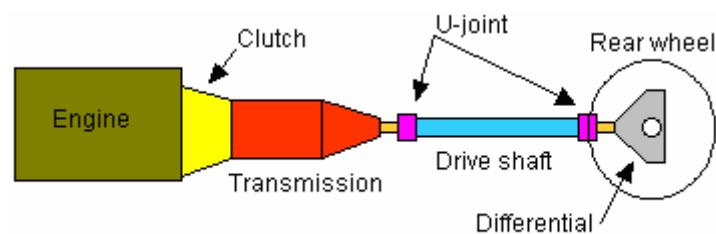


UNIT I

CLUTCH AND GEAR BOX

TRANSMISSION SYSTEM

Chief function of the device is to receive power at one torque and angular velocity and to deliver it at another torque and the corresponding angular velocity.



LAYOUT OF AUTOMOBILE POWER TRANSMISSION SYSTEM

REQUIREMENTS OF TRANSMISSION SYSTEM

1. To provide for disconnecting the engine from the driving wheels.
2. When the engine is running, to enable the connection to the driving wheels to be made smoothly and without shock.
3. To enable the leverage between the engine and driving wheels to be varied.
4. It must reduce the drive-line speed from that of the engine to that of the driving wheels in a ratio of somewhere between about 3:1 and 10:1 or more, according to the relative size of engine and weight of vehicle.
5. Turn the drive, if necessary, through 90° or perhaps otherwise re-align it.
6. Enable the driving wheels to rotate at different speeds.
7. Provide for relative movement between the engine and driving wheels.

CLUTCH

The clutch is housed between the engine and transmission where it provides a mechanical coupling between the engine's flywheel and the transmission input shaft. The clutch is operated by a linkage that extends from the passenger compartment to the clutch housing. The purpose of the clutch is to disconnect the engine from the driven wheels when a vehicle is changing gears or being started from rest.

Disengaging the clutch separates the flywheel, the clutch plate and the pressure plate from each other. The flywheel is bolted to the end of the crankshaft and rotates with it. The clutch plate is splined to the gearbox in order for both to rotate together and the pressure plate clamps the clutch plate to the flywheel. When the pressure is released by depressing the clutch pedal, the crankshaft and gearbox input shaft rotate independently. When the foot is taken off they rotate as one.

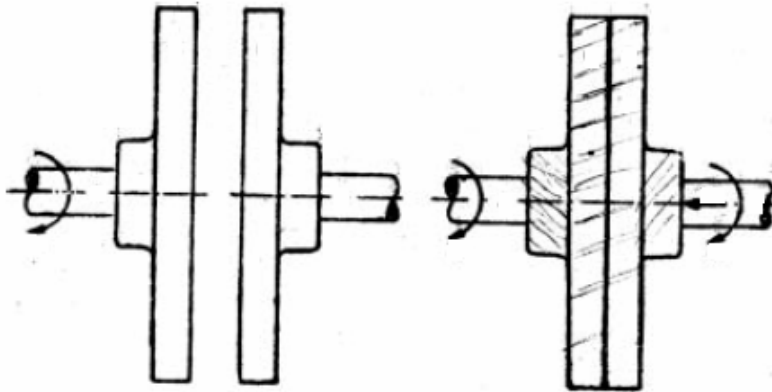
REQUIREMENTS OF A CLUTCH

The clutch must

1. Pick up its load smoothly without grab or clatter.
2. Have a driven disc of low moment of inertia to permit easy shifting.
3. Damp out any vibration of the crankshaft to prevent gear clatter.
4. Require little pedal pressure to operate it.
5. Be easy to adjust and service.
6. Be cheap to manufacture.

BASIC PRINCIPLE OF THE FRICTION TYPE CLUTCH

To understand the working principle of clutch, let's take two sanding discs, first one driven by a power drill corresponds to the flywheel of a car, driven by the engine. If a second sanding disc is brought into contact with the first, friction makes it revolve too but more slowly. But when the second disc pressed against the first disc which is connect to the power drill, as the pressure increases the two discs revolve as one. This is how a friction clutch works.



BASIC PRINCIPLE OF THE FRICTION TYPE CLUTCH

TYPES OF CLUTCHES

MULTI COIL SPRING SINGLE PLATE CLUTCH

CONSTRUCTION

A typical clutch actuated by a number of coil springs on a pitch circle nears the periphery is shown. The driven shaft which normally is a forward extension of gearbox primary shaft is supported at its front end in ball bearing in a hole in the centre of flywheel web, which is spigot and bolted on to a flange at the rear end of the crankshaft.

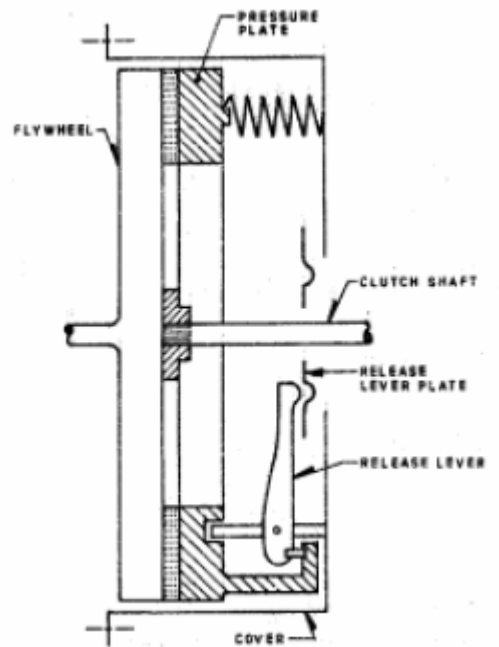
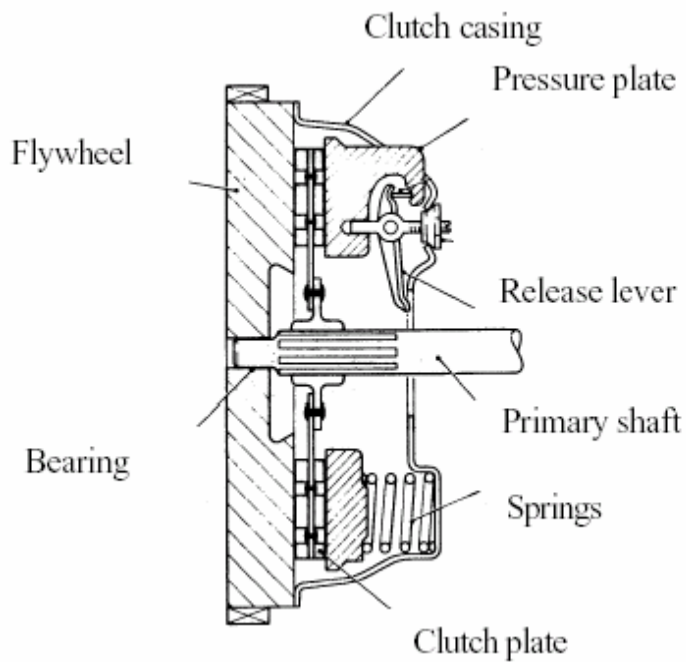
In this clutch, the coil springs force the pressure plate forwards to clamp the driven plate between it and the rear face of the flywheel. Three lugs extend rearwards from periphery of pressure plate both to rotate the pressure plate and to cause it to rotate with the rest of the assembly. The driven plate of course is splined onto the shaft.

There are three release levers pressing the coil springs at the outer end. The inner ends of the levers can be forced forward by means of thrust bearing made of graphite and slide along the clutch shaft when clutch pedal is depressed. The driven plate mounted between flywheel and pressure plate makes the clutch shaft to rotate to transmit power. It has the clutch facing made of friction materials around the periphery of disc.

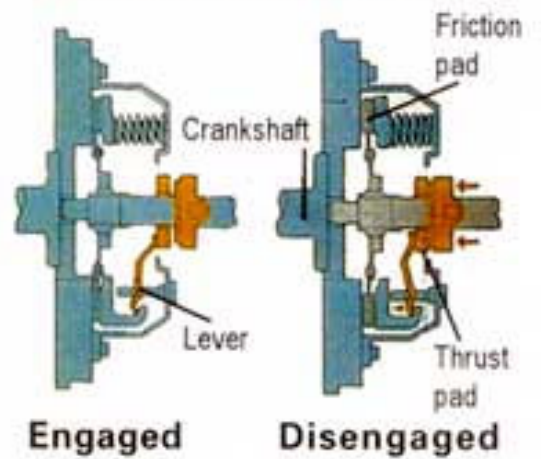
WORKING

When the clutch is engaged, the clutch plate is gripped between the flywheel and pressure plate. The friction linings are on both sides of clutch plate. Due to friction between flywheel, clutch plate and pressure plate, the clutch plate revolves with the flywheel. As clutch plate revolves the clutch shaft also revolves. Thus, engine power is transmitted to the clutch shaft.

When the clutch pedal is pressed the pressure plate moves back against the spring force and clutch plate becomes free between flywheel and pressure plate. Thus flywheel remains rotating as long as the clutch pedal is pressed, the clutch is said to be disengaged and clutch shaft speed reduces slowly and finally it stops rotating.



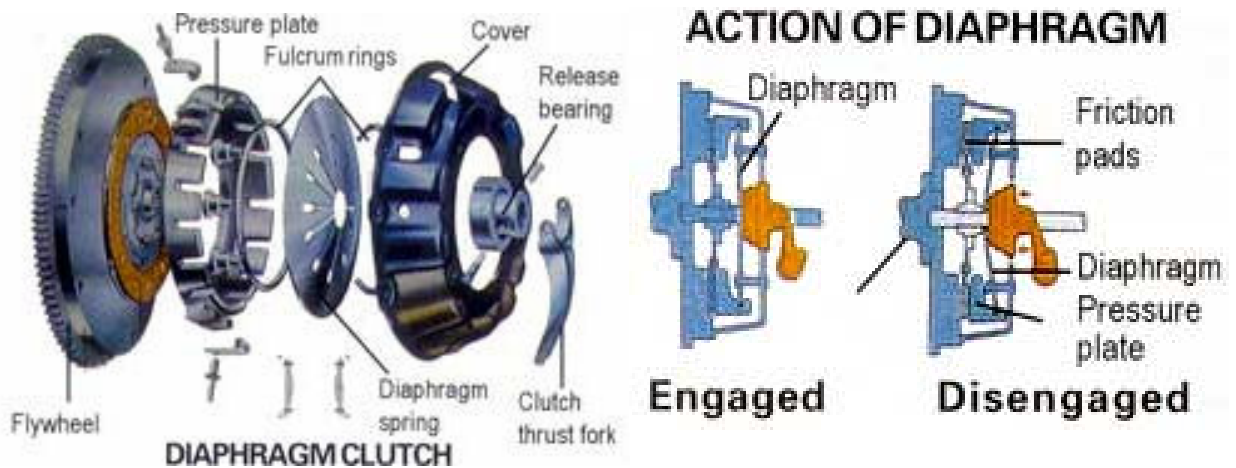
THRUST PAD OPERATION

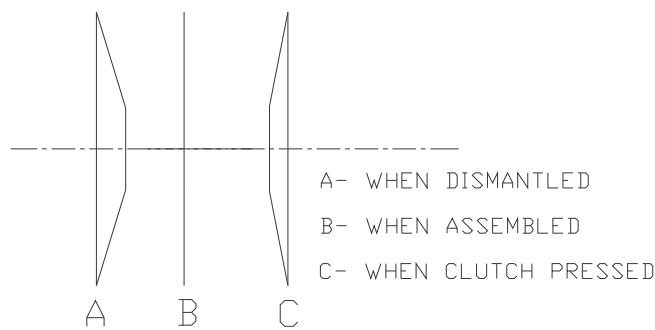
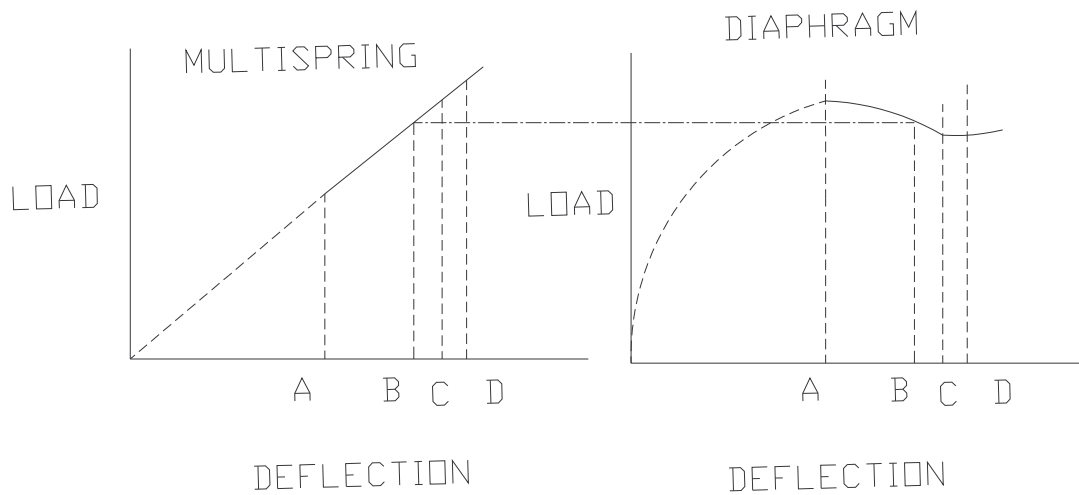


DIAPHRAGM SPRING SINGLE PLATE CLUTCH

Diaphragm spring pressure plate assemblies are widely used in most modern cars. The diaphragm spring is a single thin sheet of metal which yields when pressure is applied to it. When pressure is removed the metal springs back to its original shape. The centre portion of the diaphragm spring is slit into numerous fingers that act as release levers. During disengagement of the clutch the fingers are moved forward by the release bearing.

The spring pivots over the fulcrum ring and its outer rim moves away from the flywheel. The retracting spring pulls the pressure plate away from the clutch plate thus disengaging the clutch. When engaged the release bearing and the fingers of the diaphragm spring move towards the transmission. As the diaphragm pivots over the pivot ring its outer rim forces the pressure plate against the clutch disc so that the clutch plate is engaged to the flywheel.

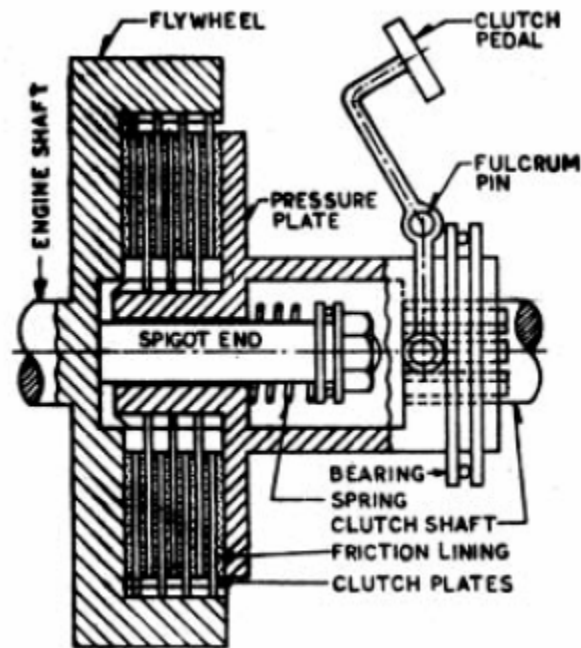




ADVANTAGES OF DIAPHRAGM SPRING CLUTCH

1. It is more compact than other designs.
2. It is easier to balance rotationally and is less subjected to unwanted effects due to centrifugal force at high rotational speeds.
3. It gives uniformly distributed pressure on pressure plate.
4. It needs no release levers.
5. Minimum effort is sufficient to disengage the clutch.
6. It provides minimum number of moving components and hence minimum internal friction is experienced.
7. This is very commonly used in cars, light Lorries and mini trucks but is not much used in heavy vehicles

MULTIPLATE CLUTCH



The multi-plate clutch is an extension of single plate type where the number of frictional and the metal plates are increased. The increase in the number of friction surfaces obviously increase capacity of the clutch to transmit torque, the size remaining fixed. Alternatively, the overall diameter of the clutch is reduced for the same torque transmission as a single plate clutch. This type of clutch is, used in some heavy transport vehicles, in epicyclic gearboxes and racing cars where high torque is to be transmitted. Besides, this finds applications in case of scooters and motorcycles, where space available is limited.

Extension of flywheel is a drum; which on its inner circumference is splined to carry a number of thin metal plates. These must consequently revolve with drum but are able to slide axially. Interleaved with these outer plates are a number of inner plates that are splined to an inner drum which is coupled rotationally to the gearbox shaft.

This drum is supported on a spigot extension of crankshaft. Between the web of inner drum and sleeve in inner drum is a strong coil spring. The inner drum is thus pressed to left being provided with a flange it squeezes the inner and outer plates together so that friction between them transmits driving torque from outer to inner drum.

The clutch is disengaged by pulling inner drum right against spring force. The plates of multi-plate clutch were at one time made alternately of steel and phosphor bronze but now are all of steel or one set may be lined with a friction material. With metal contact lubrication is essential and so clutch is made oil-tight and partly filled with oil. The oil tends to make the plates drag when clutch is disengaged and so some mean should be provided to avoid this drag.

DRY MULTIPLATE CLUTCH

Multi plate clutches are also made to work dry, without any oil. The driving plates are then lined on each side with a friction fabric. In such clutches, the driving plates are sometimes carried on a number of studs screwed into the web of flywheel in the same way as the outer plate of a Single Plate Clutch is carried. This construction is inconvenient when oil is used. Several small springs can be used instead of a single spring.

AUTOMATIC CLUTCH

Many attempts have been made to produce motor vehicles that can be controlled by the accelerator pedal and brakes only. This can be done in several ways. A centrifugal clutch which automatically disengages itself when the speed falls below and which re-engages when the speed rises above some predetermined values may be used. Alternatively, a fluid coupling, fluid torque converter may be employed.

CENTRIFUGAL CLUTCH

In this type of clutches the springs are eliminated altogether and only the centrifugal force is used to apply the required pressure for keeping the clutch in engagement position.

The advantage of the centrifugal clutch is that no separate clutch pedal is required. The clutch is operated automatically depending upon the engine speed. This means that car can be stopped in gear without stalling the engine. Similarly while starting, the driver can first select the gear, put the car into the gear and simply press the accelerator pedal. This makes the driving operation very easy.

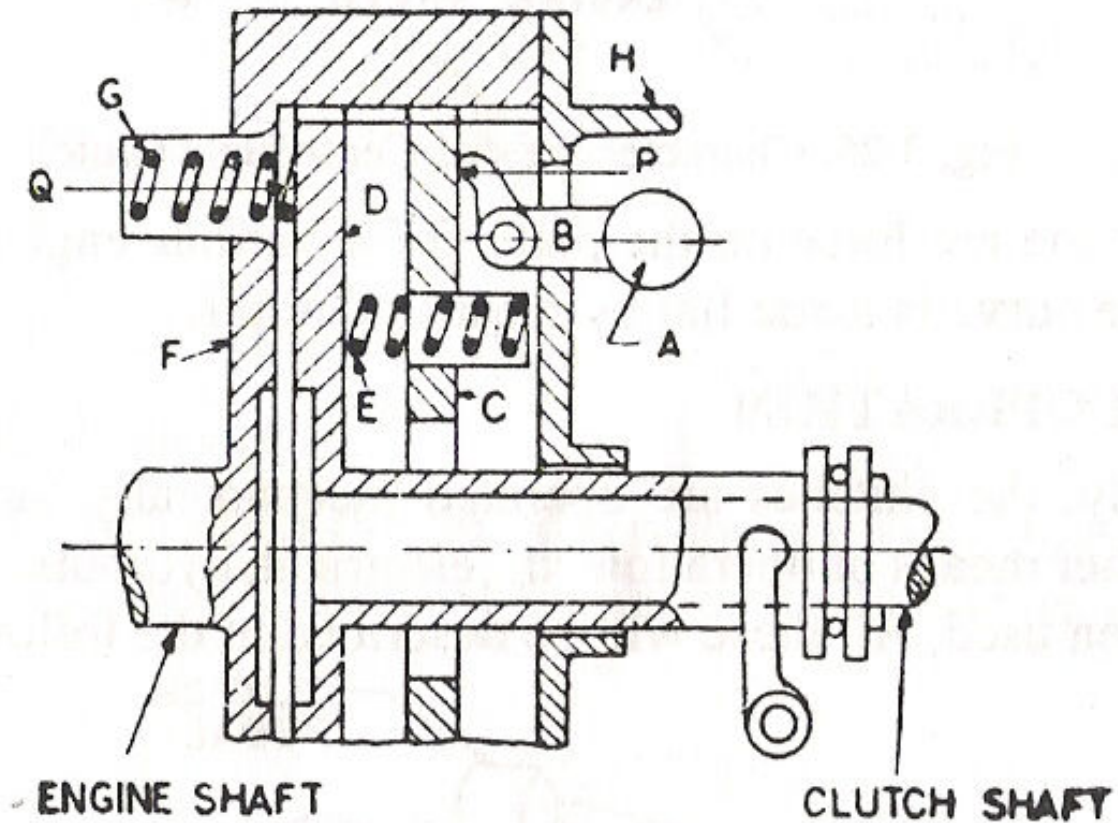
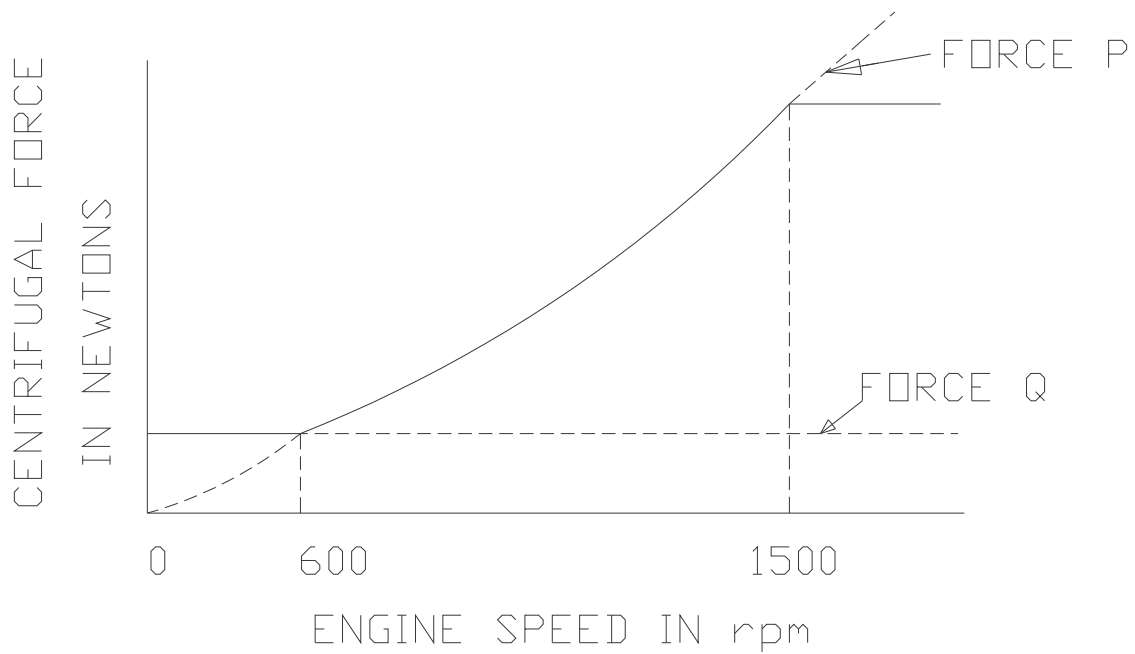


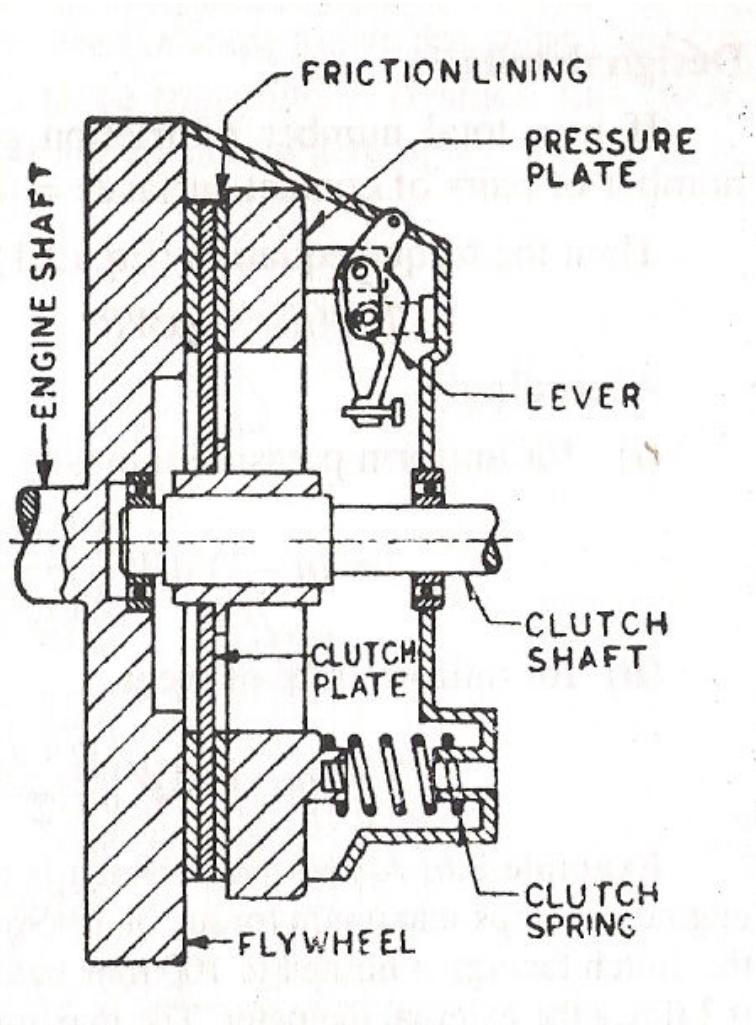
Figure shows a schematic diagram of a centrifugal clutch. As the speed increases, the weight A fly off, thereby operating the bell crank lever B that presses the plate C. This force is transmitted to the plate D by means of springs E. The plate D containing friction lining is thus pressed against the flywheel F thereby engages the clutch. Spring G serves to keep the clutch disengaged at low speed say 500 rpm. The stop H limits the amount of centrifugal force.



The operating characteristics of this type of clutch will be then as shown in figure. Force P is proportional to the centrifugal force at a particular speed, while force Q exerted by spring G is constant at all speeds. The firm line in the figure shows that net force on the plate D for various engine speeds. At the upper end the curve is made flat by means of stop H.

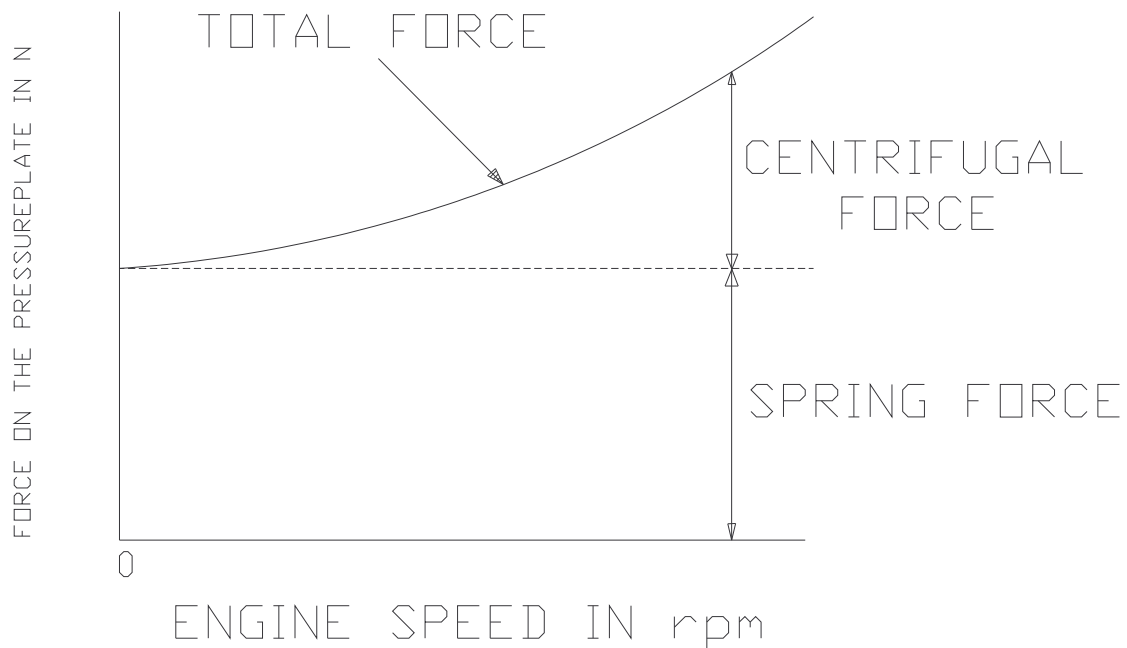
SEMI CENTRIFUGAL CLUTCH

It uses both centrifugal and spring force for keeping it in an engaged position. The springs are designed to transmit torque at normal speed, while centrifugal force assists in torque transmission at high speed. This clutch consists of three hinged and weighted levers and three clutch springs alternately arranged at equal spaces on the pressure plate. At low speeds the springs keep the clutch engaged and the weighted levers do not have any pressure on pressure plate. At high speeds when power transmission is high, weights fly off and the levers also exert pressure on plate, keeping the clutch firmly engaged.



When the speed decreases the weights do not exert any pressure on the pressure plate. Only spring pressure is exerted on pressure plate which keeps the clutch engaged. An adjusting screw is provided at the end of the lever by means of which the centrifugal force on pressure plate can be adjusted. At low speeds pressure on the spring is sufficient to transmit the torque required.

However at high speeds, the centrifugal force due to weight moves about the fulcrum thereby pressing the pressure plate. The centrifugal force is proportional to the square of speed so that adequate pressure level is attained.



Graph shows the variation of force on the pressure plate as speed increases. At low speeds spring along applies the force on the pressure plate. But when speed of the engine raises the centrifugal force also applied by the weights.

TORQUE CAPACITY OF A SINGLE PLATE CLUTCH

Assume the clutch has a single disc and it may be coil spring clutch or diaphragm spring clutch.

$$\text{Area of each friction surface of the disc } A = \pi/4 (D_o^2 - D_i^2)$$

$$\text{Normal force applied by the pressure springs} = \pi/4 (D_o^2 - D_i^2) p$$

$$\text{Net axial force applied by pressure springs} = \pi/4 (D_o^2 - D_i^2) p \eta_{\text{mech}} \quad \text{Kgf}$$

$$\text{Frictional force applied to each frictional face} = \pi/4 (D_o^2 - D_i^2) \mu p \eta_{\text{mech}} \quad \text{Kgf}$$

$$\text{Torque due to both surface} = \pi/2 (D_o^2 - D_i^2) \mu p \eta_{\text{mech}} R_m \quad \text{Kgf-cm} \text{-----1}$$

Where

μ = coefficient of friction,

p = permissible unit pressure on friction surfaces

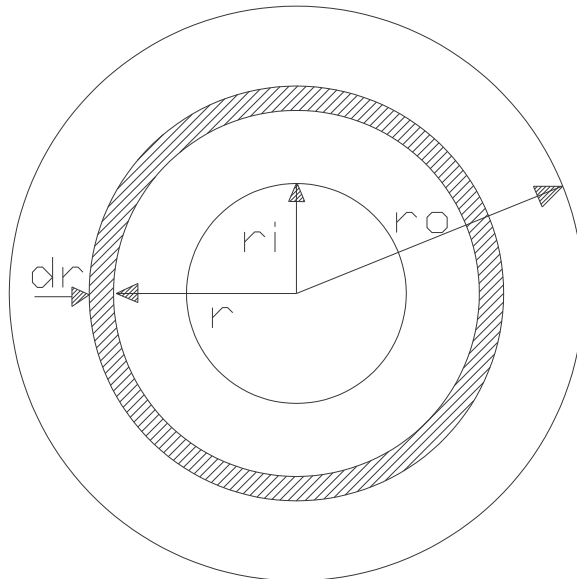
D_i = inner outer diameter of disc

D_o = outer diameter of disc

R_m = mean radius of the friction surface in cm

R_F = coefficient of reserve (R_F value = 1.4)

η_{mech} = coefficient of mechanical efficiency (η_{mech} value = 0.85)



Consider a small element at radius 'r' and of thickness 'dr'.

$$\text{Area of the element } a = 2\pi r dr \text{ cm}^2$$

$$\text{Normal force} = 2\pi r dr p \text{ Kgf}$$

$$\text{Frictional force} = 2\pi r dr p \mu \text{ Kgf}$$

$$\text{Torque transmitted by each elemental area} = 2\pi r^2 dr p \mu \text{ Kgf-cm}$$

$$\text{Torque transmitted by each friction surface} = \int 2\pi r^2 dr p \mu \text{ Kgf-cm}$$

$$= 2\pi \mu p [r^3 / 3] \text{ Kgf-cm}$$

$$= 2\pi \mu p [D_o^3 - D_i^3 / 8 \times 3] \text{ Kgf-cm}$$

Torque transmitted

$$\text{By both friction surfaces} = 2 \times 2\pi \mu p [D_o^3 - D_i^3 / 8 \times 3] \text{ Kgf-cm}$$

$$= \pi/6 \mu p [D_o^3 - D_i^3] \text{ Kgf-cm} \text{ -----2}$$

Comparing equations 1 and 2,

$$\pi/2 (D_o^2 - D_i^2) \mu p R_m = \pi/6 \mu p [D_o^3 - D_i^3]$$

$$R_m = 1 / 3 [[D_o^3 - D_i^3 / D_o^2 - D_i^2]$$

Now substitute Value of R_m in equation----- 1.

Torque transmitted

$$\text{By both friction surfaces} = \pi/2 \mu p \eta_{\text{mech}} R_m (D_o^2 - D_i^2) \text{ Kgf-cm}$$

$$= \pi/6 \mu p \eta_{\text{mech}} [D_o^3 - D_i^3] \text{ Kgf-cm}$$

$$\text{Torque capacity of clutch } T = \pi/600 \mu p \eta_{\text{mech}} [D_o^3 - D_i^3] \text{ Kgf-m}$$

$$\text{Rated torque capacity of a clutch} = T / R_F$$

$$\text{Then Rated torque capacity} = \pi/600 \mu p \eta_{\text{mech}} [D_o^3 - D_i^3] / R_F \text{ Kgf-m}$$

DESIGN ASPECTS OF CLUTCH

In passenger cars, pedal effort should be in the range of 10-12 kgf and in commercial vehicles the range should be 20-24 kgf.

Let

E1 be energy required to compress the clutch for disengagement

E2 be energy required to overcome friction of clutch components

E3 be elastic deformation of clutch disengaging components

Then $E2 + E3 = 0.6E1$

Thus total energy required = $E1 + E2 + E3 = 1.6 E1$

CLUTCH GRAB

Grab is the sudden jerky motion of the vehicle when clutch pedal is released.

CLUTCH CLATTER

Clatter is the alternative movement of the clutch disc between flywheel and pressure plate. (Or) A shaking or shuddering of the vehicle as the clutch is operated.

CLUTCH DRAG

A problem in which the clutch disc does not come to a complete stop after the clutch pedal is depressed.

CUSHION SPRINGS

The clutch clatter is avoided by providing cushion spring between friction facings. Cushioning device consists of waved cushion springs to which the facings are riveted. These springs compress slightly as the clutch engages producing a cushioning effect.

DAMPER SPRINGS

Torsional vibration damper hub is provided in steel disc to prevent the torsional vibration of the engine from being transmitted through clutch which would result in gear and driveline noises. Coil springs are generally used to provide torque build up as hub is rotated with respect to the clutch plate. The pressure of the coil spring is designed to produce a counter torque as high as the torque transmitted by the engine.

FRICTION MATERIALS

Mill board type

1. Woven type- a) laminated b) solid woven
2. Moulded or composition type

MOULDED TYPE

This type of lining is composed of asbestos fibers in their natural state mixed with a bonding material and then moulded in dies under pressure and at elevated temperature. Metallic wires are sometimes included only to increase the wear qualities and to eliminate scoring of metal faces against which the lining rubs.

CLUTCH COMPONENTS MATERIALS

Components	Material	Components	Material
Cover plate	Mild steel	Damper spring	Spring steel
Diaphragm spring	Spring steel	Clutch hub	Mild steel
Coil spring	Spring steel	Bolts to fasten flywheel and cover plate	Steel
Pressure plate	Cast iron	Rivets on cushion spring	Brass
Clutch disc	Mild steel	Retainer spring	Spring steel
Friction facing	Asbestos base	Damper spring retainer plate	Mild steel
Rivets on facing	Aluminum brass	Cushion spring	Mild steel

PROBLEMS ON CLUTCH DESIGN

Torque transmitting capacity of clutch

According to Uniform pressure theory

$$\text{Torque transmitted by clutch} = T = \frac{2}{3} \times \mu \times W \times n \times \frac{(r_2^3 - r_1^3)}{(r_2^2 - r_1^2)} \quad \text{N-m}$$

According to Uniform wear theory

$$\text{Torque transmitted by clutch} = T = \mu \times W \times n \times \frac{(r_2 + r_1)}{2} \quad \text{N-m}$$

$$C = pr = \text{Constant} = \frac{W}{2\pi(r_2 - r_1)}$$

Where,

W = Axial load exerted by actuating spring in N

μ = coefficient of friction between the contact surfaces.

P = normal pressure in Pa

r_1 and r_2 = internal and external radii of contact surface in m

n = No. of contact surfaces

$$n = n_1 + n_2 - 1$$

n_1 and n_2 are no. of driving plates and no. of driven plates respectively

1. A motorcar engine develops 5.9 b.k W at 2100rpm. Find the suitable size of clutch plate having friction linings riveted on both sides, to transmit the power, under the following conditions:

- (i) Intensity of pressure on the surface not to exceed 6.87×10^4 pa.
- (ii) Slip torque and losses due to wear etc. is 35% of engine torque.
- (iii) Coefficient of friction on contact surface is 0.3.
- (iv) Inside diameter of the friction plate is 0.55 times the outside diameter.

$$T = \frac{P_w \times 60000}{2\pi N} = \frac{60000 \times 5.9}{2\pi \times 2100} = 26.84 \text{ N-m}$$

Taking account of the losses, the total torque is

$$T = 26.84 \times 1.35 = 36.23 \text{ N-m}$$

We, have

$$T = 2\pi \mu C (r_2^2 - r_1^2)$$

$$36.23 = 3.14 \times 0.3 \times 6.87 \times 10^4 \times r_1 [(r_1/0.55)^2 - r_1^2] \times 2$$

$$= 3.14 \times 4.122 \times 10^4 [(1/0.303) - 1] r_1^3$$

$$r_1^3 = \frac{36.23 \times 0.303}{3.14 \times 4.122 \times 10^4 \times 0.697} = \frac{1.22}{10^4} \text{ m}^3$$

$$r_1 = 49.5 \text{ mm} \quad r_2 = 90 \text{ mm}$$

Hence,

$$\text{Inside diameter} = 99 \text{ mm}$$

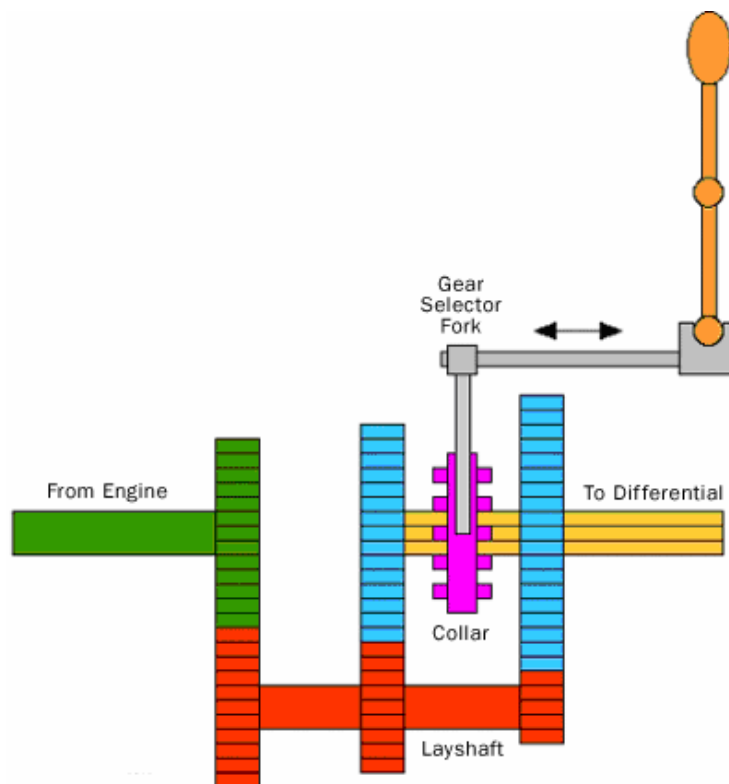
$$\text{Outside diameter} = 180 \text{ mm}$$

NECESSITY OF GEAR BOX IN AN AUTOMOBILE

The gear box is necessary in the transmission system to maintain engine speed at the most economical value under all conditions of vehicle movement. An ideal gear box would provide an infinite range of gear ratios, so that the engine speed should be kept at or near that the maximum power is developed what ever the speed of the vehicle.

FUNCTION OF A GEAR BOX

1. Torque ratio between the engine and wheels to be varied for rapid acceleration and for climbing gradients.
2. It provides means of reversal of vehicle motion.
3. Transmission can be disconnected from engine by neutral position of gear box.



TYPES OF GEAR BOX

PROGRESSIVE TYPE GEAR BOX

Usually this gear boxes are used in motor cycles. In this gear boxes the gears pass through the intervening speeds while shifting from one speed to another. There is a neutral position between two positions. These gear boxes are a combination of sliding and constant mesh gear boxes. The various gear speeds are obtained by sliding the dog clutch or gear to the required position.

EPICYCLIC (OR) PLANETARY TYPE GEAR BOX

The epicyclic or planetary type transmission uses no sliding dogs or gears to engage but different gear speeds are obtained by merely tightening brake-bands on the gear drums, which simplify gear changing. A planetary gear set consists of ring gear or annular wheel, sun gear and planet gears with carrier. In order to obtain different speeds any one of these three units can be held from rotation by means of brake bands.

SELECTIVE TYPE GEAR BOX

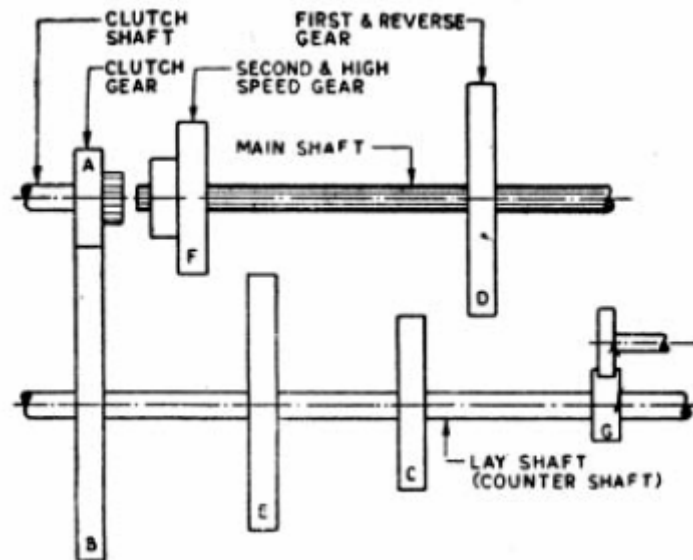
It is the transmission in which any speed may be selected from the neutral position. In this type of transmission neutral position has to be obtained before selecting any forward or reverse gear. Some selective type gear boxes are,

1. Constant mesh gear box with positive dog clutch.
2. Constant mesh gear box with synchromesh device.
3. Sliding mesh gear box.

SLIDING MESH GEAR BOX

It is the simplest and oldest type of gear box.

1. The clutch gear is rigidly fixed to the clutch shaft.
2. The clutch gear always remains connected to the drive gear of countershaft.
3. The other lay shaft gears are also rigidly fixed with it.
4. Two gears are mounted on the main shaft and can be sliding by shifter yoke when shifter is operated.
5. One gear is second speed gear and the other is the first and reverse speed gears. All gears used are spur gears.
6. A reverse idler gear is mounted on another shaft and always remains connected to reverse gear of counter shaft.



FIRST GEAR

By operating gearshift lever, the larger gear on main shaft is made to slide and mesh with first gear of countershaft. The main shaft turns in the same direction as clutch shaft in the ratio of 3:1.

SECOND GEAR

By operating gear shaft lever, the smaller gear on the main shaft is made to slide and mesh with second gear of counter shaft. A gear reduction of approximately 2:1 is obtained.

TOP GEAR

By operating gearshift lever, the combined second speed gear and top speed gear is forced axially against clutch shaft gear. External teeth on clutch gear mesh with internal teeth on top gear and the gear ratio is 1:1.

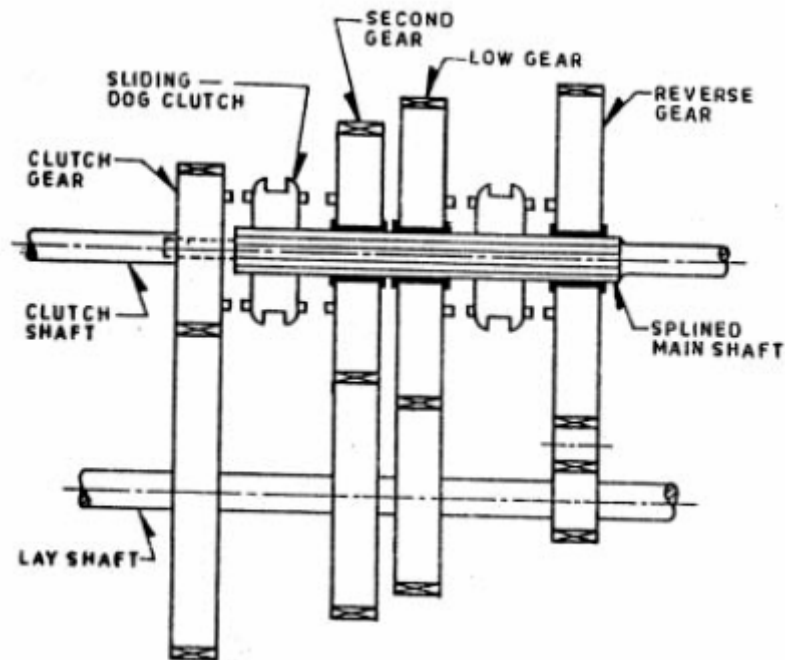
REVERSE GEAR

By operating gearshift lever, the larger gear of main shaft is meshed with reverse idler gear. The reverse idler gear is always on the mesh with counter shaft reverse gear. Interposing the idler gear, between reverse and main shaft gear, the main shaft turns in a direction opposite to clutch shaft.

NEUTRAL GEAR

When engine is running and the clutch is engaged, clutch shaft gear drives the drive gear of the lay shaft and thus lay shaft also rotates. But the main shaft remains stationary as no gears in main shaft are engaged with lay shaft gears.

CONSTANT MESH GEARBOX



In this type of gearbox, all the gears of the main shaft are in constant mesh with corresponding gears of the countershaft. The gears on the main shaft which are bushed are free to rotate. The dog clutches are provided on main shaft. The gears on the lay shaft are, however, fixed.

When the left Dog clutch is slid to the left by means of the selector mechanism, its teeth are engaged with those on the clutch gear and we get the direct gear. The same dog clutch, however, when slid to right makes contact with the second gear and second gear is obtained.

Similarly movement of the right dog clutch to the left results in low gear and towards right in reverse gear. Usually the helical gears are used in constant mesh gearbox for smooth and noiseless operation.

SYNCHROMESH GEARBOX

This type of gearbox is similar to the constant mesh type gearbox. Instead of using dog clutches here synchronizers are used. The modern cars use helical gears and synchronmesh devices in gearboxes, that synchronize the rotation of gears that are about to be meshed.

SYNCHRONIZERS

This type of gearbox is similar to the constant mesh type in that all the gears on the main shaft are in constant mesh with the corresponding gears on the lay shaft. The gears on the lay shaft are fixed to it while those on the main shaft are free to rotate on the same. Its working is also similar to the constant mesh type, but in the former there is one definite improvement over the latter. This is the provision of synchronmesh device which avoids the necessity of double-declutching. The parts that ultimately are to be engaged are first brought into frictional contact, which equalizes their speed, after which these may be engaged smoothly.

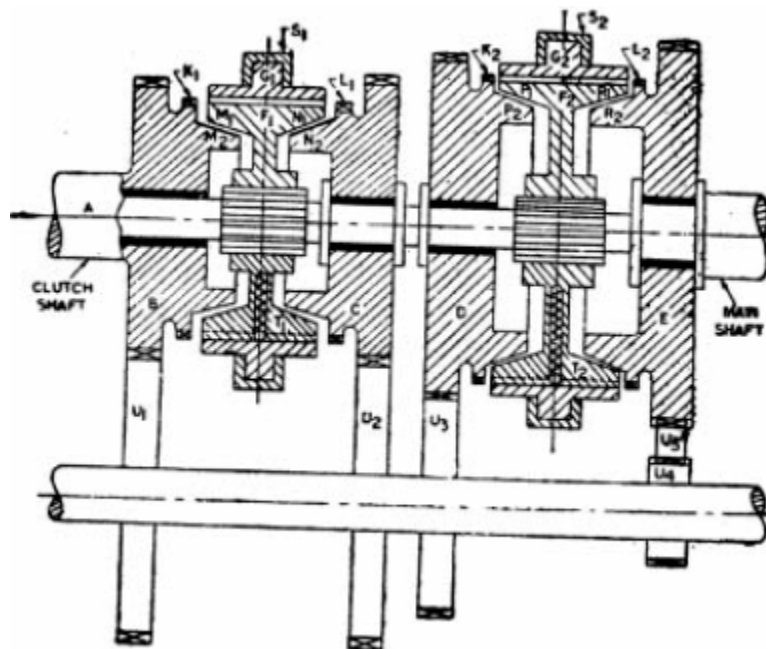
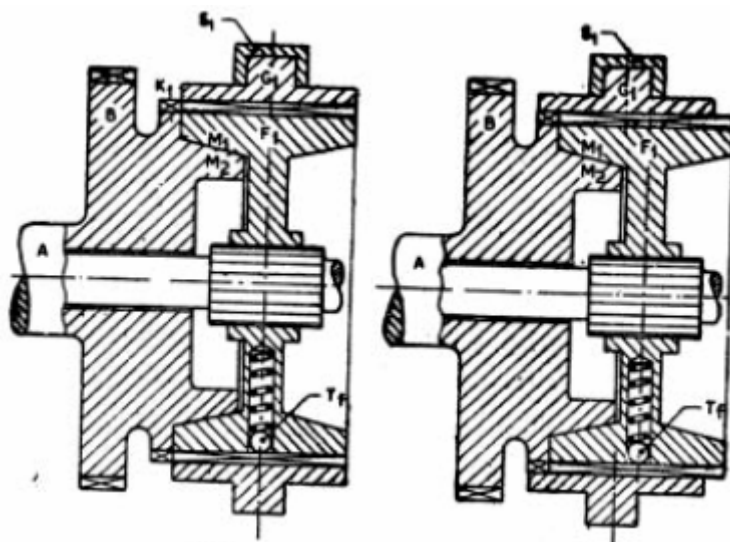


Figure shows the construction and working of a synchromesh gearbox. In most of the cars, however, the synchromesh devices are not fitted to all the gears as is shown in this figure. They are fitted only on the high gears and on the low and reverse gears ordinary dog clutches are only provided. This is done to reduce the cost.

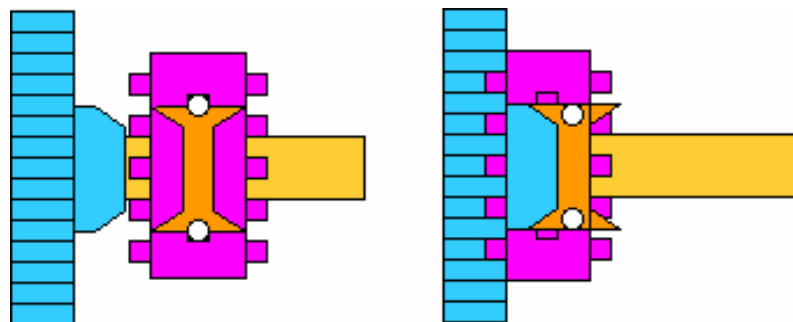
In figure A is the engine is the engine shaft, Gears B, C, D, E are free on the main shaft and are always in mesh with corresponding gears on the lay shaft. Thus all the gears on main shaft as well as on lay shaft continue to rotate so long as shaft A is rotating. Members F1 and F2 are free to slide on splines on the main shaft. G1 and G2 are ring shaped members having internal teeth fit onto the external teeth members F1 and F2 respectively. K1 and K2 are dogteeth on B and D respectively and these also fit onto the teeth of G1 and G2. S1 and S2 are the forks. T1 and T2 are the balls supported by spring. These tend to prevent the sliding of members G1 (G2) on F1 (F2). However when the force applied on G1 (G2) slides over F1 (F2). These are usually six of these balls symmetrically placed circumferentially in one synchromesh device. M1, M2, N1, N2, P1, P2, R1, R2 are the frictional surfaces.



To understand the working of this gearbox, consider figure which shows in steps how the gears are engaged. For direct gear, member G1 and hence member F1 (through spring-loaded balls) is slid towards left till cones M1 and M2 rub and friction makes their speed equal. Further pushing the member G1 to left causes it to overdrive the balls and get engaged with dogs K1. Now the drive to the main shaft is direct from B via F1 and the splines. However, if member G1 is pushed too quickly so that there is not sufficient time for synchronization of speeds, a clash may result. Likewise defect will arise in case springs supporting the balls T1 have become weak.

Similarly for second gear the members F1 and G1 are slid to the right so that finally the internal teeth on G1 are engaged with L1. Then the drive to main shaft will be from B via U1, U2, C, F1 and splines. For first gear, G2 and F2 are moved towards left. The drive will be from B via U1, U2, D, F2 and splines to the main shaft. For reverse gear, G2 and F2 are slid towards right. In this case the drive will be from B via U1, U2, U5, E, F2 and splines to the main shaft.

A synchro's purpose is to allow the collar and the gear to make frictional contact before the dog teeth make contact. This lets the collar and the gear synchronize their speeds before the teeth need to engage, like this: The cone on the blue gear fits into the cone-shaped area in the collar, and friction between the cone and the collar synchronize the collar and the gear. The outer portion of the collar then slides so that the dog teeth can engage the gear.



DETERMINATION OF GEAR RATIOS

From one point of view, the ideal type of transmission is the so called indefinitely variable gear, in which torque ratio can be varied continuously within wide limits, because it permits of operating the engine at all times under optimum conditions with respect to both fuel consumption and wear and tear. However in an ordinary transmission only a small number of gear ratios can be provided to cut down expenses and weight. The most desirable number of gear changes depends in part on the use to which the transmission is to be put, and each field of application must be considered separately.

The most desirable number of gear ratios depends on the difference between highest and lowest gear ratios. The larger the ratio between corresponding gear ratios, the more difficult it is to make the change from one gear to another, it must be done by either shifting gears into mesh laterally or securing a gear to its shaft by means of a jaw clutch. A ratio of 2:1 is about the limit and is frequently used in trucks although a ratio of 1.5:1 is considered better from a standpoint of ease shifting.

Formerly a ratio of 1:8 was used in passenger cars and high speed was a direct drive with a ratio of 1:1, the intermediate speed a reduction gear with a 1:8 ratio and the low speed a reduction gear with a ratio of $(1.8 \times 1.8): 1$ or 3.24:1. When gear ratios are arranged in such an order they form a geometrical series which offers certain advantages from the standpoint of operation. In most automotive transmission the ratios of the different gears come fairly close to forming a geometric series. A slight deviation from the series is made at high speed end. There are certain limitations on the number of gear teeth which can be provided, and it is therefore not always possible to obtain an exact geometrical series of ratios, even if that should be desired.

UNIT III

AUTOMATIC TRANSMISSION

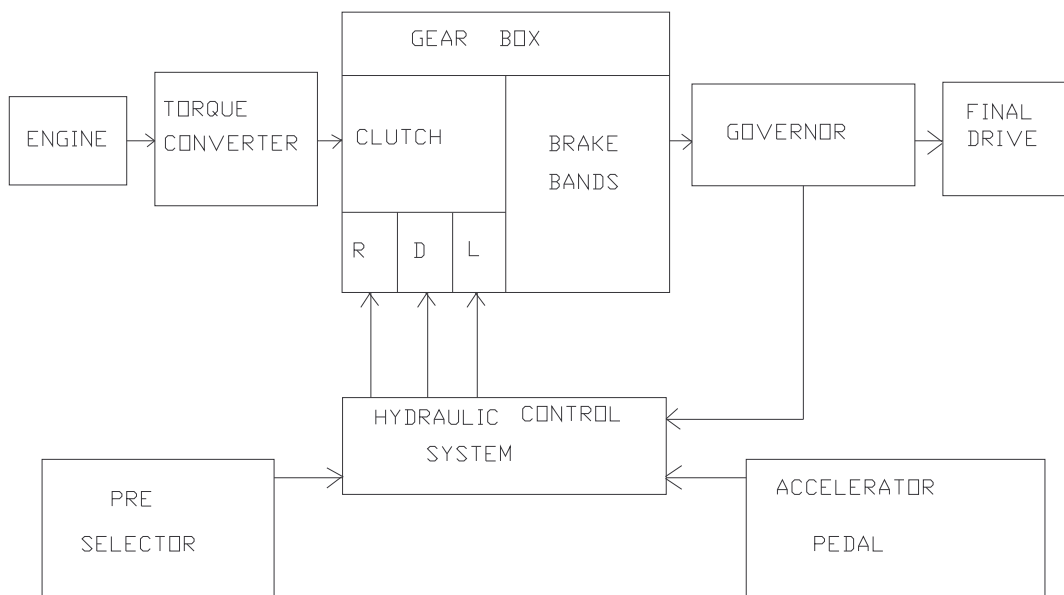
AUTOMATIC TRANSMISSION

It is the transmission which automatically provides varying gear ratios to suit operating conditions. In this case gear changes are effected automatically without manual assistant.

MAIN COMPONENTS OF AUTOMATIC TRANSMISSION SYSTEM

1. Torque converter.
2. Gearbox of planetary type with friction brake bands and multiple disc clutch operated by hydraulic system.
3. Hydraulic Control System. This system has a source of hydraulic pressure servo units and control valves.

BLOCK DIAGRAM OF AUTOMATIC TRANSMISSION



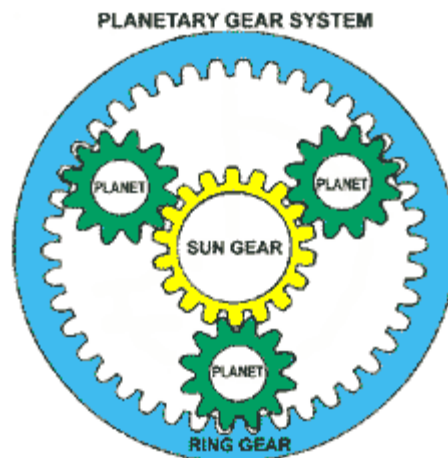
ADVANTAGES OF AUTOMATIC TRANSMISSION SYSTEM

1. Smooth operation.
2. Ease of control, i.e. it relieves the driver from fatigue due to the elimination of clutch and gear controls.
3. Numerous numbers of gear ratios are available.
4. Quick change of gear ratios effected automatically.
5. Minimum interruption of power during gear shifts.

DISADVANTAGES OF AUTOMATIC TRANSMISSION SYSTEM

1. High cost.
2. Complicated design.
3. Possibility for oil leakage.

EPICYCLIC GEAR TRAIN



Sun gear (S) = 30 teeth (A)

Annulus (or) Ring gear (A) = 75 teeth (B)

FOR HIGH REDUCTION	SUN = DRIVER CARRIER = DRIVEN ANNULUS = FIXED
GEAR RATIO	$A / (A+B) = 30 / (30 + 75)$ GEAR RATIO = 3.5 : 1

FOR LOW REDUCTION	SUN = DRIVEN CARRIER = DRIVER ANNULUS = FIXED
GEAR RATIO	$(A+B) / A = (30 + 75) / 30$ GEAR RATIO = 1.4 : 1

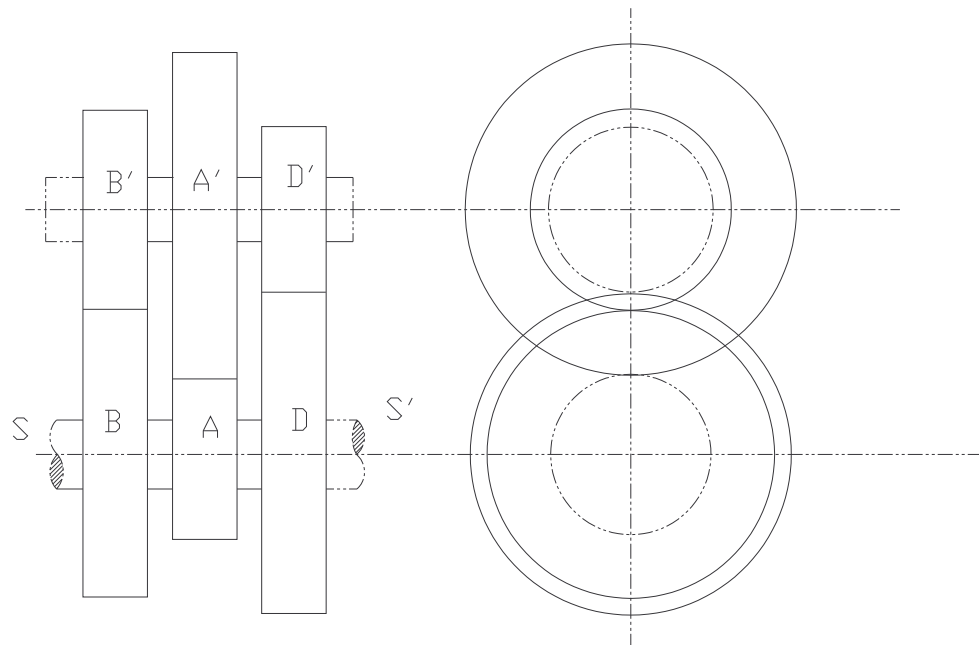
FOR DIRECT DRIVE	SUN = DRIVER CARRIER AND ANNULUS ARE INTER LOCKED WITH EACH OTHER
GEAR RATIO	GEAR RATIO = 1 : 1

FOR OVER DRIVE	SUN = FIXED CARRIER = DRIVER ANNULUS = DRIVEN
GEAR RATIO	$(A+B) / A = (30 + 75) / 30$ GEAR RATIO = 1 : 1.4

FOR REVERSE DRIVE	SUN = DRIVER CARRIER = FIXED ANNULUS = DRIVEN
GEAR RATIO	$A / B = 30 / 75$ GEAR RATIO = 2.5 : 1

FOR NEUTRAL	SUN = DRIVER CARRIER = FREE ANNULUS = DRIVEN
GEAR RATIO	GEAR RATIO = --

PRINCIPLE OF THE ALL-SPUR TYPE



One form of the “all-spur” type of planetary transmission is illustrated in diagram. It comprises three adjacent independent gears A, B, D, Concentric with the driving and driven shafts S and S', which are telescoped, and three corresponding pinions A', B', D', forming a rigid planetary unit. There are usually three such units, spaced uniformly around the gears mounted on the driving and driven shafts. Gear A is the driving and gear D the driven member.

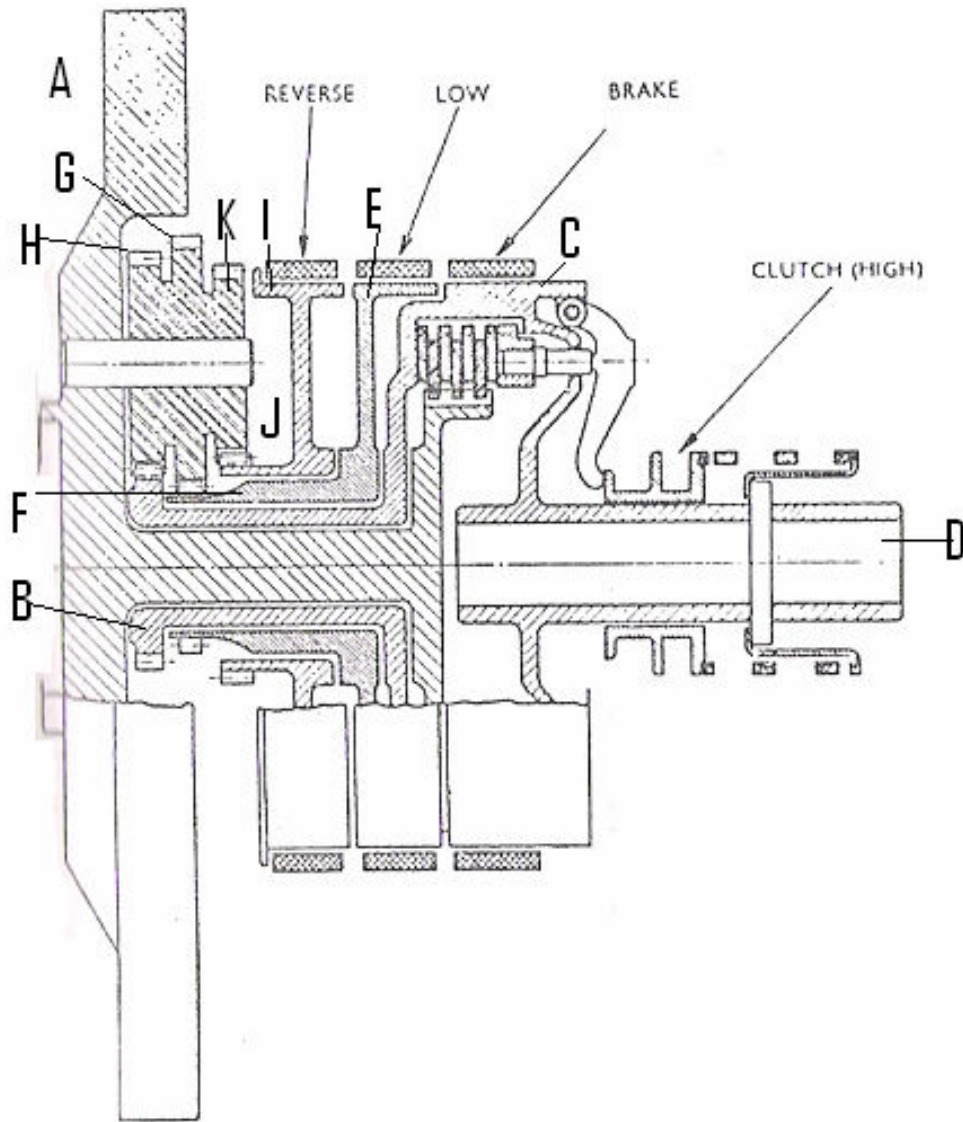
Low forward speed is obtained in a simple manner by holding the planet carrier from rotation, by means of a brake drum provided on it for the purpose. Power then enters at gear A and is transmitted through the pairs of back gears A', D', to gear D, hence no planetary motion is involved in the low speed forward drive. The low-speed reduction ratio is $a'd / ad'$. For high forward speed the driving shafts is coupled to the driven shaft by means of a friction clutch incorporated in the assembly, which gives a direct drive.

For the reverse motion gear B, which is free on the driving shaft, is held from rotation. Assume that the planet carrier rotates left-handedly, causing pinion B' to roll on B. For one left-handed revolution of the planet carrier, the gear cluster A', B', D', makes b/b' left-hand revolutions around its own axis. The reduction ratio then is $d(b-a) / a(d-b)$. It will be seen at once that this reduction is positive if $b > a$ and $d > b$.

FORD T-MODEL

An example of the “all-spur” type of planetary transmission is the ford model T, with which millions of cars have been equipped. A sectional view of this transmission is shown in figure. Here, instead of power being applied through one of the sun pinions, it is applied to the planet carrier.

The flywheel rim A serves as planet carrier and driving member, having lateral studs secured into it which carry triple planetary pinions. Gear B is the driven member, being keyed to the hub of clutch drum C, which in turn is secured to driven shaft D. By applying a brake band to drum E, gear F is held stationary, pinion G rolls on it, and the smaller pinion H causes gear B to turn slowly in the same direction as pinion carrier A. By applying a brake band to drum I, gear J is held stationary; pinion H turns gear B slowly in the reverse direction. For the high gear or direct drive, the friction clutch locks clutch drum C to the engine crankshaft, and the gear rotates as a unit.



The three pedals control the transmission and brakes. When the left pedal is pushed down all the way, the car is in low gear. To remain in low gear, you must continue pushing on the left pedal. (It's been said that you push a Model T up a hill in low gear with your left foot!). If the left pedal is pushed to the halfway position, the car is in neutral. When the left pedal is completely released (not depressed at all), the car is in high gear. If the car is in neutral (either by depressing the left pedal halfway or by moving the lever to the left of the pedals to an upright position) the middle pedal can be pushed to engage reverse gear. The right pedal is a brake that acts on the transmission when pushed. Operating the brake and transmission sounds more difficult than it really is. After some practice, most drivers don't give it a second thought.

Interestingly, the Model T has a planetary transmission that's the forerunner of the automatic transmission. It's very similar to an automatic transmission except you use foot pedal pressure to operate the bands rather than hydraulic pressure and it doesn't have a torque converter. The lever at the right and under the steering wheel is the hand throttle. It controls the speed, much like the control found on the tractor or riding mower.

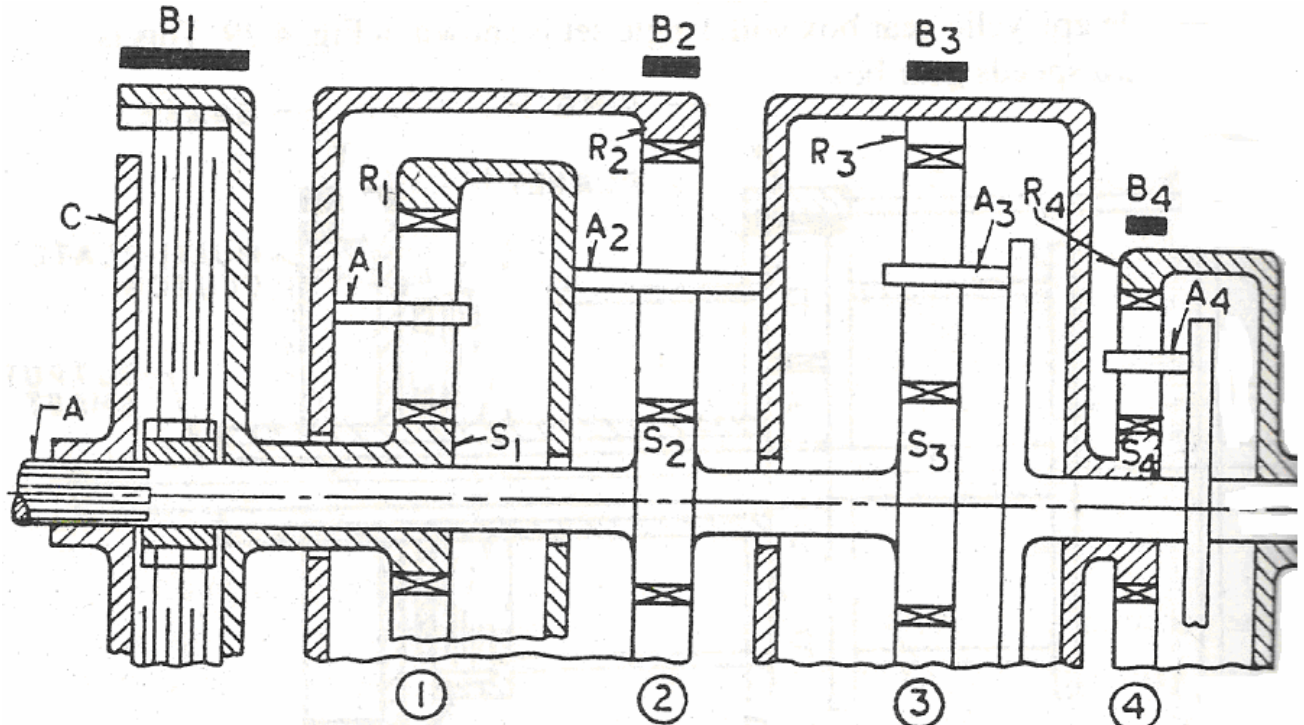
Model T's were made from 1908 until 1927. Over 15 million Model T's were produced. Far more were produced than any other car until the Volkswagen Beetle overtook its production in the 1970's, at a time when two-car families became the norm. The impact of the Model T's in its day is hard for us to imagine, but in the early 1920s, half of all cars on the road worldwide were Model T's. A Model T in good mechanical condition will cruise all day at 30 to 35 mph. Most can go 45 to 50 mph but the engine is working pretty hard at these speeds, so most drivers go this fast only briefly. These figures also depend upon the body style and weight of the car, with roadsters and sedans and depot hacks the heaviest.

WILSON GEAR BOX

CONSTRUCTION

The gearbox comprises of three subassemblies, the running gear, the brake harness and the control mechanism housed in an oil tight container. This consists of a four epicyclic trains of gear inter connected, so that different ratios and a reverse can be obtained. The direct drive is achieved by engaging the clutch.

One train of epicyclic gearing is used for all the various ratios, its sun S_1 being secured to a shaft D coupled permanently to the engine and its arm R_1 to the shaft E which is coupled permanently to the driving road wheels and the various ratios are obtained by driving the annulus A_1 at different speeds in relation to the engine speed.



OPERATION

FIRST GEAR

First gear is obtained by applying a brake to the first gear train annulus A1. So that it is held stationary. The engine will then, be turning the sun gear S1. So that the planet gears will be rolling round inside the annulus A1 carrying their arm R1 round with them. As this arm R1 fixed to the output shaft its motion is imported to it.

First Gear ratio:

Engine speed = 1000 rpm = sun S1 speed

$$\begin{aligned}\text{Arm R1 speed} = \text{wheel speed} &= 1000 \times s / (a + s) \\ &= 1000 \times 25 / (100 + 25) \\ &= 200\end{aligned}$$

Where $s = 25 =$ sun wheel teeth

$a = 100 =$ annulus teeth

Gear ratio:

$$1000 : 200$$

$$5 : 1$$

SECOND GEAR

Second gear is obtained by holding the second gear train annulus A2 stationary by its brake. The main sun gear S2 still turned by the engine cause the planet gear to revolve and their arm R2. But this arm R2 is connected to the first gear train annulus A1 which therefore turns, speeding up the rotation of the planet gear and arm R1 and it turning the output shaft faster than was the case in first gear, i.e. less reduction.

Second Gear ratio:

Engine speed = 1000 rpm = sun gear S1

Annulus speed = 100 rpm

Gear ratio: 1000 : 280

$$3.57 : 1$$

THIRD GEAR

Third gear is obtained by holding the third gear sun wheel S3 by brake drum holding. Which is interconnected further the annulus A3 is an integral part of the second gear planet arm R2 which is in turn connected to the first gear annulus A1. The third gear arm R3 is connected to the second gear annulus A2 so driving it in same direction as the engine. i.e. increasing its speed so the drive is taken back through the second gear planets and arm R2 and the first gear train annulus A1 both of which are speeded up. The result is to speed up the first gear train arm R1 which are connected to output shaft. In other words by interconnecting the second and third arms, an increase of speed is obtained at the first gear train annulus, which increases the speed of the arm R1.

Third Gear ratio:

Sun wheel S1 speed = 1000 rpm

Arm R1 speed = 360 rpm

Gear ratio: 1000 : 360

2.78 : 1

TOP GEAR

In the top gear all the gear trains are locked together and revolve as a solid block driving the output shaft at engine speed. This is brought about by the engagement of this driving member to the clutch which is the drum and sun gear S3 gear train. So that locking the third gear sun to the driving shaft.

Those are all the sun gear will be revolving at the same speed. Since the first and second gear train sun wheel are fixed to the shaft (output) and their will not be any individual action of the various gear train. All the brake bands being loose their annulus.

Gear ratio:

1000 : 1000

1 : 1

REVERSE GEAR

The first gear annulus A1 is connected to the sun gear S4 of the reverse gear train and hence drive output shaft opposite to engine rotation. When the brake is applied to the reverse gear annulus A4 the reverse gear planet wheels turned by the reverse sun gear connected to the first gear annulus and therefore turning opposite to the engine speed output shaft. As the arm A4 connected to the output shaft the direction of rotation of the propeller shaft reversed.

Reverse Gear ratio:

Sun wheel S1 = 25

Annulus A1 = 100

Sun wheel S4 = 40

Annulus A4 = 80

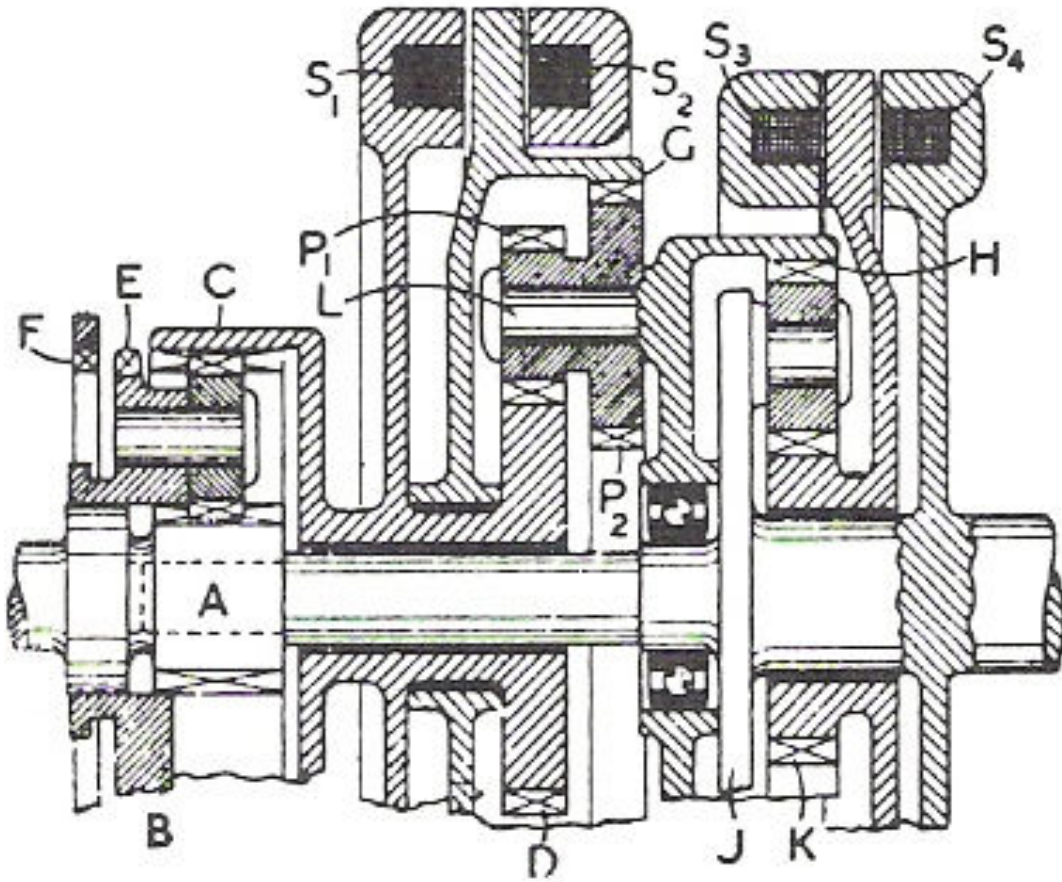
Gear ratio: 7 :1

COTAL EPICYCLIC GEAR BOX

CONSTRUCTION

The wheel A is integral with the engine shaft and meshes with pinions carried by a spider A which is free to slide along the outside of the engine shaft. When the spider B is slid to the left, its teeth E mesh with teeth F of an annulus which is fixed to the gearbox casing. The pins of the spider then form fixed bearings for the pinions, and so the annulus C with which the latter mesh is driven in the opposite direction to the wheel A. This gives the reverse drives. When the spider B is slid to the right its teeth E engage the teeth of the annulus C and then the wheel A, pinions, spider B and annulus C revolve 'solid'. This gives the forward drives.

The four forward ratios are obtained by means of two epicyclic trains arranged in tandem. One consists of the sun D (fixed to the annulus C), the compound planets P1, P2 (carried on the pins L of the arm which is integral with the annulus H of the second train) and the annulus G which can be held fixed.



When this is done the annulus H is driven in the same direction as the sun D but at a lower speed. The second train consists of the annulus H, the sun K and the arm J which is fixed on the output shaft. The sun K can be held at rest so that the train gives a reduction between the annulus H and the arm J and it can also be locked to the output shaft so that the train must revolve solid. The annulus G can also be locked to the sun D so that the first train must revolve solid.

WORKING

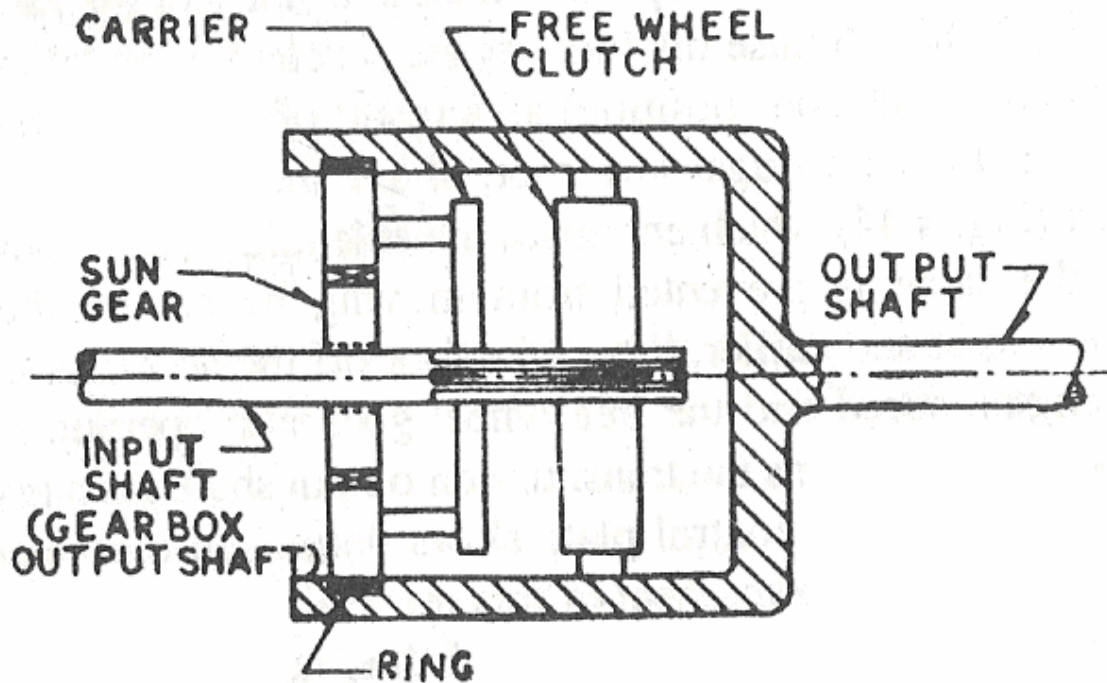
The fixing and locking of the members is done by electromagnets whose windings S1 S2 S3 S4 are energized as may be required. For first gear S2 and S3 are energized and both epicyclic trains provide a reduction since both annulus G and sun K are fixed. For second gear, S2 and S4 are energized and the second train revolves solid, the only reduction being in the first train. For third gear, S1 and S3 are energized, the first train is locked solid and the only reduction occurs in the second train. For fourth gear S1 and S4 are energized and both trains revolves solid so that a direct drive is obtained.

The windings S1 and S4 are carried by parts that sometimes rotate and so these windings are connected to slip rings on which brushes bear. The current for energizing the windings is supplied by the battery or generator of the car and is between two and three amperes.

The control is extremely simple consisting merely of a switch which connects the appropriate winding to the battery. This switch is usually mounted at the center of the steering wheel.

AUTOMATIC OVER DRIVE

PRINCIPLE OF THE OVER DRIVE



The principle of the overdrive is illustrated in figure, which shows a simple epicyclic gear train. In this the gear box output shaft is connected to the planet carrier ring while the annulus or ring gear is attached to the outer race of the free-wheel unit and from this to the output shaft; the latter drives the propeller shaft through the front universal coupling. The sun wheel is the reaction member of the gear train and when it is locked to the gear box output -or overdrive input-shaft the planet carrier which is driven by the gear box shaft rotates about the sun wheel carrying with it the pinion wheels which in turn, drive the annulus gear and with it the output shaft to the propeller shaft. The annulus rotates more slowly than the planet carrier thus giving a lower gear ratio than for direct drive. The position in which the sun wheel is locked to the gear box shaft, which has just been described, gives the overdrive requirements.

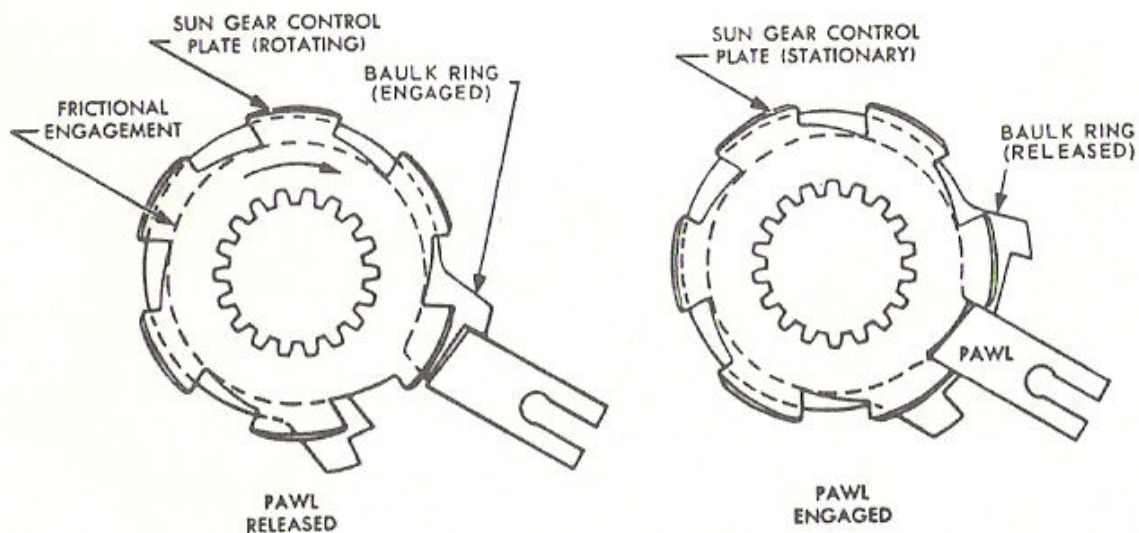
OVER DRIVE OPERATIONS

DIRECT DRIVE CONDITIONS

To obtain direct drive the overdrive gear must be locked in some manner so as to rotate as one solid unit. This is done by connecting the sun gear solidly with the planet carrier. The drive from the gear box shaft is then taken through the locked epicyclic gear train and thence through the free-wheel to the overdrive output shaft. Therefore, when in direct drive the propeller shaft side can rotate faster than the gear box shaft. Since, however, the direct drive is only used at speeds below about 28 to 30 m.p.h, this advantage is only realized at these at these lower speeds.

COMPLETE OVERDRIVE

Having explained the principle of this overdrive, the purposes of the various components should be better understood by reference to the illustration of a complete overdrive, as used on cars fitted with three-speed gear boxes. The annulus or ring gear and the outer race of the free-wheel are splined to the overdrive main shaft.



The overdrive unit includes a device to lock the sun gear and hold of stationary. To do this a centrifugal governor, driven by the overdrive output shaft is used to close the contacts of an electrical circuit which contains the windings of the solenoid. When the road speed of the car increases to about 28 to 30 m.p.h., the governor closes the contacts and thus energizes the solenoid which forces its plunger outwards and therefore pushes the pawl member towards a notched ring around the overdrive input shaft; this pawl is shown as the gear plate. Since it would not be advisable for the pawl to engage with one of the notches in the gear plate during the rotation of the plate the pawl is not allowed, initially, to enter a notch, being prevented from doing so by a baulk ring which is a friction fit to the gear plate.

To obtain a smooth engagement of the pawl it is necessary to decelerate the overdrive input shaft, momentarily, by releasing the accelerator pedal, so that the engine begins to slow down. Since the momentum of the car will cause the ring gear still to rotate, the ring gear will rotate the pinions, driving the sun gear and baulk ring in a reverse direction. This movement of the baulk ring allows the pawl to move from the step and engage one of the gear plate notches, thus bringing the overdrive into operation. The engagement, after the accelerator pedal release, is so quick that the gear plate rotates only about one-third of a turn before full engagement of the pawl with a notch. This method of engagement depends upon synchronizing the pawl movement with the momentary stopping of the gear plate.

To release the overdrive the accelerator should be released so that the road speed falls by 2 to 4 m.p.h below the cut-in speed; so that the governor's contacts open and the solenoid actuated pawl is released; the direct drive is then re-engaged, automatically.

LOCKING THE OVERDRIVE

As mentioned earlier, in order to obtain normal top gear as given by the gear box, it is necessary to lock the epicyclic gear so that it will rotate as a solid unit. The sun gear which is integral with the pinion shaft is therefore locked to the planet carrier and the whole unit revolves solidly.

THE KICK-DOWN CONTROL SWITCH

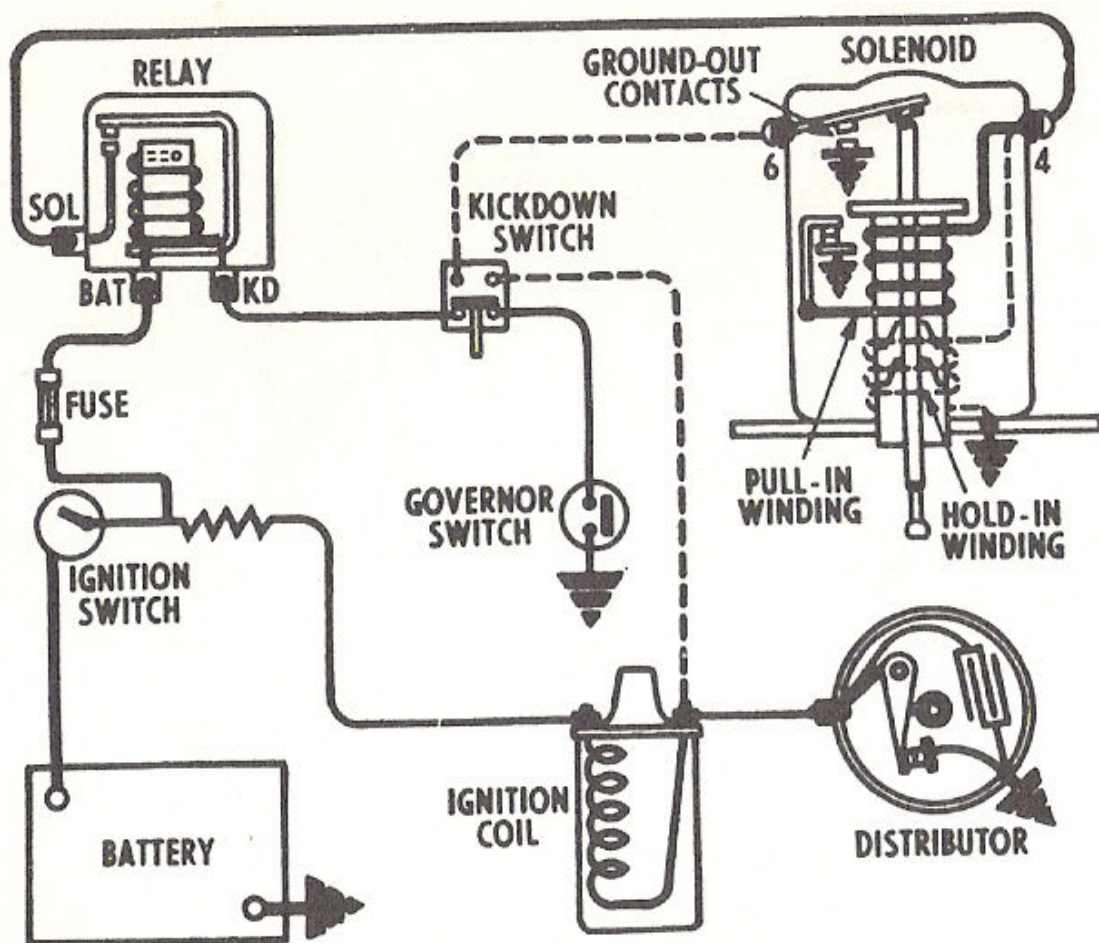
When the car is running in overdrive, due to the fact that the gear ratio is then lower than for direct top gear there is, at any given speed, below maximum, less engine power available for the overdrive than for direct drive at the same speed. Therefore, should the driver wish to accelerate, for traffic passing purposes, it is possible for him to do so, simply by depressing the accelerator pedal quickly and to its full extent. The result is to release the solenoid current but owing to the torque reaction the pawl does not leave its notch in the gear plate and cannot do so until the torque is released momentarily. This is done with the kick-down switch which is operated when the accelerator pedal is depressed fully, thus switching off the ignition so that the pawl readily leaves its notch and also the gear plate; the direct drive is then in operation through the over running clutch.

When the driver wished to return to the overdrive he lifts his foot from the accelerator pedal for a few seconds, when the over drive engages automatically, unless the car speed falls below about 28 to 30 m.p.h. For convenience the overdrive switch is often mounted on the carburettor, so that it can be actuated by the accelerator pedal rod connection at the throttle lever.

SOLENOID UNIT

The solenoid comprises (i) a heavy traction type of winding which is provided to move the plunger outwards until the pawl is almost engaged with the gear ring; (2) a hold-in winding taking much less current than (i) which holds the plunger in its outward position; (3) relay contacts.

One set of contacts operates when the governor switch closes at 28 to 30 m.p.h., the other contacts in series with the windings (i) opens and cuts out these windings. Since the solenoid remains energized during the whole period of over drive engagement the current consumption is reduced appreciably since it flows through the holding winding, only.



ELECTRICAL CIRCUITS

There are three circuits associated with the operation of the overdrive, the circuits are shown diagrammatically.

THE CONTROL CIRCUIT

This circuit includes the solenoid relay and the electrical contacts in the kick-down and governor switches. These two contacts must be closed for the solenoid to operate. The relay is a type of switch operated electromagnetically, such that when current flows through a relay winding the relay contacts are closed. If either of the other two set of contacts are opened, the relay contacts will be broken.

THE SOLENOID CIRCUIT

This circuit includes the solenoid winding and also the relay contacts. When the contacts are closed the solenoid pull-in winding is energized, and the pawl is moved towards the notched gear ring as explained earlier. As the solenoid plunger completes its outward movement it opens a set of contact in series with the pull-in winding. The current in the pull-in winding is thus switched off and at the same time current is switched on to the holding winding to keep the solenoid plunger in its outward position.

THE KICK-DOWN OR IGNITION CIRCUIT

This circuit, which is indicated by the dotted lines in figure, is connected in parallel with the ignition distributor's contact breaker points. It also includes the normally open solenoid contacts which close when the overdrive is engaged. When the kick down switch is operated the ignition coil primary winding is earthed through these two sets of contacts thus cutting off the ignition H.T current to the sparking plugs and thus stopping the engine momentarily. The control circuit is opened at the same time, to allow the overdrive to be cut out of action as explained previously.

ADVANTAGES OF OVER DRIVE

1. This device permits the engine to operate at only about 70% of the propeller shaft speed when the car is operating in the higher speed ranges. i.e., over drive engine speed about 30%.
2. Because the engine is not required to turn over fast at high car speed, the use of over drive reduces engine wear and vibration and saves gasoline.
3. Usually a slightly higher rear –axle gear ratio is employed with an over –drive than without one.

DRAWBACKS OF OVER DRIVE

1. In descending long steep hills where the braking effect of the engine would be lost due to slip in ORC. To avoid over drive should be locked.
2. The driving force available at the wheels is less in case of vehicles with over-drive.

REVERSE LOCK-UP

Since the free-wheel unit cannot power in reverse, provision is made in the Gear-shift linkage to shift the rail and fork assembly into the lock-up position whenever the conventional transmission is shifted take reverse gear.

OVER-DRIVE LUBRICATION

The over-drive unit is connected to the transmission and uses the same type of lubricant, SAE 80 (or) 90 gear oil (or) SAE Engine Oil.

OVER-DRIVE GEAR RATIO

Driving member	-	Planet Carrier (A)
Driven member	-	Ring Gear (R)
Fixed member	-	Sun Gear (S)

RING GEAR	SUN	ARM
S/R	-1	0
1	1	1
(R+S) / R	0	1

$$\begin{aligned}\text{Over-Drive gear ratio} &= \text{Driving / Driven} \\ &= 1 / ((R+S) / R) \\ &= R / (R+S)\end{aligned}$$

If number of teeth in ring gear is 140 and sun gear are 60.

Then,

$$\begin{aligned}\text{The Over-Drive gear ratio} &= 140 / (140 + 60) = 0.7. \\ &= 0.7 : 1\end{aligned}$$

UNIT II

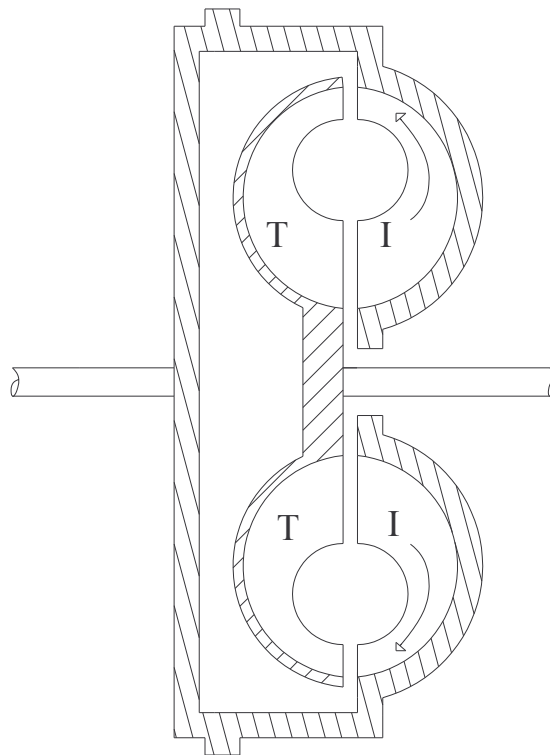
HYDRODYNAMIC DRIVE

FLUID COUPLING

Fluid coupling is a device which is used to transmit torque from engine to gear box with fluid as working medium. The purpose of fluid coupling is to act as flexible power transmitting coupling.

CONSTRUCTION DETAILS AND PRINCIPLE OF OPERATION

The function of the FC is to act as an automatic clutch between engine and gearbox. It allows the engine to idle when the car is stationary but takes up the drive smoothly and progressively when the driver speeds up the engine by depressing the accelerator pedal.



There are two main rotating parts; an impeller driven by the engine and a turbine which drives gearbox. Each is bowl shaped and contains a number of partitions called vanes. The two bowls are placed face to face in a casing filled with oil, and they are separated by a small clearance so that no rubbing contact between them.

The basic form of the fluid drive known as fluid flywheel or fluid coupling is used in place of friction clutch in cars with pre-selector gearboxes. It generally consists of an impeller and a turbine with oil continuously circulated between the two when engine is running. When the engine is idling, the oil is flung from the impeller by centrifugal force. Directed forward by the vanes, it enters turbine which remains stationary because the force of oil is not yet sufficient to turn it.

When the driver depresses the accelerator pedal, impeller speed increases and turning effect derived from fast moving oil becomes great enough to overcome the resistance of the turbine, which begins to rotate so setting the car in motion. After giving up the energy to turbine, oil reenters the impeller and is circulated back to the impeller again. If the speed of engine continues to increase, the difference between the rotational speeds of impeller and turbine gradually diminishes until the slip between them is reduced to as little as 2%. The limitations of FC is that torque applied to turbine can never be greater than that delivered by impeller.

IDLING

The driving part of FC is attached to the engine and faces the driven part from which it is separated by small clearances. At idling speed, there is insufficient centrifugal force for the oil to turn turbine and to move the car.

LOW TO MEDIUM REVOLUTIONS

As the engine speeds up, centrifugal force pushes oil into turbine and some turning effort is transmitted. But there is still a large degree of slip in the unit. The output shaft is thus rotating more slowly than input shaft.

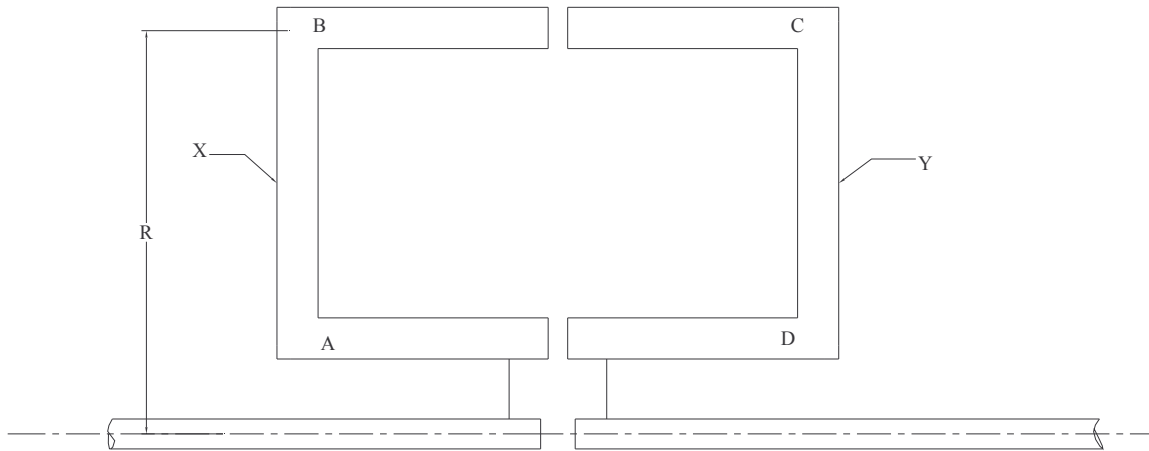
MEDIUM TO HIGH REVOLUTIONS

Once the engine reaches a preset speed, the oil forces is sufficient to transmit full power. This gives in effect a direct drive with output shaft rotating at about 98%of speed of input shaft.

ADVANTAGES OF FLUID COUPLING

1. It provides acceleration pedal control to effect automatic disengagement of drive to gearbox at a predetermined speed.
2. Vibrations from engine side are not transmitted to wheels and similarly shock loads from transmission side will not be transmitted to engine.
3. The engine will not stall if it is overloaded.
4. No wear on moving parts and no adjustments to be made.
5. No jerk on transmission when gear engages. It damps all shocks and strains incident with connecting a revolving engine to transmission.
6. Vehicle can be stopped in gear and move off by pressing acceleration only.
7. There is no direct firm connection between engines and wheels. So when engine is overloaded, it will not stop. But it results in slip within coupling.
8. Unlike friction clutch, slip within coupling does not cause damage within working components.
9. In case of FC, engine is not forced to operate at very low speeds when it is overloaded.
- 10.No wear is experienced on impeller or turbine blades.

TORQUE CAPACITY OF FLUID COUPLING



When slip is below 3%

The change in KE depends on,

1. weight of fluid particles
2. speed of impeller
3. outer radius of coupling

i.e., change in KE Proportional to W

Proportional to N^2

Proportional to R^2

Centrifugal force acting on impeller F_1 is proportional to N^2

Centrifugal force acting in turbine F_2 is proportional to n^2

When slip is constant,

n is proportional to N

i.e., F_2 is proportional to N^2

Resultant centrifugal force $F_1 - F_2$ is proportional to N^2 ,

But resistance to flow = $F_1 - F_2$

i.e., Resistance to flow is proportional to v_f^2 .

$\Rightarrow N^2$ is proportional to v_f^2

Therefore, N is proportional to v_f

Number of flow circuits per unit time is proportional to v_f

Is also proportional to N

Power transferred = Energy per cycle x Number of cycles per unit time.

Energy per cycle is proportional to N^2 and W

Number of cycles per unit time is proportional to N and R^2

Case 1

Power transferred is proportional to N^2 and N

Power transferred is proportional to N^3

Torque transmitted = P/N ,

i.e., T is proportional to N^2

Case 2

Power transferred is proportional to $W \times R^2$

Wt of particle = volume x density i.e., W is proportional to D^3

Power transferred per cycle is proportional to $D^3 \times D^2$

Total power transferred in a number of cycles is proportional to $D^5 \times N$

Torque transmitted is proportional to D^5

From (1) and (2); T is proportional to N^2 and D^5

Thus, $T = CN^2D^5$

Where, N = Impeller speed in hundreds of rpm

D = Outer diameter of coupling in m

C = coupling constant = 5.25

When Slip is upto 10%

$$\text{K.E.} = \frac{1}{2} \times W/g (2 \times 3.14 \times N)(R^2 - r^2)$$

Power transferred = Energy per cycle x Number of cycles per unit time

$$P \text{ is proportional to } WN^2 (R^2 - r^2) N$$

$$P = f(T, N) \leq WN^2 (R^2 - r^2) N$$

$$\text{i.e., Torque } T \text{ is proportional to } WN^2 (R^2 - r^2)$$

Up to 10% of slip,

Torque is directly proportional to slip

i.e., T is proportional to S

$$\text{Therefore, } T \text{ is proportional to } SWN^2 (R^2 - r^2)$$

$$\text{Torque capacity } T = C_1 SWN^2 (R^2 - r^2) \text{ Kgf-m}$$

Where, $c_1 = 0.241$ and S is in decimal.

$$\text{If slip is 10\%, then } S = 0.1$$

$$R = (0.5(R_1^2 + R_2^2))^{1/2} \text{ m}$$

$$r = (0.5(r_1^2 + r_2^2))^{1/2} \text{ m}$$

DISADVANTAGE OF FLUID COUPLING

Even when the engine is idling, runner shaft rotates i.e., some torque is transmitted during the engine is in idling condition. Due to this the gear shifting is somewhat difficult and the vehicle will tend to move when it is parked in gear.

DRAG TORQUE

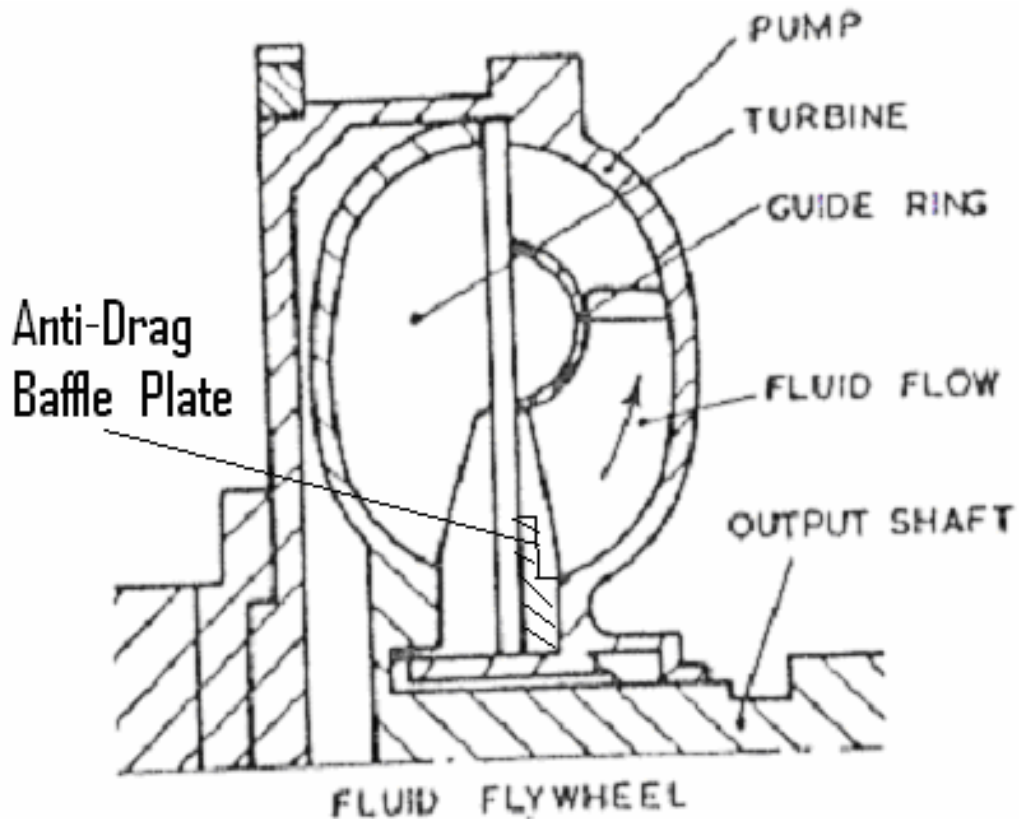
The torque transmitted when the engine is in idling condition is known as drag torque. Even when %slip is 100% there is a drag on GB shafts, which renders gear changing with ordinary type of gearbox very difficult and vehicle will tend to move when it is parked.

To reduce the drag torque, following methods are used,

- 1) Using anti drag torque baffle.
- 2) Using fluid reservoir.
- 3) Using combination of both.
- 4) Fluid coupling is used on combination with epicyclic gear box.
- 5) Fluid coupling with conventional synchromesh gearbox

ANTI-DRAG BAFFLE

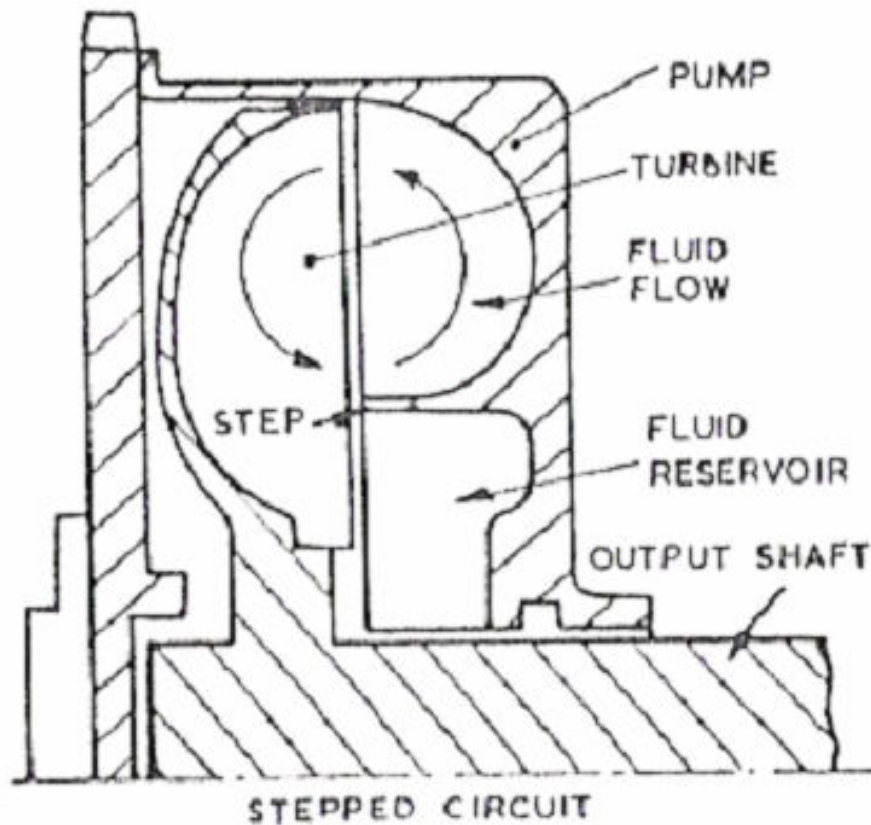
It is a circular plate secured to either the impeller or the runner at the inner circumference of the torus, and projecting into latter, thus obstructing the path of fluid flow. At normal operating speeds the liquid is carried toward the circumference of coupling by centrifugal force, and it is found that under these conditions the baffle does not affect circulation.



Under idling and starting conditions, however, the velocity of the runner is low and the liquid then is forced by the centrifugal force of the vertex motion, so that its flow is greatly affected by the plate. As the speed of the runner increases, the centrifugal force due to runner rotation increases, while that due to vertex motion decreases, and the effect of baffle then gradually vanishes or reduced.

FLUID RESERVOIR

At high speeds, when the liquid is forced toward the other circumference of the coupling by centrifugal force, the fluid coupling will be completely filled with oil and the fluid coupling will transmit maximum possible torque in the reservoir.



The chief purpose of the reservoir is to reduce the drag while the engine is idling, by partly emptying the torus by returning the oil to the torus automatically with an increase in speed. It ensures low slip and relatively high efficiency of the coupling at crossing speeds. Here we can safely ensure 100% filling of coupling. Hence the maximum torque and efficiency is increased. But the fluid reservoir increases the overall axial length of coupling.

COMBINING ANTI-DRAG BAFFLE AND RESERVOIR

Using both the anti-drag baffle and the reservoir in the same vehicle we can reduce the drag torque by about 85 to 90%.

FLUID COUPLING WITH EPICYCLIC GEARBOX

The fluid flywheel is generally used in combination with epicyclic gearboxes which eliminates the drag torque.

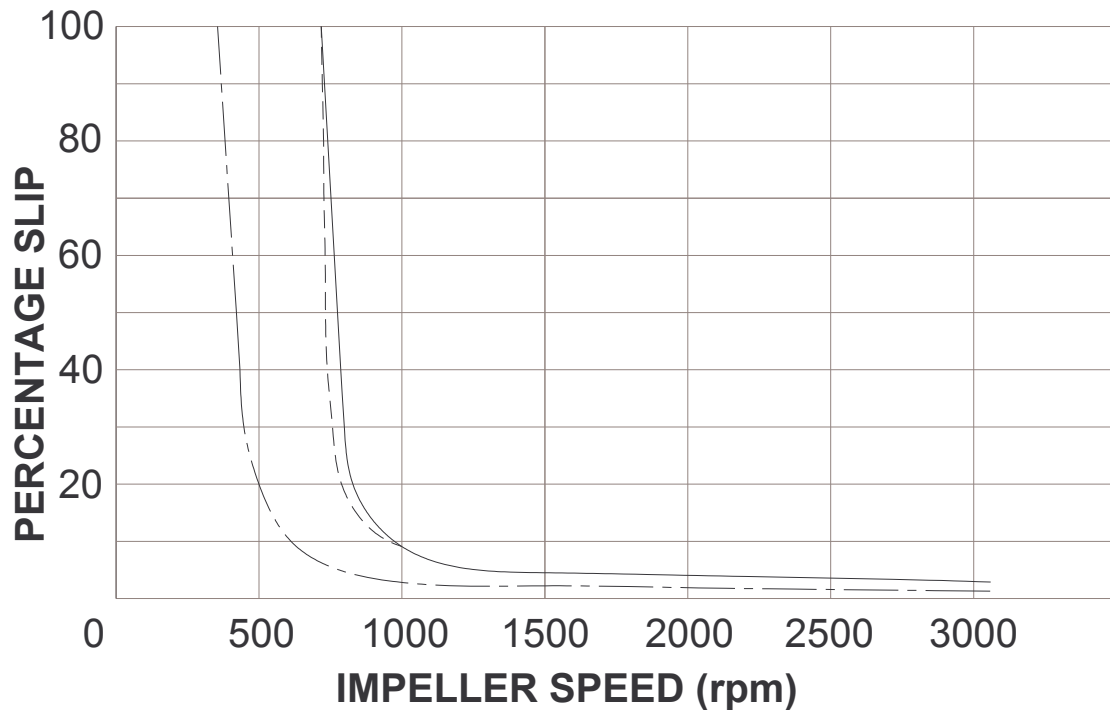
FLUID COUPLING WITH CONVENTIONAL SYNCHROMESH GEARBOXES

Auxiliary clutches are used during gear shift operation. This type of transmission was used in Dodge fluid drive in 1950.

CHARACTERISTICS OF THE FLUID COUPLING

The characteristic which is of chief importance is the way in which percentage slip varies with speed of rotation. The percentage slip is defined as the quantity $((N-n)/N)*100$ and is a measure of the difference in speed of two rotors. If 'n' were equal to 'N', then % slip would be zero and if $n=0$ then % slip would be 100.

When percentage slip is plotted against engine speed (N), the resulting graph is the form as shown. At any speed less than about 600 rpm (this speed can be made to have any desired value by suitably modifying the design), the % slip is 100, i.e., the GB shaft is stationary and we have equivalent of completely disengaged clutch.



As the engine speed increases from 600 to 1000 rpm, the % slip falls rapidly to about 12%. This corresponds to the period of actual engagement of an ordinary clutch, the speed of GB shaft being rapidly brought up to roughly the same value as engine speed. From 1000 rpm up to maximum speed, the % slip decreases comparatively slowly from 12 down to possibly as little as 2. The % slip at any engine speed depends however on the torque being transmitted. The curve being based on assumption that engine exerts full torque at every speed. If the engine torque is reduced below the full torque (open throttle) value, the % slip will be reduced.

Thus % slip may be less than 1%. The chain dotted curve indicates the variation of % slip with the speed for such condition, i.e., level road and no head wind. Whatever the value of % slip, however it represents a direct loss of energy and thus an increase in fuel consumption. Thus, it is an abuse of fluid flywheel to allow the engine speed to fall to region between 100 to 600 rpm, full throttle when %slip becomes ordinary clutch which also increases fuel consumption, the difference being that whereas an ordinary friction clutch would be damaged by prolonged slipping but FC will not suffer any damage although it may become so hot as to burn ones hand if he touches it.

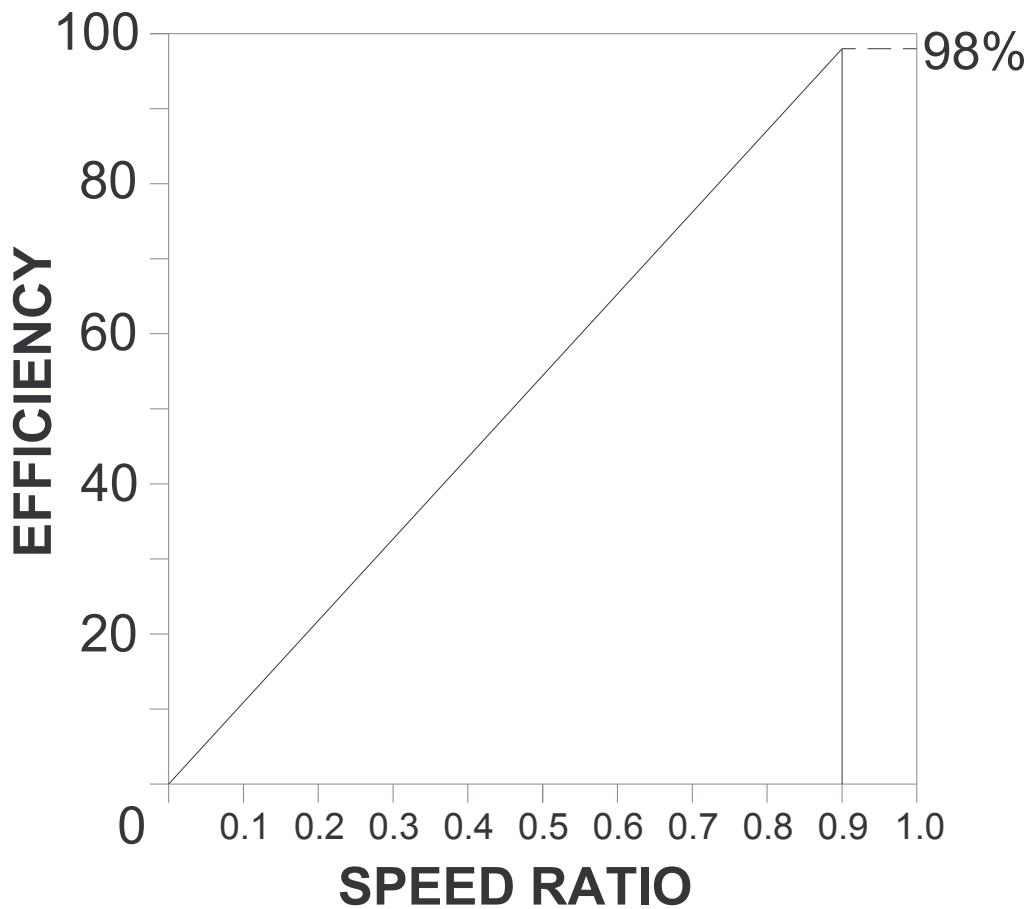


Figure shows that when speed ratio (n / N) is 0.9, the efficiency is maximum.

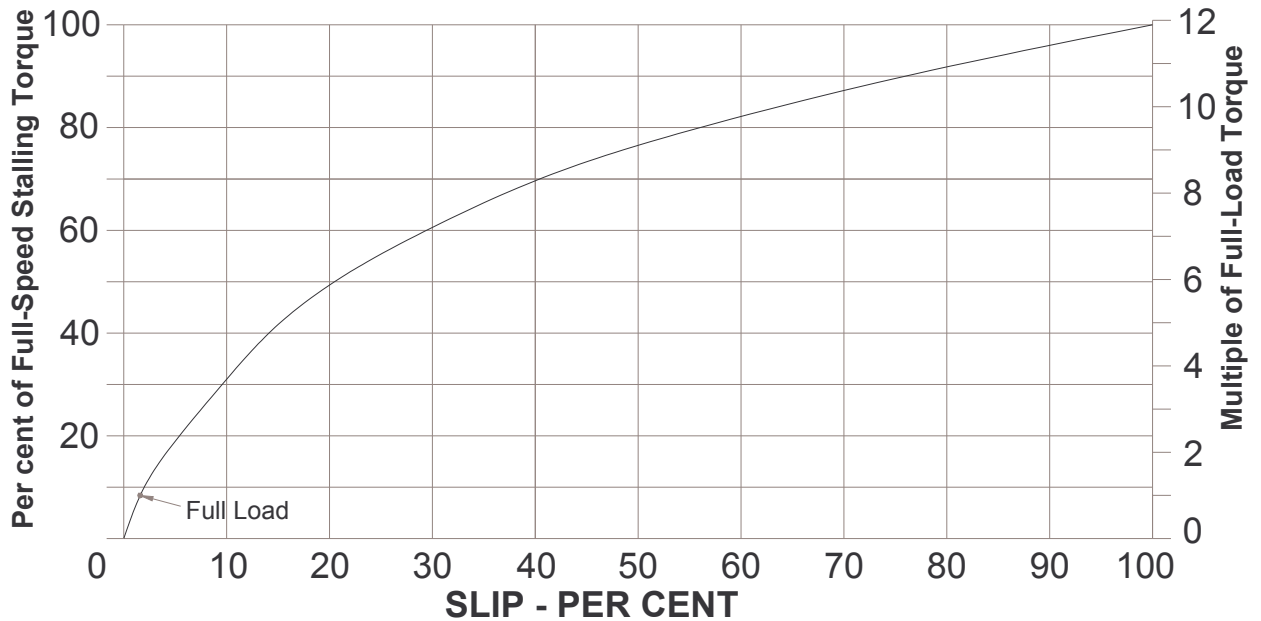


Figure shows Up to 10% of slip the torque is directly proportional to % slip.

PROBLEM ON FLUID COUPLING

1. A petrol engine develops 10 N-m torque at maximum BHP speed of 4000rpm. Determine the diameter of the impeller required to transmit the torque for a slip of 3%.

Torque transmitting capacity of the fluid coupling with in 3% slip

$$T = C D^5 N^2$$

Where

C = A coefficient = 5.25

D = Diameter of impeller in m

N = Maximum speed of the impeller in hundreds of rpm.

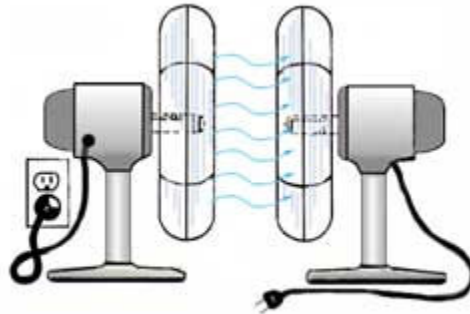
$$T = 10 \text{ N-m} = 5.25 \times D^5 \times 40^2$$

The diameter of the impeller $D = 0.26 \text{ m}$

TORQUE CONVERTER

BASIC PRINCIPLE OF TORQUE CONVERTER

On automatic transmissions, the torque converter takes the place of the clutch found on standard shift vehicles. It is there to allow the engine to continue running when the vehicle comes to a stop. The principle behind a torque converter is like taking a fan that is plugged into the wall and blowing air into another fan which is unplugged. If you grab the blade on the unplugged fan, you are able to hold it from turning but as soon as you let go, it will begin to speed up until it comes close to the speed of the powered fan. The difference with a torque converter is that instead of using air, it uses oil or transmission fluid, to be more precise.

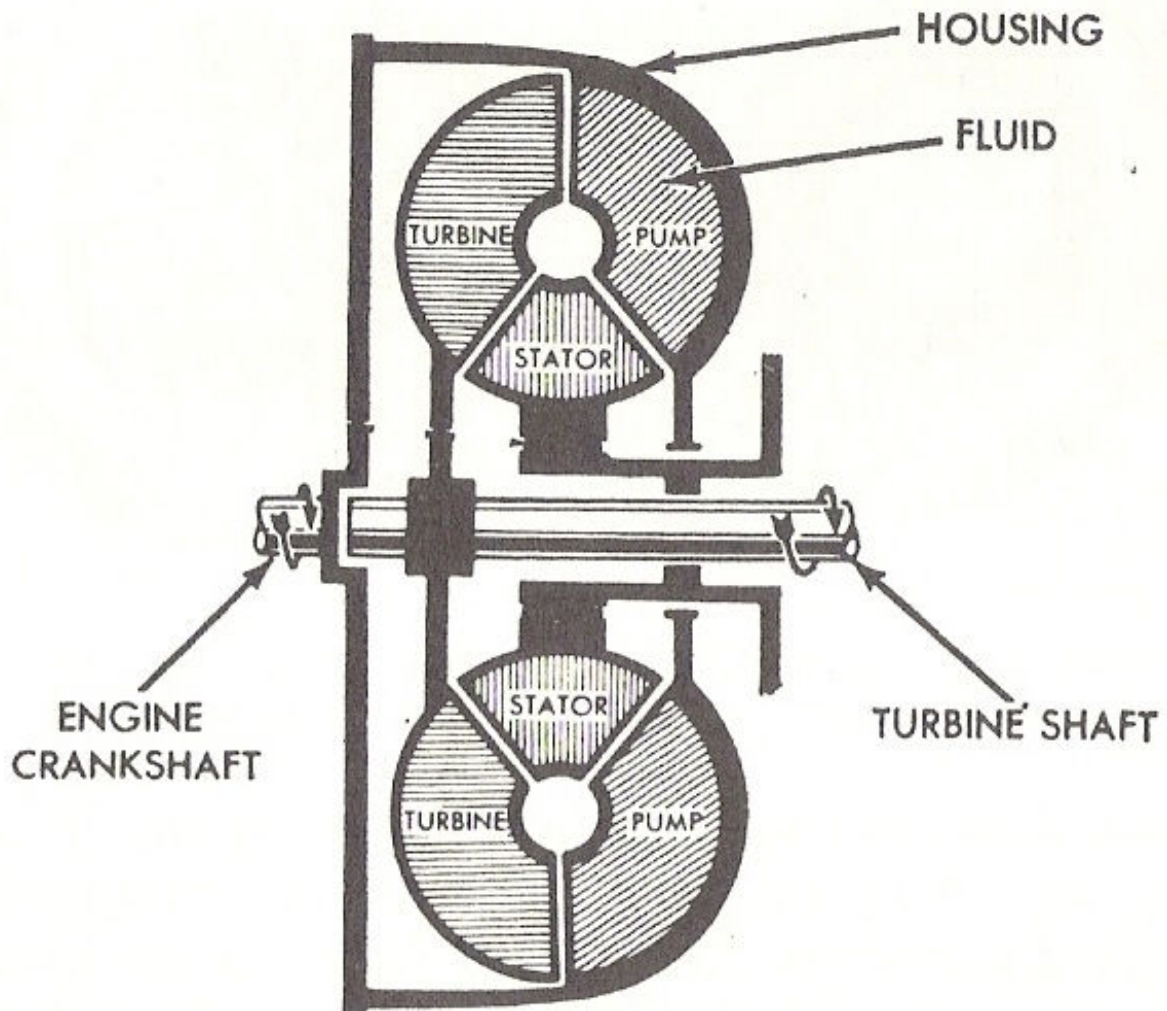


TORQUE CONVERTER

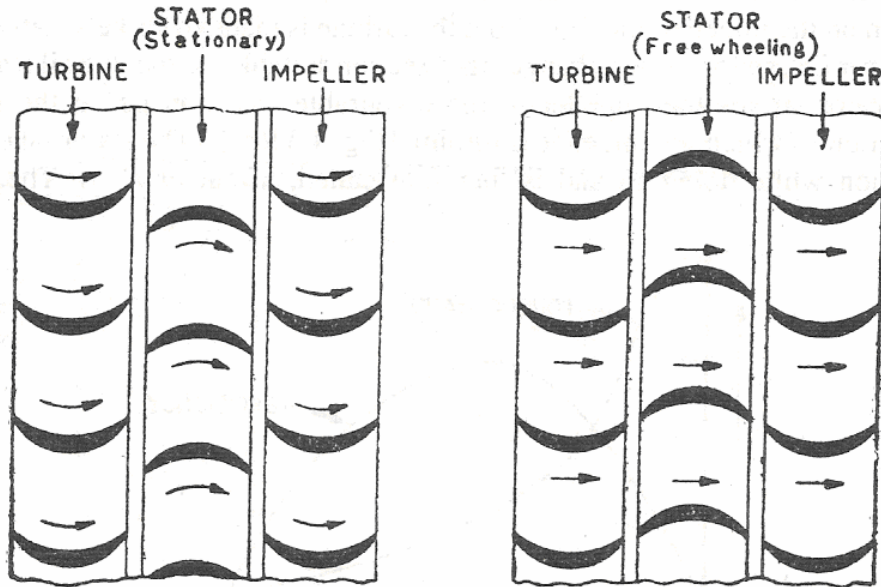
Most cars with automatic transmission use a form of fluid drive known as torque converter as the name implies, it converts the torque or turning effort of engine into higher torque needed by cars at low road speed. An increase in torque has same effect as changing to a lower gear; so a TC is also a gear reducer, acting like an extra set of gears before engine drive reaches gear box.

Like fluid flywheel, TC has an engine driven impeller and a turbine connected to GB input shaft. It is also able to deliver a higher torque than that engine produces, because it is also able to deliver a higher torque and a small vane wheel known as reactor (stator). A one way clutch (ORC) lock reactor to gear box casing at lower engine speed.

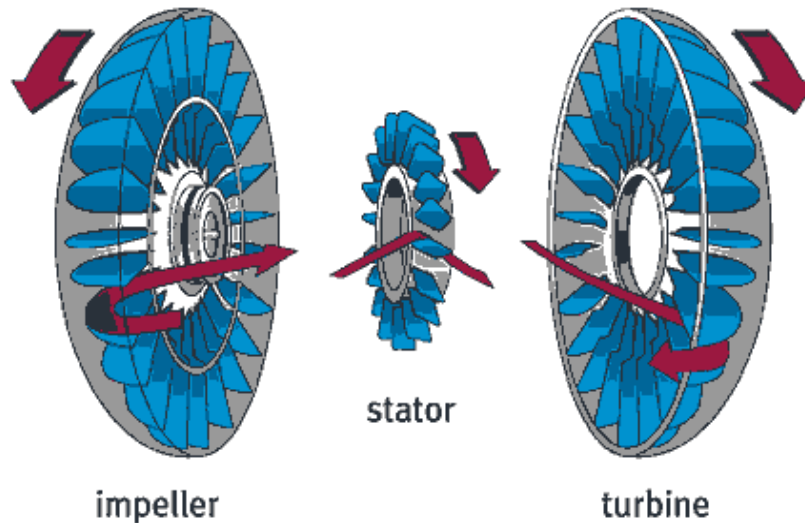
In a fluid flywheel, oil returning from turbine tends to curb the speed of impeller. But in TC, the vanes of locked reactor direct oil along a torque favorable path back to the centre of impeller enabling it to give extra thrust to turbine blades.



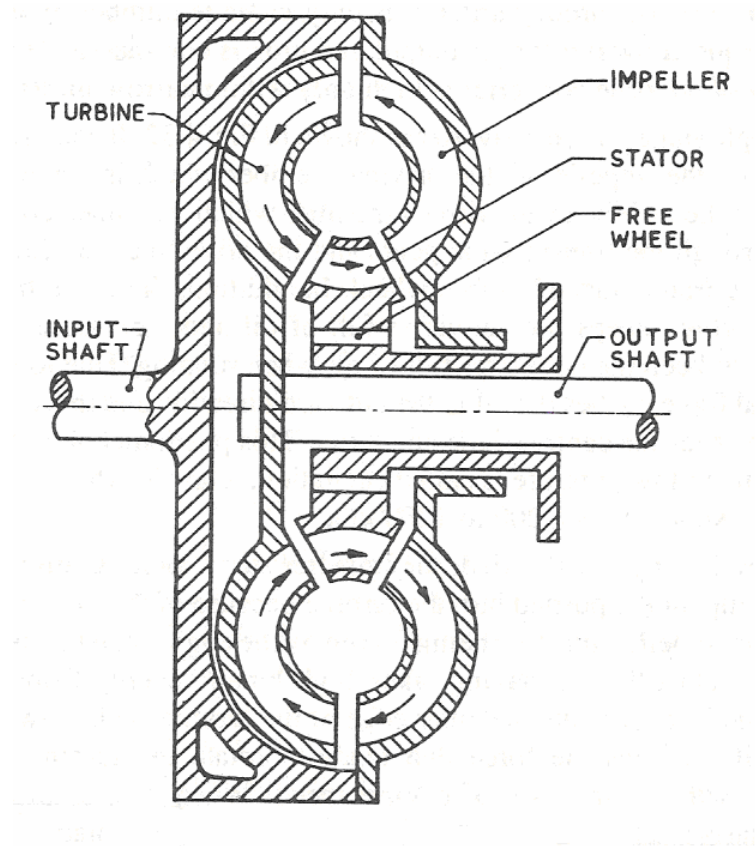
At pull away speeds, Torque Converter double the effort produced by engine. As engine picks up speed, this 2:1 increase in torque is reduced until at cruising speed, there is no torque increase at all. The reactor is spun round by oil at some rate as turbine. TC now acts like a fluid flywheel with reactor 'free wheeling' and having no torque increasing effort. Neither FC nor TC can be epicycle transmission which aloe gear changing without disconnecting.



An alternative used on few models, is to provide a friction clutch in addition to TC. This enables a synchromesh gear box to be used, as friction clutch disconnects engine when gears are being changed.



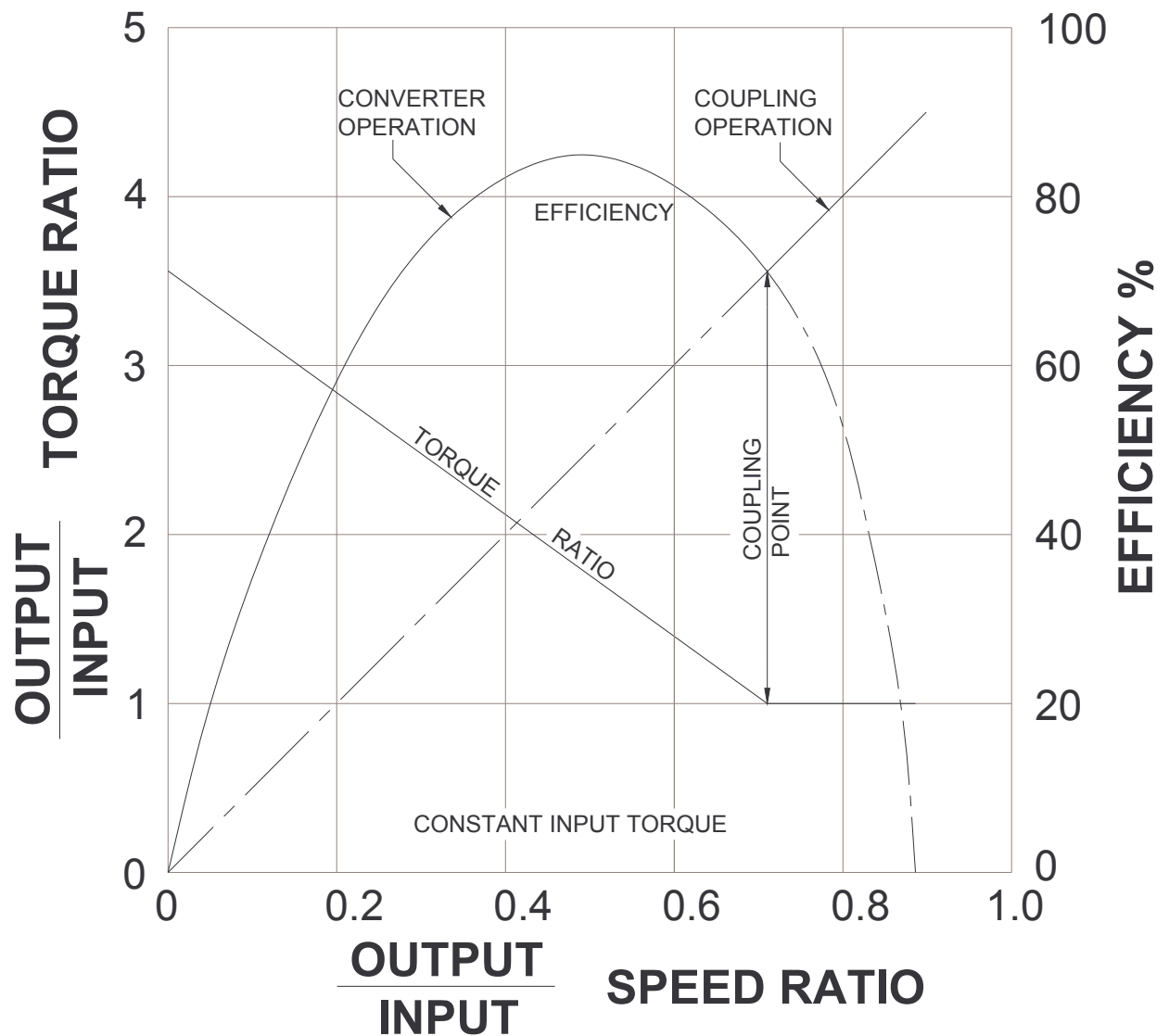
CONVERTER COUPLING



The converter coupling is a next type of fluid drive to consider. This type represents an attempt to combine the best part of converter performance with those of Fluid Coupling. Most forms of converter can be employed as coupling by means of small changes to the reactors.

The converter is made to operate as a coupling by releasing the reactors from external constraint at the time when the applied torque on this member reverses direction. The reactor then rotates slowly so as to give least resistance to the fluid flow. As a result, any torque applied to the pump alters flow conditions in a way that induces an equal torque on turbine member. Various methods of controlling the reactor have been utilized but most convenient is the use of “free wheel” or “over running clutch” unit into reactor so that it is free to rotate in some direction as the turbine but is locked against rotation in opposite direction.

Conditions of equilibrium, ensures that torque on reactor changes direction when o/p torque has fallen to value of i/p, so that introduction of free wheel unit provides an automatic method of changing from converter to coupling operation at correct time. This change over point is normally referred to as “coupling point”. The resultant characteristics is shown where it is seen that best parts of converter and coupling efficiency curves have been successfully combined.



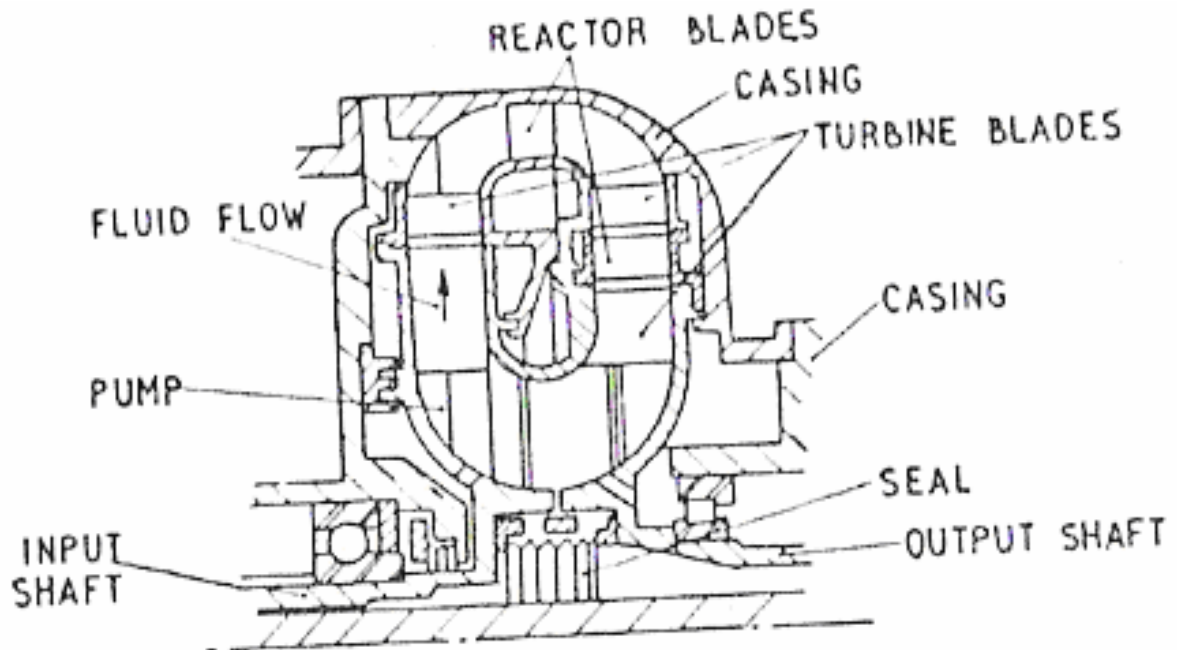
It is noted that there is a slip in efficiency curve around coupling point and effort to remove this efficiency resulted in some interesting innovations. One very simple improvement involves a change in design point which is shifted to a higher speed ratio, when it will be seen that a much improved efficiency curve is obtained for the higher speed ratio at the expense of performance at lower speed ratio. More important, however is the rather serious reduction in torque conversion which detracts from the usefulness of design if larger o/p torque ratios are required. It is fair to say, however that several American automobiles manufacturers appears to be concentrating on this type of converter-coupling, used in conjunction with a number of gear ratios to amplify o/p torque. One feature of converter coupling which is not relived by graphs plotted to a speed ratio base is the fact that transition from converter to coupling after reaching the coupling point is a drawn out process which usually to be accelerated by increased input speed.

The reason for this sluggish coupling operation is that the curved blade which is required, in turbine for converter operation is not so suitable when operating as a coupling. Moreover, the free-wheeling reactor introduces some additional friction in fluid circuit so that it is very difficult to obtain a curve on a basis of turbine o/p speed revolves more accurately the most of the transition from converter to coupling and performance of converter coupling as been redrawn to new abscissa. That graph illustrates conditions which apply for full throttle acceleration. Normal occurring conditions where engine runs at reduced o/p torque will take place at or near coupling point. At this point i/p and o/p torque are equal but o/p speed will be about 10% less than i/p speed, depending on converter design.

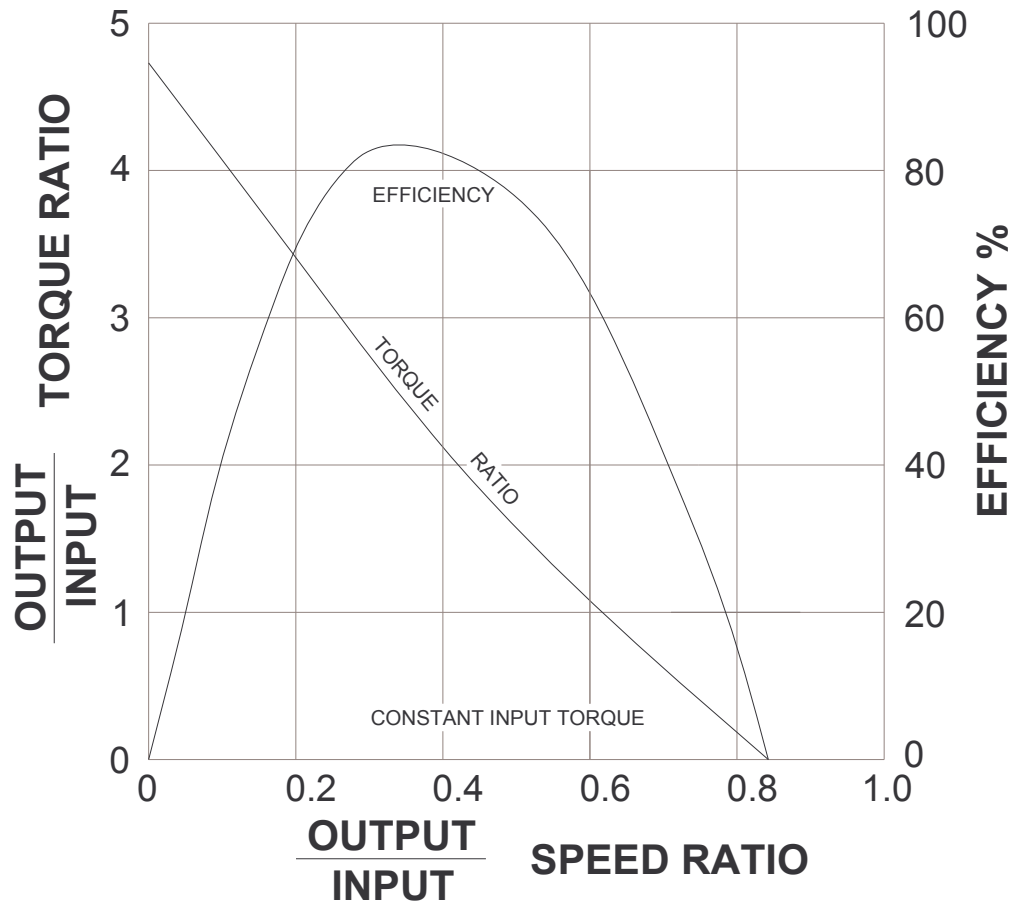
MULTISTAGE TORQUE CONVERTERS

For the highest torque ratios, it is necessary to employ multi-stage converter. The M.S converter is one in which, the circulating fluid impinges two or more turbine members separated by reactors. First stage of conversion is reached when the fuel has traveled through the impeller, turbine and reactors and extra stages are sometimes added to obtain a particular type of performance.

The provision of an additional turbine is referred to as an extra stage and thus conforms to steam turbine practice. It is necessary to emphasize that additional turbine member must be separated by a reactor from a previous turbine to create an extra stage, and should not be confuse with practice of dividing the reactor member for instance into several parts, so as to carry out operation of reactor in a number of phases.



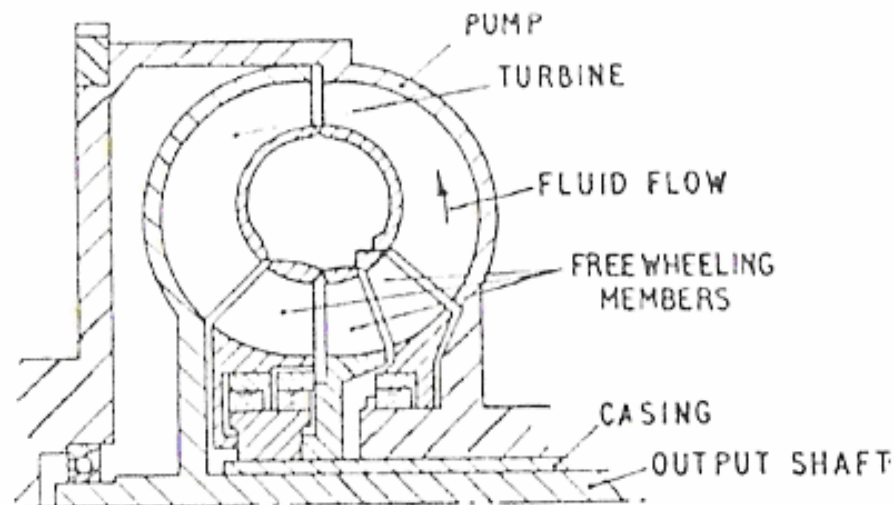
The reason for using an increased number of stages is usually to increase torque conversion ratio but certain other advantages are obtainable. A multistage converter having a turbine immediately preceding impeller has advantage that as the vehicle accelerates, fluid can be delivered at greater velocity head to turbine which is enabled to rotate at faster speed. This fact extends useful range of converter and increases its power rating. The increased number of stages may increase fluid friction on account of longer circulation path, and efficiency of multistage converter tends to fall off rather sharply and racing usually occurs at lower speed ratios than for a single-stage converter.



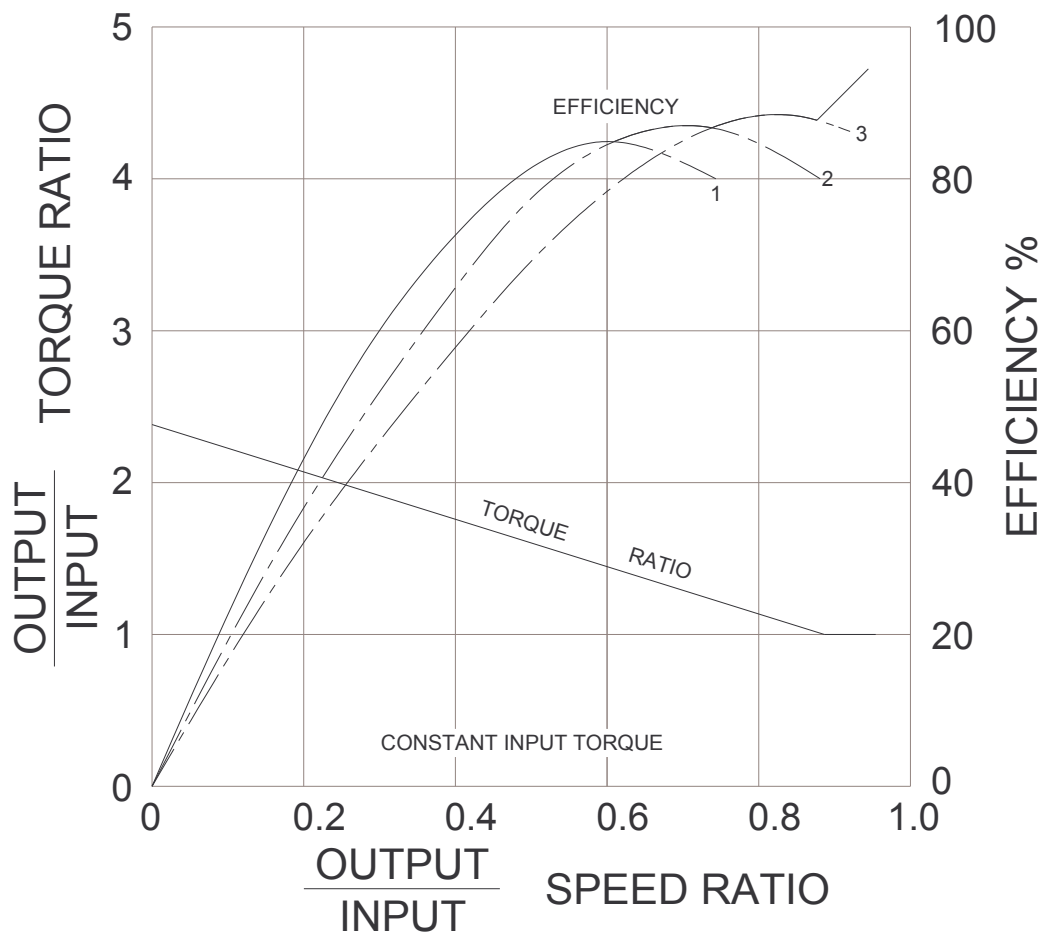
Thus, although increased torque ratios are obtained with multistage converter, it is noticed that all the forms of converter discussed so far exhibit a similar o/p characteristic which is roughly parabolic in shape. Variations in blade design or number of stages has the effect of moving the peak of curve towards low ratio (stall) end of o/p speed range or towards the higher ratios (racing).consequently a typical Multi Stage converter will develop an o/p torque equal to about $2 \frac{1}{2}$ times the value of i/p torque at its maximum efficiency point. But as the turbine accelerates, the η falls rapidly and o/p torque soon falls to zero. This type of converters thus would give a good initial acceleration of vehicle from rest but would be inefficient for a normal cruising which is mainly carried out at unity torque ratio (direct drive). For this reason, most MS or high torque ratio converters are used in conjunction with a friction clutch are some other way of obtaining an efficient direct drive.

POLY PHASE CONVERTER-COUPLING

The next type of fluid drive represents an attempt to combine the best operating characteristics of two or more different designs of converters into a single converter. These ideas are usually incorporated into converter couplings, but it is converter operation it takes place in several phases.



A poly phase converter coupling is a variation of a normal 3-element machine in which at least one of the 3 basic members is divided into further elements. The reactor has to deal with fluid flow from widely changing entrance direction and this member can be divided into a number of elements which adjusts them selves to the changing flow. The practice of dividing a rotating member into several elements is widely used at the present time and it is usual to have 2&3 phases, each of which is represented by a bladed ring element which is also rotate freely when the fluid flow has changed the direction by a given amount. In this way, the elements of reactor enables the operation of redirecting the flow to be carried out in a number of distinct phases, giving rise to the use of the term poly phase converter, or poly phase converter coupling.

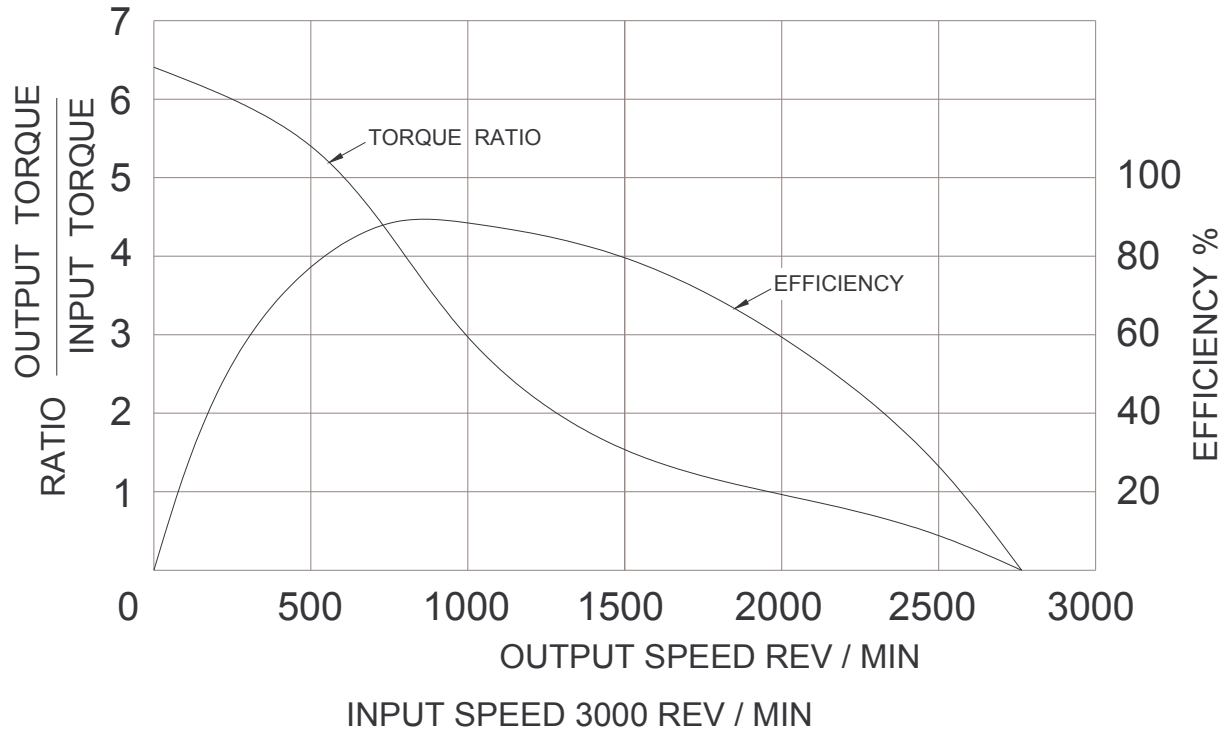


This method is not confined to reaction member, and impeller or turbine member may also be divided into a number of these elements which may rotate at different speed by the introduction of free-wheel units. Such free-wheeling blade elements detach themselves from parent member and rotate at speeds that least resistance to fluid flow occurs. At this point, the detached members can be considered as turbines at raising speed. From figure, it is seen that efficiency curve combines the most useful parts of curves of three different converter designs as the fluid drive effects a three phase transition from converter to coupling. Each η peak is related to a particular design point which is referred to as one of the three separate phases of particular design. The poly phase converter coupling represents modern practice in which entrance losses are almost completely removed for a useful range of speed ratios. There still remains a need to extend this useful range of ratio and most transmission use and auxiliary gear box to give increased o/p torque for improved acceleration and for climbing steeped gradient.

Although there are a few transmission which include a certain features not covered by foregoing 5 basic types, all known design can be properly classified into one of three categories.

CHARACTERISTICS OF TORQUE CONVERTER

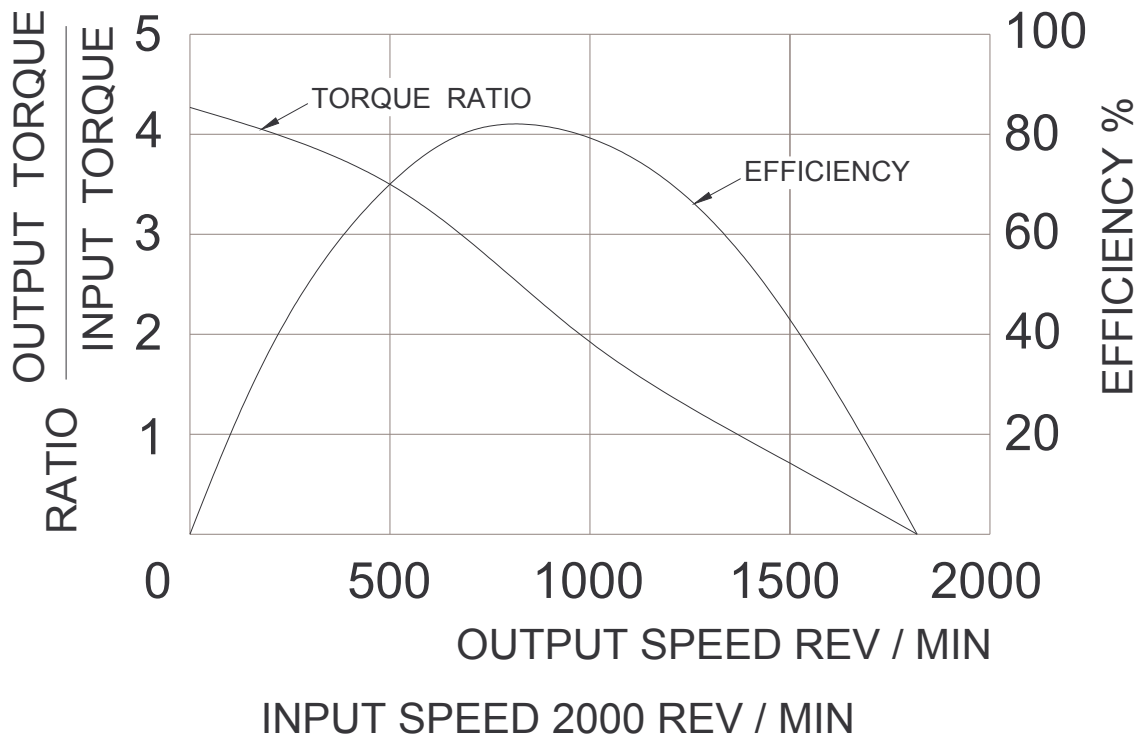
The graph (a) shows the manner in which torque-increases and efficiency vary when rotor speed varies from zero to maximum value (2700 rpm) the impeller speed being constant at 300 rpm. When the rotor speed is zero (because the resistance opposing its motion is large enough to hold it fixed), the torque tending to rotate will be yearly $6 \frac{1}{2}$ times the torque developed by engine at its speed of 3000 rpm.



If the resistance to the motion of rotor now decreases, so that rotor starts to rotate, then as it gathers speed so the driving torque driving action on it falls off, At a rotor speed of 1000, for eg: the driving torque would have fallen off to about 3 times the engine torque, at 1900 rpm. The driving torque would be just equal to engine torque, while at 2700 rpm, then driving torque would have fallen to zero.

The efficiency on the other hand starts at zero, when the rotor speed is zero, because although torque acting on the rotor is then larger, no rotation occurs and torque does not work, thus no work is being got out of converter but a lot of work being put in by the engine and so the efficiency is zero, as the rotor speed increases, so does the efficiency and a peak efficiency of 86-90% is reached at a rotor speed of about 1000 rpm.

As the rotor speed continuous to increase, the efficiency falls off again and at 2700 rpm becomes zero one more, this time because although rotor is revolving rapidly, the driving torque on it is zero, and no work can be out of it.



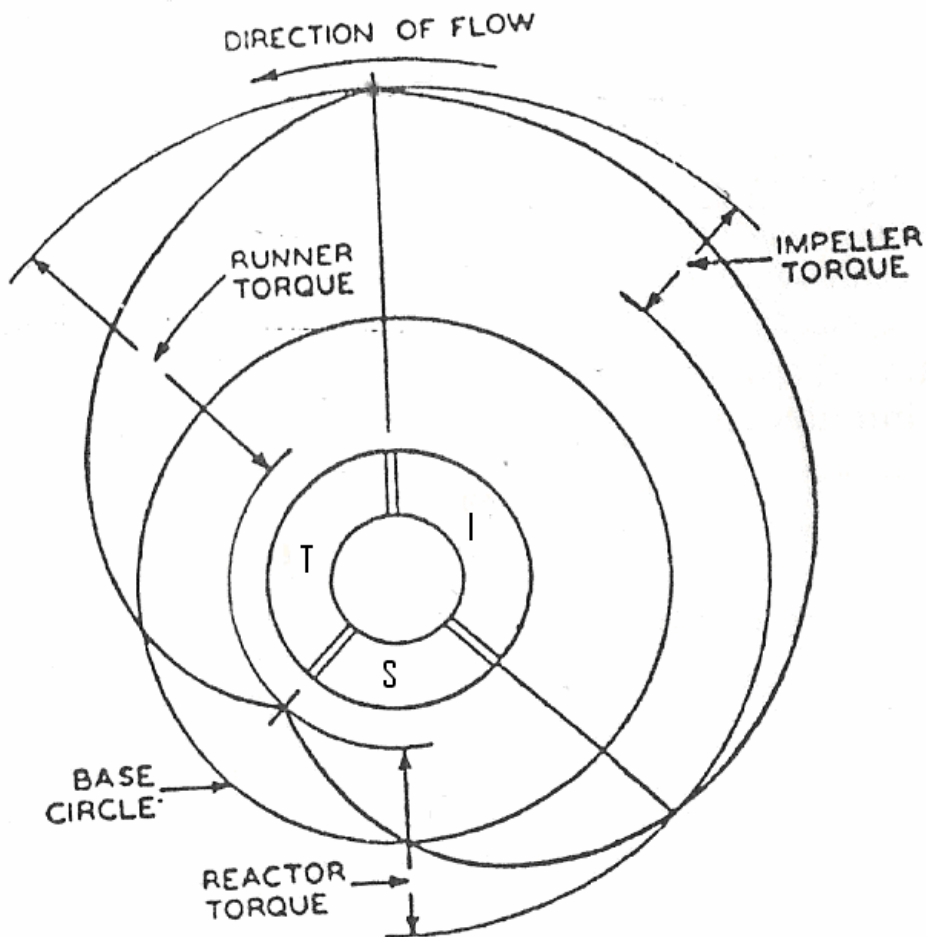
The graph (b) shows the same things but for on impeller speed of 2000 rpm instead of 3000 rpm. It will be seen that the driving torque acting on the rotor when it is stalled, i.e., held at rest, is now only 4 1/3 times of engine torque and it falls of to zero at a rotor speed of 1800 rpm. The efficiency reaches a maximum value of only about 80% instead of 85-90%. It is thus that only a rather range of rotor speeds, if the efficiency resonantly good and it must be borne in mind that if efficiency is (say)60%, then 40% of power developed by engine is wasted, being converted into heat which raises the temperature of TC fluid and which has to dissipate by some means, commonly a radiator.

The falling off of the efficiency at low speed end of range can be treated because these speeds are normally used only for short period when starting and climbing several hill, but the fall off at higher speeds cannot be tolerated and must be circumvented.

There are two principle ways in which this can be done

- a) By substituting a direct drive for TC at high speeds.
- b) By making the TC function as a fluid flywheel at the higher speeds.

PRINCIPLE OF HYDRODYNAMIC TORQUE CONVERSION



The phenomenon of torque conversion by hydrodynamic means is illustrated graphically by polar diagram, where distance radially outward from the base circle represent forward or positive moments of momentum of the spinning fluid and distance inward from that circle negative moments. The spinning velocity of fluid and its moment momentum both are maximum at the runner entrance at the top of diagram. When the runner is at rest or nearly so the spinning velocity and moment of momentum of the fluid are quickly reduced to zero, and owing curvature of the running blades a reverse or negative spinning motion is imparted to the fluid. The torque imposed on the runner is proportional to the sum of the forward moment of momentum at the entrance and reverse moment of momentum at the exit as shown. But while fluid undergoes a great change of motion in runner and as a result subjects the latter to heavy torque it gives up little energy to the runner because latter is either standstill or turning at a very low speed. On leaving the runner the fluid therefore still possesses most of the kinetic energy with which it entered the member.

In reaction member the spinning motion is reversed and the fluid leaves that member with a positive moment of momentum nearly equal to the negative moment of momentum with which it entered. The reaction member is subjected to a torque opposite in direction to that on the runner and proportional to the difference between the negative moment of momentum at entrance and positive moment of momentum at exit, of course it is proportional to the sum of the two. The fluid now enters the impeller already possessed of a certain forward spinning velocity and the impeller needs to add only enough to bring the total velocity up to that at which the fluid entered the runner. In the case represented by the diagram, if the moment of momentum added to the fluid in the impeller is represented by 1, the change of moment of momentum in the reactor is about 1.4, and the change of moment of momentum in the runner is 2.4. Therefore torque ratio in this particular case is 2.4.