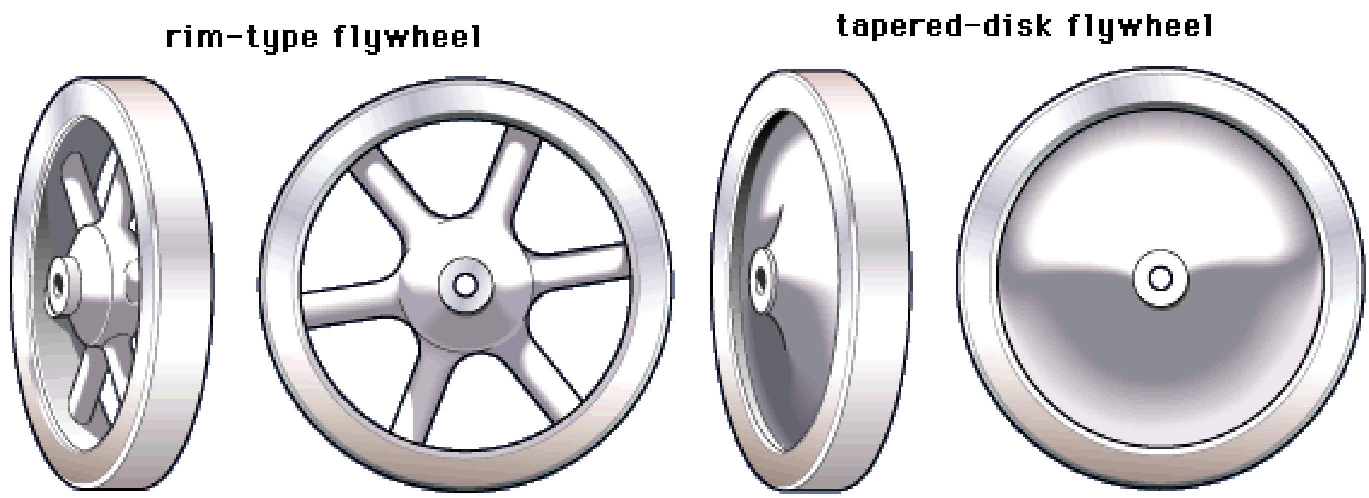
CHAPTER-7

Flywheel & Camshaft

Flywheel



Flywheel

In a combustion engine, & especially in one with one or two cylinders, energy is imparted to the crankshaft intermittently, & in order to keep it rotating at a fairly uniform speed under a substantially constant load, it is necessary to provide it with a flywheel.

In a **single cylinder engine(4 Stroke)**, in which there is only one power stroke in two revolutions of the crankshaft, a considerable fraction of energy generated per cycle is stored in the flywheel, & the proportion thus stored decreases with an increase in the No. of cylinders

In a **4 cylinder engine** about 40% of the energy of the cycle is temporarily stored.
However, not all of this energy goes into flywheel

During the 1st half of the power stroke, when energy is being supplied in excess by the burning gases, all of the reciprocating parts of the engine are being accelerated & absorb energy; besides, the rotating parts other than the flywheel also have some flywheel capacity, & this reduces the proportion of the energy of the cycle which must be stored in the flywheel.

In a 6 cylinder engine the proportion of the energy which must be absorbed & returned by the moving parts amounts to about 20%.

The greater the No. of cylinders the smaller the flywheel capacity required per unit of piston displacement, because the overlap of power strokes is greater & besides other rotating parts of the engine have greater inertia.

However, the flywheel has by far the greatest inertia even in a multi cylinder engine.

Aside from its principle function, the fly wheel serves as a member of the friction clutch, & it usually carries also the ring gear of the electric starter.

A flywheel used in machines serves as a reservoir which stores energy during the period when the supply of energy is more than the requirement and releases, it during the period when the requirement of energy is more than supply.

In case of steam engines, internal combustion engines, reciprocating compressors and pumps, the energy is developed during one stroke and the engine is to run for the whole cycle on the energy produced during this one ,stroke. For example; in I.C. engines, the energy is developed, only during power stroke which is much more than the engine load; and no energy is being developed during suction, compression and exhaust strokes in case of four, stroke engines and during compression in case of two stroke engines. The excess energy developed during power stroke is absorbed by the flywheel and releases it to the crankshaft during other strokes in which no energy is developed, thus rotating the crankshaft at a uniform speed. A little, consideration will show that when the flywheel absorbs energy, its speed increases and when it releases, the speed decreases. Hence a flywheel does not maintain a constant speed, it simply reduces the fluctuation of speed.

In machines where the operation is intermittent like punching machines, shearing machines, riveting machines, crushers etc., the flywheel stores energy from the ,power source during the greater portion, of the operating cycle and gives it up during a small period of the cycle. Thus the energy from the power source to machines is supplied practically at a constant rate throughout the operation.

Note: The function of a governor in engine is entirely different from that of a flywheel .It regulates the mean speed of an engine when there are variations in the load, e.g., when the load on the engine increases it becomes necessary to increase the supply of Working fluid. On the other hand, when the load decreases, less working fluid is required. The governor automatically; controls the supply, of working fluid to the engine with the varying load condition and keeps the mean speed within certain limits.

As discussed above, the flywheel does not maintain constant speed. It simply reduces the fluctuation of speed. In other words *a flywheel controls the speed variations caused by the fluctuation of the engine turning moment during each cycle of operation. It does not control the speed variations caused by the varying load*.

Coefficient of Fluctuation of Speed

The difference between max. & min. speeds during a cycle is called ***maximum fluctuation of speed.***

The ratio of the max. fluctuation of speed to mean speed is called ***coefficient of fluctuation of speed.***

Let N_1 =Maximum speed in r.p.m. during the cycle,

N_2 = Minimum speed in r.p.m. during the cycle, and

$$N = \text{Mean speed in r.p.m.} = \frac{N_1 + N_2}{2}$$

∴ Coefficient of fluctuation of speed,

$$C_S = \frac{N_1 - N_2}{N} = \frac{2(N_1 - N_2)}{N_1 + N_2}$$

$$= \frac{\omega_1 - \omega_2}{\omega} = \frac{2(\omega_1 - \omega_2)}{\omega_1 + \omega_2} \quad \text{...(In terms of angular speeds)}$$

$$= \frac{v_1 - v_2}{v} = \frac{2(v_1 - v_2)}{v_1 + v_2} \quad \text{...(In terms of linear speeds)}$$

The coefficient of fluctuation of speed is a limiting factor in the design of flywheel. It varies depending upon the nature of service to which the flywheel is employed. The following table shows the permissible values for coefficient of fluctuation of speed for some machines.

Note : The reciprocal of coefficient of fluctuation of speed is known as *coefficient of steadiness* and it is denoted by m.

$$\therefore \quad m = \frac{1}{C_S} = \frac{N}{N_1 - N_2} = \frac{\omega}{\omega_1 - \omega_2} = \frac{v}{v_1 - v_2}$$

Table Permissible values for coefficient of fluctuation of speed (C_S).

S.No.	*Type of machine or class of service*	*Coefficient of fluctuation of speed* (C_S)
1.	Crushing machines	0.200
2.	Electrical machines	0.003
3.	Electrical machines (direct drive)	0.002
4.	Engines with belt transmission	0.030
5.	Gear wheel transmission	0.020
6.	Hammering machines	0.200
7.	Pumping machines	0.03 to 0.05
8.	Machine tools	0.030
9.	Paper making, textile and weaving machines	0.025
10.	Punching, shearing and power presses	0.10 to 0.15
11.	Spinning machinery	0.10 to 0.020
12.	Rolling mills and mining machines	0.025

Fluctuation of Energy

The fluctuation of energy may be determined by the turning moment diagram for one complete cycle of operation. Consider a turning moment diagram for a single cylinder double acting steam engine as shown in Figure The vertical ordinate represents the turning moment and the horizontal ordinate (abscissa) represents the crank angle.

A little consideration will show that the turning moment is zero when the crank angle is zero. It rises to a maximum value when crank angle reaches 90° and it is again zero when crank angle is 180°. This is shown by the curve *abc* in Figure and it represents the turning moment diagram for outstroke. The curve *cde* is the turning moment diagram for instroke and is somewhat similar to the curve *abc*.

Since the work done is the product of the turning moment and the angle turned, therefore the area of the turning moment diagram represents the work done per revolution. In actual practice, the engine is assumed to work against the mean resisting torque, as shown by a horizontal line *AF*. The height of the ordinate *a A* represents the mean height of the turning moment diagram. Since it is assumed that the work done by the turning moment per revolution is equal to the work done against the mean resisting torque, therefore the area of the rectangle *aA Fe* is proportional to the work done against the mean resisting torque.

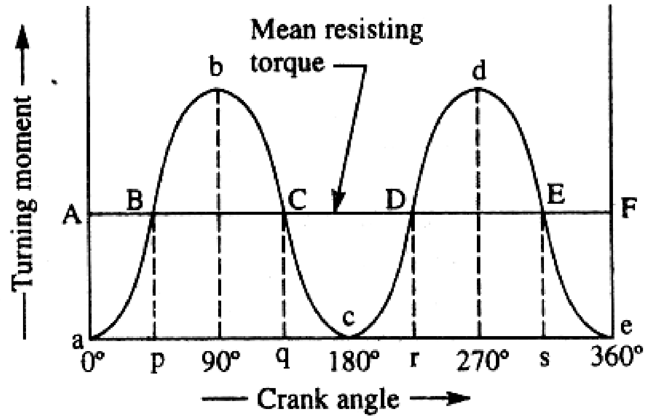


Figure Turning moment diagram for a single cylinder double acting steam engine.

We see in Figure , that the mean resisting torque line *AF* cuts the turning moment diagram at points *B, C, D* and *E*. When the crank moves from '*a*' to '*p*' the work done by the engine is equal to the area *aBp*, whereas the energy required is represented by the area *aABp*. In other words, the engine has done less work (equal to the area *aAB*) than the requirement. This amount of energy is taken from the flywheel and hence the speed of the flywheel decreases. Now the crank moves from *p* to *q*, the work done by the engine is equal to the area *pBbCq*, whereas the requirement of energy is represented by the area *pBCq*. Therefore the engine has done more work than the requirement. This excess work (equal to the area *BbC*) is stored in the flywheel and hence the speed of the flywheel increases while the crank moves from *p* to *q*.

Similarly when the crank moves from *q* to *r*, more work is taken from the engine than is developed. This loss of work is represented by the area *CcD*. To supply this loss, the flywheel gives up some of its energy and thus the speed decreases while the crank moves from *q* to *r*. As the crank moves from *r* to *s*, excess energy is again developed given by the area *DdE* and the speed again increases. As the piston moves from *s* to *e*, again there is a loss of work and the speed decreases. The variations of energy above and below the mean resisting torque line are called *fluctuation of energy*. The areas *BbC, CcD, DdE* etc. represent fluctuations of energy.

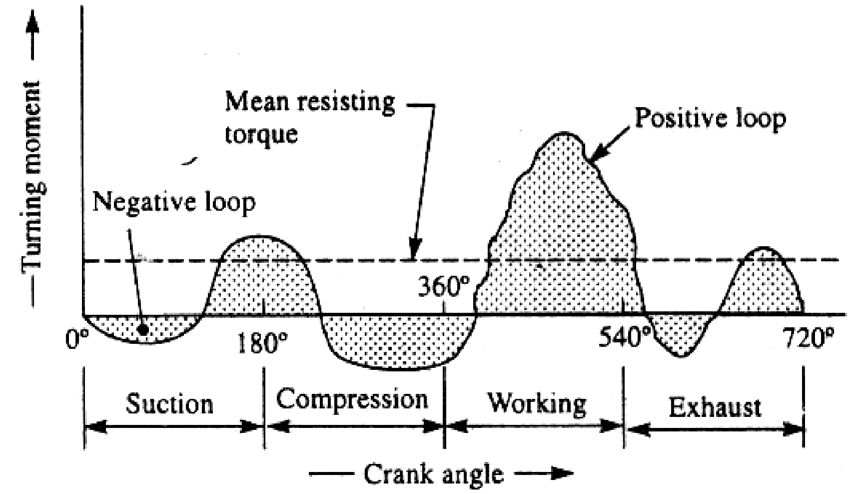


Figure Turning moment diagram for a four stroke internal combustion engine.

A little consideration will show that the engine has a maximum speed either at q or at s. This is due to the fact that the flywheel absorbs energy while the crank moves from p to q and from r to s. On the other hand, the engine has a minimum speed either at p or at r. The reason is that the flywheel gives out some of its energy when the crank moves from a to p and q to r. The difference between the maximum and the minimum energies is known as *maximum fluctuation of energy*.

A turning moment diagram for a four stroke internal combustion engine is shown in Fig.

We know that in a four stroke internal combustion engine, there is one working stroke after the crank has turned through 720° (or 4π radians). Since the pressure inside the engine cylinder is less than the atmospheric pressure during suction stroke, therefore a negative loop is formed as shown in Figure. During the compression stroke, the work is done on the gases, therefore a higher negative loop is obtained. In the working stroke, the fuel burns and the gases expand, therefore a large positive loop is formed. During exhaust stroke, the work is done on the gases, therefore a negative loop is obtained.

A turning moment diagram for a compound steam engine having three cylinders and the resultant turning moment diagram is shown in Figure. The resultant turning moment diagram

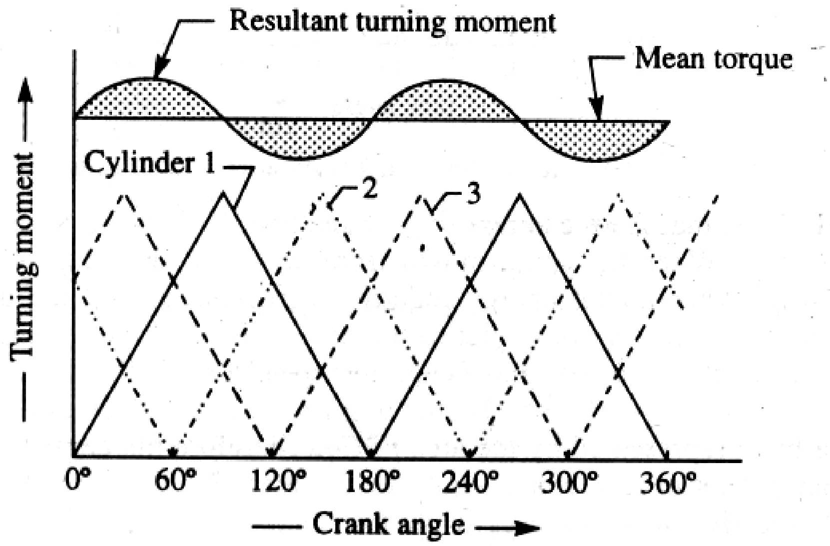


Figure Turning moment diagram for a compound steam engine.

is the sum of the turning moment diagrams for the three cylinders. It may be noted that the first cylinder is the high pressure cylinder, second cylinder is the intermediate cylinder and the third cylinder is the low pressure cylinder. The cranks, in case of three cylinders are usually placed at 120° to each other.

Maximum Fluctuation of Energy

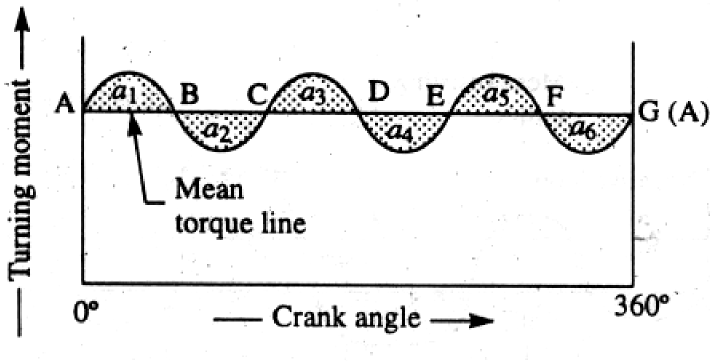


Figure Turning moment diagram for a multi-cylinder engine.

A turning moment diagram for a multi-cylinder engine is shown by a wavy curve in Fig. The horizontal line AG represents the mean torque line. Let a_1, a_3, a_5 be the areas above the mean torque line and a_2, a_4 and a_6 be the areas below the mean torque line. These areas represent some quantity of energy which is either added or subtracted from the energy of the moving parts of the engine.

Let the energy in the flywheel at $A = E$, then from Figure , we have

Energy at $B = E + a_1$

Energy at $C = E + a_1 - a_2$

Energy at $D = E + a_1 - a_2 + a_3$

Energy at $E = E + a_1 - a_2 + a_3 - a_4$

Energy at $F = E + a_1 - a_2 + a_3 - a_4 + a_5$

Energy at $G = E + a_1 - a_2 + a_3 - a_4 + a_5 - a_6$ = Energy at A

Let us now suppose that the maximum of these energies is at B and minimum at E.

$\therefore$ Maximum energy in the flywheel

$$= E + a_1$$

and minimum energy in the flywheel

$$= E + a_1 - a_2 + a_3 - a_4$$

$\therefore$ Maximum fluctuation of energy,

$$\Delta E = \text{Maximum energy} - \text{Minimum energy}$$
$$= (E + a_1) - (E + a_1 - a_2 + a_3 - a_4) = a_2 - a_3 + a_4$$

Coefficient of Fluctuation of Energy

It is defined as the ratio of the maximum fluctuation of energy to the work done per cycle. It is usually denoted by C_E. Mathematically, coefficient of fluctuation of energy,

$$C_E = \frac{\text{Maximum fluctuation of energy}}{\text{Work done per cycle}}$$

The workdone per cycle may be obtained by using the following relations :

1. Workdone/cycle $= T_{mean} \times \theta$

where T_{mean} = Mean torque, and

θ = Angle turned in radians per revolution

= 2π, in case of steam engines and two stroke internal combustion engines.

= 4π, in case of four stroke internal combustion engines.

The mean torque (T_{mean}) in N-m may be obtained by using the following relation *i.e.*,

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{P}{\omega}$$

where P = Power transmitted in watts,

N = Speed in r.p.m., and

ω = Angular speed in rad/s $= 2\pi N / 60$

2. The workdone per cycle may also be obtained by using the following relation :

$$\text{Workdone/cycle} = \frac{P \times 60}{n}$$

where n = Number of working strokes per minute.

= N, in case of steam engines and two stroke internal combustion engines.

= $N/2$, in case of four stroke internal combustion engines.

The following table shows the values of coefficient of fluctuation of energy for steam engines and internal combustion engines.

Table **Coefficient of fluctuation of energy (C_E) for steam and internal combustion engines.**

S.No.	*Type of engine*	*Coefficient of fluctuation of energy (C_E)*
1.	Single cylinder, double acting steam engine	0.21
2.	Cross-compound steam engine	0.096
3.	Single cylinder, single acting, four stroke gas engine	1.93
4.	Four cylinder, single acting, four stroke gas engine	0.066
5.	Six cylinder, single acting, four stroke gas engine	0.031

Energy Stored in a Flywheel

A flywheel is shown in Figure, We have already discussed that when a flywheel absorbs energy its speed increases and when it gives up energy its speed decreases.

Let m = Mass of the flywheel in kg,

k = Radius of gyration of the flywheel in metres,

I = Mass moment of inertia of the flywheel about the axis of rotation in kg-m^2 $= m.k^2$,

N_1 and N_2 = Maximum and minimum speeds during the cycle in r.p.m.,

ω_1 and ω_2 = Maximum and minimum angular speeds during the cycle in rad/s,

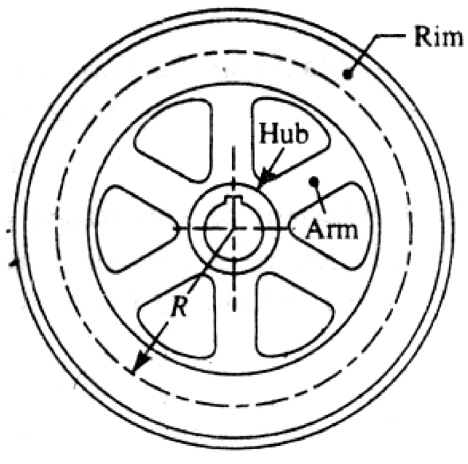


Figure Flywheel.

N = Mean speed during the cycle in r.p.m. $= \dfrac{N_1 + N_2}{2}$,

ω = Mean angular speed during the cycle in rad/s $= \dfrac{\omega_1 + \omega_2}{2}$,

C_S = Coefficient of fluctuation of speed $= \dfrac{N_1 - N_2}{N}$ or $\dfrac{\omega_1 - \omega_2}{\omega}$

We know that mean kinetic energy of the flywheel,

$$E = \frac{1}{2} \times I.\omega^2 = \frac{1}{2} \times m.k^2.\omega^2 \qquad \text{(in N-m or joules)}$$

As the speed of the flywheel changes from ω_1 to ω_2, the maximum fluctuation of energy,

$$\Delta E = \text{Maximum K.E.} - \text{Minimum K.E.} = \frac{1}{2} \times I\,(\omega_1)^2 - \frac{1}{2} \times I\,(\omega_2)^2$$

$$= \frac{1}{2} \times I \left[(\omega_1)^2 - (\omega_2)^2\right] = \frac{1}{2} \times I\,(\omega_1 + \omega_2)\,(\omega_1 - \omega_2)$$

$$= I.\omega\,(\omega_1 - \omega_2) \qquad \ldots\left(\because\ \omega = \frac{\omega_1 + \omega_2}{2}\right) \ldots(i)$$

$$= I.\omega^2 \left(\frac{\omega_1 - \omega_2}{\omega}\right) \qquad \text{...[Multiplying and dividing by } \omega]$$

$$= I.\omega^2.C_S = m.k^2.\omega^2.C_S \qquad ...(\because I = m.k^2) \qquad ...(ii)$$

$$= 2\ E.C_S \qquad ...\left(\because\ E = \frac{1}{2} \times I\,.\,\omega^2\right) \qquad ...(iii)$$

The radius of gyration (k) may be taken equal to the mean radius of the rim (R), because the thickness of rim is very small as compared to the diameter of rim. Therefore substituting $k = R$ in equation (ii), we have

$$\Delta\ E = m.R^2.\omega^2.C_S = m.v^2.C_S \qquad ...(\because v = \omega.R)$$

From this expression, the mass of the flywheel rim may be determined.

Notes : 1. In the above expression, only the mass moment of inertia of the rim is considered and the mass moment of inertia of the hub and arms is neglected. This is due to the fact that the major portion of weight of the flywheel is in the rim and a small portion is in the hub and arms. Also the hub and arms are nearer to the axis of rotation, therefore the moment of inertia of the hub and arms is very small.

2. The density of cast iron may be taken as 7260 kg/m^3 and for cast steel, it may taken as 7800 kg/m^3.

3. The mass of the flywheel rim is given by

$$m = \text{Volume} \times \text{Density} = 2\,\pi R \times A \times \rho$$

From this expression, we may find the value of the cross-sectional area of the rim. Assuming the cross-section of the rim to be rectangular, then

$$A = b \times t$$

where b = Width of the rim, and

t = Thickness of the rim.

Knowing the ratio of $b\,/\,t$ which is usually taken as 2, we may find the width and thickness of rim.

4. When the flywheel is to be used as a pulley, then the width of rim should be taken 20 to 40 mm greater than the width of belt.

Example 1. *The turning moment diagram for a petrol engine is drawn to the following scales : Turning moment, 1 mm = 5 N-m ; Crank angle, 1 mm = 1°.*

The turning moment diagram repeats itself at every half revolution of the engine and the areas above and below the mean turning moment line, taken in order are 295, 685, 40, 340, 960, 270 mm².

Determine the mass of 300 mm diameter flywheel rim when the coefficient of fluctuation of speed is 0.3% and the engine runs at 1800 r.p.m. Also determine the cross-section of the rim when the width of the rim is twice of thickness. Assume density of rim material as 7250 kg/m³.

Solution. Given : D = 300 mm or R = 150 mm = 0.15 m ; C_S = 0.3% = 0.003 ; N = 1800 r.p.m. or $\omega = 2\pi \times 1800/60 = 188.5$ rad/s ; $\rho = 7250$ kg/m³

Mass of the flywheel

Let m = Mass of the flywheel in kg.

First of all, let us find the maximum fluctuation of energy. The turning moment diagram is shown in Figure

Since the scale of turning moment is 1 mm = 5 N-m, and scale of the crank angle is 1 mm = 1° = π / 180 rad, therefore 1 mm² on the turning moment diagram.

$$= 5 \times \pi / 180 = 0.087 \text{ N-m}$$

Let the total energy at A = E. Therefore from Figure, we find that

Energy at $B = E + 295$

Energy at $C = E + 295 - 685 = E - 390$

Energy at $D = E - 390 + 40 = E - 350$

Energy at $E = E - 350 - 340 = E - 690$

Energy at $F = E - 690 + 960 = E + 270$

Energy at $G = E + 270 - 270 = E$ = Energy at A

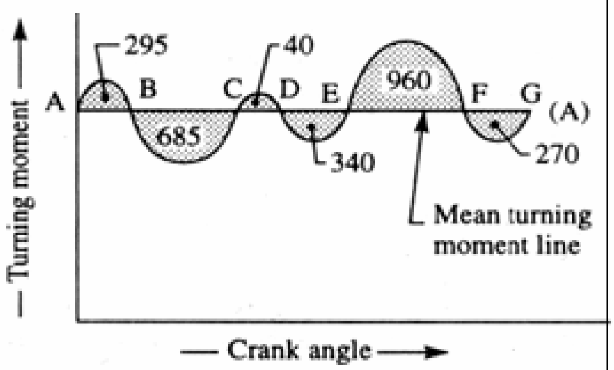


Fig.

From above we see that the energy is maximum at B and minimum at E.

∴ Maximum energy $= E + 295$

and minimum energy $= E - 690$

We know that maximum fluctuation of energy,

$$\Delta E = \text{Maximum energy} - \text{Minimum energy}$$
$$= (E + 295) - (E - 690) = 985 \text{ mm}^2$$
$$= 985 \times 0.087 = 86 \text{ N-m}$$

We also know that maximum fluctuation of energy (ΔE),

$$86 = m.R^2.\omega^2.C_S = m\,(0.15)^2\,(188.5)^2\,(0.003) = 2.4\,m$$

∴ $m = 86 / 2.4 = 35.8$ kg **Ans.**

Cross-section of the flywheel rim

Let t = Thickness of rim in metres, and

b = Width of rim in metres = $2\,t$ ...(Given)

∴ Cross-sectional area of rim,

$$A = b \times t = 2\,t \times t = 2\,t^2$$

We know that mass of the flywheel rim (m),

$$35.8 = A \times 2\pi R \times \rho = 2\,t^2 \times 2\pi \times 0.15 \times 7250 = 13\,668\,t^2$$

∴ $t^2 = 35.8 / 13\,668 = 0.0026$ or $t = 0.051$ m = 51 mm **Ans.**

and $b = 2\,t = 2 \times 51 = 102$ mm **Ans.**

Example 2. *The intercepted areas between the output torque curve and the mean resistance line of a turning moment diagram for a multicylinder engine, taken in order from one end are as follows :* $-35, +410, -285, +325, -335, +260, -365, +285, -260\ mm^2$.

The diagram has been drawn to a scale of 1 mm = 70 N-m and 1 mm = 4.5°. The engine speed is 900 r.p.m. and the fluctuation in speed is not to exceed 2% of the mean speed.

Find the mass and cross-section of the flywheel rim having 650 mm mean diameter. The density of the material of the flywheel may be taken as 7200 kg / m³. The rim is rectangular with the width 2 times the thickness. Neglect effect of arms, etc.

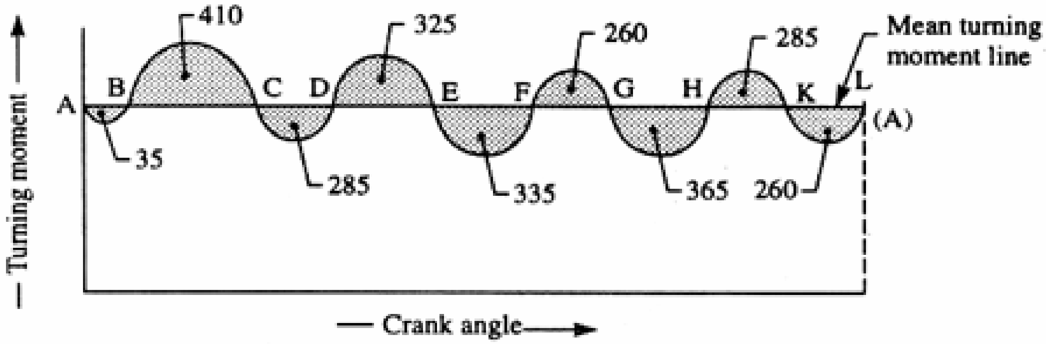
Solution. Given : N = 900 r.p.m. or $\omega = 2\pi \times 900/60 = 94.26$ rad/s ; $\omega_1 - \omega_2 = 2\%\ \omega$ or $\frac{\omega_1 - \omega_2}{\omega} = C_S = 2\% = 0.02$; D = 650 mm or R = 325 mm = 0.325 m ; $\rho = 7200\ kg / m^3$

Mass of the flywheel rim

Let m = Mass of the flywheel rim in kg.

First of all, let us find the maximum fluctuation of energy. The turning moment diagram for a multi-cylinder engine is shown in Figure

Since the scale of turning moment is 1 mm = 70 N-m and scale of the crank angle is 1 mm = 4.5° = $\pi / 40$ rad, therefore 1 mm² on the turning moment diagram = $70 \times \pi / 40 = 5.5$ N-m



Let the total energy at $A = E$. Therefore from Figure , we find that

Energy at $B = E - 35$

Energy at $C = E - 35 + 410 = E + 375$

Energy at $D = E + 375 - 285 = E + 90$

Energy at $E = E + 90 + 325 = E + 415$

Energy at $F = E + 415 - 335 = E + 80$

Energy at $G = E + 80 + 260 = E + 340$

Energy at $H = E + 340 - 365 = E - 25$

Energy at $K = E - 25 + 285 = E + 260$

Energy at $L = E + 260 - 260 = E$ = Energy at A

From above, we see that the energy is maximum at E and minimum at B.

∴ Maximum energy = $E + 415$ and minimum energy = $E - 35$

We know that maximum fluctuation of energy,

$$= (E + 415) - (E - 35) = 450\ mm^2 = 450 \times 5.5 = 2475 \text{ N-m}$$

We also know that maximum fluctuation of energy (ΔE),

$$2475 = m.R^2.\omega^2.C_S = m(0.325)^2\ (94.26)^2\ 0.02 = 18.77\ m$$

$$\therefore \quad m = 2475/18.77 = 132 \text{ kg} \quad \textbf{Ans.}$$

Cross-section of the flywheel rim

Let t = Thickness of the rim in metres, and b = Width of the rim in metres = $2\ t$ ...(Given)

∴ Area of cross-section of the rim, $A = b \times t = 2\ t \times t = 2\ t^2$

We know that mass of the flywheel rim (m),

$$132 = A \times 2\ \pi\ R \times \rho = 2\ t^2 \times 2\ \pi \times 0.325 \times 7200 = 29\ 409\ t^2$$

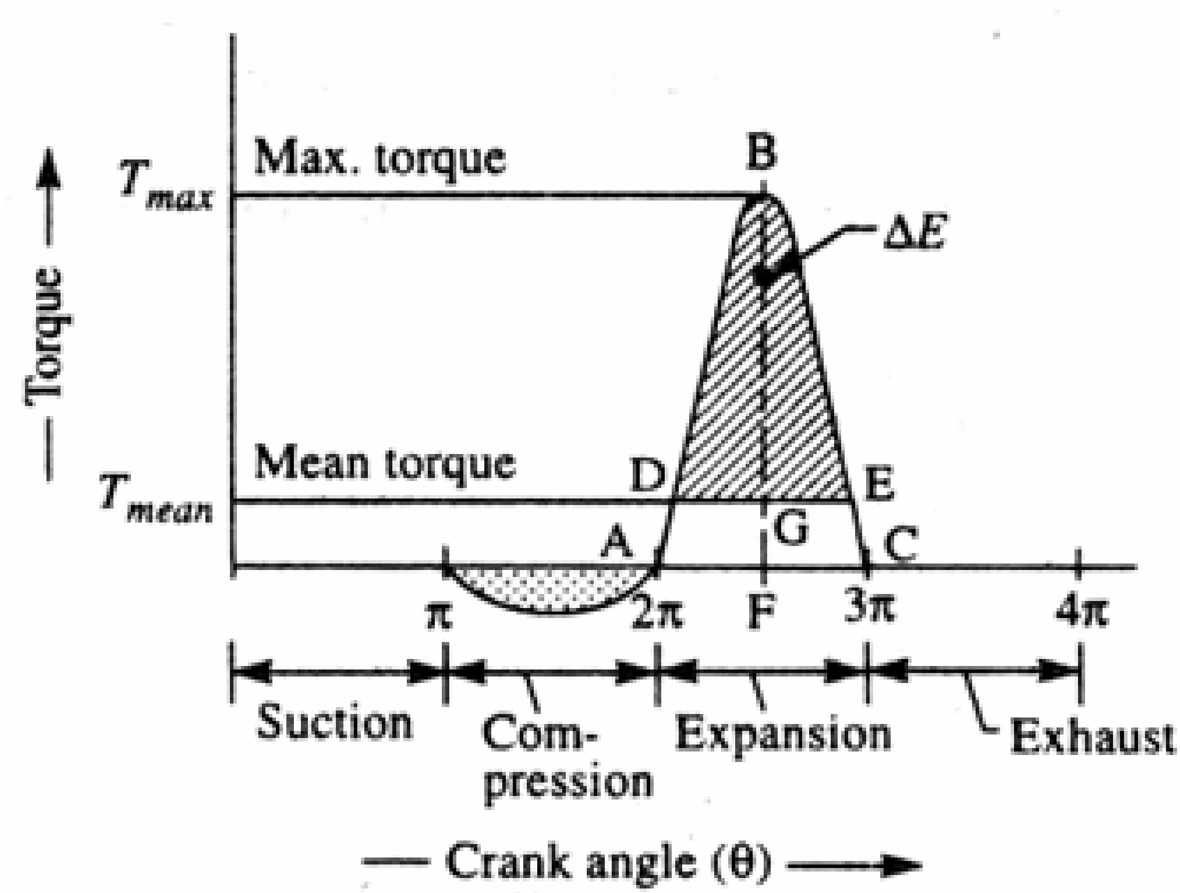
$$\therefore \quad t^2 = 132 / 29\ 409 = 0.0044 \quad \text{or} \quad t = 0.067 \text{ m} = 67 \text{ mm} \quad \textbf{Ans.}$$

and $$b = 2\ t = 2 \times 67 = 134 \text{ mm} \quad \textbf{Ans.}$$

Example 3 *A single cylinder, single acting, four stroke oil engine develops 20 kW at 300 r.pm. The workdone by the gases during the expansion stroke is 2.3 times the workdone on the gases during the compression and the workdone during the suction and exhaust strokes is negligible. The speed is to be maintained within ± 1%. Determine the mass moment of inertia of the flywheel.*

Solution. Given : P = 20 kW = 20×10^3 W ; N = 300 r.p.m. or $\omega = 2\pi \times 300/60 = 31.42$ rad/s ; $\omega_1 - \omega_2 = \pm 1\%\ \omega$

First of all, let us find the maximum fluctuation of energy (ΔE). The turning moment diagram for a four stroke engine is shown in Figure. It is assumed to be triangular during compression and expansion strokes, neglecting the suction and exhaust strokes.



We know that mean torque transmitted by the engine,

$$T_{mean} = \frac{P \times 60}{2\,\pi N} = \frac{20 \times 10^3 \times 60}{2\,\pi \times 300} = 636.5 \text{ N-m}$$

and *workdone per cycle = $T_{mean} \times \theta = 636.5 \times 4\,\pi = 8000$ N-m ...(i)

* The workdone per cycle may also be calculated as follows :

We know that for a four stroke engine, number of working strokes per cycle

$n = N/2 = 300/2 = 150$

$\therefore$ Workdone per cycle = $P \times 60/n = 20 \times 10^3 \times 60/150 = 8000$ N-m

Let W_C = Workdone during compression stroke, and

W_E = Workdone during expansion stroke.

Since the workdone during suction and exhaust strokes is negligible, therefore net work done per cycle

$$= W_E - W_C = W_E - W_E / 2.3 = 0.565\ W_E \qquad ...(ii)$$

From equations (*i*) and (*ii*), we have

$$W_E = 8000 / 0.565 = 14\ 160 \text{ N-m}$$

The workdone during the expansion stroke is shown by triangle *ABC* in Figure, , in which base $AC = \pi$ radians and height $BF = T_{max}$.

∴ Workdone during expansion stroke (W_E),

$$14\ 160 = \frac{1}{2} \times \pi \times T_{max} = 1.571\ T_{max}$$

or $$T_{max} = 14\ 160 / 1.571 = 9013 \text{ N-m}$$

We know that height above the mean torque line,

$$BG = BF - FG = T_{max} - T_{mean}$$

$$= 9013 - 636.5 = 8376.5 \text{ N-m}$$

Since the area *BDE* shown shaded in Figure I above the mean torque line represents the maximum fluctuation of energy (ΔE), therefore from geometrical relation,

$$\frac{\text{Area of } \Delta\, BDE}{\text{Area of } \Delta\, ABC} = \frac{(BG)^2}{(BF)^2}, \text{ we have}$$

Maximum fluctuation of energy (*i.e.* area of Δ *BDE*),

$$^*\Delta E = \text{Area of } \Delta\, ABC \left(\frac{BG}{BF}\right)^2 = W_E \left(\frac{BG}{BF}\right)^2$$

$$= 14\ 160 \left(\frac{8376.5}{9013}\right)^2 = 12\ 230 \text{ N-m}$$

*The maximum fluctuation of energy (ΔE) may also be obtained as discussed below :

From similar triangles *BDE* and *BAC*,

$$\frac{DE}{AC} = \frac{BG}{BF} \quad \text{or} \quad DE = \frac{BG}{BF} \times AC = \frac{8376.5}{9013} \times \pi = 2.92 \text{ rad}$$

∴ Maximum fluctuation of energy (*i.e.*, area of Δ *BDE*),

$$\Delta E = \frac{1}{2} \times DE \times BG = \frac{1}{2} \times 2.92 \times 8376.5 = 12\ 230 \text{ N-m}$$

Since the speed is to be maintained within ± 1% of the mean speed, therefore total fluctuation of speed

$$\omega_1 - \omega_2 = 2\ \%\ \omega = 0.02\ \omega$$

and coefficient of fluctuation of speed,

$$C_S = \frac{\omega_1 - \omega_2}{\omega} = 0.02$$

Let I = Mass moment of inertia of the flywheel in kg-m^2.

We know that maximum fluctuation of energy (ΔE),

$$12\ 230 = I.\omega^2.C_S = I\ (31.42)^2\ 0.02 = 19.74\ I$$

∴ $$I = 12\ 230 / 19.74 = 619.5 \text{ kg-m}^2 \quad \textbf{Ans.}$$

Stresses in a Flywheel Rim

A flywheel, as shown in Figure , consists of a rim at which the major portion of the mass or weight of flywheel is concentrated, a boss or hub for fixing the flywheel on to the shaft and a number of arms for supporting the rim on the hub.

The following types of stresses are induced in the rim of a flywheel:

1. Tensile stress due to centrifugal force,
2. Tensile bending stress caused by the restraint of the arms, and
3. The shrinkage stresses due to unequal rate of cooling of casting. These stresses may be very high but there is no easy method of determining. This stress is taken care of by a factor of safety.

We shall now discuss the first two types of stresses as follows :

1. *Tensile stress due to the centrifugal force*

The tensile stress in the rim due to the centrifugal force, assuming that the rim is unstrained by the arms, is determined in a similar way as a thin cylinder subjected to internal pressure.

Let b = Width of rim,

t = Thickness of rim,

A = Cross-sectional area of rim = $b \times t$,

D = Mean diameter of flywheel

R = Mean radius of flywheel,

ρ = Density of flywheel material,

ω = Angular speed of flywheel,

v = Linear velocity of flywheel, and

σ_t = Tensile or hoop stress.

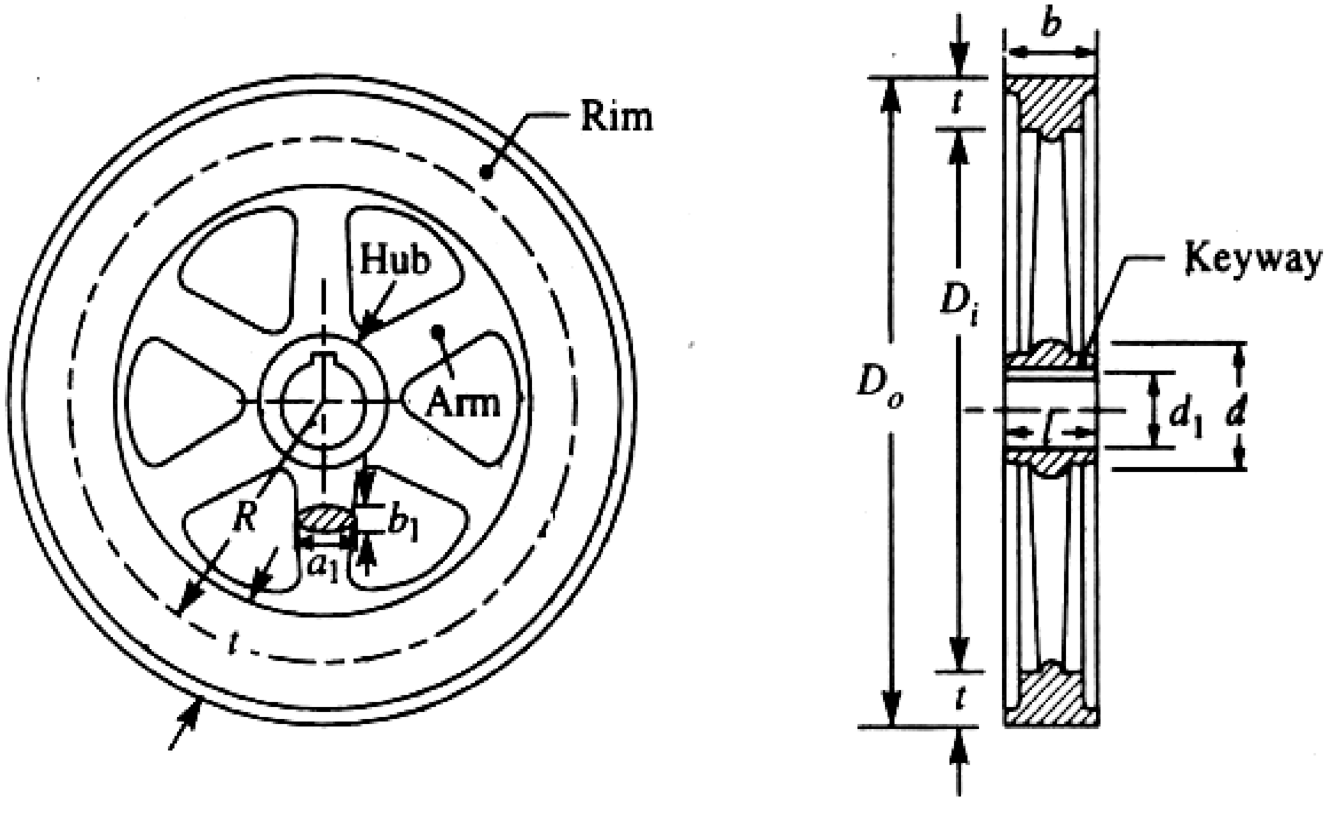
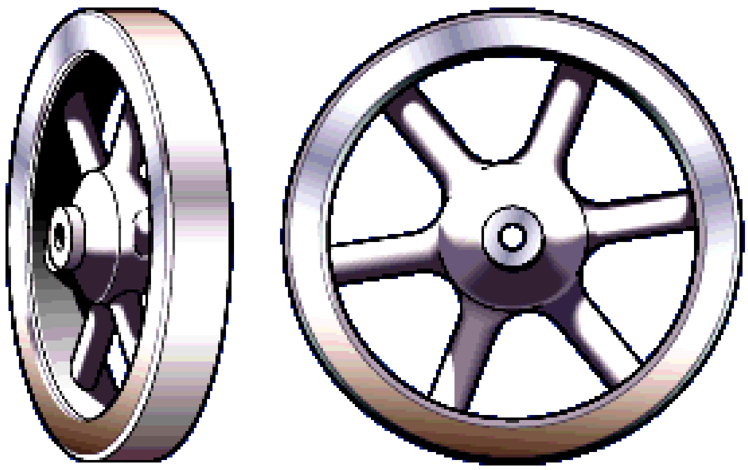


Fig. Flywheel.



Consider a small element of the rim as shown shaded in Figure. Let it subtends an angle $\delta\theta$ at the centre of the flywheel.

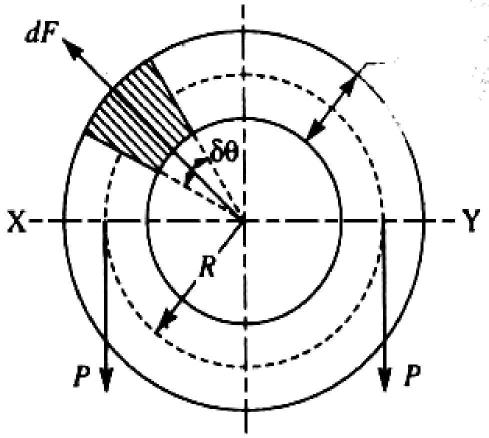


Figure Cross-section of a flywheel rim.

Volume of the small element

$$= A.R.\delta\theta$$

∴ Mass of the small element,

$$dm = \text{Volume} \times \text{Density}$$
$$= A.R.\delta\theta.\rho = \rho.A.R.\delta\theta$$

and centrifugal force on the element,

$$dF = dm.\omega^2.R = \rho.A.R.\delta\theta.\omega^2.R$$
$$= \rho.A.R^2.\omega^2.\delta\theta$$

Vertical component of dF

$$= dF.\sin\theta$$
$$= \rho.A.R^2.\omega^2.\delta\theta \sin\theta$$

∴ Total vertical bursting force across the rim diameter X-Y,

$$= \rho.A\ R^2.\omega^2 \int_0^{\pi} \sin\theta\, d\theta$$

$$= \rho.A.R^2.\omega^2 \left[-\cos\theta\right]_0^{\pi} = 2\ \rho.A.R^2.\omega^2 \qquad ...(i)$$

This vertical force is resisted by a force of $2P$, such that

$$2P = 2\sigma_t \times A \qquad ...(ii)$$

From equations (i) and (ii), we have

$$2\ \rho\ A.R^2.\omega^2 = 2\ \sigma_t \times A$$

$$\therefore \qquad \sigma_t = \rho.R^2.\omega^2 = \rho.v^2 \qquad ...(\because v = \omega.R) \qquad ...(iii)$$

when ρ is in kg/m^3 and v is in m/s, then σ_t will be in N/m^2 or Pa.

Note : From the above expression, the mean diameter (D) of the flywheel may be obtained by using the relation,

$$v = \pi D.N/60$$

2. *Tensile bending stress caused by restraint of the arms*

The tensile bending stress in the rim due to the restraint of the arms is based on the assumption that each portion of the rim between a pair of arms behaves like a beam fixed at both ends and uniformly loaded, as shown in Figure , such that length between fixed ends,

$$l = \frac{\pi D}{n} = \frac{2\pi R}{n}, \text{ where } n = \text{Number of arms.}$$

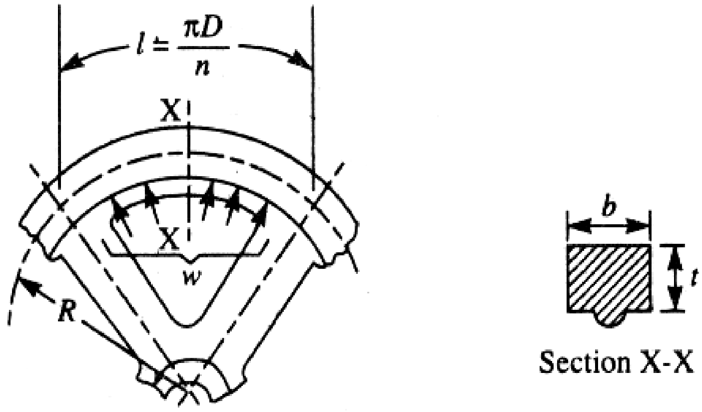
The uniformly distributed load (w) per metre length will be equal to the centrifugal force between a pair of arms.

$$\therefore \qquad w = b.t.\rho.\omega^2.R \text{ N/m}$$

We know that maximum bending moment,

$$M = \frac{w.l^2}{12} = \frac{b.t.\rho.\omega^2.R}{12}\left(\frac{2\pi R}{n}\right)^2$$

and section modulus, $$Z = \frac{1}{6} b \times t^2$$



Figure

$\therefore$ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{b.t.\rho.\omega^2.R}{12}\left(\frac{2\pi R}{n}\right)^2 \times \frac{6}{b \times t^2}$$

$$= \frac{19.74\,\rho.\omega^2.R^3}{n^2.t} = \frac{19.74\,\rho.v^2.R}{n^2.t} \qquad \text{...(iv)}$$

...(Substituting $\omega = v/R$)

Now total stress in the rim,

$$\sigma = \sigma_t + \sigma_b$$

If the arms of a flywheel do not stretch at all and are placed very close together, then centrifugal force will not set up stress in the rim. In other words, σ_t will be zero. On the other hand, if the arms are stretched enough to allow free expansion of the rim due to centrifugal action, there will be no restraint due to the arms, *i.e.*, σ_b will be zero.

It has been shown by G. Lanza that the arms of a flywheel stretch about $\frac{3}{4}$ th of the amount necessary for free expansion. Therefore the total stress in the rim,

$$= \frac{3}{4}\sigma_t + \frac{1}{4}\sigma_b = \frac{3}{4}\rho.v^2 + \frac{1}{4} \times \frac{19.74\,\rho.v^2.R}{n^2.t} \qquad \text{...(v)}$$

$$= \rho.v^2\left(0.75 + \frac{4.935\,R}{n^2.t}\right)$$

Example 4 *A multi-cylinder engine is to run at a constant load at a speed of 600 r.p.m. On drawing the crank effort diagram to scale of 1 m = 250 N-m and 1 mm = 3°, the areas in sq mm above and below the mean torque line are as follows :*

+ 160, – 172, + 168, – 191, + 197, – 162 sq mm

The speed is to be kept within ± 1% of the mean speed of the engine. Calculate the necessary moment of inertia of the flywheel.

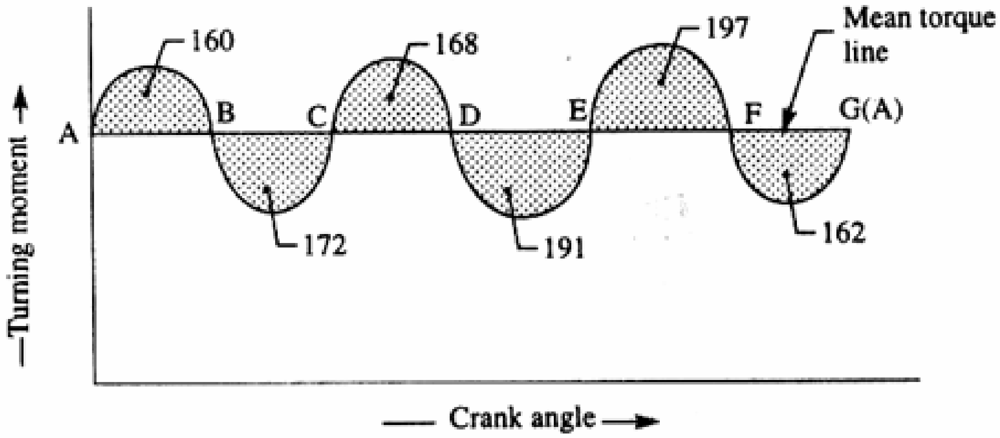
Determine suitable dimensions for cast iron flywheel with a rim whose breadth is twice its radial thickness. The density of cast iron is 7250 kg/m³, and its working stress in tension is 6 MPa. Assume that the rim contributes 92% of the flywheel effect.

Solution. Given : = N = 600 r.p.m. or $\omega = 2\pi \times 600/60 = 62.84$ rad/s ; $\rho = 7250$ kg/m³ ; $\sigma_t = 6$ MPa $= 6 \times 10^6$ N/m²

Moment of inertia of the flywheel

Let I = Moment of inertia of the flywheel.

First of all, let us find the maximum fluctuation of energy. The turning moment diagram is shown in Figure.



Since the scale for the turning moment is 1 mm = 250 N-m and the scale for the crank angle is 1 mm = 3° = $\frac{\pi}{60}$ rad, therefore

1 mm² on the turning moment diagram $= 250 \times \frac{\pi}{60} = 13.1$ N-m

Let the total energy at $A = E$. Therefore from Fig. we find that

Energy at $B = E + 160$

Energy at $C = E + 160 - 172 = E - 12$

Energy at $D = E - 12 + 168 = E + 156$

Energy at $E = E + 156 - 191 = E - 35$

Energy at $F = E - 35 + 197 = E + 162$

Energy at $G = E + 162 - 162 = E$ = Energy at A

From above, we find that the energy is maximum at F and minimum at E.

∴ Maximum energy $= E + 162$ and minimum energy $= E - 35$

We know that the maximum fluctuation of energy,

$$\Delta E = \text{Maximum energy} - \text{Minimum energy}$$
$$= (E + 162) - (E - 35) = 197 \text{ mm}^2 = 197 \times 13.1 = 2581 \text{ N-m}$$

Since the fluctuation of speed is ± 1% of the mean speed (ω), therefore total fluctuation of speed, $\omega_1 - \omega_2 = 2\%\ \omega = 0.02\ \omega$

and coefficient of fluctuation of speed,

$$C_S = \frac{\omega_1 - \omega_2}{\omega} = 0.02$$

We know that the maximum fluctuation of energy (ΔE),

$$2581 = I.\omega^2.C_S = I\,(62.84)^2\,0.02 = 79\,I$$

$$\therefore \quad I = 2581 / 79 = 32.7 \text{ kg-m}^2 \quad \textbf{Ans.}$$

Dimensions of a flywheel rim

Let t = Thickness of the flywheel rim in metres, and

b = Breadth of the flywheel rim in metres = $2\,t$ ...(Given)

First of all let us find the peripheral velocity (v) and mean diameter (D) of the flywheel.

We know that tensile stress (σ_t),

$$6 \times 10^6 = \rho.v^2 = 7250 \times v^2$$

$$\therefore \quad v^2 = 6 \times 10^6 / 7250 = 827.6 \quad \text{or} \quad v = 28.76 \text{ m/s}$$

We also know that peripheral velocity (v),

$$28.76 = \frac{\pi\,D.N}{60} = \frac{\pi\,D \times 600}{60} = 31.42\,D$$

$$\therefore \quad D = 28.76 / 31.42 = 0.915 \text{ m} = 915 \text{ mm} \quad \textbf{Ans.}$$

Now let us find the mass of the flywheel rim. Since the rim contributes 92% of the flywheel effect, therefore the energy of the flywheel rim (E_{rim}) will be 0.92 times the total energy of the flywheel (E). We know that maximum fluctuation of energy (ΔE),

$$2581 = E \times 2\,C_S = E \times 2 \times 0.02 = 0.04\,E$$

$$\therefore \quad E = 2581 / 0.04 = 64\,525 \text{ N-m}$$

and energy of the flywheel rim,

$$E_{rim} = 0.92\,E = 0.92 \times 64\,525 = 59\,363 \text{ N-m}$$

Let m = Mass of the flywheel rim.

We know that energy fo the flywheel rim (E_{rim}),

$$59\,363 = \frac{1}{2} \times m \times v^2 = \frac{1}{2} \times m\,(28.76)^2 = 413.6\,m$$

$$\therefore \quad m = 59\,363 / 413.6 = 143.5 \text{ kg}$$

We also know that mass of the flywheel rim (m),

$$143.5 = b \times t \times \pi D \times \rho = 2\,t \times t \times \pi \times 0.915 \times 7250 = 41\,686\,t^2$$

$$\therefore \quad t^2 = 143.5 / 41\,686 = 0.003\,44$$

or $t = 0.0587$ say $0.06 \text{ m} = 60 \text{ mm}$ **Ans.**

and $b = 2\,t = 2 \times 60 = 120 \text{ mm}$ **Ans.**

Notes : The mass of the flywheel rim may also be obtained by using the following relations. Since the rim contributes 92% of the flywheel effect, therefore using

1. $I_{rim} = 0.92\,I_{flywheel}$ or $m.k^2 = 0.92 \times 32.7 = 30 \text{ kg–m}^2$

Since radius of gyration, $k = R = D/2 = 0.915/2 = 0.4575$ m, therefore

$$m = \frac{30}{k^2} = \frac{30}{(0.4575)^2} = \frac{30}{0.209} = 143.5 \text{ kg}$$

2.
$$(\Delta\,E)_{rim} = 0.92\,(\Delta\,E)_{flywheel}$$
$$m.v^2.C_S = 0.92\,(\Delta\,E)_{flywheel}$$
$$m\,(28.76)^2\,0.02 = 0.92 \times 2581$$
$$16.55\,m = 2374.5 \quad \text{or} \quad m = 2374.5 / 16.55 = 143.5 \text{ kg}$$

Example 5 *The areas of the turning moment diagram for one revolution of a multi-cylinder engine with reference to the mean turning moment, below and above the line, are*

– 32, + 408, – 267, + 333, – 310, + 226, – 374, + 260 and – 244 mm².

The scale for abscissa and ordinate are : 1 mm = 2.4° and 1 mm = 650 N-m respectively. The mean speed is 300 r.p.m. with a percentage speed fluctuation of ± 1.5%. If the hoop stress in the material of the rim is not to exceed 5.6 MPa determine the suitable diameter and cross-section for the flywheel, assuming that the width is equal to 4 times the thickness. The density of the material may be taken to be 7200 kg/m³. Neglect the effect of the boss and arms.

Solution. Given : $N = 300$ r.p.m. or $\omega = 2\pi \times 300/60 = 31.42$ rad / s ; $\sigma_t = 5.6$ MPa $= 5.6 \times 10^6$ N/m² ; $\rho = 7200$ kg/m³

Diameter of the flywheel

Let D = Diameter of the flywheel in metres.

We know that peripheral velocity of the flywheel,

$$v = \frac{\pi D.N}{60} = \frac{\pi D \times 300}{60} = 15.71\ D \text{ m/s}$$

We also know that hoop stress (σ_t),

$$5.6 \times 10^6 = \rho \times v^2 = 7200\ (15.71\ D)^2 = 1.8 \times 10^6\ D^2$$

$$\therefore \quad D^2 = 5.6 \times 10^6 / 1.8 \times 10^6 = 3.11 \quad \text{or} \quad D = 1.764 \text{ m} \quad \textbf{Ans.}$$

Cross-section of the flywheel

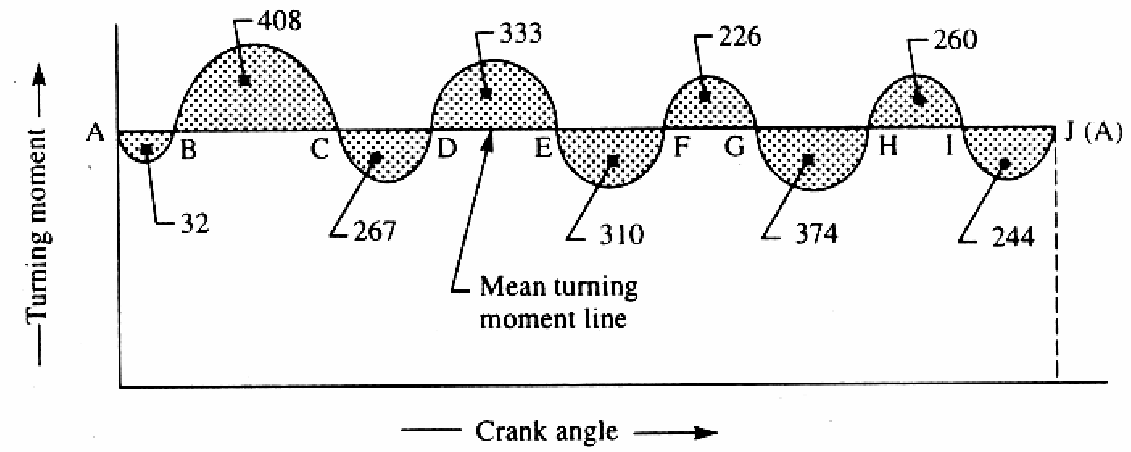
Let t = Thickness of the flywheel rim in metres, and

b = Width of the flywheel rim in metres = $4\,t$ ...(Given)

∴ Cross-sectional area of the rim,

$$A = b \times t = 4\,t \times t = 4\,t^2 \text{ m}^2$$

Now let us find the maximum fluctuation of energy. The turning moment diagram for one revolution of a multi-cylinder engine is shown in Figure.



Figure

Since the scale of crank angle is 1 mm $= 2.4° = 2.4 \times \frac{\pi}{180} = 0.042$ rad, and the scale of the turning moment is 1 mm = 650 N-m, therefore

1 mm² on the turning moment diagram

$$= 650 \times 0.042 = 27.3 \text{ N-m}$$

Let the total energy at $A = E$. Therefore from Fig., we find that

Energy at $B = E - 32$

Energy at $C = E - 32 + 408 = E + 376$

$$\text{Energy at } D = E + 376 - 267 = E + 109$$
$$\text{Energy at } E = E + 109 + 333 = E + 442$$
$$\text{Energy at } F = E + 442 - 310 = E + 132$$
$$\text{Energy at } G = E + 132 + 226 = E + 358$$
$$\text{Energy at } H = E + 358 - 374 = E - 16$$
$$\text{Energy at } I = E - 16 + 260 = E + 244$$
$$\text{Energy at } J = E + 244 - 244 = E = \text{Energy at } A$$

From above, we see that the energy is maximum at E and minimum at B.

$\therefore$ Maximum energy $= E + 442$

and minimum energy $= E - 32$

We know that maximum fluctuation of energy,

$$\Delta E = \text{Maximum energy} - \text{Minimum energy}$$
$$= (E + 442) - (E - 32) = 474 \text{ mm}^2$$
$$= 474 \times 27.3 = 12\ 940 \text{ N-m}$$

Since the fluctuation of speed is $\pm$ 1.5% of the mean speed, therefore total fluctuation of speed,

$$\omega_1 - \omega_2 = 3\% \text{ of mean speed} = 0.03\ \omega$$

and coefficient of fluctuation of speed,

$$C_S = \frac{\omega_1 - \omega_2}{\omega} = 0.03$$

Let m = Mass of the flywheel rim.

We know that maximum fluctuation of energy (ΔE),

$$12\ 940 = m.R^2.\omega^2.C_S = m\left(\frac{1.764}{2}\right)^2 (31.42)^2\ 0.03 = 23\ m$$

$\therefore$ $m = 12\ 940 / 23 = 563$ kg **Ans.**

We also know that mass of the flywheel rim (m),

$$563 = A \times \pi\ D \times \rho = 4\,t^2 \times \pi \times 1.764 \times 7200 = 159\ 624\ t^2$$

$\therefore$ $t^2 = 563 / 159\ 624 = 0.003\ 53$

or $t = 0.0594$ m $= 59.4$ say 60 mm **Ans.**

and $b = 4\,t = 4 \times 60 = 240$ mm **Ans.**

Example 6 . *An otto cycle engine develops 50 kW at 150 r.p.m. with 75 explosions per minute. The change of speed from the commencement to the end of power stroke must not exceed 0.5% of mean on either side. Design a suitable rim section having width four times the depth so that the hoop stress does not exceed 4 MPa. Assume that the flywheel stores 16 / 15 times the energy stored by the rim and that the workdone during power stroke is 1.40 times the workdone during the cycle. Density of rim material is 7200 kg/m³.*

Solution. Given : $P = 50$ kW $= 50 \times 10^3$ W ; $N = 150$ r.p.m. ; $n = 75$; $\sigma_t = 4$ MPa $= 4 \times 10^6$ N/m² ; $\rho = 7200$ kg/m³

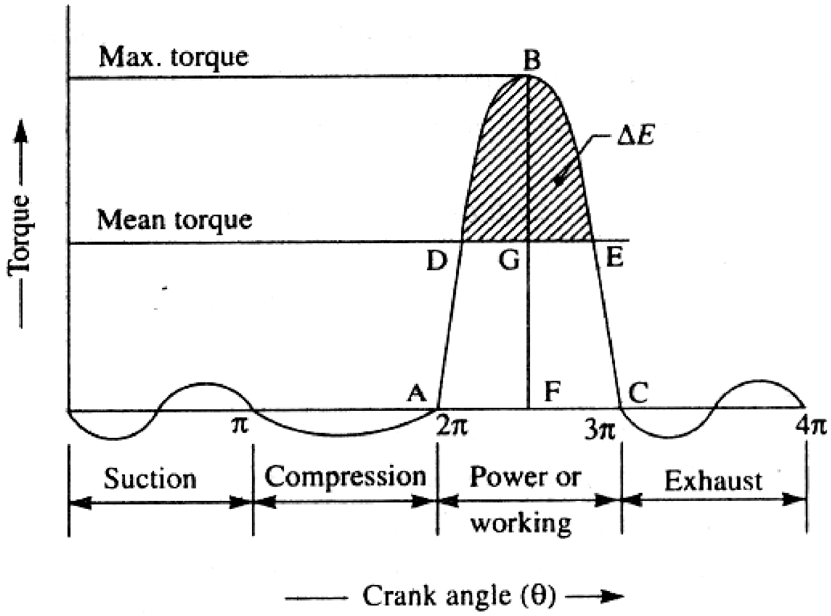
First of all, let us find the mean torque (T_{mean}) transmitted by the engine or flywheel. We know that the power transmitted (P),

$$50 \times 10^3 = \frac{2\pi N \times T_{mean}}{60}$$

$$= \frac{2\pi \times 150 \times T_{mean}}{60} = 15.71\ T_{mean}$$

$$\therefore \quad T_{mean} = 50 \times 10^3/15.71 = 3182.7 \text{ N-m}$$

Since the explosions per minute are equal to $N/2$, therefore the engine is a four stroke cycle engine. The turning moment diagram of a four stroke engine is shown in Figure.



Figure

We know that *workdone per cycle

$$= T_{mean} \times \theta = 3182.7 \times 4\pi = 40\ 000 \text{ N-m}$$

***The workdone per cycle for a four stroke engine is also given by**

$$\text{Workdone/cycle} = \frac{P \times 60}{\text{Number of explosion/min}} = \frac{P \times 60}{n} = \frac{50\,000 \times 60}{75} = 40\ 000 \text{ N-m}$$

$\therefore$ Workdone during power or working stroke

$$= 1.4 \times 40\ 000 = 56\ 000 \text{ N-m} \quad \ldots(i)$$

The workdone during power or working stroke is shown by a triangle ABC in Figure in which base $AC = \pi$ radians and height $BF = T_{max}$.

$\therefore$ Workdone during working stroke

$$= \frac{1}{2} \times \pi \times T_{max} = 1.571\ T_{max} \quad \ldots(ii)$$

From equations (i) and (ii), we have

$$T_{max} = 56\ 000 / 1.571 = 35\ 646 \text{ N-m}$$

Height above the mean torque line,

$$BG = BF - FG = T_{max} - T_{mean} = 35\ 646 - 3182.7 = 32\ 463.3 \text{ N-m}$$

Since the area BDE (shown shaded in Fig) above the mean torque line represents the maximum fluctuation of energy (ΔE), therefore from geometrical relation

$$\frac{\text{Area of } \Delta\, BDE}{\text{Area of } \Delta\, ABC} = \frac{(BG)^2}{(BF)^2}, \text{ we have}$$

Maximum fluctuation of energy (*i.e.*, area of triangle BDE),

$$\Delta E = \text{Area of triangle } ABC \times \left(\frac{BG}{BF}\right)^2 = 56\ 000 \times \left(\frac{32\ 463.3}{35\ 646}\right)^2$$

$$= 56\ 000 \times 0.83 = 46\ 480 \text{ N-m}$$

Mean diameter of the flywheel

Let D = Mean diameter of the flywheel in metres, and

v = Peripheral velocity of the flywheel in m/s.

We know that hoop stress (σ_t),

$$4 \times 10^6 = \rho . v^2 = 7200 \times v^2$$

$$\therefore \quad v^2 = 4 \times 10^6 / 7200 = 556 \quad \text{or} \quad v = 23.58 \text{ m/s}$$

We also know that peripheral velocity (v),

$$23.58 = \frac{\pi D . N}{60} = \frac{\pi D \times 150}{60} = 7.855 \text{ D}$$

$$\therefore \quad D = 23.58 / 7.855 = 3 \text{ m} \quad \textbf{Ans.}$$

Cross-sectional dimensions of the rim

Let t = Thickness of the rim in metres, and

b = Width of the rim in metres = $4\ t$ ...(Given)

$\therefore$ Cross-sectional area of the rim,

$$A = b \times t = 4\ t \times t = 4\ t^2$$

First of all, let us find the mass of the flywheel rim.

Let m = Mass of the flywheel rim, and

E = Total energy of the flywheel.

Since the fluctuation of speed is 0.5% of the mean speed on either side, therefore total fluctuation of speed,

$$N_1 - N_2 = 1\% \text{ of mean speed} = 0.01\ N$$

and coefficient of fluctuation of speed,

$$C_S = \frac{N_1 - N_2}{N} = 0.01$$

We know that the maximum fluctuation of energy (ΔE),

$$46\ 480 = E \times 2\ C_S = E \times 2 \times 0.01 = 0.02\ E$$

$$\therefore \quad E = 46\ 480 / 0.02 = 2324 \times 10^3 \text{ N-m}$$

Since the energy stored by the flywheel is $\frac{16}{15}$ times the energy stored by the rim, therefore the energy of the rim,

$$E_{rim} = \frac{15}{16} E = \frac{15}{16} \times 2324 \times 10^3 = 2178.8 \times 10^3 \text{ N-m}$$

We know that energy of the rim (E_{rim}),

$$2178.8 \times 10^3 = \frac{1}{2} \times m \times v^2 = \frac{1}{2} \times m\ (23.58)^2 = 278\ m$$

$$\therefore \quad m = 2178.8 \times 10^3 / 278 = 7837 \text{ kg}$$

We also know that mass of the flywheel rim (m),

$$7837 = A \times \pi D \times \rho = 4\ t^2 \times \pi \times 3 \times 7200 = 271\ 469\ t^2$$

$$\therefore \quad t^2 = 7837 / 271\ 469 = 0.0288 \text{ or } t = 0.17 \text{ m} = 170 \text{ mm} \quad \textbf{Ans.}$$

and $b = 4\ t = 4 \times 170 = 680$ mm **Ans.**

Stresses in Flywheel Arms

The following stresses are induced in the arms of a flywheel.

1. Tensile stress due to centrifugal force acting on the rim.

2. Bending stress due to the torque transmitted from the rim to the shaft or from the shaft to the rim.

3. Shrinkage stresses due to unequal rate of cooling of casting. These stresses are difficult to determine.

We shall now discuss the first two types of stresses as follows :

1. *Tensile stress due to the centrifugal force*

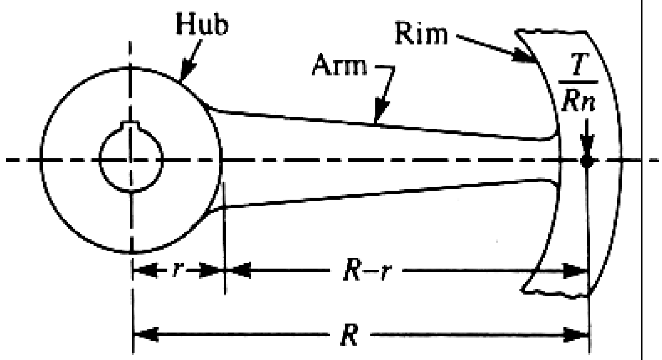
Due to the centrifugal force acting on the rim, the arms will be subjected to direct tensile stress whose magnitude is same as discussed in the previous article.

∴ Tensile stress in the arms,

$$\sigma_{t1} = \frac{3}{4}\sigma_t = \frac{3}{4}\rho \times v^2$$

2. *Bending stress due to the torque transmitted*

Due to the torque transmitted from the rim to the shaft or from the shaft to the rim, the arms will be subjected to bending, because they are required to carry the full torque load. In order to find out the maximum bending moment on the arms, it may be assumed as a centilever beam fixed at the hub and carrying a concentrated load at the free end of the rim as shown in Figure



Figure

Let T = Maximum torque transmitted by the shaft,

R = Mean radius of the rim,

r = Radius of the hub,

n = Number of arms, and

Z = Section modulus for the cross-section of arms.

We know that the load at the mean radius of the rim,

$$F = \frac{T}{R}$$

$$\therefore \text{ Load on each arm } = \frac{T}{R\,.\,n}$$

and maximum bending moment which lies on the arm at the hub,

$$M = \frac{T}{R\,.\,n}(R - r)$$

∴ Bending stress in arms,

$$\sigma_{b1} = \frac{M}{Z} = \frac{T}{R.n.Z}(R - r)$$

∴ Total tensile stress in the arms at the hub end,

$$\sigma = \sigma_{t1} + \sigma_{b1}$$

Notes : 1. The total stress on the arms should not exceed the allowable permissible stress.

2. If the flywheel is used as a belt pulley, then the arms are also subjected to bending due to net belt tension $(T_1 - T_2)$, where T_1 and T_2 are the tensions in the tight side and slack side of the belt respectively. Therefore the bending stress due to the belt tensions,

$$\sigma_{b2} = \frac{(T_1 - T_2)(R - r)}{\frac{n}{2} \times Z}$$

... ($\because$ Only half the number of arms are considered to be effective in transmitting the belt tensions)

$\therefore$ Total bending stress in the arms at the hub end,

$$\sigma_b = \sigma_{b1} + \sigma_{b2}$$

and the total tensile stress in the arms at the hub end,

$$\sigma = \sigma_{t1} + \sigma_{b1} + \sigma_{b2}$$

Design of Flywheel Arms

The cross-section of the arms is usually elliptical with major axis as twice the minor axis, as shown in Figure, and it is designed for the maximum bending stress.

Let a_1 = Major axis, and

b_1 = Minor axis.

$\therefore$ Section modulus,

$$Z = \frac{\pi}{32} \times b_1 (a_1)^2 \quad \text{...}(i)$$

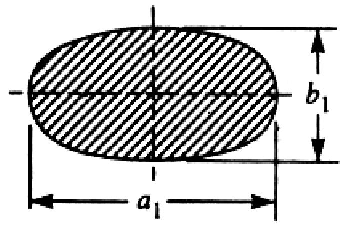


Fig. Elliptical cross section of arms.

We know that maximum bending moment,

$$M = \frac{T}{R.n}(R - r)$$

$\therefore$ Maximum bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{T}{R.n.Z}(R - r) \quad \text{...}(ii)$$

Assuming $a_1 = 2\,b_1$, the dimensions of the arms may be obtained from equations (*i*) and (*ii*).

Notes : 1. The arms of the flywheel have a taper from the hub to the rim. The taper is about 20 mm per metre length of the arm for the major axis and 10 mm per metre length for the minor axis.

2. The number of arms are usually 6. Sometimes the arms may be 8, 10 or 12 for very large size flywheels.

3. The arms may be curved or straight. But straight arms are easy to cast and are lighter.

4. Since arms are subjected to reversal of stresses, therefore a minimum factor of safety 8 should be used. In some cases like punching machines amd machines subjected to severe shock, a factor of safety 15 may be used.

5. The smaller flywheels (less than 600 mm diameter) are not provided with arms. They are made web type with holes in the web to facilitate handling.

Design of Shaft, Hub and Key

The diameter of shaft for flywheel is obtained from the maximum torque transmitted. We know that the maximum torque transmitted,

$$T_{max} = \frac{\pi}{16} \times \tau \, (d_1)^3$$

where d_1 = Diameter of the shaft, and

τ = Allowable shear stress for the material of the shaft.

The hub is designed as a hollow shaft, for the maximum torque transmitted. We know that the maximum torque transmitted,

$$T_{max} = \frac{\pi}{16} \times \tau \left(\frac{d^4 - d_1^{\,4}}{d} \right)$$

where d = Outer diameter of hub, and

d_1 = Inner diameter of hub or diameter of shaft.

The diameter of hub is usually taken as twice the diameter of shaft and length from 2 to 2.5 times the shaft diameter. It is generally taken equal to width of the rim.

A standard sunk key is used for the shaft and hub. The length of key is obtained by considering the failure of key in shearing. We know that torque transmitted by shaft,

$$T_{max} = L \times w \times \tau \times \frac{d_1}{2}$$

where L = Length of the key,

τ = Shear stress for the key material, and

d_1 = Diameter of shaft.

Example 7. *Design and draw a cast iron flywheel used for a four stroke I.C engine developing 180 kW at 240 r.p.m. The hoop or centrifugal stress developed in the flywheel is 5.2 MPa, the total fluctuation of speed is to be limited to 3% of the mean speed. The work done during the power stroke is 1/3 more than the average work done during the whole cycle. The maximum torque on the shaft is twice the mean torque. The density of cast iron is 7220 kg/m³.*

Solution. Given : $P = 180$ kW $= 180 \times 10^3$ W ; $N = 240$ r.p.m. ; $\sigma_t = 5.2$ MPa $= 5.2 \times 10^6$ N/m² ; $N_1 - N_2 = 3\%\, N$; $\rho = 7220$ kg/m³

First of all, let us find the maximum fluctuation of energy (ΔE). The turning moment diagram of a four stroke engine is shown in Fig.

We know that mean torque transmitted by the flywheel,

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{180 \times 10^3 \times 60}{2\pi \times 240} = 7161 \text{ N-m}$$

and *workdone per cycle $= T_{mean} \times \theta = 7161 \times 4\pi = 90\,000$ N-m

*The workdone per cycle may also be obtained as discussed below :

Workdone per cycle $= \dfrac{P \times 60}{n}$, where n = Number of working strokes per minute

For a four stroke engine, $n = N/2 = 240/2 = 120$

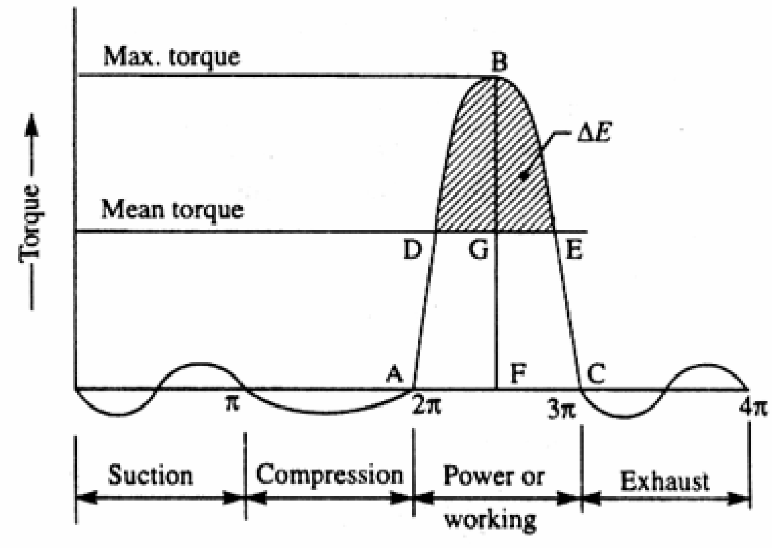
$\therefore$ Workdone per cycle $= \dfrac{180 \times 10^3 \times 60}{120} = 90\,000$ N-m

Since the workdone during the power stroke is 1/3 more than the average workdone during the whole cycle, therefore,

Workdone during the power (or working) stroke

$$= 90\,000 + \tfrac{1}{3} \times 90\,000 = 120\,000 \text{ N-m} \quad ...(i)$$

The workdone during the power stroke is shown by a triangle ABC in Figure in which the base $AC = \pi$ radians and height $BF = T_{max.}$



$\therefore$ Workdone during power stroke $= \frac{1}{2} \times \pi \times T_{max}$ ...(ii)

From equations (i) and (ii), we have

$\frac{1}{2} \times \pi \times T_{max} = 120\,000 \quad \therefore \quad T_{max} = \dfrac{120\,000 \times 2}{\pi} = 76\,384$ N-m

Height above the mean torque line,

$$BG = BF - FG = T_{max} - T_{mean} = 76\,384 - 7161 = 69\,223 \text{ N-m}$$

Since the area BDE shown shaded in Figure above the mean torque line represents the maximum fluctuation of energy (ΔE), therefore from geometrical relation,

$$\frac{\text{Area of } \Delta\, BDE}{\text{Area of } \Delta\, ABC} = \frac{(BG)^2}{(BF)^2}, \text{ we have}$$

*Maximum fluctuation of energy (*i.e.* area of $\Delta\, BDE$),

$$\Delta E = \text{Area of } \Delta\, ABC \times \left(\frac{BG}{BF}\right)^2 = 120\ 000 \left(\frac{69\ 223}{76\ 384}\right)^2 = 98\ 555 \text{ N-m}$$

*The approximate value of maximum fluctuation of energy may be obtained as discussed below:

Workdone per cycle = 90 000 N-m ...(as calculated above)

Workdone per stroke = 90 000 / 4 = 22 500 N-m ...($\because$ of four stroke engine)

and workdone during power stroke = 120 000 N-m

$\therefore$ Maximum fluctuation of energy,

$$\Delta E = 120\ 000 - 22\ 500 = 97\ 500 \text{ N-m}$$

1. *Diameter of the flywheel rim*

Let D = Diameter of the flywheel rim in metres, and

v = Peripheral velocity of the flywheel rim in m/s.

We know that the hoop stress developed in the flywheel rim (σ_t),

$$5.2 \times 10^6 = \rho.v^2 = 7220 \times v^2$$

$\therefore$ $v^2 = 5.2 \times 10^6 / 7220 = 720$ or $v = 26.8$ m/s

We also know that peripheral velocity (v),

$$26.8 = \frac{\pi D \,.\, N}{60} = \frac{\pi D \times 250}{60} = 13.1\ D$$

$\therefore$ $D = 26.8 / 13.1 = 2.04$ m **Ans.**

2. *Mass of the flywheel rim*

Let m = Mass of the flywheel rim in kg.

We know that angular speed of the flywheel rim, $\omega = \frac{2\pi \times N}{60} = \frac{2\pi \times 250}{60} = 25.14$ rad/s

and coefficient of fluctuation of speed, $C_S = \frac{N_1 - N_2}{N} = 0.03$

We know that maximum fluctuation of energy (ΔE),

$$98\ 555 = m.R^2.\omega^2.C_S = m\left(\frac{2.04}{2}\right)^2 (25.14)^2\ 0.03 = 19.73\ m$$

$\therefore$ $m = 98\ 555 / 19.73 = 4995$ kg **Ans.**

3. *Cross-sectional dimensions of the rim*

Let t = Depth or thickness of the rim in metres, and

b = Width of the rim in metres = $2\ t$ ...(Assume)

$\therefore$ Cross-sectional area of the rim, $A = b.t = 2\ t \times t = 2\ t^2$

We know that mass of the flywheel rim (m),

$$4995 = A \times \pi D \times \rho = 2\ t^2 \times \pi \times 2.04 \times 7220 = 92\ 556\ t^2$$

$\therefore$ $t^2 = 4995 / 92\ 556 = 0.054$ or $t = 0.232$ say 0.235 m

= 235 mm **Ans.**

and $b = 2\ t = 2 \times 235 = 470$ mm **Ans.**

4. *Diameter and length of hub*

Let d = Diameter of the hub, d_1 = Diameter of the shaft, and l = Length of the hub.

Since the maximum torque on the shaft is twice the mean torque, therefore maximum torque acting on the shaft, $T_{max} = 2 \times T_{mean} = 2 \times 7161 = 14\,322$ N-m $= 14\,322 \times 10^3$ N-mm

We know that the maximum torque acting on the shaft (T_{max}),

$$14\,322 \times 10^3 = \frac{\pi}{16} \times \tau\,(d_1)^3 = \frac{\pi}{16} \times 40\,(d_1)^3 = 7.855\,(d_1)^3$$

...(Taking $\tau = 40$ MPa $= 40$ N/mm^2)

$\therefore \quad (d_1)^3 = 14\,322 \times 10^3 / 7.855 = 1823 \times 10^3$

or $\quad d_1 = 122$ say 125 mm **Ans.**

The diameter of the hub is made equal to twice the diameter of shaft and length of hub is equal to width of the rim.

$\therefore \quad d = 2\,d_1 = 2 \times 125 = 250$ mm $= 0.25$ m

and $\quad l = b = 470$ mm $= 0.47$ mm **Ans.**

5. *Cross-sectional dimensions of the elliptical arms*

Let a_1 = Major axis, b_1 = Minor axis $= 0.5\,a_1$ n = Number of arms = 6 ...(Assume)

σ_b = Bending stress for the material of arms = 15 MPa = 15 N/mm^2 ...(Assume)

We know that the maximum bending moment in the arm at the hub end, which is assumed as cantilever is given by

$$M = \frac{T}{R.n}(R - r) = \frac{T}{D\,.\,n}(D - d) = \frac{14\,322}{2.04 \times 6}(2.04 - 0.25) \text{ N-m}$$

$$= 2094.5 \text{ N-m} = 2094.5 \times 10^3 \text{ N-mm}$$

and section modulus for the cross-section of the arm,

$$Z = \frac{\pi}{32} \times b_1\,(a_1)^2 = \frac{\pi}{32} \times 0.5\,a_1\,(a_1)^2 = 0.05\,(a_1)^3$$

We know that the bending stress (σ_b),

$$15 = \frac{M}{Z} = \frac{2094.5 \times 10^3}{0.05(a_1)^3} = \frac{41\,890 \times 10^3}{(a_1)^3}$$

$\therefore \quad (a_1)^3 = 41\,890 \times 10^3/15 = 2793 \times 10^3$ or $a_1 = 140$ mm **Ans.**

and $\quad b_1 = 0.5\,a_1 = 0.5 \times 140 = 70$ mm **Ans.**

6. *Dimensions of key*

The standard dimensions of rectangular sunk key for a shaft of diameter 125 mm are as follows :

Width of key, $\quad w = 36$ mm **Ans.**

and thickness of key $\quad = 20$ mm **Ans.**

The length of key (L) is obtained by considering the failure of key in shearing.

We know that the maximum torque transmitted by the shaft (T_{max}),

$$14\,322 \times 10^3 = L \times w \times \tau \times \frac{d_1}{2} = L \times 36 \times 40 \times \frac{125}{2} = 90 \times 10^3\,L$$

$\therefore \quad L = 14\,322 \times 10^3 / 90 \times 10^3 = 159$ say 160 mm **Ans.**

Let us now check the total stress in the rim which should not be greater than 15 MPa. We know that total stress in the rim,

$$\sigma = \rho.v^2\left(0.75 + \frac{4.935\,R}{n^2\,.\,t}\right)$$

$$= 7220\,(26.8)^2\left[0.75 + \frac{4.935\,(2.04/2)}{6^2 \times 0.235}\right] \text{N/m}^2$$

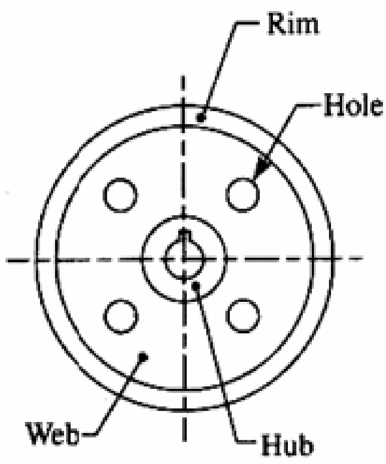
$$= 5.18 \times 10^6\,(0.75 + 0.595) = 6.97 \times 10^6 \text{ N/m}^2 = 6.97 \text{ MPa}$$

Since it is less than 15 MPa, therefore the design is safe.

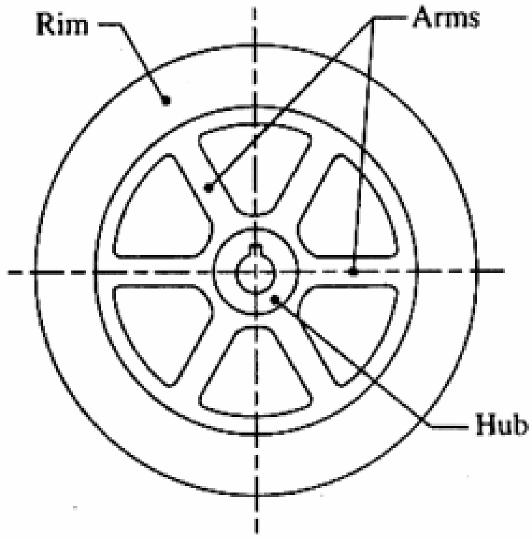
Construction of Flywheels

The flywheels of smaller size (upto 600 mm diameter) are casted in one piece. The rim and hub are joined together by means of web as shown in Figure (*a*). The holes in the web may be made for handling purposes.

In case the flywheel is of larger size (upto 2.5 metre diameter), the arms are made instead of web, as shown in Figure (*b*). The number of arms depends upon the size of flywheel and its speed of rotation. But the flywheels above 2.5 metre diameter are usually casted in two piece. Such a flywheel is known as *split flywheel*. A *split flywheel* has the advantage of relieving the shrinkage stresses in the arms due to unequal rate of cooling of casting. A flywheel made in two halves should be spilt at the arms rather than between the arms, in order to obtain better strength of the joint. The two halves of the flywheel are connected by means of bolts through the hub, as shown in Figure. The two halves are also joined at the rim by means of cotter joint (as shown in Figure) or shrink links (as shown in Figure). The width or depth of the shrink link is taken as 1.25 to 1.35 times the thickness of link. The slot in the rim into which the link is inserted is made slightly larger than the size of link.



(*a*) Flywheel with web.



(*b*) Flywheel with arms.

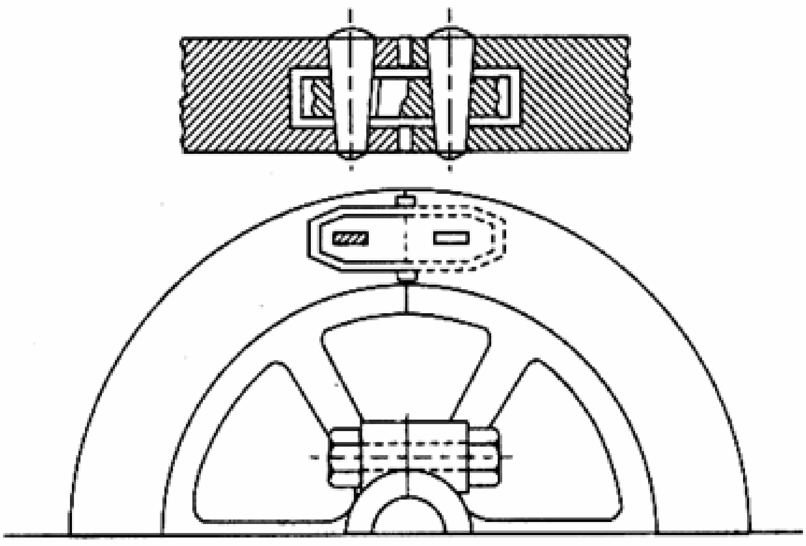


Fig. Split flywheel

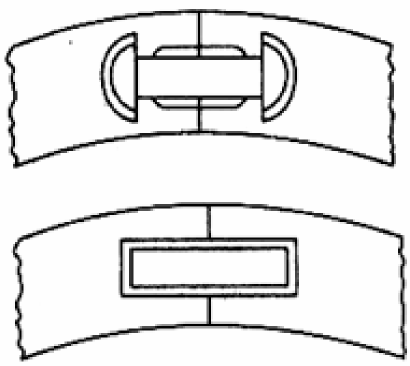


Fig. Shrink links.

- <u>References;</u>

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* Machine Design -R S Khurmi, *Auto Design -R B Gupta
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Camshaft

A cam is a mechanical member for transmitting a desired motion to a follower by direct contact. The driver is called the cam, and the driven member is called the follower. The cam, may remain stationary, or translate oscillate or rotate whereas the follower may translate or oscillate.

Types of Cams

Cams may be classified in the following three ways:

(i) Follower motion,

e.g. dwell-rise-dwell (D-R-D); dwell-rise return-dwell (D-R-R-D) ; or rise-return-rise (R-R-R).

(ii) Cam shape

e.g. wedge, radial, globoidal, cylindrical, conical, spherical.

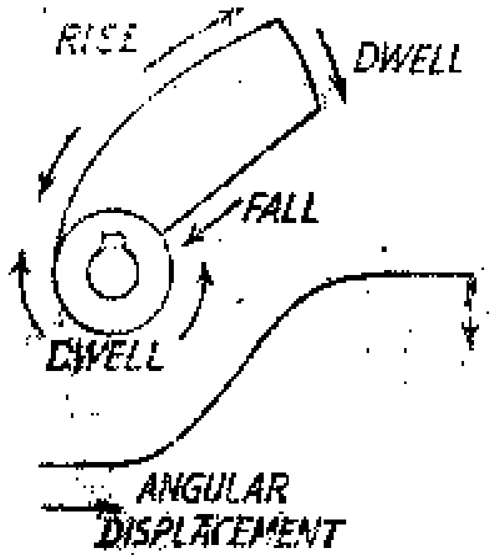
(iii) Manner of constraint of the follower.

Constraint may be obtained either by spring loading to keep the follower in contact with the cam surface, or by position drive.

(i) Classification Based on follower motion

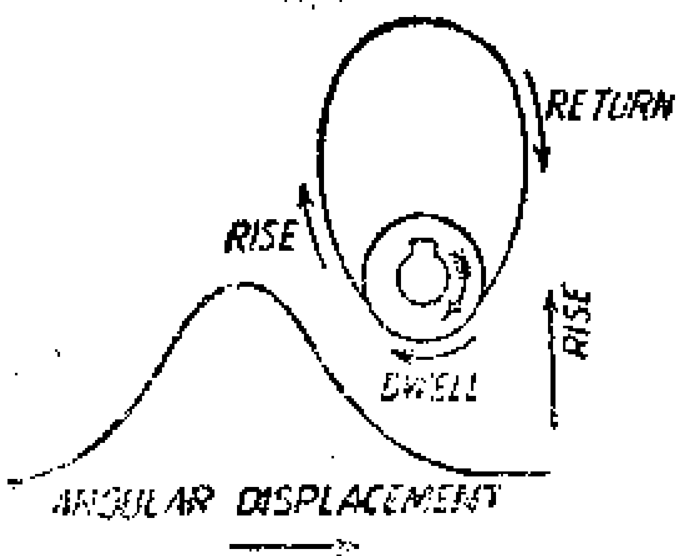
(a) Dwell-rise-dwell cam.

The zero displacement part of the cam is ca1led dwell. There is zero displacement followed by a rise contour, to another dwell, period, It is used very frequently in machinery, In this case rise is followed by a fall or it may be dwell-rise-dwell return.



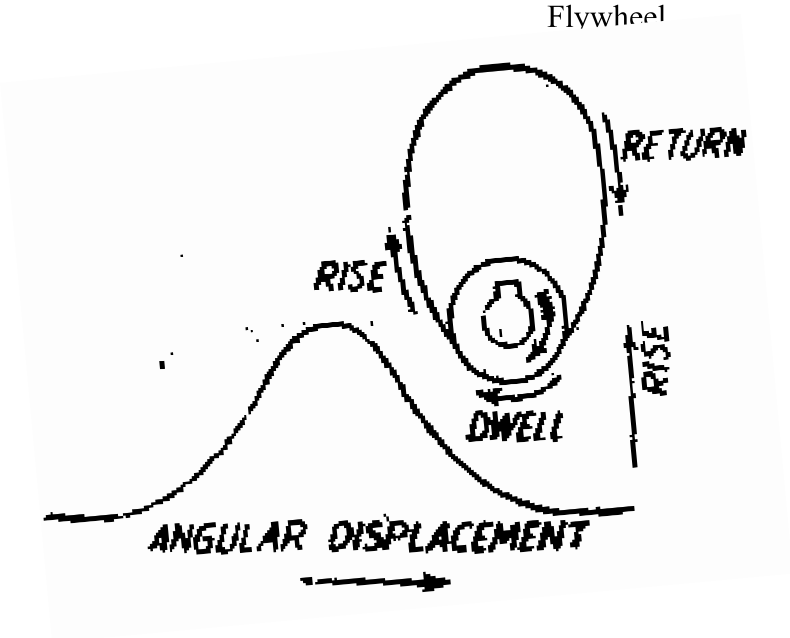
(b) Dwell-rise return-dwell cam (D-R-R-D)

In this type the rise and return are preceded & followed by dwells



(c) Rise-return-rise cam (R-R-R).

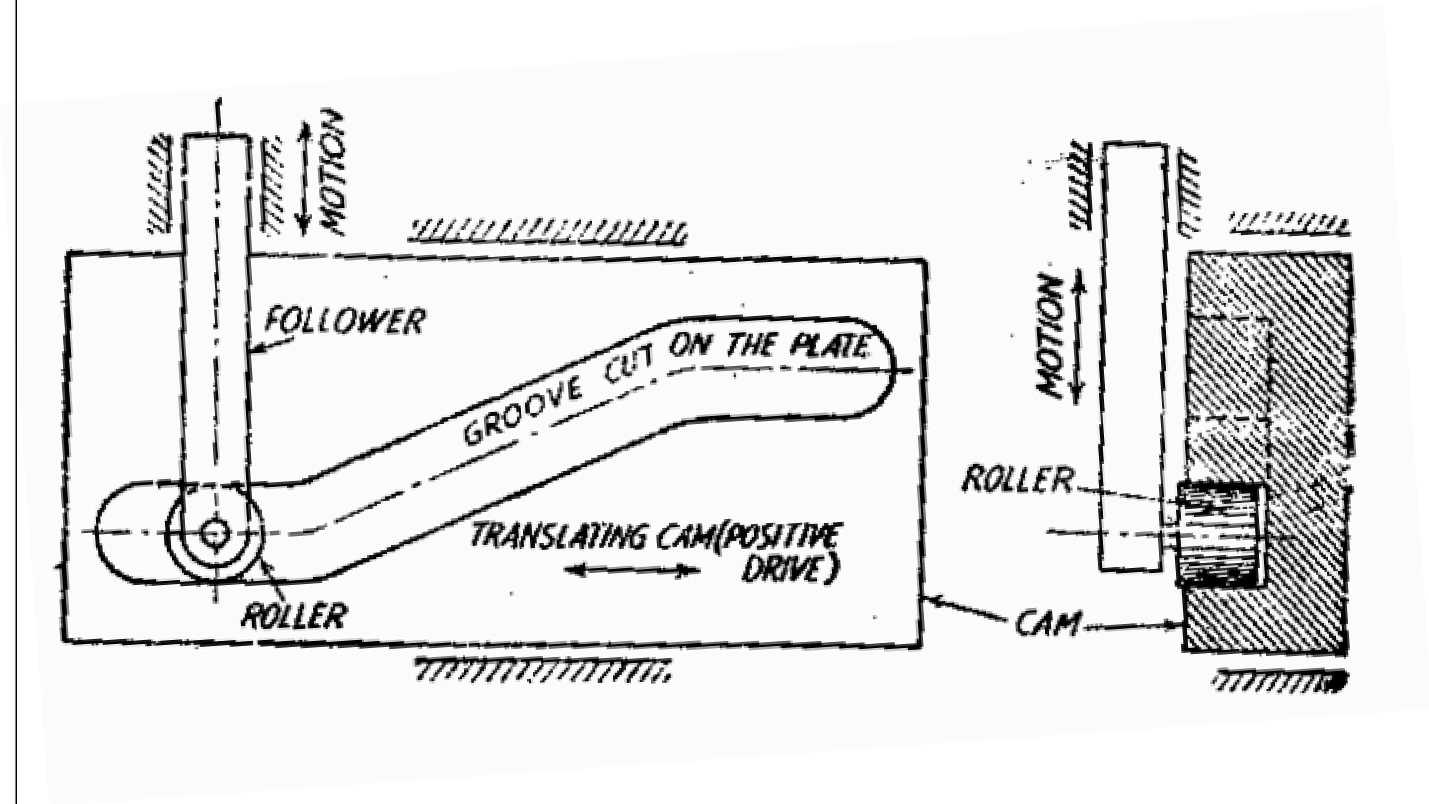
This has no dwells .its application is limited. An eccentric mechanism is suggested in place of such a contour cam



(ii) Classification Based on the shape of the cam

(a) The translation, wedge or .flat cam,

This is the-simplest type of cam. The plate cam moves back and forth imparting a translatory motion to the follower. The follower could be held in the groove & thus has a positive motion. The follower could be held by a spring & the plate cut to give a desired motion with out any groove cut. The follower will in that case move on the surface of the plate.



(b) The radial or disc cam.

The position of the follower is determined by the radial distance from the cam axis. The follower is held in contact by spring, or gravity. (Fig B)

In Fig.B1 &2, the difference is only of constraint. The follower has positive constraint. The radial type of cams are the most popular because of their simplicity and compactness.

The cam shown in Fig.B1 is also called the yoke cam.

The cam shown in Fig.B2 is conjugate supplementary or double disc cam. In this type, one roller is preloaded against the other. This imparts very precise motion.

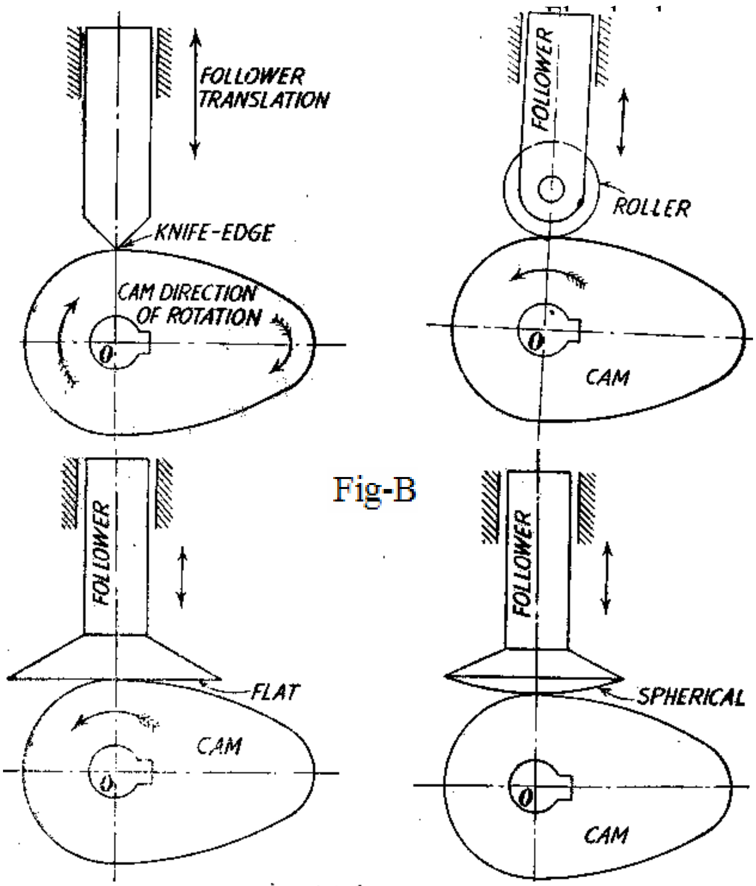


Fig-B

A particular type of <u>radial plate cam called circular arc cam used in Automobile,</u> has its contour composed of radii of circles.

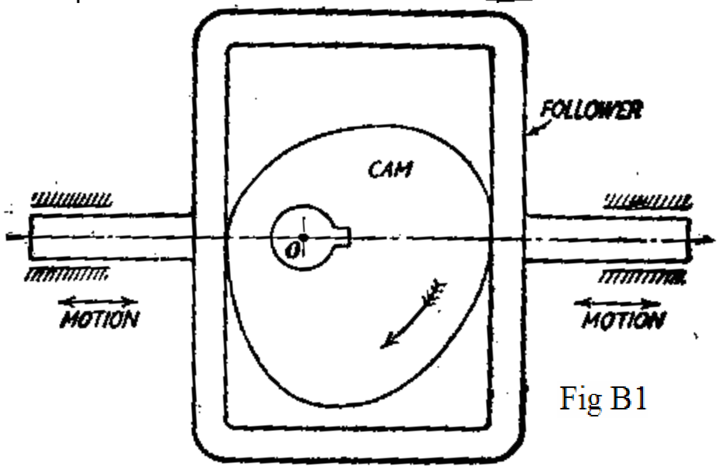
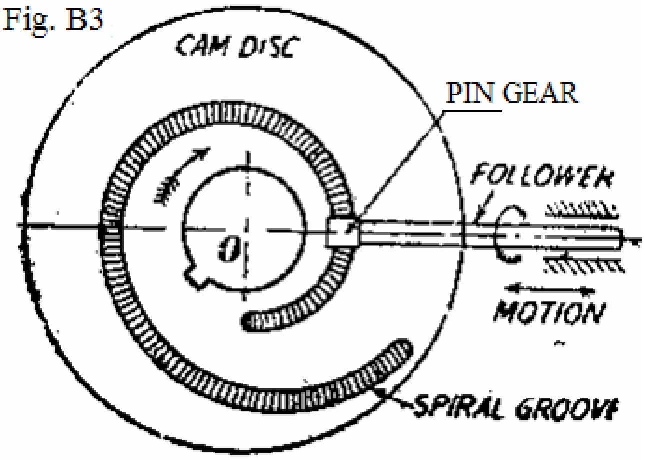


Fig B1

The spiral cam shown in Fig.B3 is a form of face cam having a spiral groove cut in it. Pin gear follower is driven by teeth in the groove; the follower velocity is a function of the radial distance of the groove from the axis of the cam.

<u>This cam is used in computers,</u>

Fig. B3



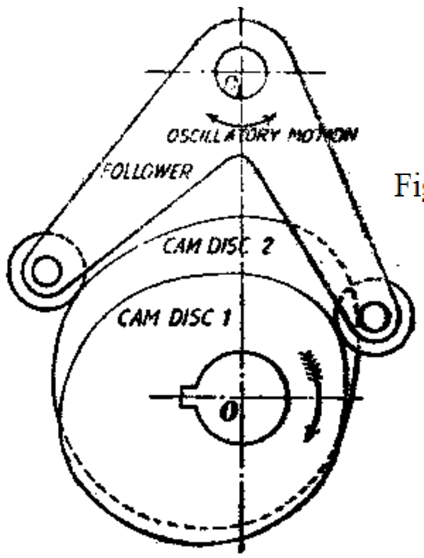


Fig B2

(c) The cylindrical or drum cam.

It has a circumferential contour cut in the surface of the cylinder which rotates about its axis. The follower may translate or oscillate in the direction of this axis, Two types of cylindrical cams, one in which the groove is cut on the surface and roller has constrained positive motion: and the other in which the end of the cylinder is the working surface are shown in Figs C1 and C2, The follower in Fig, C1 is having an oscil1atory motion and Fig, C2 it is having translatory motion.

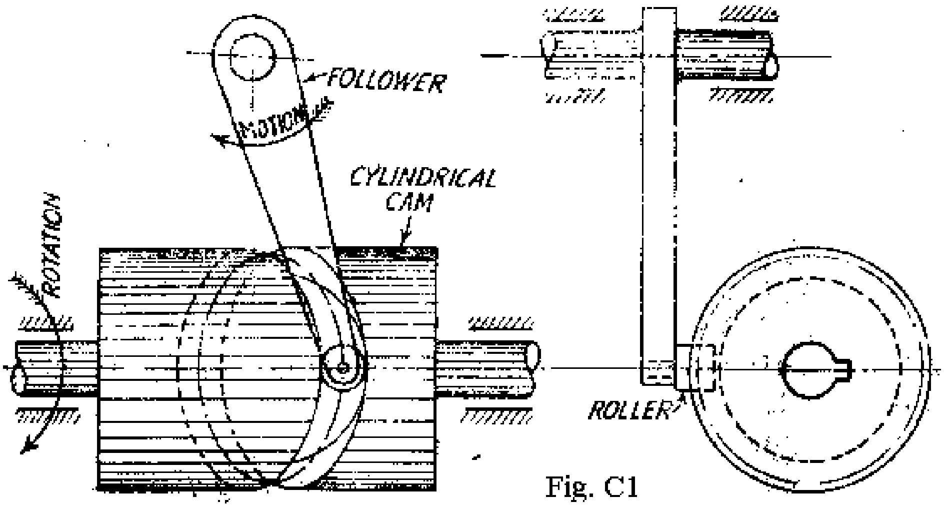


Fig. C1

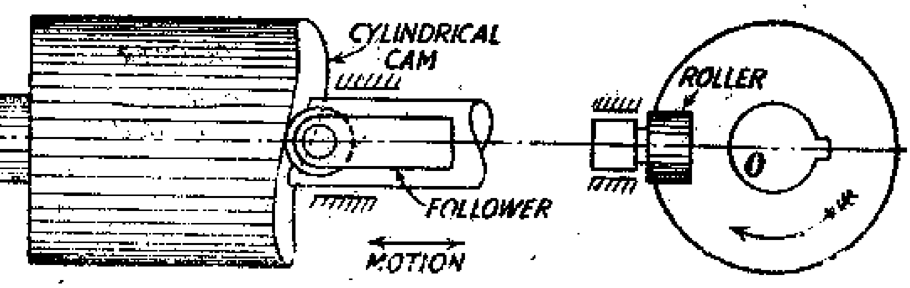


Fig. C2

(d) The globoidal cam,

It is similar to the cylindrical cam. Convex or concave globoids, as shown in Figs.D1 and D2 replace the cylinder. These cams, though of much academic interest have not been popular. These cams are used for indexing and other such purposes. These are used for much specialized application.

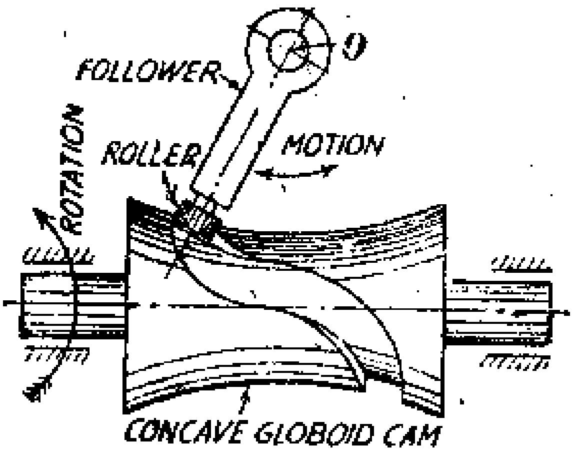


Fig. D1

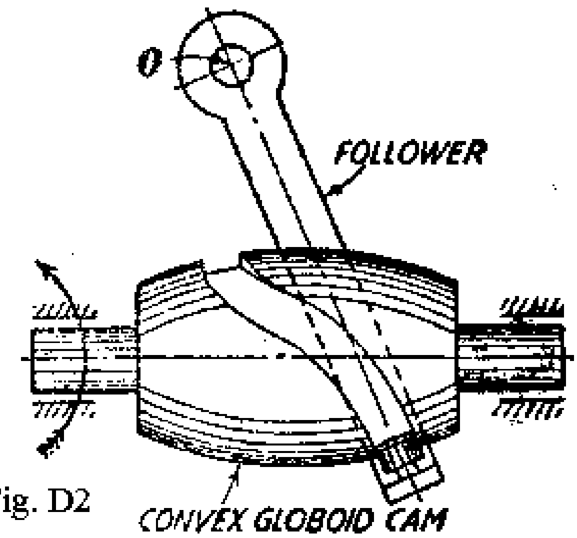


Fig. D2

(iii) Classification based on constraint

(a) Spring loaded or pre-loaded.

In the radial cams of the types shown in Fig. A1, the follower is to be held by an external force provided by pre-compression of the spring or hydraulic load or gravity. The cylindrical cam shown in Fig. A2 is also a case of successful constraint.

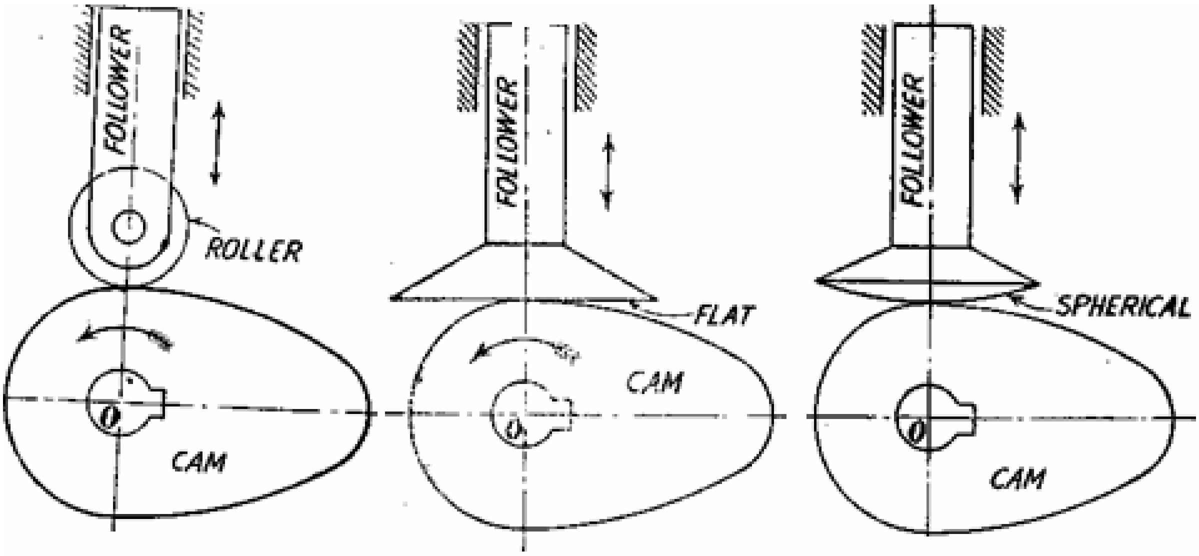


Fig. A1

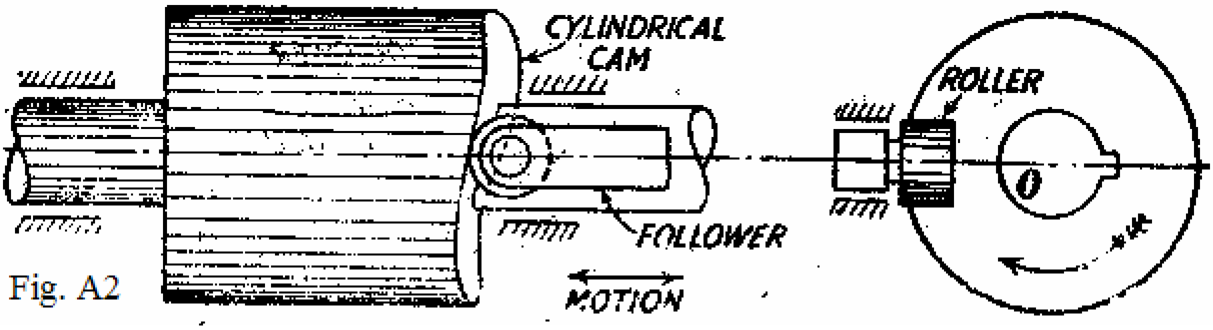


Fig. A2

(iii)

(b)The cylindrical cam shown in Fig. B1, the globoidal cams shown in Figs. B2 and B3, and the yoke cams (Fig.B4) are all cases of positive constraint. No external force is; required to keep the follower and the cams surface in contact

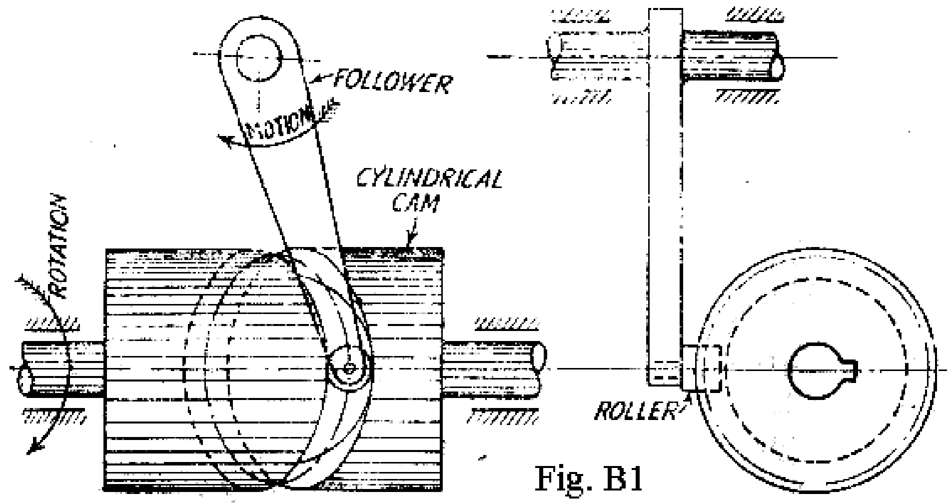


Fig. B1

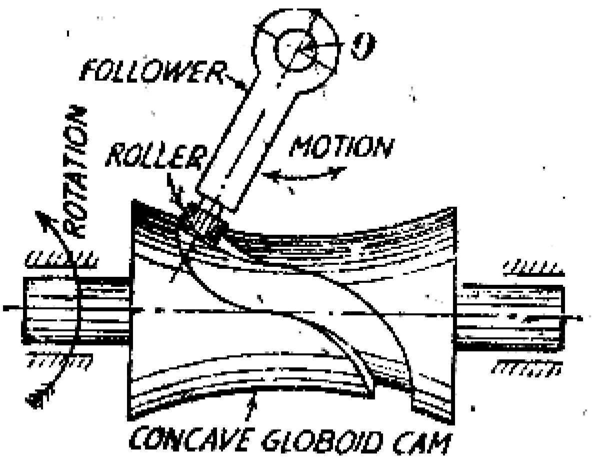


Fig B2

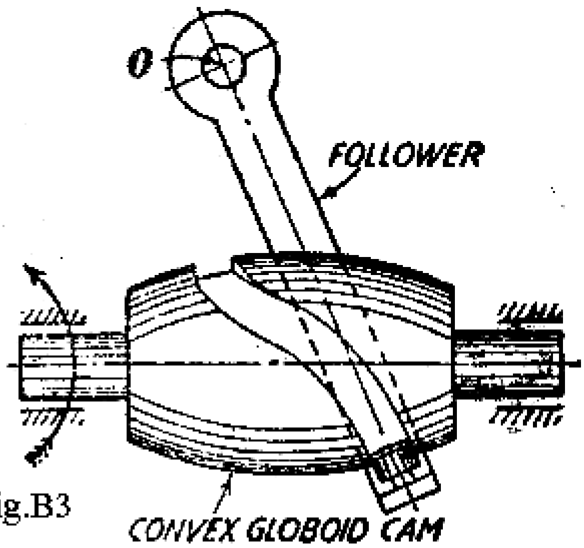
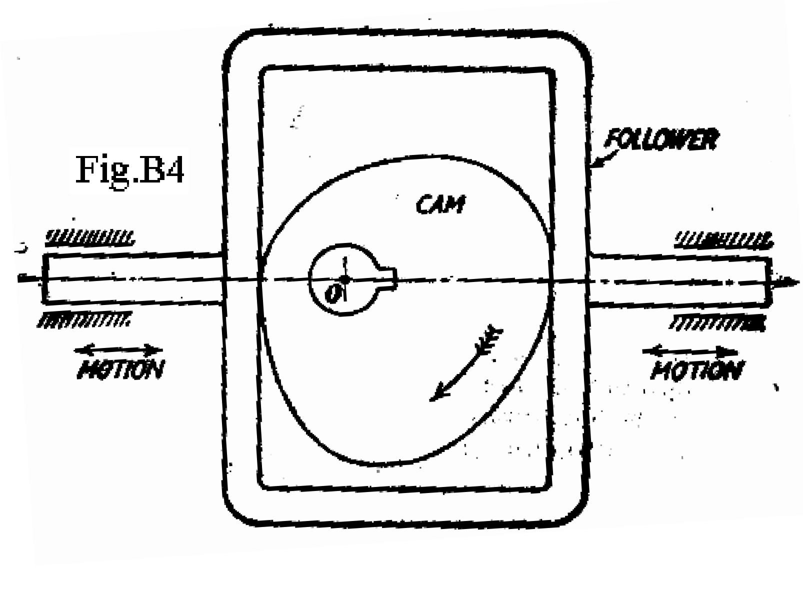


Fig.B3

Fig.B4



<u>Typesof Followers</u>.

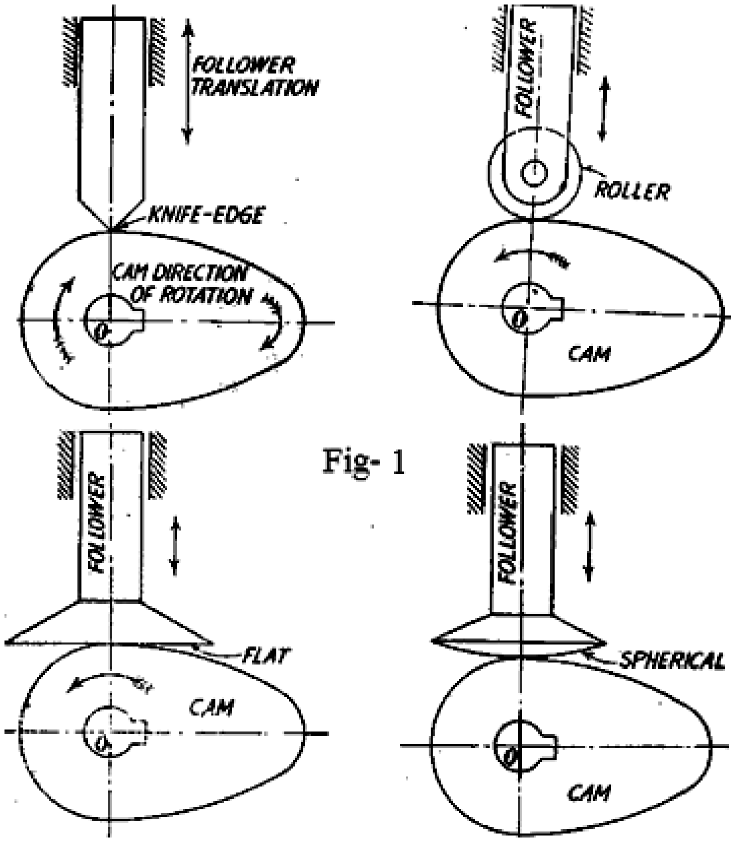


Fig- 1

Followers may be classified in the following three ways:

(i) Construction of the surface in contact

e.g. knife-edge, roller, flat-faced mushroom or special shape mushroom.

(ii) The type of motion-translatory and oscillatory.

(iii)Location of the line of motion, with respect to cam centre, radial & offset

(i) Classification based on the surface in contact (refer figure 1)

(a) Knife-edge,

A sharp knife edge is in contact with cam. It is the simplest in construction, but of very little practical use due to the fact that extreme wear of the cam surface and the contact will take place.

(b) A roller follower

It is a cylindrical roller held by a pin to the follower assembly. At low speeds, a perfect rolling contact is possible, though some sliding occurs at high speeds. This type is very extensively used. Aircraft engines employ roller followers to limit wear at high cam peripheral velocities.

(C)Mushroom follower.

It may be flat-faced having perfectly flat plane. This causes high surface stresses. To minimise these stresses a spherical shape having a surface of large sphere radius is used.

Automobile engines use flat-faced mushroom followers with spherical curvature in preference to rollers because of limited space and pin weakness.

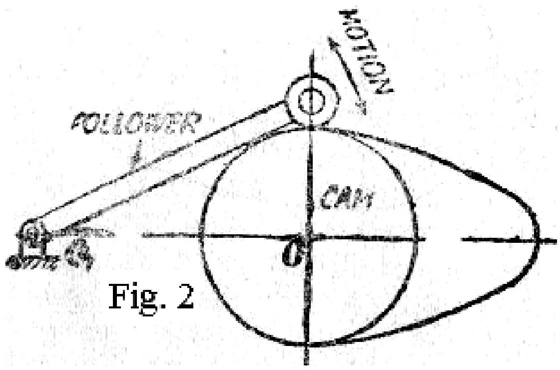


Fig. 2

(ii) Classification based on type of movement

(a) Translation.

The followers shown in above Fig.1 are all translatory followers. The cam rotates uniformly and the follower reciprocates in the guides.

(b) Oscillation.

A uniform rotary motion of the cam is converted to a pre-determined oscillatory motion of the follower which is pivoted at a suitable point to the fixed frame.

(iii) Classification based on line of motion

(a) Radial follower, (refer figure 1)

Radial follower is one in which the follower translates along an axis passing through the cam centre of rotation.

(b) Offset follower,

Offset follower is one in which the axis of the follower movement is displaced from the cam, centre of rotation. The off-set may be to the right or to the left.

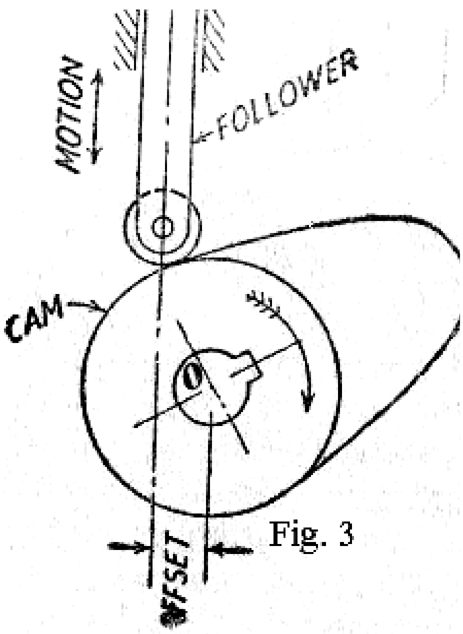


Fig. 3

Camshaft

The camshaft provides a means of actuating the opening and controlling the period before closing, both for the inlet as well as the exhaust valves. It also provides a drive for the ignition distributor and the mechanical fuel pump.

The camshaft receives its motion from the crankshaft, from which all of the accessories also must be driven.

In the case of passenger car engines these accessories usually include

a fuel pump,
an oil pump,
a water pump,
a generator,
an ignition unit, and
a fan.

In cars equipped with hydraulic power steering a drive must be provided for an additional pump.

On truck and bus engines there may be an air compressor, a governor, and a magneto in addition, though the latter item of equipment is now rarely found on road vehicles. .A drive must be provided also for the starting motor.

The camshaft consists of a number of cams at suitable angular positions for operating the valves at approximate timings relative to the piston movement and in a sequence according to the selected firing order. There are two lobes on the camshaft for each cylinder of the engine; one to operate the intake valve and the other to operate the exhaust valve. A number of integral bearing journals support the shaft in bearings. Camshaft bearing journals are always larger than the cam lobes so that the camshaft may be installed in the engine through the cam bearings. To provide room for lubrication and metal expansion, a clearance of 0.05 to 0.125 mm is usually provided between the bearing journals and the bores.

Usually there is an integral spiral toothed gear on the camshaft to drive the distributor and the oil pump. The fuel pump is operated from an integral eccentric or a bolt-on eccentric. In some cases, however, distributors are driven directly from the camshaft end.

Endwise movement of the camshaft is limited by a thrust plate between the front bearing journal and the drive gear or the sprocket. This thrust plate is bolted to the engine. However, in some engines instead of the thrust plate, the same is achieved by the tapered spiral teeth of the distributor oil pump drive.

Material

The camshaft is forged from alloy steel or cast from hardenable cast iron and is case hardened. A typical cast iron alloy for a camshaft would consist of

3.3% carbon,
2% silicon,
0.65 % manganese,
0.65 % chromium,
0.25% molybdenum and the remainder iron.

In modern engines, cam lobes are ground with a slight taper across the face. The tappets used with such camshafts have spherical base and are slightly offset from the cam face. This provides tappet rotation and a wear pattern preventing edge loading which is a major cause of failure. Camshaft is supported in a number of bearings.

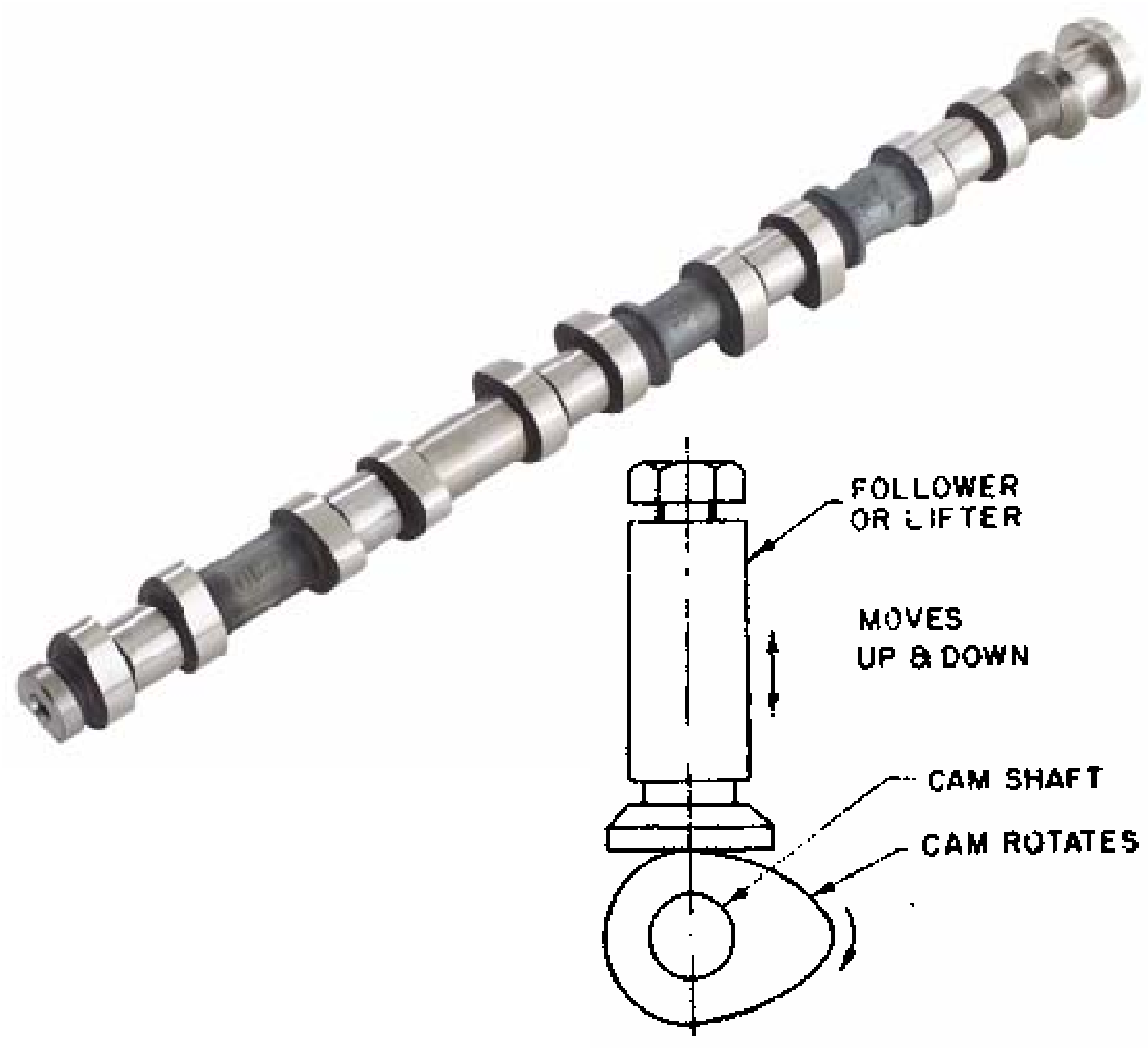
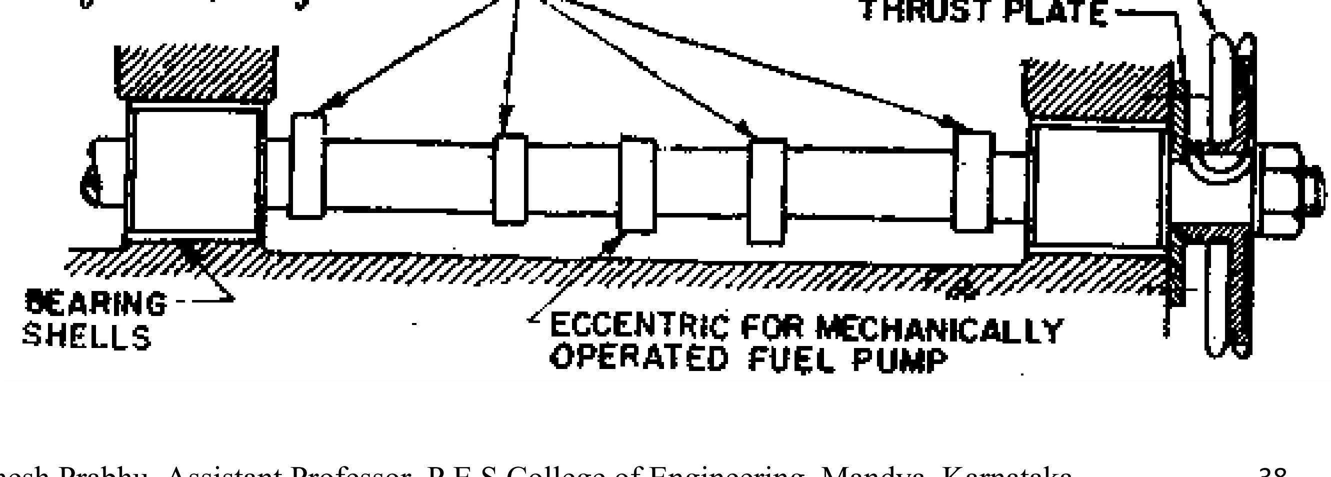


Fig. Cam and Follower.

Fig. Cam shaft.



Camshaft drive

The camshaft rotates at half the crankshaft speed so as to open and close the valves once in every two revolutions of crankshaft.

In designing a drive for the camshaft and accessories of passenger- car engines, silence of operation is an important consideration. Most of the early automobile engines had a plain spur-gear drive, but in 1905 to eliminate the noise of poppet valves, it was found that, in order to achieve any real improvement, other sources of noise also to be eliminated as well, and so for the half-time shaft a toothed-chain drive was used.

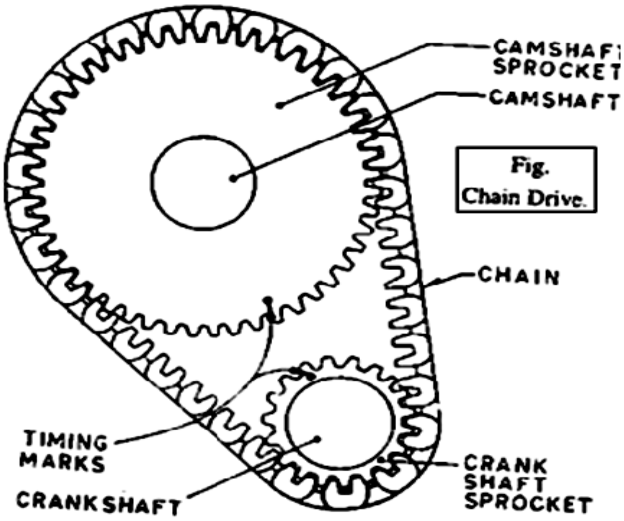


Fig. Chain Drive.

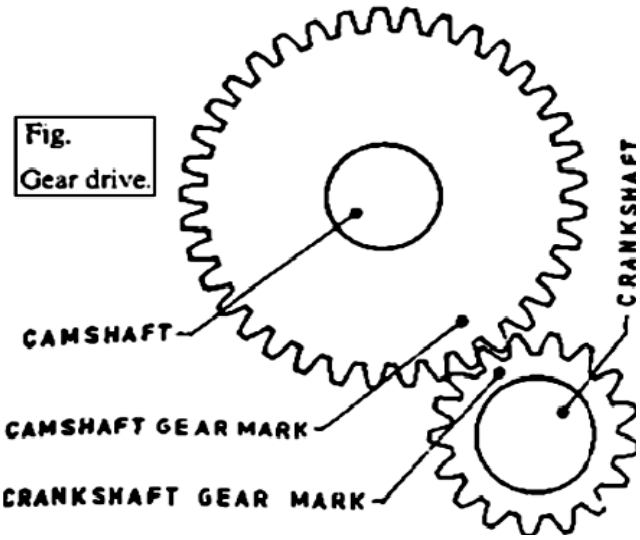


Fig. Gear drive.

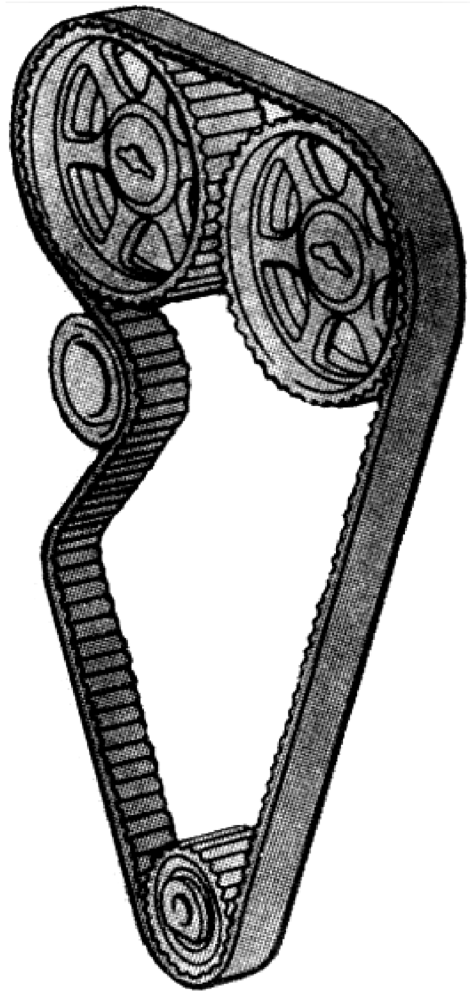


Fig. Toothed Drive belt for doubleoverhead camshafts

With increasing engine speeds the problem of reducing gear noise became constantly more urgent, and from 1911 on camshaft gears on the more expensive cars were cut with helical teeth, the pinion on the crankshaft at that time being made of steel and the gear on the camshaft of cast iron. Most truck engines still use such metal gears with helical teeth, while the majority of passenger car engines now have a chain drive, and those that have gear drive have one of the gears made of non-metallic material.

The drive from crankshaft to the camshaft may be either chain drive or gear drive where camshaft gear or sprocket wheel is twice as large as the crankshaft gear or sprocket wheel. There are timing marks on the sprockets or gears of the camshaft and the crankshaft to ensure correct valve timing. In a chain drive a separate idler gear has to be provided. Moreover, a long chain drive tends to whip to avoid which a tensioning device is used, which may be a roller or spring-loaded steel strip or a rubber pad attached to a spring-loaded piston.

Gear drives, on the other hand, need no tensioner, but are noisy and are suitable only for camshafts mounted close to the crankshafts. Obviously gear is not suitable for overhead camshafts.

The latest type of drive is, however, by means of a toothed rubber belt, which is made of rubber molded onto a non-stretching cord Such belts operating relatively silently, do not need any lubrication and are becoming increasingly popular for both petrol as well as diesel engines with overhead camshaft. Periodic tensioning checks are all that is required for these.

Timing gears are made of cast iron, steel, aluminum or laminated fiber. Chains are made with links of alloy steel with ground and case hardened pins.
Sprockets are made of nylon, aluminum, steel or iron.

Oil Pump and Ignition-Unit Drive

The ignition unit combines an interrupter and a distributor which must be driven at camshaft speed. A very common practice is to provide on the engine a shaft, either vertical or inclined, which is driven from the camshaft by helical gearing, generally located at the middle of the length of the engine, to the upper end of which shaft the ignition unit is secured, and to the lower end the oil pump. The drive at both ends is through an integral transverse key and slot. This arrangement has the advantages that it makes one drive do for two accessories, places the gearing where it is well lubricated and well protected against dirt, and brings the ignition unit into an accessible position at a minimum average distance from the spark plugs. If the shaft is arranged vertically the pump will be immersed in oil and will not have to lift it by suction, whereas if it is inclined the pump will be outside the crankcase and therefore more accessible. An ignition-unit and oil-pump drive for an engine in which the pump is outside the crankcase is shown in Figure.

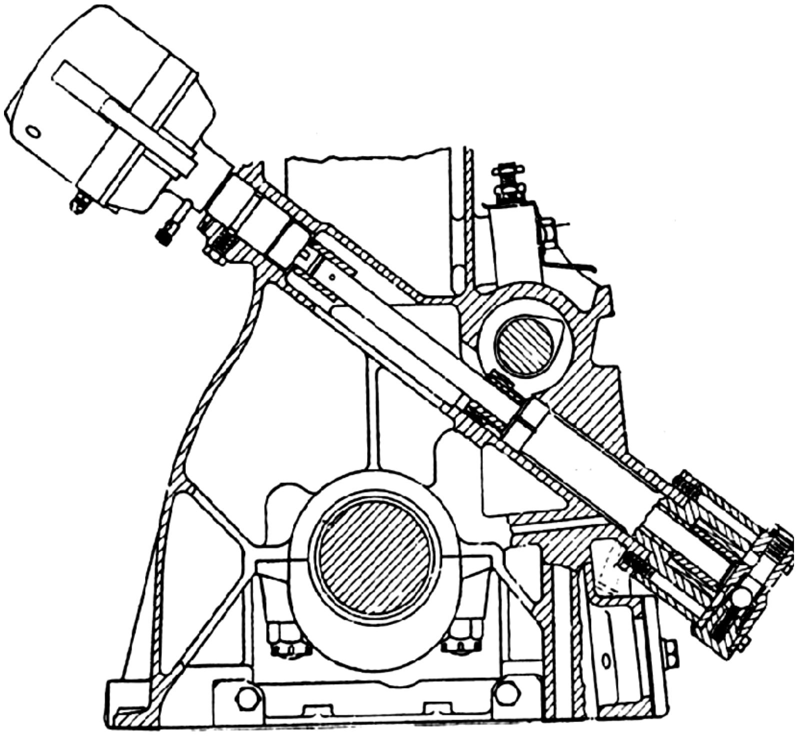


Fig. Drive for ignition unit and oil pump.

Cast Camshafts.

For many years camshafts were made exclusively of low-carbon steel and case-hardened on the cam surfaces, but during the early thirties one engine manufacturer began to use camshafts of cast iron, and the practice soon became quite general.

Cast camshafts offer a number of advantages from the production standpoint. A material saving results from the use of patterns instead of dies in the production of the blanks, and if changes in design become necessary, they can be incorporated in the patterns much more quickly and with less expense than in the dies. With conventional cast camshafts the processes of copper plating, carburizing and heat treating are eliminated. Rough-turning of the cams and eccentric are unnecessary, these parts being finished directly by grinding. As a rule, no finish is allowed on the sides of bearings and cams, and on the shaft diameter. The material used is a superior grade of cast iron melted in the electric furnace. Originally both the cams and the journals were cast in chills and sufficiently close to size so that only grinding was required to finish them. These chilled surfaces are said to offer great resistance to wear than the case-hardened surfaces of the usual steel cams.

The use of numerous chills in the mold is troublesome, and it was later found that the necessary hardness on the cam surfaces can be produced without the use of chills. The cam surfaces are merely heated with a gas torch and air-hardened and large numbers of cast-iron camshafts are now made in this way. In one case where chilling is employed the iron used has the following composition:

Total carbon, 3.30-3.65%;
silicon, 0.45-0.55%;
manganese, 0.15-0.35%;
copper, 2.5-3.0%;
chromium, 0.25% max.;
phosphorus, 0.05% max.;
sulphur. 0.06% max.

These camshafts in the "as cast" condition show a Brinell hardness of about 300 on the bearings and 450 on the chilled cam surfaces. Where chills are used they must be accurately located, as the chilling effect does not penetrate very deeply, and if there are any material inaccuracies in. the cam part of the shaft, more material may have to be ground away on one side of the lobe than on the other, and all of the chilled material may be removed on one side. On the chilled cam surfaces a stock allowance of 1/32-3/64 in. is made, and on the bearing surface, 1/l6 in.

Camshaft Manufacture

Camshafts of drop-forged steel, after being rough-turned, are cemented or carburized at the cam portions and then quenched from a temperature slightly above the critical point of the steel, so as to render the cam surfaces very hard and resistant to wear. To prevent carburization of the rest of the surfaces, the shaft is first copper-plated, and the layer of copper is then removed by machining where it is desired to carburize the surface. Carbon will not penetrate the layer of copper. Another plan consists in leaving those portions of the shaft which are to remain soft, over-size until after the carburizing process, and then turning off the carburized layer.

Machining Camshaft

All of the turning operations on camshafts with the exception of forming the contour of the cams are often performed in Lo-Swing lathes. With the proper tooling, all of these surfaces oil an average automobile camshaft can be turned in a minute and a half.

With the attachment used with the Lo-Swing lathe for the spacing of the cams, the sides of the cams and the bearings are all properly spaced and faced, and any chamfering or necking operations are also performed.

As high, as thirty-five and forty tools are used on a camshaft simultaneously. Of course, if all these tools were to act at once and all of them were to take the maximum cut, the shaft would be sprung so that it would be worthless. To overcome this, each slide carrying a set of tools is fed into the work a little ahead of the slide next to it. In this manner the tools in the first slide, for instance, will be through their heaviest cutting by the time the tools in the second slide are ready to take their heavy cut, and so on to the third and fourth slide.

After the camshafts are spaced and faced with this attachment, they are put back in the lathe and the portion between the cams is turned. Generally this is done with simple tooling, although all bearings and spaces are turned simultaneously, thus reducing the time considerably.

Cam-Grinding

It may be pointed out that cam-grinding machines designed to handle integral camshafts work on practically the same principle. The cam to be ground is secured on an arbor inserted into a spindle which carries the master cam at its other end. The spindle is carried in a head provided with an upwardly extending arm whereby it is swung from the body or frame of the attachment, which is clamped to the grinder frame. The swinging head is forced to one side by a coiled spring so that the master cam presses against a stationary shoe plate. The spindle carrying the cams is driven by a V belt from a jackshaft which in turn is driven by a belt from the headstock face plate. Only the hub of the V belt pulley on the spindle is shown in the drawing. To make the necessary allowance for wear of the grinding wheel, a set of different master-cam shoe plates are furnished with the attachment, which are inserted one after another as the wheel wears down, the plates having contact surfaces of different curvature; By means of this attachment it is possible also to cut the master cam from a template. A method of grinding the cam with the flat side of the wheel has been developed, whereby inaccuracies due to change in curvature of the grinding surface with wear are eliminated.

To grind an integral camshaft it is necessary to first make a master camshaft with all of the cams correctly spaced angularly and longitudinally. This camshaft is driven in synchronism with the one to be ground, and acts on a roller carried by a cross slide carrying the grinding wheel, the roller being drawn against the master cam by a spring acting on the cross slide.

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