \quad for\ \ 20^0 \quad full\ \ depth\ \ involute\ system$$

$$= 0.175 - \frac{0.841}{T} \quad for\ 20^0 \quad stub\ system$$

$$y - tooth\ \ form\ \ factor$$

3.find the static load W_S:

$$W_S = S_0 \times b \times p_c \times y = S_0 \times b \times \pi m \times y$$

W_S should be greater than W_D

4. for steady loads $\quad Ws \geq 1.25\ W_D$

For pulsating loads $\quad Ws \geq 1.35\ W_D$

For shock loads $\quad Ws \geq 1.5\ W_D$

Ex 1:

Two parallel shafts, about 600 mm apart are to be connected by spur gears, one shaft is to run at 360 R.P.M , and the other at 120 R.P.M , design the gears if the circular pitch is to be 25 mm.

Sol:

$$\frac{N_1}{N_2}=\frac{d_2}{d_1}=\frac{360}{120}=3$$

$$d_2=3d_1$$

$$x=\frac{d_1+d_2}{2}$$

$$600=\frac{d_1+d_2}{2}$$

$$d_1+d_2=1200$$

$$d_1+3d_1=1200$$

$$d_1=\frac{1200}{4}=300mm \qquad \Rightarrow d_2=3d_1=3\times 300=900mm$$

$$T_1=\frac{\pi d_1}{P_c}=\frac{\pi\times 300}{25}=37.7$$

$$T_2=\frac{\pi d_2}{P_c}=\frac{\pi\times 900}{25}=113.1$$

The number of teeth on both gears should be in complete numbers, then for the first gear we take T_1 = 38

And the number of the second gear should be T2 = 3 × 38 = 114

Now the exact pitch circle diameter of the first gear is

$$d_1{}'=\frac{T_1\times P_c}{\pi}=\frac{38\times 25}{\pi}=302.36mm$$

And the exact pitch circle diameter of the second gear is

$$d_2{}'=\frac{T_2\times P_c}{\pi}=\frac{114\times 25}{\pi}=907.1mm$$

Exact distance between the two shafts is

$$x' = \frac{d_1' + d_2'}{2} = \frac{302.26 + 907.1}{2} = 604.73mm$$

Ex2:

A bronze spur pinion rotating at 600 R.P.M. drives a cast iron spur gear at a transmission ratio of 4 : 1 the allowable static stresses for the bronze pinion is 48 MPs and for cast iron gear is 105 MPs.

The pinion has 16 standard 20^0 full depth involute teeth of module 8 mm. the face width of both the gears is 90 mm. find the power that can be transmitted.

$$m = \frac{D}{T}$$

$$D_P = m.T_P = 8 \times 16 = 128mm = 0.128\ m$$

$$\therefore v = \frac{\pi\, D_P\, .N_P}{60} = \frac{\pi \times 0.128 \times 600}{60} = 4.02\, m/s$$

$$v\ \ is\ lessthan\ 12.5\ m/s$$

$$\therefore C_v = \frac{3}{3+v} = \frac{3}{3+4.02} = 0.427$$

$$for\, 20^0\ \ full\ depth\ involute\ tooth,$$

$$y_P = 0.154 - \frac{0.912}{T_P} = 0.154 - \frac{0.912}{16} = 0.097$$

$$W_T = S_{WP}.b.\pi\ \ m.y_P = (S_{0P} \times C_v)\, b.\pi\, m.y_P$$

$$= 84 \times 0.427 \times 90 \times \pi \times 8 \times 0.097 = 7870N$$

$$W_T = \frac{P}{v} \times C_S$$

$$for\ \ steady\ load\ 8-10\, hours\, per\ \ day \quad C_S = 1$$

$$P = W_T \times v = 7870 \times 4.02 = 31640 Watt\ =\ 31.64\ \ KW$$

Ex3:

A gear drive is required to transmit a maximum power of 22.5 KW. The velocity ratio is 1:2 and R.P.M of the pinion is 200. The centre distance between the shafts is 600 mm. the teeth has 20^0 stub involute profiles. The static stress for the gear material may be taken as 60 MPs, and face width as 10 times the module. Find the module, face width and number of teeth on each gear.

Sol:

The centre distance L = 600 mm

$$L = \frac{D_P}{2} + \frac{D_G}{2}$$

$$velocity\ ratio = \frac{D_G}{D_P} = 2 \qquad \Rightarrow \qquad D_G = 2\,D_P$$

$$600 = \frac{D_P}{2} + \frac{2\,D_P}{2} = 1.5\,D_P$$

$$D_P = \frac{600}{1.5} = 400\ mm = 0.4\ m$$

$$D_G = 2\ D_P = 2 \times 400 = 800\ mm = 0.8\ m$$

$$v = \frac{\pi\ D_P\ N_P}{60}\ \frac{\pi \times 0.4 \times 200}{60} = 4.2\ m/s$$

$$v\ is\ less\ than\ 12.5\ m/s$$

$$\therefore \quad C_v = \frac{3}{3+v} = \frac{3}{3+4.5} = 0.417$$

$$m = \frac{D_P}{T_P} \qquad \Rightarrow \qquad T_P = \frac{D_P}{m} = \frac{400}{m}$$

$$the\ tooth\ form\ factor\ is:$$

$$y = 0.175 - \frac{0.841}{T_P} = 0.175 - \frac{0.841 \times m}{400}$$

$$= 0.175 - 0.0021\,m$$

$$assuming\ steady\ load\ 8-10\,hours\ of\ service\ per\ day$$

$$C_S = 1$$

$$W_T = \frac{P}{v} \times C_S = \frac{22500}{4.2} \times 1 = 5375\ N$$

$$Lewis\ equation\ for\ pinion$$

$$W_T = S_{WP}.b.\pi\,m.y_P = S_{OP} \times C_V.b.\pi\,m.y_P$$

$$5375 = 60 \times 0.417 \times 10\,m \times \pi\,m \times (0.175 - 0.0021m)$$

$$= 137.6\ m^2 - 1.65\ m^3$$

$$m = 6.5 \qquad \cong\ 8\ mm$$

$$b = 10\ m = 10 \times 8 = 80\ mm$$

$$T_P = \frac{D_P}{m} = \frac{400}{8} = 50$$

$$T_G = \frac{D_G}{m} = \frac{800}{8} = 100$$

Cams

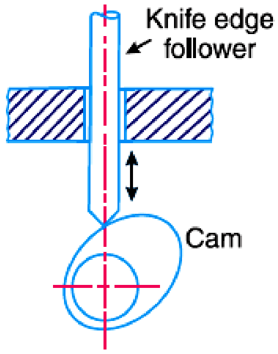
A cam is a rotating machine element which gives reciprocating or oscillating motion to another element known as follower .

the cams are usually rotated at uniform speed by a shaft , but the follower motion is predetermined and will be according to the shape of the cam .

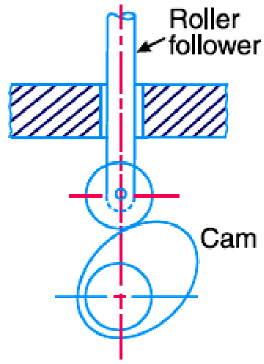
the cams are widely used for operating the inlet and exhaust valves for internal combustion engines, automatic attachment of machineries, paper cutting machines, feed mechanism of automatic lathes etc..

classification of cams

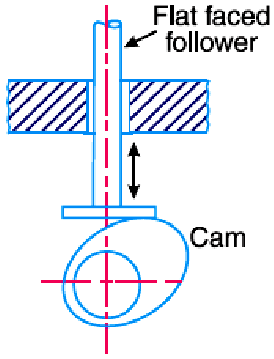
1.radial or disc cam



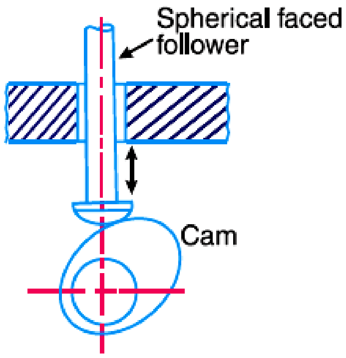
(*a*) Cam with knife edge follower.



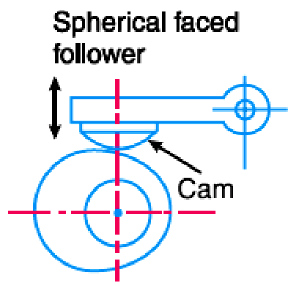
(*b*) Cam with roller follower.



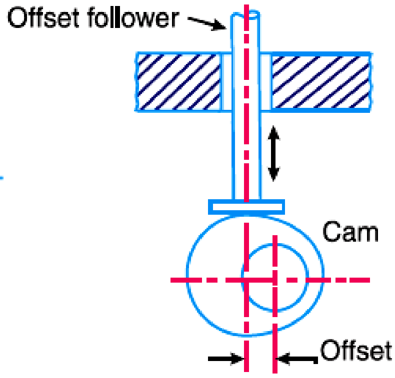
(*c*) Cam with flat faced follower.



(*d*) Cam with spherical faced follower.

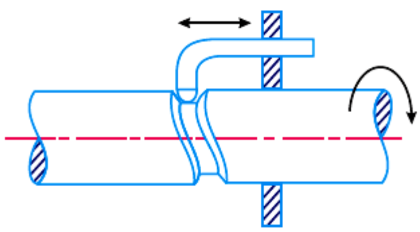


(*e*) Cam with spherical faced follower.

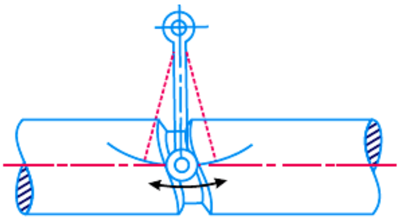


(*f*) Cam with offset follower.

2.cylindrical cam



(*a*) Cylindrical cam with reciprocating follower.



(*b*) Cylindrical cam with oscillating follower.

Cylindrical cam.

Classification of the followers

1.according to the surface in contact

a. knife edge follower

b. roller follower

c. flat face follower

d. spherical face follower

2.according to the motion of the follower

a. reciprocating follower

b. oscillating follower

3.according to the path of motion of the follower

a. radial follower

b. off-set follower

terms used in radial cams

1. base circle: it’s the smallest circle that can be drawn to the cam profile.

2. trace point: it’s a reference point on the follower and it used to generate the pitch curve.

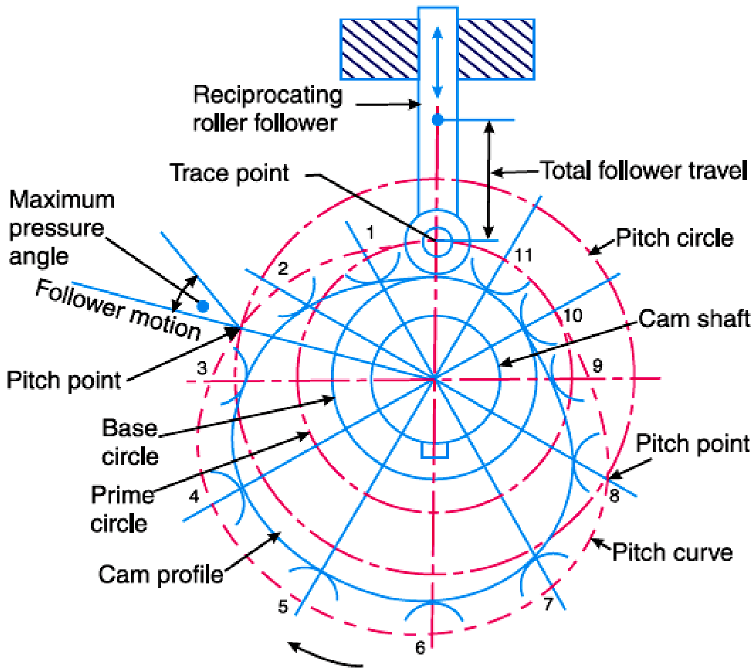
3. pitch curve : it’s the curve generated by trace point as the follower moves relative to the cam.

4. pressure angle : it’s the angle between the direction of the follower motion and a normal to the pitch curve.

5. pitch point :it’s a point on the pitch curve having the maximum pressure angle.

6. prime circle : it’s the smallest circle that can be drawn from the center of the cam and tangent to the pitch curve.

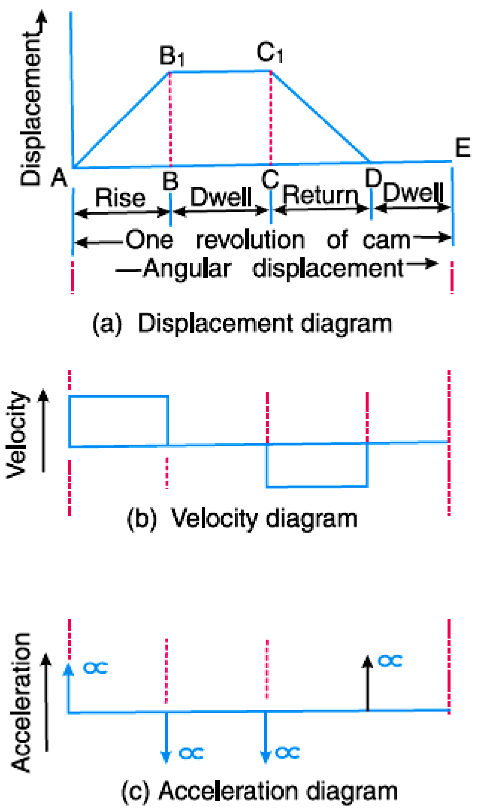
7. stroke : it’s the maximum travel of the follower from its lowest position to the topmost position.

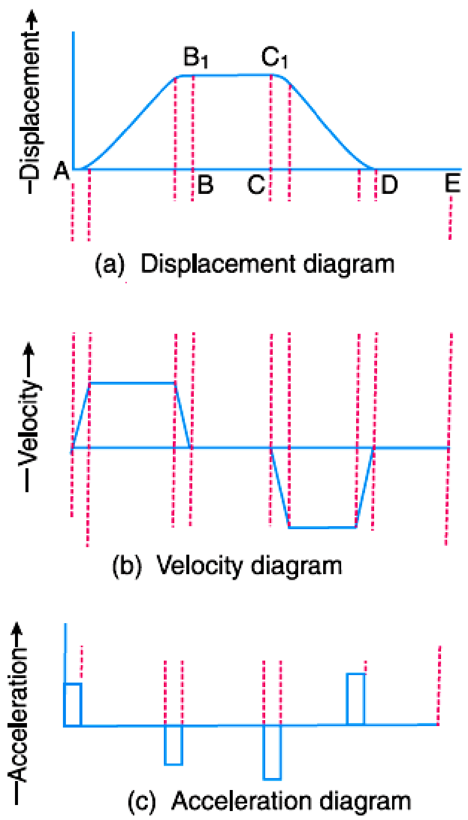


Motion of the follower

1. uniform velocity 2. simple harmonic motion 3. Uniform acceleration and retardation and 4. Cycloidal motion

Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity

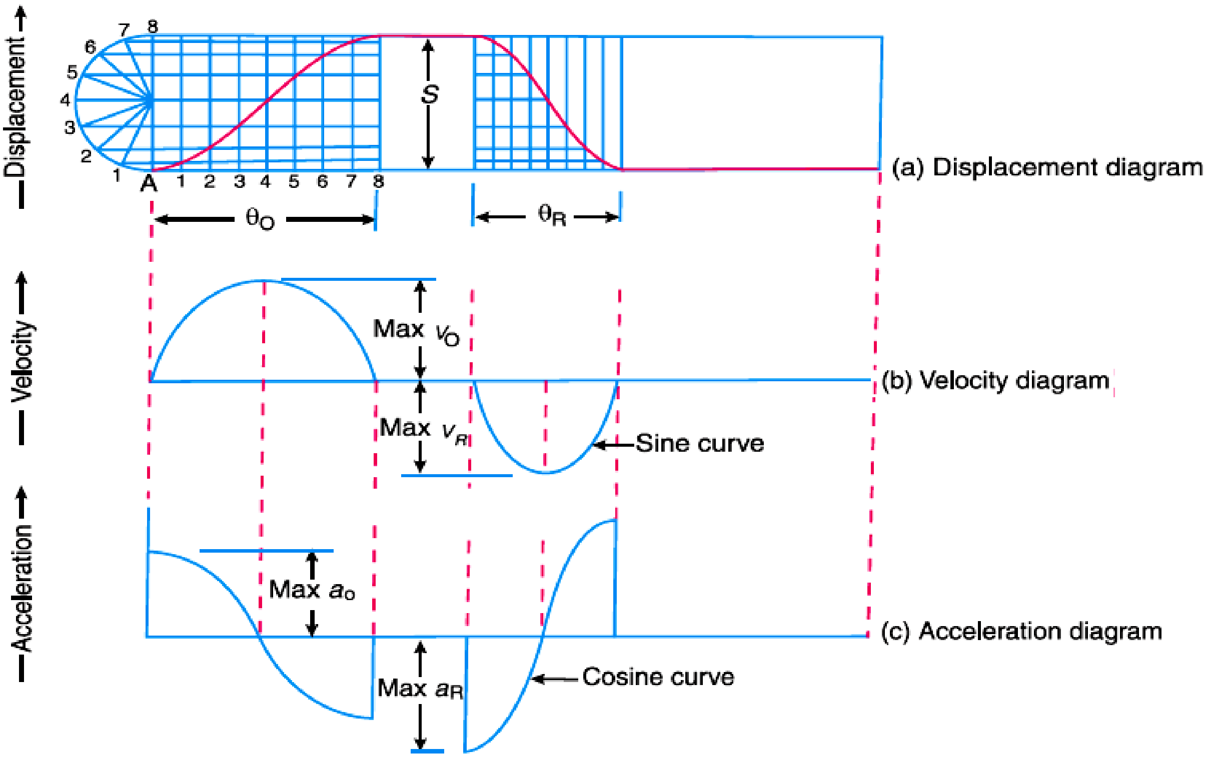




Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity.

Modified displacement, velocity and acceleration diagrams when the follower moves with uniform velocity.

Displacement, velocity and acceleration diagrams when the follower moves with simple harmonic motion



Displacement, velocity and acceleration diagrams when the follower moves with simple harmonic motion.

Let S – stroke of the follower

$\theta_o - \theta_R$ - angular displacement of the cam during out stroke and return stroke of the follower in Radians

ω - angular velocity of the cam in *Rad/s*

t_o - time required for the out stroke of the follower in seconds

$$\omega = \frac{\theta_o}{t_o}$$

$$t_o = \frac{\theta_o}{\omega}$$

The point which defines the motion of the follower on the cam moves at uniform speed round the circumference of a circle of diameter *S* in time $2t_o$

$$\therefore \quad v_o = \frac{\pi\ S}{2\ t_o} = \frac{\pi\ S}{2 \times \dfrac{\theta_o}{\omega}} = \frac{\pi\ S}{2} \times \frac{\omega}{\theta_o} = \frac{\pi\,\omega\,S}{2\theta_o}$$

$$v_o = v_p$$

Maximum acceleration of the follower on the outstroke

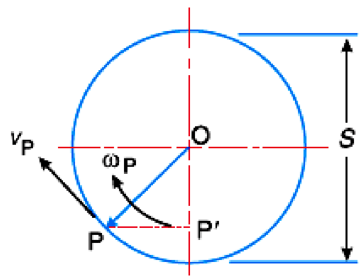
$$a_o = a_p = \frac{(v_p)^2}{OP} = \frac{\left(\dfrac{\pi\,\omega\,S}{2\ \theta_o}\right)^2}{\dfrac{S}{2}} = \frac{\pi^2\omega^2\ S}{2\ \theta_o^{\ 2}}$$

Similarly, maximum velocity of the follower on the return stroke

$$v_R = \frac{\pi\omega S}{2\theta_R}$$

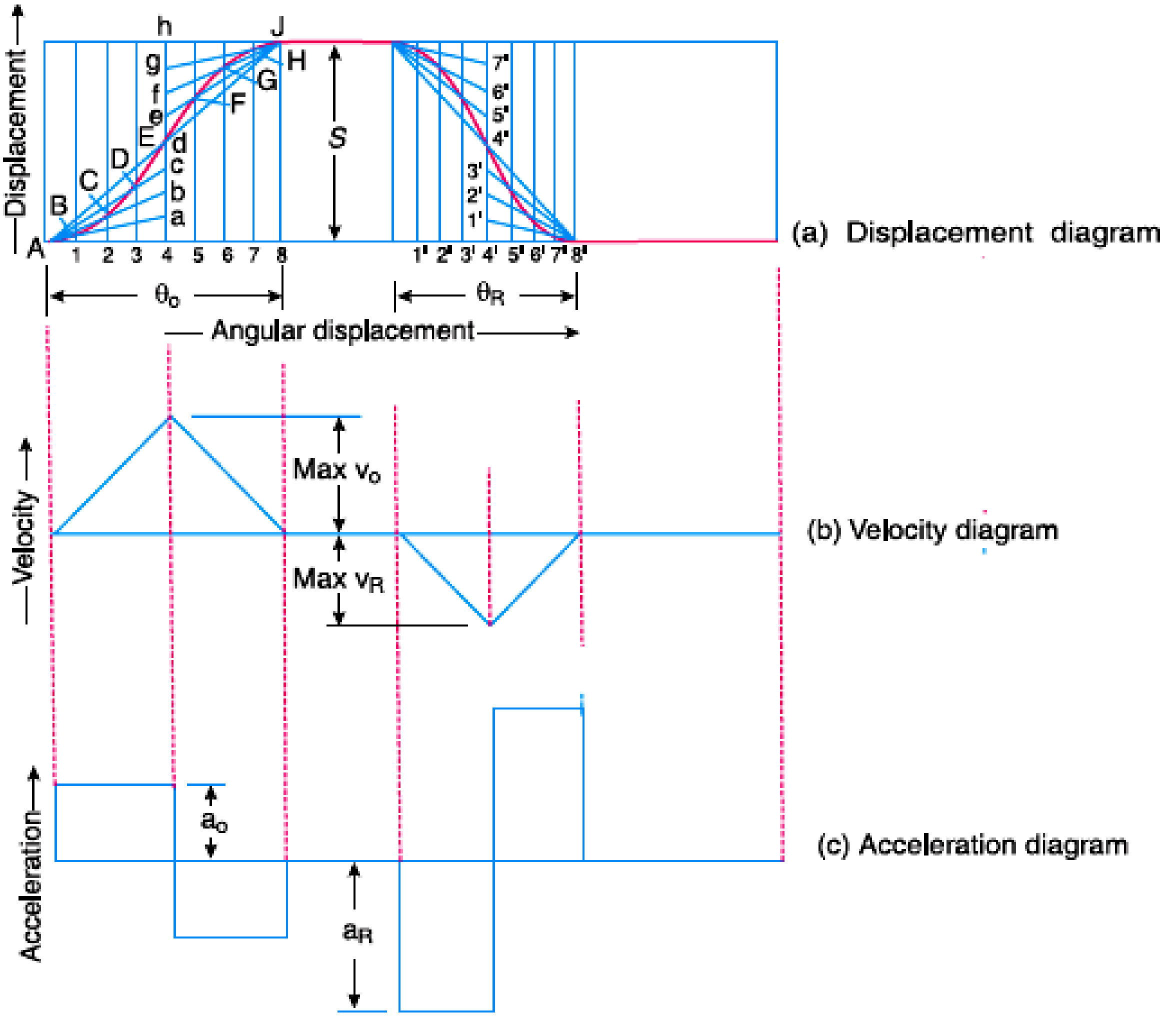
Maximum acceleration of the follower on the return stroke

$$a_R = \frac{\pi^2\omega^2\ S}{2\ \theta_R^{\ 2}}$$



Motion of a point.

Displacement, velocity and acceleration diagrams when the follower moves with uniform acceleration and retardation



Displacement, velocity and acceleration diagrams when the follower moves with uniform acceleration and retardation.

Mean velocity of the follower during out stroke $v_o = \dfrac{S}{t_o}$

Mean velocity of the follower during return stroke $v_R = \dfrac{S}{t_R}$

The maximum velocity of the follower is equal to twice mean velocity

$$v_o = \frac{2\,S}{t_o} = \frac{2\,\omega\,S}{\theta_o}$$

$$v_R = \frac{2\,\omega\,S}{\theta_R}$$

The maximum velocity of the follower is reached after the time $\frac{t_o}{2}$ during the out stroke and $\frac{t_R}{2}$ during the return stroke

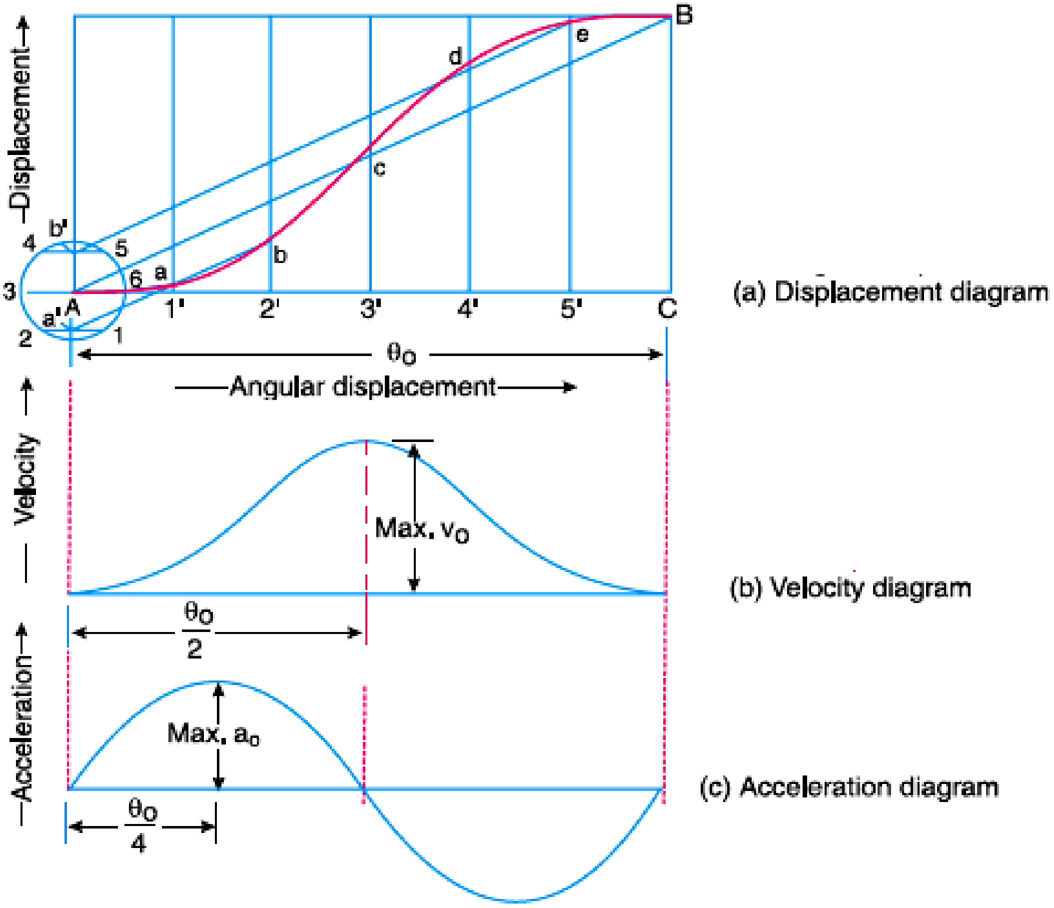
Maximum acceleration of the follower on the outstroke

$$a_o = \frac{v_o}{\frac{t_o}{2}} = \frac{2\,v_o}{t_o} = \frac{2\times 2\omega\,S}{t_o\,\theta_o} = \frac{4\,\omega^2\,S}{\theta_o^{\,2}}$$

Similarly, maximum acceleration of the follower on the return stroke

$$a_R = \frac{4\,\omega^2\,S}{\theta_R^{\,2}}$$

Displacement, velocity and acceleration diagrams when the follower moves with cycloidal motion



Displacement, velocity and acceleration diagrams when the follower moves with cycloidal motion.

Maximum velocity of the follower during out stroke is : $v_o = \dfrac{2\omega S}{\theta_o}$

Similarly, maximum velocity of the follower during return stroke is :

$$v_R = \frac{2\,\omega\,S}{\theta_R}$$

Maximum acceleration of the follower during out stroke is :

$$a_o = \frac{2\,\pi\,\omega^2\,S}{(\theta_o)^2}$$

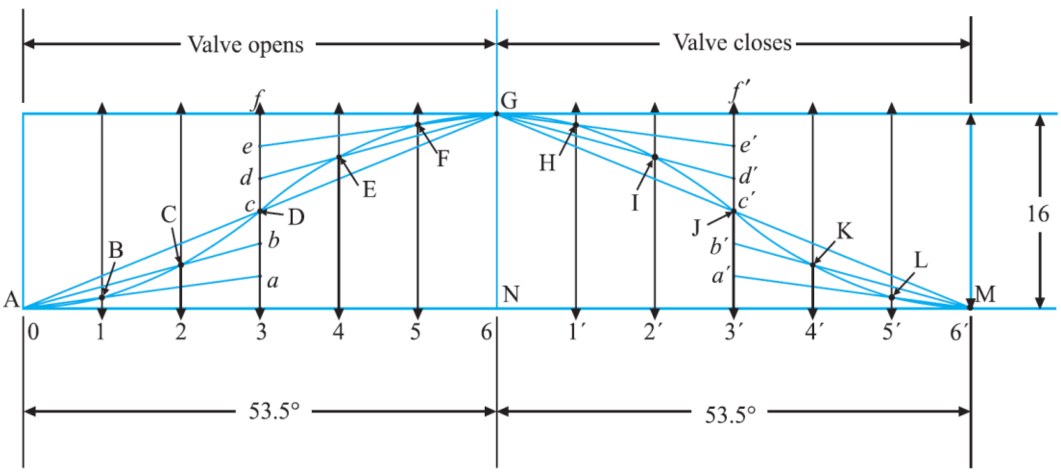
Similarly, maximum acceleration of the follower during

return stroke is : $a_R = \frac{2\pi\omega^2 S}{(\theta_R)^2}$

Design of cam

The cam is forged as one piece with the camshaft. It is designed as discussed below : The diameter of camshaft (*D*2) is taken empirically as *D*2 = 0.16 × Cylinder bore + 12.7 mm = 0.16 × 140 + 12.7 = 35.1 say 36 mm
The base circle diameter is about 3 mm greater than the camshaft diameter.
4 Base circle diameter = 36 + 3 = 39 say 40 mm
The width of cam is taken equal to the width of roller, *i.e.* 14 mm.
The width of cam (*w*2) is also taken empirically as
w 2 = 0.09 × Cylinder bore + 6 mm = 0.09 × 140 +6 = 18.6 mm
Let us take the width of cam as 18 mm.
Now the *cam is drawn according to the procedure given below :
First of all, the displacement diagram, as shown in Fig. 32.32, is drawn as discussed in the following steps :

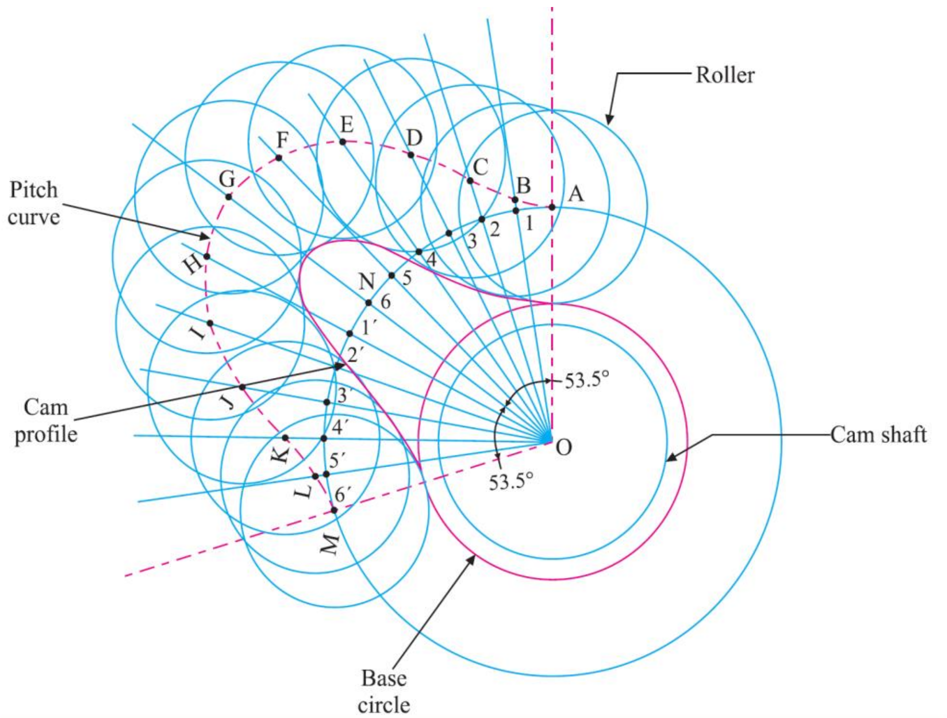
1. Draw a horizontal line *ANM* such that *AN* represents the angular displacement when valve opens (*i.e.* 53.5º) to some suitable scale. The line *NM* represents the angular displacement of the cam when valve closes (*i.e.* 53.5º).
2. Divide *AN* and *NM* into any number of equal even parts (say six).
3. Draw vertical lines through points 0, 1, 2, 3 etc. equal to the lift of valve *i.e.* 16 mm.
4. Divide the vertical lines 3 – *f* and 32 – *f* 2 into six equal parts as shown by points *a*, *b*, *c* ... and *a*2, *b*2, *c*2 .... in Fig. 32.32.
5. Since the valve moves with equal uniform acceleration and decleration for each half of the lift, therefore, valve displacement diagram for opening and closing of valve consists of double parabola.



6. Join *Aa*, *Ab* , *Ac* intersecting the vertical lines through 1, 2, 3 at *B*, *C*, *D* respectively.

7. Join the points *B*, *C*, *D* with a smooth curve. This is the required parabola for the half of valve opening. Similarly other curves may be drawn as shown in Fig. 32.32.
8. The curve *A*, *B*, *C*, ..., *G*, *K*, *L*, *M* is the required displacement diagram. Now the profile of the cam, as shown in Fig. 32.32, is drawn as discussed in the following steps:
1. Draw a base circle with centre *O* and diameter equal 40 mm (radius = 40/2 = 20 mm)
2. Draw a prime circle with centre *O* and radius, *OA*= Min. radius of cam +1 /2

Diameter of roller = 20+1/2× 28 = 20 + 14 = 34 mm
3. Draw angle *AOG* = 53.5º to represent opening of valve and angle *GOM* = 53.5º to represent closing of valve.
4. Divide the angular displacement of the cam during opening and closing of the valve (*i.e* angle *AOG* and *OM*) into same number of equal even parts as in displacement diagram.
5. Join the points 1, 2, 3, etc. with the centre *O* and produce the lines beyond prime circle as shown in Fig. 2.33.
6. Set off points 1*B*, 2*C*, 3*D*, etc. equal to the displacements from displacement diagram.
7. Join the points *A*, *B*, *C*, ...*L*, *M*, *A*. The curve drawn through these points is known as ***pitch curve.***
8. From the points *A*, *B*, *C*, ...*K*, *L*, draw circles of radius equal to the radius of the roller.



9. Join the bottoms of the circle with a smooth curve as shown in Fig. 32.33. The is the required profile of cam.

Ex 1:

a cam is to give the following motion to a knife-edged follower :

1.out stroke during 60^{o} of cam rotation

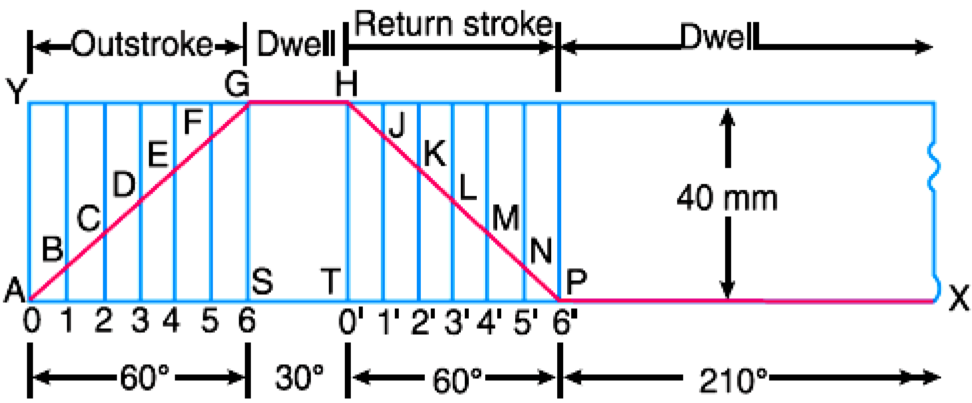
2.dwell for the next 30^{o} of cam rotation

3.return stroke during next 60^{o} of cam rotation, and

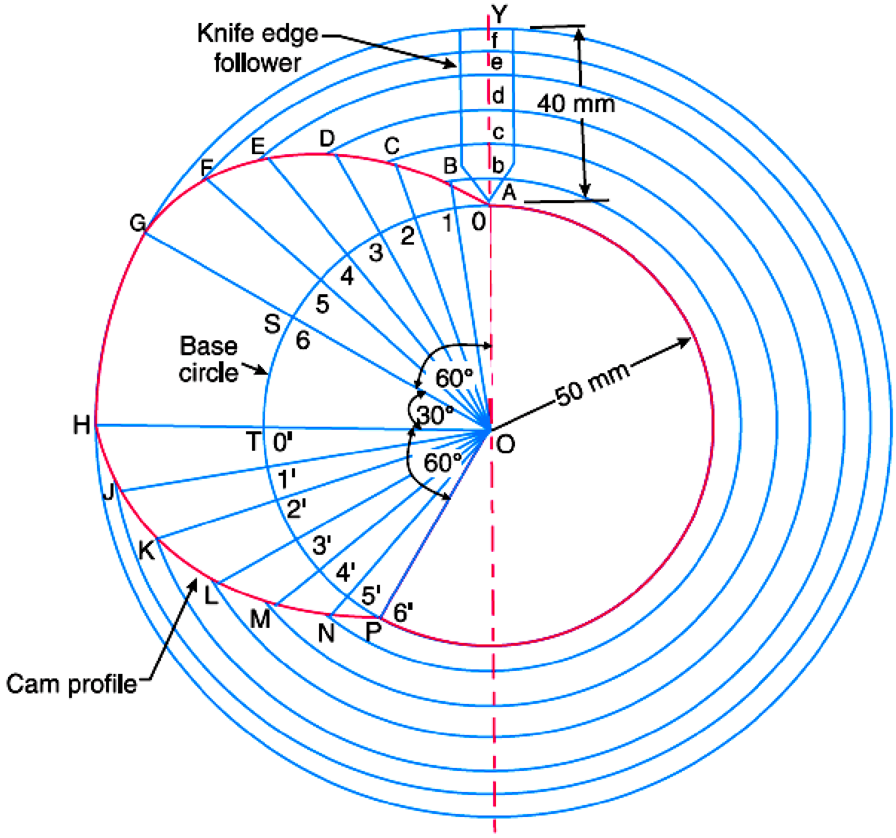
4.dwell for the remaining 210^{o} of cam rotation

The stroke of the follower is 40 mm and the minimum radius of the cam is 50 mm. the follower moves with uniform velocity during both the outstroke and return strokes. Draw the profile of the cam when(a) the axis of the follower passes through the axis of the cam shaft. (b) the axes of the follower is offset by 20 mm from the axis of cam shaft.

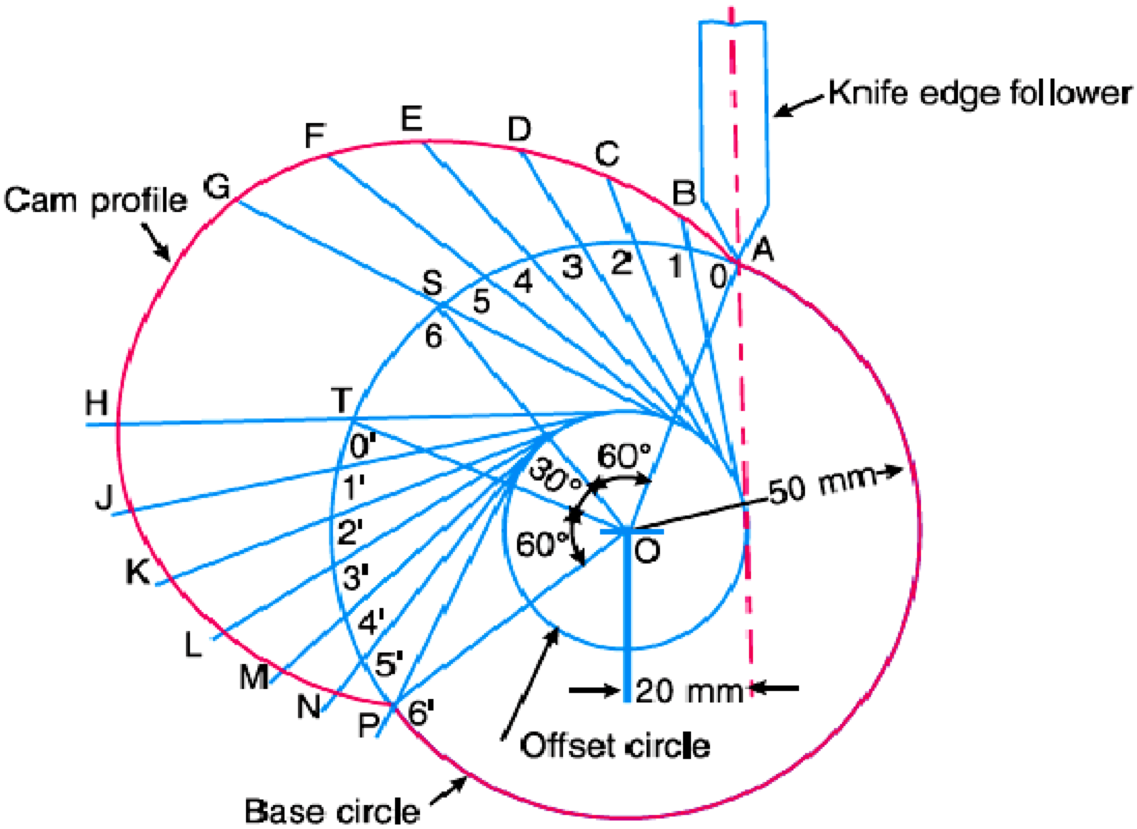
Sol:



(a)profile of the cam when the axis of follower passes through the axis of cam shaft



(b)profile of the cam when the axis of follower is offset by 20 mm from the axis of the cam shaft



Ex2:

A cam is to be designed for a knife edge follower with following data:

1.cam lift = 40 mm during 90° of cam rotation with simple harmonic motion.

2.dwell for the next 30°.

3.during the next 60° of cam rotation, the follower return to its original position with simple harmonic motion.

4.dwell during the remaining 180° .

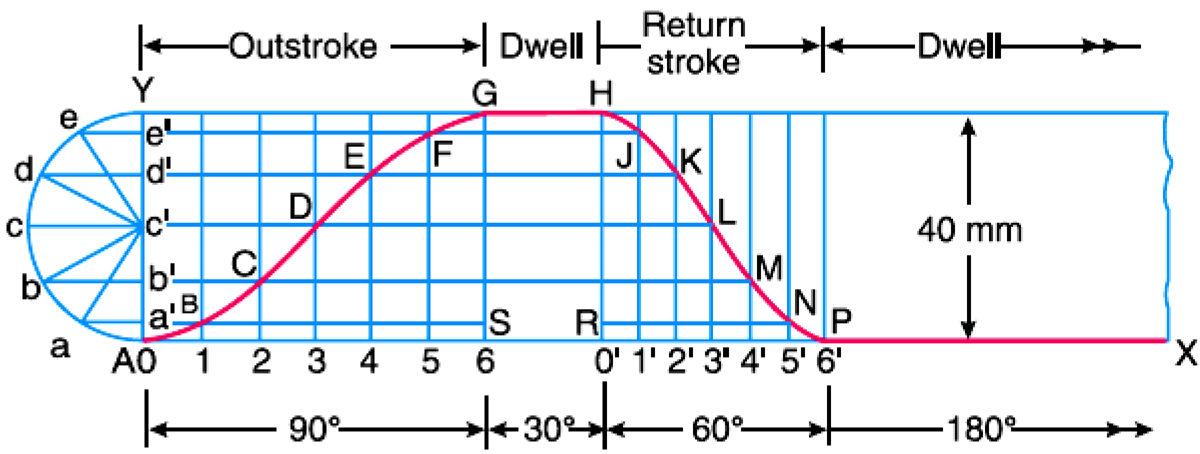
Draw the profile of the cam when:

(a)the line of stroke of the follower passes through the axis of cam shaft, and

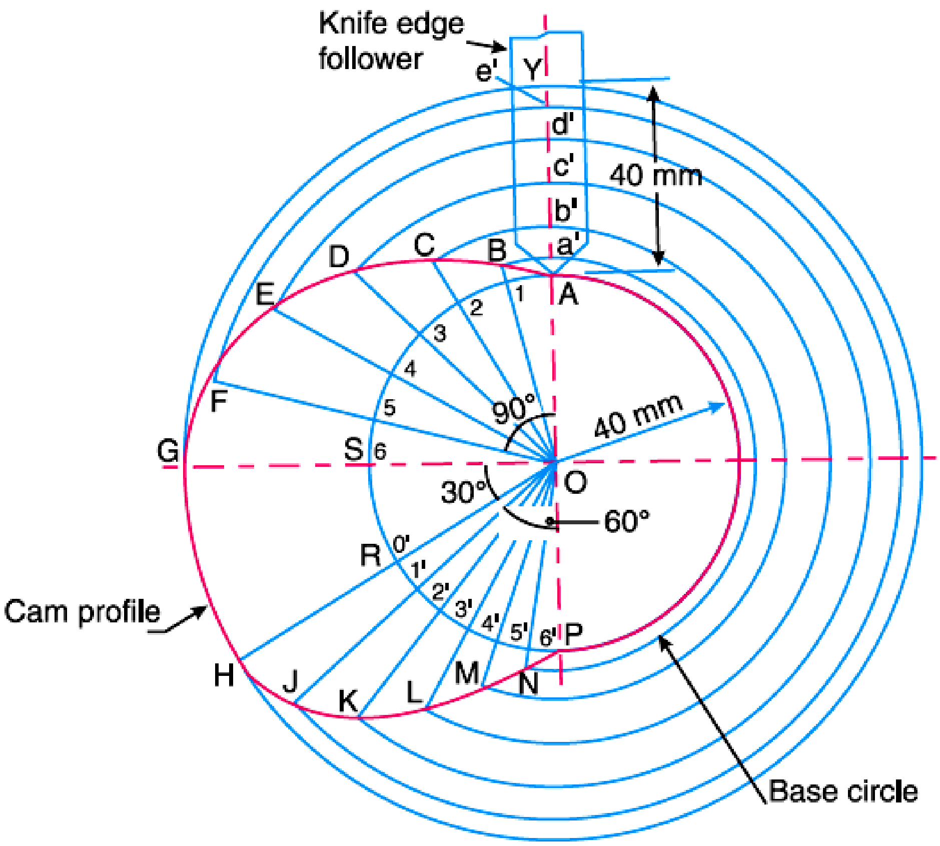
(b)the line of stroke is offset 20 mm from the axis of the cam shaft.

The radius of the base circle of the cam is 40 mm. determine the maximum velocity and acceleration of the follower during its ascent and descent, if the cam rotates at 240 R.P.M.

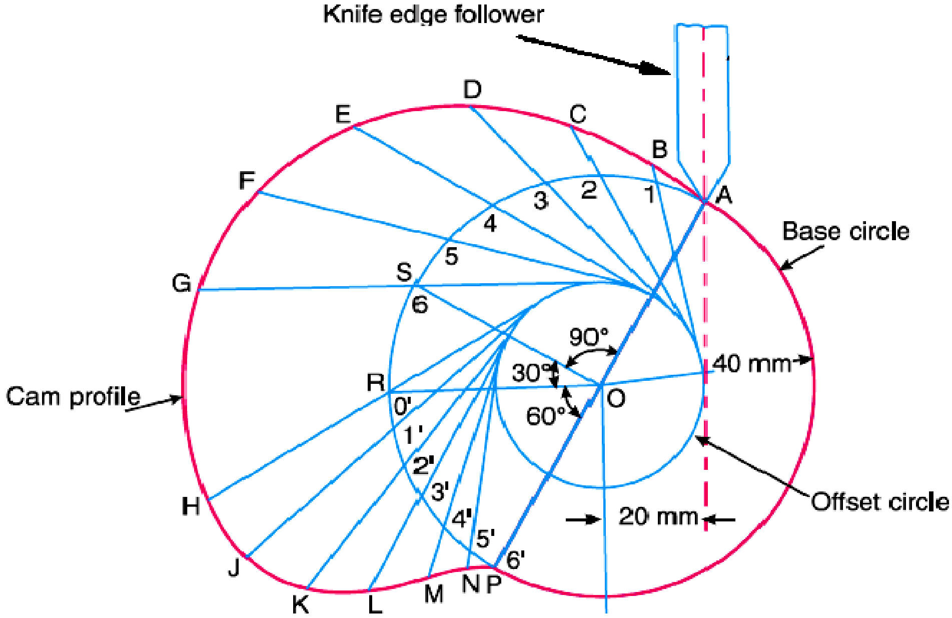
Sol:



(a) profile of cam:



(b) profile of the cam with offset:



MAXIMUM VELOCITY OF THE FOLLOWER(during ascent and descent)

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 240}{60} = 25.14 rad/s$$

$$\theta_o = 90^o = \frac{\pi}{2} rad = 1.571 rad$$

$$v_o = \frac{\pi\omega S}{2\theta_o} = \frac{\pi \times 25.14 \times 0.04}{2 \times 1.571} = 1 m/s$$

$$\theta_R = 60^o = \frac{\pi}{3} rad = 1.047 rad$$

$$v_R = \frac{\pi\omega S}{2\theta_R} = \frac{\pi \times 25.14 \times 0.04}{2 \times 1.047} = 1.51 m/s$$

MAXIMUM ACCELERATION OF THE FOLLOWER(during ascent and descent)

$$a_o = \frac{\pi^2\omega^2 S}{2(\theta_O)^2} = \frac{\pi^2 \times (25.14)^2 \times 0.04}{2 \times (1.571)^2} = 50.6 m/s^2$$

$$a_R = \frac{\pi^2\omega^2 S}{2(\theta_R)^2} = \frac{\pi^2 \times (25.14)^2 \times 0.04}{2 \times (1.047)^2} = 113.8 m/s^2$$

- المصادر الاساسية :
- Strength of Material by Ferdinal L .Singer
- Strength of Materials by R.S.Khurmi
- Machine Design by R.S. Khurmi, J.K. Gupta
- Machine Design by Paul H.Black
- المصادر المقترحة:
- **Schaums Outline Series of Machine Design by Hall , Holowenko , Laughin**