AUTOMOBLE - TRANSMISSION

THESIS SUBMITTED TO THE DEPARTMENT OF
MECHANICAL ENGINEERING IN PARTIAL
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FOR THE DEGREE OF BACHELOR
OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

BY

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A C K N O W L E D G E M E N T

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Dated: 16th May, 1960 S. K. Shome

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POWER TRANSMISSION IN AUTOMOBILES

In an automobile the power is produced at the engine and is then transmitted to the wheels to obtain traction. The mechanical system which connects the engine to the driving wheels is called the TRANSMISSION LINE. It includes all devices which aid or regulate the flow of power. Some of the functions of the transmission line are listed below.

FIRST :

One should be able to run the engine on no load. That is, one should be able to detach the driving wheels. This makes the starting of the engine easier. The CLUTCH and the GEAR BOX serve this function.

SECOND :-

After the engine has been started one should be able to connect it to the driving wheels gradually and without shock. This function is fulfilled by the CLUTCH.

THIRD :-

To maintain a constant speed the tractive effort at the wheels should equal the resistance to motion. Since the resistance varies, say, while going along a plain road and up a gradient, one must have some mean to change the tractive effort too. For a given horsepower the torque and hence the tractive effort will vary with speed. Hence we should be able to change the speed. It is also known that for maximum economy or power

CHAPTER - I

1. Introduction
2. Functions
3. Types of Transmission

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FOURTH :-

In general the engine position is such that the axis of rotation of the wheels are perpendicular to the axis of rotation of crank-shaft. Hence to turn the drive through 90° a level DIFFERENTIAL GEAR is used.

FIFTH :-

When negotiating a curve the outer wheel will travel a greater distance in the same time than the inner wheel. The diameter of the two wheels being same they will move at different speeds (r.p.m) This is taken care of by the DIFFERENTIAL GEAR.

SIXTH :-

The wheels are connected to the chassis through springs so that the road bumps ' are not transmitted directly to the chassis. Hence there is relative movement between engine and the rear wheels. Since the engine is rigidly fixed to the chassis. The propeller shaft should be such as to be able to accomodate this change. This is done with the SLIDING JOINT.

SEVENTH :-

The transmission line has to be suitably changed to accommodate the various auxilliaries like, air-conditioning pump, generator etc.

To deal with the various conditions under which the automobile is likely to be used as also for economic conditions, various types of transmissions have been deviced.

(1) Rear wheel Drive

(a) Chain Drive

(b) De Dion Drive

(c) Propeller shaft drive.

(2) Front wheel Drive (F. W. D.)

(3) Four wheel Drive

(3) Six wheel Drive

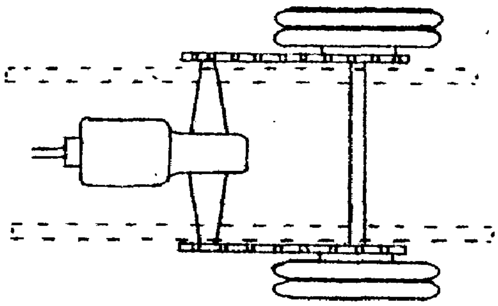
(4) Rear Engine Drive

(5) Under floor Engines.

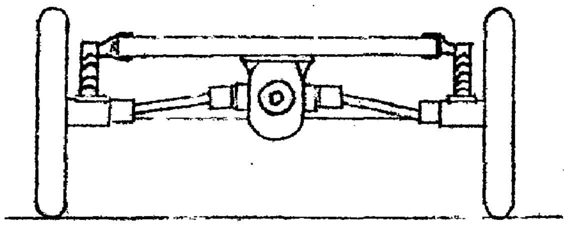
REAR WHEEL DRIVE

(a) CHAIN DRIVE :-

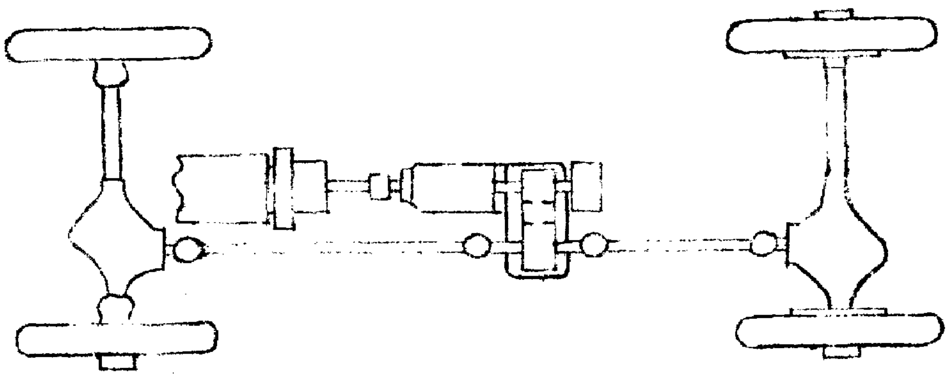
It is now used only on special vehicles. It is an example of the dead-axle drive. In this the drive passes from engine to clutch through the gear box and finally to the differential. The differential turns the drive through 90°. There are sprockets at the end which transmits the power to the wheels through chains. The chains are rather noisy in running and are not easy to keep clean and properly lubricated unless a chain case is fitted. It gives a good and long service even without a chain case. Its chief advantage lies in its simplicity. It



CHAIN DRIVE



DE DION DRIVE



4-WHEEL DRIVE

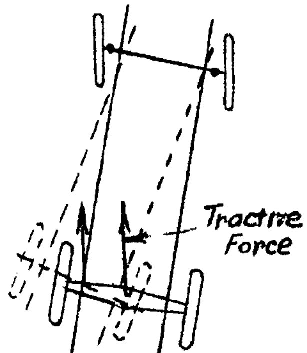
also reduces unspring weight to a minimum and introduces a certain degree of flexibility which saves wear and tear. The chain drive offers scope for some further reduction and is hence useful when very large reductions are required.

(b) DE DION DRIVE

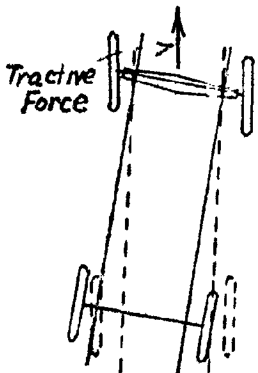
This uses a shaft drive with the difference that both the engine and the final drive ~~the engine~~ casing are fixed to the frame. The wheels are independently suspended. To allow for relative movement between the chassis and wheel axles the power is transferred from the final drive through two universal joints and an intermediate shaft. This is not now so common but a somewhat similar arrangement is used in cars having independantly spring wheels. It was used in the perrari which won the formula II race at Nurburgring in 1950.

(c) PROPELLER SHAFT DRIVE :-

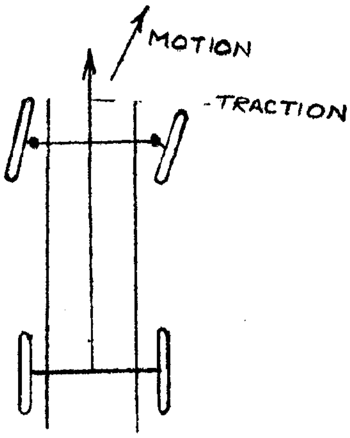
This type is the one which is most common these days. The majority of the cars have a propeller shaft to transmit power after the gear box and upto the differential gear. The construction is simple. The main advantages are that practically no maintenance is required as compared to the chain drive. The main disadvantage is that it takes up some floor space and so raises the height and centre of gravity of the car. Various devices, like the hypoid bevel gear, are used to lower the propeller shaft.



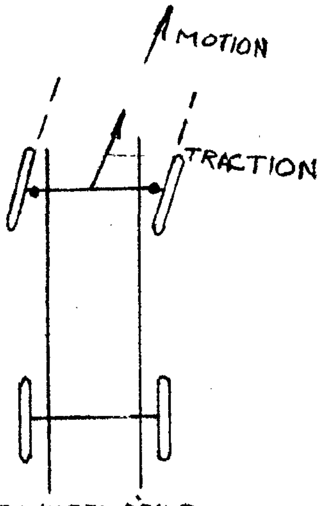
REAR WHEEL DRIVE
(TENDS TO SKID)



FRONT WHEEL
DRIVE



REAR WHEEL DRIVE



FRONT WHEEL DRIVE

FRONT WHEEL DRIVE

In this the power is transmitted to the front wheels. The examples of this type of transmission are found in Alpera Romeo truck, Zundapp, B.S.A., Derby, Citroen, Alvis, Aubum, Tracta, Stoer, Adler, Morris Mini Minor, Austin Seven etc. When the rear wheel driven car has to negotiate a corner the front wheel turn whilst the rear ones are pushed in a direction tending to throw the rear of the car in the opposite direction causing it to skid. If the car were allowed to swing round freely it would ultimately take up a position in which the driven or rear ~~wheels were actually in front or~~ rear wheels were actually in front and the front wheels behind, this being the stable condition of equilibrium. In the case of the front driven car both the power and steering are applied at the point of contact with the front wheels, the rear wheels merely trailing behind and exerting no appreciable effect on the steering. From both points of view the F. W. D car is better to steer round corners and on straight, moreover it can negotiate corners safely at a much greater speed ~~than~~ than a rear driven car. Racing cars which have had front wheel drive have been noticed to do better in races where cornering is important. Another advantage of the F. W. D. is that it enables a lower centre of gravity to be obtained for the whole car, there being no transmission members and back axle casings at rear. The lower seating position and flat undershield of the front drive are other points in its favour. The body can be made of smaller over all height whilst still giving the same headroom than in the case of the orthodox rear driven car body. The F. W. D can be made lighter in weight. Also the full range of steering angles can be obtained without difficulty. An advantage is that the front drive will pull the vehicle along while the rear drive will only push it along. The front wheel drive therefore helps in pulling the car out of ruts and obstacles which the idling front wheels of a rear driven car would have difficulty in surmounting.

Perhaps the main disadvantage is that when going up till the line of action of the weight moves nearer to the rear wheels and hence the reaction at the front wheel decreases. This reduces the frictional resistance and there may be some slipping of the wheels. Also the load on the front axle is seldom more than 35% to 45% of the gross weight and so the tractive effort is less at front wheel than it would be at the rear wheel. The F.W.D is of a more complicated design.

The usual arrangement of the F.W.D is to place the gear box in front of the engine and to incorporate the final drive bevel pivion and crown wheel as well as the differential gear inside the gear box casing. From the differential gear the drive is taken to each road wheel through a short propeller having a sliding coupling at the gear box end and an ordinary coupling at the other or wheel end. The universal couplings then operate at wheel speed and not at 4 to 5 times greater than wheel speed (depending on differential gear reduction).

POUR WHEEL DRIVE

The two rear wheel drive furnished in passenger cars and light commercial vehicles does not ~~mm~~ furnish enough traction to negotiate the steep hills, ditches, streams, trenches and rough muddy or sandy terrain encountered in military service. Since ability to obtain traction depends largely upon the load imposed upon the driving wheels. With all-wheel drive the full weight of the vehicle is carried on the driving wheels and thus permits maximum traction. Also, the combined pulling effect of the front wheels and pushing effect of the rear wheels aid in overcoming obstacles. Conditions often arise in which one set of ~~mmh~~ wheels of a four wheel drive vehicle is driven through a slipery place by the other set of wheels which are on ground affording adequate traction.

From engine, clutch, and transmission assembly a short propeller shaft transmits the power horizontally to a transfer gear box. From lower part of gear box one propeller shaft extends backward to the rear axle gear in the conventional manner and a second propeller shaft projects from one side of the transfer gear box forward to the front axle gear. To allow front wheels to ~~the frank ax~~ be turned for steering a special type of <u>constant velocity universal joints</u> are used to replace steering kunckles of conventional vehicles. Because of higher torque in axle shaft and large angular steering movements these universal joints must be of special design. Rear and front axle gears are either of hypoid or spiral-bevel type. Provision for declutching or disen-

disengaging front wheel drive when it is not necessary is usually incorporated in transfer case.

The obvious disadvantages are that it increases cost, the design is more complicated, production and assembly takes longer and weight is increased.

SIX WHEEL DRIVE

This is the same as the Four wheel Drive except that in this case two rear propeller shafts come out which go to the two rear axles. All the 2½ , 4 and 6 ton military trucks have six wheel Drive.

The advantages are that it can carry larger loads, has increased riding comforts, reduced impact on loads, freedom from skidding and wheel slip and ability to cross difficult terrain.

The disadvantages are additional dead-weight which has adverse effect on fuel consumption. Duplication of wearing parts are open to criticism both from point of view of maintenance cost and chances of breakdown.

REAR ENGINE DRIVE

From the theoretical stand point it is always desirable to use mechanical power where it is generated and thus avoid transmission losses. That is why in certain cars like G. K. W., Trojan, Burney; Tatra 87; Volkswagen etc. the engine is placed at the rear. Its other advantages are that it gives a lower over all height owing to abolition of the propeller shaft, better adhesion of the driving wheels due to weight of engine, a lower centre of gravity, freedom from engine fermes and noise, better visibility for the driver. The main draw back is the question of passenger accommodation. To make way for the engine at rear the seats have to be moved forward . The chassis narrows at the front to give space for the turning wheels and hence the front would have to be narrower. The cooling of the engine also presents considerable difficulties. The gear-box and engine controls are more complicated.

UNDER FLOOR ENGINES

Some busses have horizontal engines with the cylinders placed beneath the floor. This increases the available seating space although cooling and accessibility of engine are both more difficult than with the usual layout.

CHAPTER - II

CLUTCHES

1. Function
2. Types
3. Design Considerations
4. Cone Clutch
5. Plate Clutch
6. Design Features
7. Selection Considerations
8. Clutch Rating Factor
9. Friction Materials
10. Semi Centrifugal Clutch
11. Fully Centrifugal Clutch.
12. Electromagnetic Clutch
13. Magnetic Clutch
14. Clutch Operations
15. Free Wheeling Clutch
16. Hydraulic Clutch.

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The main things which in general will be discussed in the following pages are -

1. The Clutch
2. The Universal Joint
3. The Propeller Shaft
4. The Sliding Joint
5. The Differential Gear
6. The Rear Axle.

THE CLUTCH

Clutches are mechanisms which enable the rotary motion of one shaft to be transmitted at will to a second shaft, whose axis is coincident with that of the first. Owing to the fact that a gasoline engine will not start when carrying a load nor develope appreciable power until it has reached a certain minimum operating speed, about 500-600 r.p.m with automobile engines, some form of clutch is absolutely essential. When a car is standing and the engine is idling at about 500 r.p.m. it would be impossible to have the car speed instantly synchromising with the engine as it would mean that the car has to come to a speed of about 10 m.p.h from rest, instantly. The clutch is required to bring the car speed in line with the engine speed gradually. Another funotion of the clutch is to uncouple, temporarily, the engine and transmission so that the transmission gears can be shifted. Meshing can thus be accomplished without excessive clashing of gears.

TYPES : -

The chief types of automobile clutches that are or have been employed for cars are -

1. THE CONE CLUTCH
   A. DIRECT CONE
   B. INVERTED CONE

2. THE PLATE CLUTCH
   A. SINGLE PLATE
   B. MULTIPLE PLATE

3. THE SEMI CENTRIFUGAL CLUTCH

4. THE FULLY CENTRIFUGAL CLUTCH

5. THE ELECTRO MAGNETIC FLUID CLUTCH

6. THE MAGNETIC CLUTCH

7. THE HYDRAULIC CLUTCH

8. THE FREE-WHEEL CLUTCH

DESIGN :

In general the following points should be kept in mind when designing clutches.

1. The friction material must have a high co-efficient

under all working conditions. These coefficients must not be adversely affected by presence of oil, water or moderate temparature changes.

2. The clutch member driving the gear box main shaft must be made as light as possible to minimise inertia effects which otherwise would be detrimental to the operation of gear changing.

3. The clutch spring pressure and the clutch pedal lever age should be such that the driver does not have to exert undue physical labour to de-clutch.

[illegible]. The ordinary clutch springs used is the helical wire compression type. The disadvantage of this type of spring is that declutching requires a greater pressure than that holding the plates together. Further, the pressure increases with amount of pedal depression. To overcome this difficulty the INGERSOLL spring was designed. Its main characteristic is that the clutch release pressure diminishes as the clutch pedal is depressed. When the pressure is first applied the resistance to deflection increases substantially with the deflection. The resistance achieves a maximum pressure and any further deflection of the spring, as when declutching, only reduces the pressure.

4. The friction members should not slip under the influence of the greatest engine torques that may be applied to them. The members, however, should be capable of slipping on one another without any detrimental effect when the clutch

pedal is partially depressed.

5. The friction material should have long life and maintain its frictional properties throughout its useful life.

6. Provision should be made for rapid dissipation of heat generated by the friction surfaces.

7. Clutches for high speed service should be dynamically balanced to avoid vibrations.

8. Means for lubricating the bearings must be provided.

9. Careful design should eliminate sources of noise.

10. The means of adjustment should be simple.

11. The parts should be easily accessible.

12. The type of clutch and the friction material should be chosen keeping in view the uses to which it will be put.

CONE CLUTCH

This consists of the frustrum of a cone, so fitted to a shaft by means of a feather key that it can be pushed into an opposite engaging surface rigidly attached to the other shaft. Such clutches require some forces to put them into gear. This force, acting parallel to the shaft, produces an undesirable end thrust. Cone clutches may be faced with

(a) leather
(b) asbestos fabric
(c) cork inserts
(d) wood.

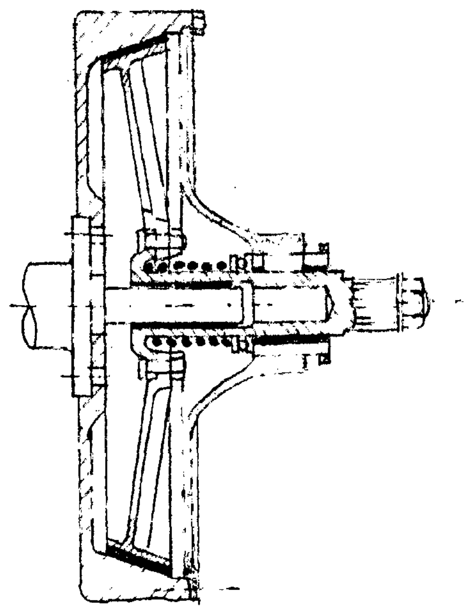
This type of clutches are not now being used in the automobile industry.

DESIGN :-

The various symbols used are -

T is torque to be transmitted
F is axial force applied
f is coefficient of friction of surfaces used.
p is unit normal pressure at contact surface

N is normal force created by F.
A is semi cone angle



CONE CLUTCH

D_1 is inner diameter of cone

D_2 is outer diameter of cone

D is mean diameter.

K is load factor

(H.P) is power transmitted

b is width of face

Q is total frictional force at mean radius

n is rotational speed.

We have $F = N.\ S \text{ in } A.$

$$\therefore N = F/S \text{ in } A.$$

$$\therefore \text{Frictional Force} = N.\ f$$
$$= \frac{F.\ f}{S \text{ in } A.}$$

The torque transmitted is -

$$T = \frac{f.\ F}{S \text{ in } A} \cdot \frac{D}{2}$$

Now $$D = \frac{D_1 + D_2}{2}$$

$$\therefore T = \frac{f.\ F.}{S \text{ in } A} \times \frac{D_1 + D_2}{4}$$

Now we also know that

$$T = \frac{63030 \times (H.P)}{n}$$

Therefore we get

$$\frac{63030 \times (H.P)}{n} = \frac{f.F}{\text{Sin } A} \cdot \frac{D + D}{4}$$

$$\therefore \quad (H.P) = \frac{f.F}{n.\text{Sin } A} \cdot \frac{D + D}{252120}$$

Now $\quad N = D.$

$$= \frac{F}{\text{Sin } A}$$

$$\therefore \quad (H.P) = \frac{f.p.b.n \ (D + D)^2}{160480}$$

On applying the load factor

$$(H.P) = \frac{f.\ p.\ 5\ n\ (D_1 + D_2)^2}{160480\ K}$$

The axial force required is

$$F = \pi . D . p . b . \text{ Sin } A$$

The clutch when being engaged requires a force larger than F as it has to overcome the friction when one cone is being pressed into another.

This force is

$$P = \pi D . p . b . \alpha (\text{Sin } A + f \cos A)$$

The actual force required to engage clutch is slightly more than P due to friction at joints.

The inclination of the slant side is highly important in order to avoid 'sticking' on one hand, and too sudden seizure on the other. This value usually ranges from 7½ to 13. The cone angle A is 12.5 for clutches having leather, asbestos or cork inserts. For clutches faced with wood the value of A varies from 15 to 25.

The mean diameter is taken from the expression

$$D = 34.2 \times \sqrt{\frac{P.k.q}{f.p.n}}$$

where $q = \frac{D}{b}$

The value of q varies from 4.5 to 8.0 . For less wear smaller value of q should be chosen.

Usually the mean diameter varies from 5d to 10d. Where 'do' is shaft diameter.

Checks should be carried out to see that peripheral speed is not excessive. For leather faced clutches maximum peripheral speed should lie between 2000 to 5000 f.p.m. For metal to metal contact the limit is only 300 to 1000 f.p.m.

Guide to selection of p and f :-

Guide to selettion of p and f :-

MATERIALS	DRY	GREASY	LUBRICATED	PRESSURE
C.I.on C.L	0.2 - 0.15	0.10-0.06	0.10 - 0.05	150-250
Bronze on C.L.	-	0.10-0.05	0.10 - 0.05	80-120
Steel on C.L.	0.3 - 0.20	0.12 -0.07	0.10 -0.06	120-200
Nood on C.L.	0.25- 0.20	0.12-0.08	-	60- 90
Fiber on metal	-	0.20-0.10	-	10-30
Cork on metal	0.35	0.30-0.25	0.25 - 0.22	8-15
Leather on metal	0.50- 0.30	0.20-0.15	0.15 - 0.12	10-20
Wire asbestos on metal	0.50- 0.35	0.30-0.25	0.25 - 0.20	40 -80
asbestos block on metel	0.48-0.40	0.30-0.25	-	40-160
abestos on metal	-	-	0.25 - 0.20	200- 300
Metal on C.L.	-	-	0.10 - 0.05	200 - 300

THE PLATE CLUTCH

The come clutch was later on entirely superseded by the single and multiple plate type of clutches, the latter having special advantages in the matter of efficiency, ease of manufacture, and long life.

The multiple disc type of clutch with metal plates running in oil was the first rival to the cone clutch, for although more expensive to manufacture it was better in action and could be made much smaller in diameter. For some time both were in use but ultimately the dry plate multiple disc clutch superseded these.

The chief disadvantage of the multiple disc type is that owing to its thin steel drive plates it was unable to dissipate satisfactorily the heat developed by friction, so that while the end plates could disperse this heat the centre plates attained a higher temperature and more rapid wear of friction material occurred. There was also a tendency for the plates to warp. This resulted in clutch drag. The all metal clutch was subsequently replaced by the single dry plate and the cork insert wet plate clutch.

The chief advantage of the single dry plate clutch are its simplicity, its ability to dissipate frictional heat to the flywheel member and its long period of service with minimum maintainence attention

DRY PLATE CLUTCH:- This type which is in most common use to-day. It was the discovery of the high coefficient of friction material such as fabric and asbestos which made this type of clutch possible. These friction materials are practically non-abrasive, retain their frictional properties over the normal range of working temperatures, practically unchanged, and also have a long life.

These clutches run unlubricated but the presence of oil or water does not seriously impair heir frictional properties. It is however always desirable to keep the plates dry.

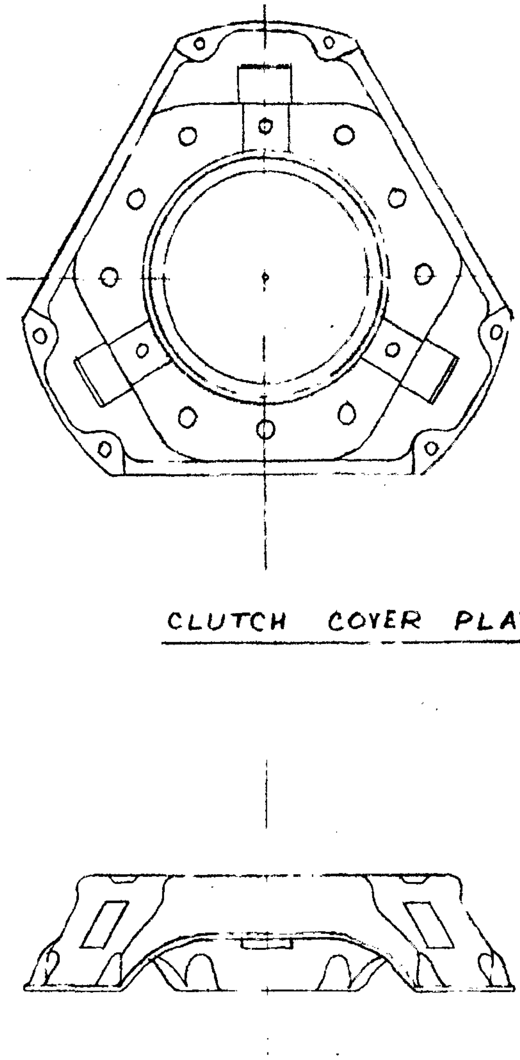
The single plate dry clutch is very simple in design and is comparitively inexpensive to manufacture. It is usually used for light and medium powered cars, as the amount of power is limited by the maximum allowable diameter of the friction material disc. For more pow r several friction discs and metal plates will have to be used.

The single plate clutch consists of a single metal plate faced on either side with a disc of friction material (i.e. there are two friction surfaces). The friction disc is located centrally between two metal surfaces formed as part of the engine flywheel unit. The two metal surfaces and the central friction disc are pressed into contact by means of a single compression spring or a series of peripheral springs. The friction discs are usually mounted on an aluminium or steel plate which, in turn, is rivetted to a bush having introduce a damping action against tosional vibrations or fariations of driving torque.

The clutch sliding member which mores along the splines of the gear box prim ry shaft has a central plate N formed integrally with the internal splined boss M. The pressure plate C is fitted round the boss but is not attached to it. Another thin plate O on the opposite side of N is fitted simil·rly but is made rigid with the boss. The driving connection between the central rigid plate N and the pressure plate C with its complimentary plate O is obtained by means of six inserted compression springs P fitted in slots in all three plates. The effect of these springs between the rigid plate (which is part of the sliding boss on the gear shaft) and the driven plate of the clutch is to give an elastic drive and torsional vibration damper between the engine and the gear box. It is usual to provide stop pins to prevent any over loading of the damping springs by limiting the degree of compression of each of these springs.

The number of springs used varies according to the size of the clutch from 6 to °O. Each of the spring is mounted tangentially to a common circle so that all the springs are at some radial distance from clutch centre.

CLUTCH COVER PLATE:- This should be designed with the object ~~nixhxthm~~ of reducing the weight of the clutch unit to a minimum,consistent with strength requirements, the clutch cover plate is now made in the form of a sheet steel pressing.



CLUTCH COVER PLATE

To minimise its weight, it is made triangular in shape as viewed along the clutch axis.

CLUTCH BALANCE:- It is important to ensure that the complete clutch is in accurate dynamic balance and that the driving and driven member components are also each in perfect in balance. Otherwise vibrations will set in which can be especially harmful if the critical frequency of vibration lies in the engine speed range i.e. upto say 5000 to 6000 r.p.m.

Usually the balancing is done on a dynamic pattern balancing machine, the surplus metal being removed by drilling.

CLUTCH SPIGOT:- This is an important feature of most clutches as it is the support bearing for the engine side of the clutch shaft. This is usually in the form of a ball bearing or'oil-less' bush pattern. The lubrication of the spigot bearing is an important matter since any wear in this bearing will give rise to clutch 'mobble' and noisy operation. Lubrication is also difficult and so the bearing is often packed with high melting point lubricants on assembly and requires no further replenishment.

DESIGN:

The same symbols are used as in the cone cluthh. The power and speed at which the load is to be transmitted is given hence the torque can be found out

$$T = \frac{(H.P) \times 5250}{N}$$

Using static load factor of 2

$$Q = \frac{48\ T}{D}$$

The total number of friction disc required is

$$i = \frac{4\ Q}{2\pi(D_2^2 - D_1^2)\ p.f.}$$

$$= \frac{504000 \times (H.P)}{n.\ \pi\ (D_1^2 - D_1^2)\ p.f.D.}$$

The mean radius D is given by

$$D = 0.707 \sqrt{(D_1^2 + D_2^2)}$$

Hence we have

$$i = \frac{504000 \times (H.P)}{n.\ \pi.p.\ f.\ 0.707\ (D_2^2 - D_1^2)(D_1^2 + D_2^2)^{\frac{1}{2}}}$$

n is the lowest speed at which the clutch is required to grip. This spedd has to be chosen from a knowledge of engine characteristic and will usually be round about 600 r.p.m From the engine characteristic curve of I.H.P. against speed we can find out the ratio of horse power to speed at various speeds. The maximum value of this is chosen for design calculations.

The material is then chosen and from its physical properties we can find out the design pressure which can be used as also its coefficient of friction under the existing workingconditions.

Now the only unknowns are D_1 D_2 and i. The ratio of outer to inner diameter usually varies from 1.3 to 1.5. The society of motor Manufacturers and Traders (S.M.M.T) have suggested standardising the dimensions of dry plate clutch rings and following provisional standards have been suggested.

EXTERNAL DIA (inches)	INTERNAL DIA (inches)	THICKNESS (inches)
6.0	4.5	0.125
6.5	4.875	
7.0	5.25	
7.5	5.625	
8.0	6.00	
8.5	6.375	
9.0	6.75	

EXTERNAL DIA (Inches)	INTERNAL DIA (Inches)	THICKNESS (Inches)
6.0	4.5	0.188
6.5	4.875	
7.0	5.25	
7.5	5.625	
8.0	6.0	
8.5	6.375	
9.0	6.75	
9.5	7.125	
10.0	7.5	
11.0	8.25	
12.0	9	
13.0	9.75	
14.0	10.5	
10	7.5	0.25
11	8.25	
12	9.0	
13	9.75	
14	10.5	
15	11.25	
16	12.0	
17	12.75	
18	13.5	

EXTERNAL DIA (Inches)	INTERNAL DIA (Inches)	THICKNESS (Inches)
12	9.0	0.313
13	9.75	
14	10.5	
15	11.25	
16	12.0	
17	12.75	
18	13.5	
14	10.5	0.375
15	11.25	
16	12.0	
17	12.75	
18	13.5	

A convenient standard diameter can be chosen and the value of "i" determined from the equation. The nearest larger whole number is chosen for the design.

Selection Considerations

Clutch requirements can be fulfilled by one or more of the commercially available units indicated. In addition, slight modifications are possible in many of the existing designs to adapt them for special installation and operating conditions. Beyond this, extensive design effor may afford the ultimate solution to a clutch problem. There are, however, many pitfalls and hidden problems likely to be encountered in clutch design that can become rather costly. Therefore, the large degree of experience represented in commercial units should be weighed carefully before entering into the design of a special clutch. Whichever the case may be, numerous factors are to be considered in the selction or design of a clutch.

CAPACITY RATING: As mentioned previously, a most serious error is the selection of an undersize clutch, which will be a trouble-maker from the beginning. Determination of clutch size involves the application of various factors which allow for the different conditions of service anticipated. To arrive at a final figure, the basic horsepower of the drive should be multiplied in turn by each service factor, resulting in an "equivalent" horsepower nominal rating. Service factors oncerning clutch rating appear in TABLE 4.

The nominal ratings of similar clutches made by different manufacturers are not always based on identical standards. Some ratings give the actual torque at which a clutch will slip when properly adjusted; others show the torque or horsepower safely transmitted in service, and may allow as much as 100 per cent safety factor.

On lineshaft installations the load demand may not always be up to the full capacity of the drive; nevertheless, it is good policy to base a lineshaft clutch rating on the full drive capacity, and in some cases to provide for future additional loads. Conversely, new machines sometimes demand greater than normal horsepower during initial running-in periods. This condition should be provided for, however temporary it may be.

HEAT DISSIPATION: If a very fast pick-up is required, despite high inertia, this obviously suggests a clutch with mechanical torque capacity increased proportionally. On the other hand, it might be necessary to provide a smooth, gradual pick-up lasting ten seconds or more, say for a wire drawing machine. An oversize clutch would again be indicated, this time to allow for the heat generated during the period of slip; in addition, the type clutch selected must be capable of close adjustment or possess inherently smooth action and self-adjustment.

If a clutch is operated once per hour or less, it will have adequate time to cool completely before the next engagement. Most catalog ratings are based on this operating frequency. More frequent engagement requires added capacity, often expressed as an extra two per cent for every additional engagement per hour up to 30. If a still higher frequency is anticipated, it is advisable to consult the manufacturer rather than to risk premature clutch failure.

AMBIENT TEMPERATURE: Usually this factor is not taken into account, but it becomes important in clutches designed for continuous slipping service or operated under unusually high or low service temperatures.

INSTALLATION CONDITIONS: In addition to basic clutch rating and heat control, there are other miscellancous points which should be considered in order to assure satisfactory clutch performance at minimum cost. These items materially affect clutch selection and service life.

ALIGNMENT: The average clutch is not a flexible coupling; therefore, to operate properly it must be installed in alignment and so maintained. Adequate bearings should be provided on each side for both radial and thrust loads.

VERTICAL OPERATION: Most clutches are designed for horizontal service. Vertical operation can impose excessive wear and result in excessive idling drag and wear unless provisions for this condition are made to counteract unbalanced weight.

PROLONGED IDLING: Adequate provision must be made for prolonged high-speed idling. In this respect it is good practice to install a clutch so that the actuating mechanism is stationary while the clutch is disengaged.

DAMPNESS: In addition to rusting, a friction clutch may be subject to friction coefficient variation under damp conditions and may either slip or grab, depending on the friction material. There are, of course, at least two ways of combating this condition, powder-metal friction material and wet operation, as discussed previously.

DUST AND DIRT: Dust causes friction elements to wear rapidly. Dirt and lint can also foul operating mechanisms. Adequate sealing will avoid these conditions.

OIL: The presence of oil in normally dry clutches lowers the coefficient of friction and causes dirt accumulation. Effective oil seats will, of course, prevent this.

FIRE HAZARD: If conditions are sufficiently critical to warrant the use of explosionproof electrical apparatus, similar precaution should be observed in selecting a clutch.

MAINTENANCE: Convenient access for periodic servicing and maintenance should be provided in all installations to avoid unusual demands on clutch durability.

Clutch Rating Factors

Service Condition	Rating Factor
Type of Prime Mover	
Electric motor or steam turbine	1.00
Steam engine or 4-cylinder gas engine	1.50
Single-cylinder gas engine	2.00
Type of Load	
Blowers, belt conveyors, generators	1.00
Hoists, heavy-duty lineshafts, rotary kilns	1.50
Crushers, rolling mills, reciprocating conveyors	2.00
Load Inertia Condition*	
Low inertia: low speeds, small pulleys, etc.	0.75 to 1.00
Average inertia	1.00 to 1.25
High inertia: ball mills, etc.	1.50 to 2.00
Clutch Speed (rpm)'	
10	10.00
100	1.00
200	0.50
400	0.25
800	0.12
1200	0.09
1800	0.07

* Low values for load-free starting; high values for starting under load.

' Based on data for an 8-inch single-plate disk clutch.

Thermal Considerations:

For a specific speed and inertia condition, and a constant clutch adjustment, heat generated in a friction cluthh is proportional to the frequendy of engagement. Since each friction surface has limiting temperature beyond which efficiency is first temporarily then permanently impaired, frictional heat must be dissipated at a rate sufficient to keep the temperature at a safe level. Excessive heat also may cause distortion and warping of the clutch members, producing hot spots and accelerating lining failure.

As a rule, and within practical limits, the most convenient safeguard against clutch overheating is to specify an oversize clutch. The larger the clutch, the greater will be the initial cost, but the lower will be the operating cost. The greater the mass to absorb the heat of engagement, the lesser will be the temperature rise. Also the greater radiating surface will speed heat dissipation. In addition, a greater area of friction lining and a larger mean effective radius will permit considerably lower contact pressures thus reducing lining wear. While there may be special reasons for holding clutch size to the minimum, such as space limitations or considerations of clutch weight or inertia, in general, thermal efficiency should not be sacrificed.

To illustrate the effect of toroque load and frequency of engagement on clutch life, one authority cites the following data from a series of friction clutch tests:

Torque Load	Frequency (starts/hr)	Satisfactory Starts
Max. Rating	180	10
Max. Rating	1	500
½ Max. Rating	180	50,000

Extensive studies have revealed that, up to the point of destruction, wear is an exponential funtion of the average rate of heat generation.

In designing for maximum capacity within a given space, the problem of overheating is met by two approaches: the use of constructions giving maximum heat dissipation and minimum warpage, and the use of friction materials that are affected by heat as little as possible. These friction materials must of couse possess adequate mechanical strength as well as good thermal properties.

Design for heat dissipation involves the use of short, highly-conductive heat transfer paths, with friction surfaces exposed liberally for air-cooling. In this connection, it is interesting to note that in one successful aircraft brake design the friction material has been reduced to a single pair of small buttons pressing against opposite sides of a steel disk. Most of the disk is exposed, and as a result, it stays cool because of the excellent convective transfer to the line surrounding air. For the same reason despite the usually small friction lining area and the high surface pressure, the friction buttons are not subject to the heat fading and excessive wear found when full-lined brakes are applied continuously. It is interesting to note that in tests this same brake failed when a second pair of buttons was added experimentally at a point diametrically opposite the first pair - heating then became cumulative.

Other means for improving heat dissipation include finned surfaces for increasing heat transfer areas and air turbulence, and roughened exterior surfaces for improving radiation. For example, a sand-cast surface has about five times the heat radiation capability of a comparable surface with a fine machined finish. Some clutches are designed to run 'wet' (in oil) to secure liquid coolin , which is very successful provided the oil does not become hot enough to gum and carbonise.

When heat warpage is found to be a problem, the solution may become involved. A typical answer is to use expansion-relieving cuts or grooves in the member supporting the friction materials, but there is no pat answer. Actually warpage is usually less of a limiting factor in clutch design than the temperature of the friction surface. Studies of this phase of the problem have proved a fruitful line of attack in increasing clutch ratings.

Friction Materials:

Some decades ago, the principal friction materials available were leather, wood (maple, elm or lignum-vitae), cork, or felt operating against metal. While these organics are reasonably durable and give a satisfactory coefficient of friction, they are adversely affected by moisture and by by comparatively small temperature rises. In most of these materia s, bleeding of the natural resins and charring become noticeable at 175 F, with resultant loss of friction and rapid wear. The energy absorption capacity of wood, for instance, is given as 250 lb-ft per minute per square inch, as compared to 1000 to 2500 lb-ft permissible with standard asbestos materials and 10,000 lb-ft permissible with certain all-metal friction materials under good heat dissipating conditions.

Cork and felt possess he special property of high resiliency and have worked out very well in friction clutches for l ght drives, such as wire recorder tension controls and sewing machines. They were used successfully in a number of early automotive clutches, although most cars now use asbestos-base materials.

The metal-to-metal d i ns of the past employed standard bearing combinations, such as bronze on steel or c cast iron on cast iron. Althou h suitable for somewhat higher temperatures than the natural nonmetallics, these metal combinations labored under the disadvantages of a low friction coeffcient, a need for careful lubrication(excepting cast iron on cast iron), and a tendency towar sparking(except bronze). Galling presented a serious problem wing bronze on steel. Accordingly, when any of the basic metal combinations are us d, clutch size, weight, and inertia will necessarily be greater than for the nonmetallic facings or linings, and utility will be limited. As clutch friction materials, the baci metals are limited in general for low-speed, low-torque applications where the cost of special facings is not justified.

The modern asbestos-base and powder-metal friction materials were developed to overcome the various limitations of their predecessors. They permit a considerable reduction in clutch size for a given load and thus more than offset their added cost. Also they are far less affected by adverse environmental conditions, wear less rapidly, and many types are available for wet or dry operation. Table presents a composite picture of clutch friction materials.

Woven Asbestos:

Asbestos fiber is woven around brass, lead, copper, or zinck wire, impregnated with asphalts, rubber, drying oils, or other resins, and then baked and heavily compressed The glazed surface is sometimes finished by grinding to shorten breaking-in time. This is rhe original asbestos friction material, and it remains in demand because of its flexibility and durability. Ordinarily it is fastened to a metal backing plate with rivets although special cements are also used. Cementing allows utilization of approximately 15 per cent mo e lining area which otherwise is wasted by the rivent method, permits wear almost through the full thickness of lining and prevents scoring f the opposing surface. Cementing and rivet ng may be used in combination in heavy-duty applications.

Molded Asbestos:

Short asbestos fiber bonded with ingredients similar to those used for woven lining are employed for molded asbestos linings. Brass chips, iron oxide, emery, etc. may be added to increase friction for use with cast iron. Graphite or mica may be added to reduce friction for use with steel which is more susceptible to scoring and cutting. Another variation may be a wire mesh backing, adding flexibility and reinforcement. Because of its versatility, low cost, and comparatively high strength and hardness, molded lining is used extensively.

Molded Semimetallics:

Developed within the last decade, this material consists of asbestos, copper powder, and a synthetic bonding resin. The lining is molded in thicness ranging from 1/64 in. to 1/8 in. directly on-to a steel shoe or other backing. Because of the copper powder and the short heat travel path, this material is rated higher than plain molded lining.It is also expensive, and therefo e, is reserved for premium applications, such as bands in automatic transmissions.

Powder Metals:

Powder-metal friction materials employ various combinations of powder iron, copper, lead, zinc, silicon, and tin, plus nonmetallic powde s such as graphite silicon carbide, silica, and alumina. The material wea s slowly and accordingly permits use of thin sections to minimize cost and wei ht. It is brittle, h wever, usually requiring a steel, copper or cast iron backing plate to which it is bonded under pressure and heat. Subsequent grinding or lapping provides a typical final thickness of 0.007 inhh facing.

Although expensive, as a result of the special processing required, sintered facin s offer many important advantages in clutch performance. Outstanding among these are: constant frictional charactersistic within a wide operating temperature range, excellent heat conductivity high resistance to galling and scoring, and insensitivity to moisture, solvents, oils, etc.

__Graphitized Carbon:__

Special mention is made of the carbon-graphite materials such as Graphitar purebon, etc. While these materials are too brittle and expensive for routine use, they are well woth considering for severe duty in small cutches and where much slippage is required. The thermal conductivity is higher than for cast iron, and the material will withstand very high pressures and speeds without galling or excessive wear. Rate of wear is indicated by the pressure velocity product: PV =K, where P= Pressure, psi, and V =velocity, fpm. The maximum value of K is 150,000 for lubricated, and 15,000 for nonlubricated surfaces. The material can be operated at a higher value of K on intermittent duty, satisfactory results having been recorded with a K value as high as 2,000,000. This value is equivalent to a 6 in. diameter clutch ring running at 4250rpm. under 300 psi loading.

A word of caution is in order concerning friction coefficients. In clutch design practice, the nominal coefficient of friction is usually taken as one-half to three quarters the value shown in Table. This provides a safety margin for wear, overloading, dirt, or other adverse conditions for contact pressure, the effective lining area should be considered rather than the grossm becuase of the spiral radial, or criss-cross grooves often provided.

Wet vs. Dry Operation as noted in the Table, many materials can be run either wet(oil) or dry. If run wet using a spray (preferred) or a shallow dip, the coefficient of friction is lowered, but it is usually possible to raise the working pressure to compensate partially for this. Grooves in the softer friction face are desirable, to improve the circulationand cooling effect of the oil, to prevent the formation of a hydrodynamic separating film, and to improve releasability.

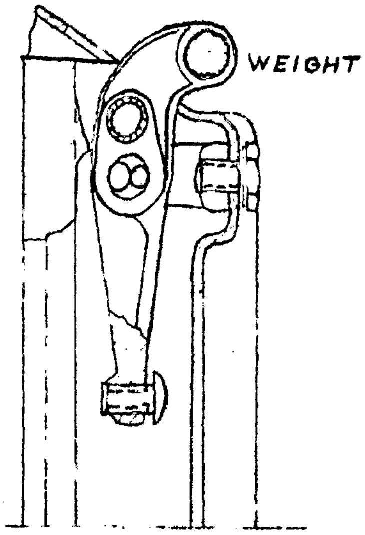
In general, a wet clutch is smoother in action, wears less, and is much better suited for rapid cycling and heavy duty, mainly because of the cooling action of the oil. Because a wet clutch is totally enclosed, protection from abrasive substances is assured and there is little or no fire hazard in the presence of lint or inflammable vapors. In constrast, a dry clutch is simpler and more compact, idles with considerably less drag and heating, and is affected much less by extreme ambient temperatues. Again grooves often are provided in the softer friction face, principally to scavenge dirt or wear particles and to permit air circulation for cooling and rapid release.

In comparing wet and dry clutches, it is helpful to recall the troubles experienced with the wet clutch in yesteryears' automobiles: gearshifting was accompanied by much clashing of teeth on cold mornings because the viscous oil would not release. Then, too, as the oil thinned out after weeks of use, the clutch would develop a tendency to grab. A change of oil often resulted in clutch slippage.

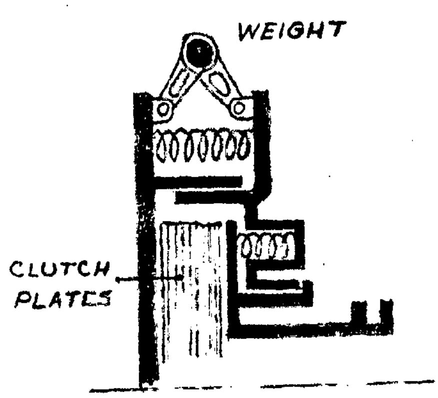
It should be noted that while Table lists most of the conventional friction materials and combinations, it does not begin to exhaust the possibilities. One unusual clutch arrangement used successfully to cushion the action of an aircraft engine starter employs tempered phosphor bronze disks against steel. Because of the high pressure required serious chattering was experienced initially as a result of transfer of bronze to the steel disks, even when the bronze was tin plated. An attempt was made to overcome the difficulty with a large number of oil reservoir holes, but this was only partially successful. Chromium plating the steel disks was tried next and found satisfactory, but the cost was considered excessive. The final solution was to add a fine abrasive material.

SEMI CENTRIFUGAL CLUTCH

In the plate clutches we have the springs exerting pressure on the friction lining and whenever we want to de-clutch we have to exert a force and compress the springs. At lower powers, the force may not be much but as the power ~~the force may~~ goes on increasing the force required to declutch is correspondingly increased. In the semi - centrifugal clutch it is so arranged that when we have to de-clutch (usually at low speeds) we have to exert a force against the springs and a negligible amount of centrifugal force. We know that power goes on increasing with speed and so a larger pressure on the friction plate is required. This excess pressure is exerted by centrifugal action on three unbalanced masses placed 120° apart. Since the centrifugal force increases proportionally to the square of the speed it always increases at a faster rate than the power and the pressure is always enough to avoid slipping.



SEMI CENTRIFUGAL CLUTCH



CENTRIFUGAL CLUTCH

(3) no slip under normal load

(4) idling drag completely eliminated at low speeds by use of shoe restraint spring designed to engage at some intermediate value.

(5) lower cost.

(6) flexible conpling action

(7) easy adjustment of tongue by varying member of shoe.

(8) automatic wear compensation

(9) no clutch pedal is necessary.

(10) automatic clutch reduces fatique and accidents.

FOUR CENTRIFUGAL CLUTCHES.

This is the same as in the Semi - centrifugal clutch except that modifications are made for the centrifugal device to operate positively from the lowest or idling speeds. The arrangement is as shown in figure. In this we have light compression springs to hold the friction plate and pressure plate apart while the engine idles at about 500 r.p.m. On accelerating the engine the centrifugal force increases and spring action is overcome and the clutch gradually engages. There is a certain amount of slip at first but it is so designed that beyond 1000 r.p.m. full engine torque is transmitted with no slip.

Most of the centrifugal clutches are of the expanding shoe type operated either directly by shoe weight or indirectly by pressure developed elsewhere . Clutches are also built by wedging balls or shoes carried outward by centrifugal force against the restraint of a garter spring. These members in turn, actuate a disc friction surface.

The advantages of the mechanical friction centrifugal clutch are :

(1) easy starting of high inertia loads.

(2) overload slip protection .

(3) no slip under normal load

(4) idling drag completely eliminated at low speeds by use of shoe restraint spring designed to engage at some intermediate value.

(5) lower cost.

(6) flexible conpling action

(7) easy adjustment of tongue by varying member of shoe.

(8) automatic wear compensation

(9) no clutch pedal is necessary.

(10) automatic clutch reduces fatique and accidents.

ELECTRO MAGNETIC FLUID CLUTCH

A new type of clutch developed recently is based upon the properties of a fluid consisting of light machine oil in which are suspended very fine particles of iron . These particles adhere to magnetic surfaces when subjected to magnetomotive forces. The fluid exhibits unique properties and has been used for clutches which give

(a) high efficiency

(b) Smooth torque variation

(c) long life

(d) finer control

(e) no clutch pedal required.

MAGNETIC CLUTCH

This is a form of clutch in which the friction surfaces are brought into contact by an electromagnet instead of by mechamical means. They may be operated at higher rotative speeds than mechanical clutches and are recommended for heavy duty. The principle of the clutch is simple. A magnet coil is carried by one member of the clutch and this on being energised attracts to itself the armature on the other member of the clutch which is carried by a spring plate so that the armature can be pulled forward. When the armature is pulled forward the friction lining comes into contact and a certain pressure is exerted on it. This transmits the torque.

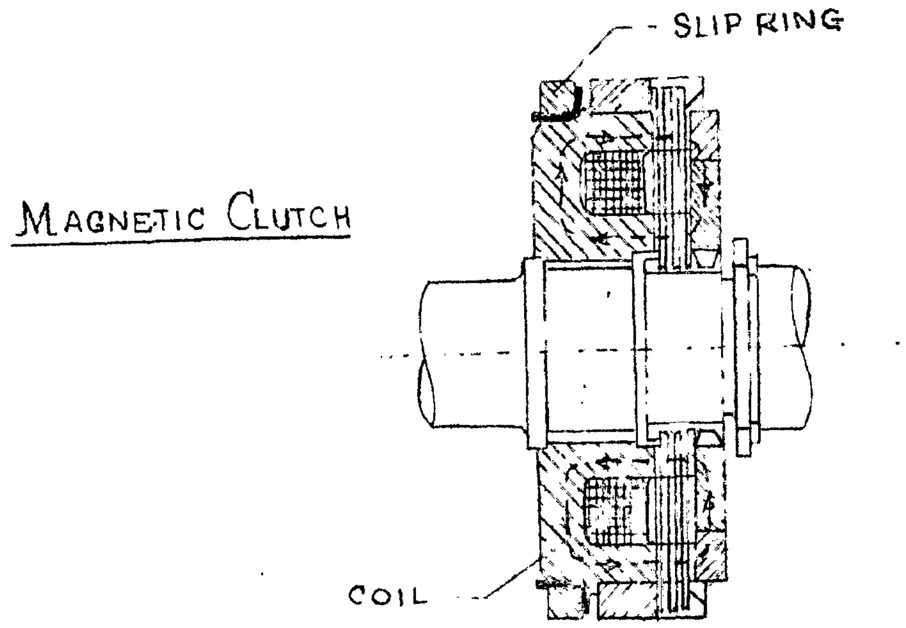
The design of the friction lining is same as has been dealt in the case of plate clutches. The clutching and de-clutching is done by gradually energising and de-energing the electromagnet. When un-clutched there is a clearance of 1/16" between the friction ring and friction lining.

The design of the magnetic coil depends on

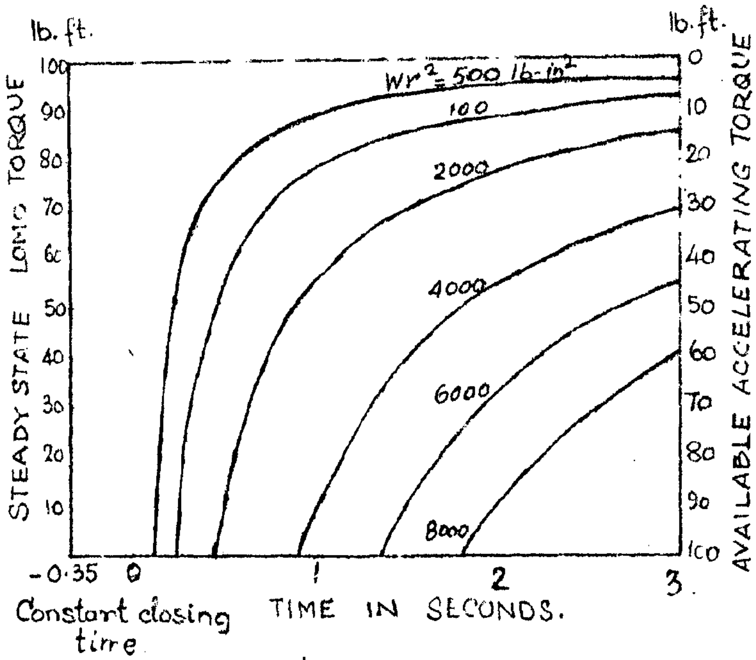
(i) the design pressure of the friction lining.

(ii) the stiffness of the spring steel

(iii) the permeability of the medium



SLIP RING
MAGNETIC CLUTCH
COIL



lb. ft.
STEADY STATE LOAD TORQUE
100
90
80
70
60
50
40
30
20
10
lb. ft.
AVAILABLE ACCELERATING TORQUE
0
10
20
30
40
50
60
70
80
90
100
$Wr^2 = 500$ lb-in^2
100
2000
4000
6000
8000
-0.35
0
1
2
3
Constant closing time
TIME IN SECONDS.

(iv) the number of coils which can possibly be accomodated.

(v) the current which can be supplied.

CLUTCH DESIGN:

The main parts of the clutch are shown. On the clutch body with the imbedded clutch coil, is shown the slip ring, and the brush for conducting current to the coil. Only only one brush is used, as one coil lead is connected to the clutch body.

The magnetic flux path is shown in figure. The flux flows transversely through the leminations, which are made of steel with relatively high permeability. Magnetic flux when the coil is energesed causes a pull between the body and armature. The armaline is forced against the body and compresses the lamination stock. The torque which the clutch can transmit is a function of the

(i) magnetic pull

(ii) coefficient of friction between making surfaces lamination stock .

(iii) No. of making surfaces.

No external operating force is transmitted as the shaft or bearing and these parts are this unaffected by closing force fo the clutch. The magnetic ~~magnetic~~ multiple disc clutch is self compensating for lamination wear.

For the clutch to disengage in reasonable time, laminations must be forced apart after magnetic force is removed. For this reason, one half of the laminations are wave formed and act as spring to force the laminations apart. In closing the clutch, magnetic pull must be greater than the spring tension in order to flatten the laminations and obtain maximmum torque.

CLUTCH OPERATION:

Before the clutch coil is energised the clutch is operating at the residual torque. This torque is present in all laminated clutches. For this clutch resideual torque is approximately 1%. When current is applied, torque increases to actual starting torque, which is maxi mum torque the clutch can start against. This torque is determined by the dyamic coefficient of friction of dry steel surfaces in motion. As speed difference between the lamination surfaces decreases, coefficient of friction increases to static value. Transition between the two Coefficient of friction is not inform but takes place in a violent oscillating pattern until the laminations lock together. This phenomenon makes the magnetic multiple disc clutch abrupt in its closing operations. When soft engagement is a requirement it should not ~~xxxx~~ be used or special consideration should be given to control circuit of the clutch.

On opening the coil the circit current decays very rapidly. However, transmitted torque is mantained for 0.055 seconds because the laminations must be forced apart.

Operation time of the clutch when closing can be divived into two parts.

(i) first is the closing time, which is the time from the energising of the clutch coil until the air gap is closed

(ii) Second is the clutching time which is the time from air gap closing to the time laminations are locked together.

Closing time is constant for any clutch type or rating. Clutch time is variable and depends on the flywheel effects of the parts being accelerated and the steady state load torque the clutch must transmit during starting.

During clutching time load of the clutch always reaches the maximum starting torque. If steady state torque of the clutch is high, torque available for acceleration of the masses to be started is small and therefore the clutching time is long. If useful torque transmitted by the clutch is equal to or greater than starting torque the starting time will be infinite. The time torque relationship is given by

$$t = \frac{wr^2 n}{44300\,(T_S - T_1)}$$

where

t = time in seconds.

wr = flywheel effect of accelerated parts in lbs in.^{2}

n = (n - n) = difference s in speed before and after clutching r.pm.

T_s = maximum starting torque lb. ft.

T_l = steady state load torque lb. ft.

To this clutching time must be added the closing time to obtain the total engaging time of the clutch.

Curves are drawn for a clutch with a rated starting forque of 70 ft. lb. Peak starting torque is 70% higher i.e. 120 lb.ft. but as this ultimate value varie from clutch to clutch a value of 100 lb.ft. has been used in the formula as Ts. giving margin of safety of about 20 to 25 % Curves are drawn for speed differentials of 1000 r.p.m. If the steady state load torque varies between 0 - 70 lb.ft. and the flywhoeel effect is small, total operating time varies very little. Above a steady state load torque 70 lb.ft. The curve flattens out very rapidly and operating time becomes long. As most clutches should not in general exceed a clutching time of 1.5 seconds this clutch is not recommended for use with torques over 70 lb.ft.

These calculations show that the clutch can handle as far as a complete start is concerned but does not indicate how often a start can be made. This is actually determined by how well Heat developed during the start is removed from clutch. Maximum operating tempareture of the coil is 125 C and of the clutch lamination about 200 C. Tempe atures above this limit will cause permanent damage to the clutch.

CONTROL :

The power consumption of these clutches are so small that telephone relays can be used for control. Thôs does not drain too much current from the battery or generator. Also telephone relays can be obtained with various kind of contact combination and time delays.

OILING AND COOLING:

The clutch is designed to operate under oily conditions. Oiling of the clutch can be done by dip, spray or running clutch slightly submerged in oil. Heat dissipation can be materially increased if the clutch is lubricated by using a hollow shaft and radiating oil from hollow centres to passages distributing oil to the inside of the laminations . From there centrifugal force will force oil through lamin tion to outer surface.

SLIP RING AND BRUSHES:-

The problem of transferring current from a stationary brush to a moving slip ~~xxx~~ ring under oily conditions is the main problem which hasn't yet been satisfactorily solved. The oil is an insulator and electric current cannot pass through and oil film. Arcing must in all cases be avoided as it leads to rapid deterioration of both brush and slip ring. A solution is to use brushes made from very fine woven metallic wire mesh and then arrange two brushes concentric with each other. Both brushes make contact with the slip ring and each brush is forced against the slip ring by independent springs. As masses of the two brushes are different, the two brushes have different natural frequency of Oscillations and the possibility of both the brush leaving contact is very remote. Further the outer brush acts as an oil scraper and removes the oil film from the slip ring so that the centre brush makes metallic contact under dry surface conditions.

FREEWHEELING CLUTCHES

Units, also know as one-way clutches or overrunning clutches, are used in many types of machiner. Recent accelerated development of automatic transmissions for automotive vehicles, and mechanical drives for rotary wing aircraft and certain types of multiengine fixed-wing aircraft, have given a major boost to the development of this machine element.

Usually a freesheeling clutch is defined as a form of connection between rotating units that transmits full power in one direction of rotation and none in the other. More generalized, it is a unit which transmits greater torque in one direction than in the other. In the majority of of designs, the greater or driving torce is held as high as possible, while the overrunning or freosheeling torous is held as small as possible. The freeshweeling torous may never disappear cmmpletely, for if it did, the unit would not be ready to re-engage when the speed of the input member tends to exceed the spedd of the driven member.

In most freewheeling drives the direction of rotation is the same in all operating conditions, and only the relative speeds of driging and driven members control the overrunning or driving effect. On an elactric generator coupled to two prime movers by freewheeling clutches, one prime mover may be running while the other one is stoppod. In this case, one clutch is driving and the other is overrunning with its input member standing still. In an automotive torque converter drive, one member of the hydrodynamic unit be coupled to the shaft by a freewheeling unit, and under certain operating conditions. it will then idle along with the freewheeling unit over-running. Helicocopters contain freewheeling units which permit the rotors to turn under the influence of external aerodynamic forces when the engine is throttled and the ship is fliding in a manner comparable to a fixed-wing airplace

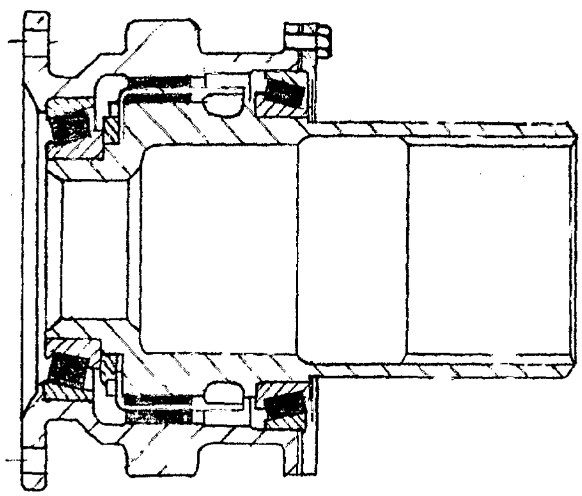
Where the operating conditions of a drive require careful limitation of the stresses in the drive elements, an essential requirement is to reduce or eleminate shock loads and vibrations. Hence, it is of utmost importance to reduce lost motion in the freewheeling unit to a minimum. In the transition from overrunningto driving, the driving member speed should not be allowed to exceed the speed of the driven components. Otherwise, stored kinetic onergy can produce dangerous shock loads in the drivo systom. Tho amount of lost motion in tho driving direction of a simplo ratchet typo unit depends on the touth spacing. In the sprag and Roller typo clutch, or in any other stepless unit, an almost immediate torque build-up takes place and the lost motion is governed largely by elastic deformation of the lutch parts under load.

Possible designs for producing freewheeling effect are almost infinite in number. Howver, as the requirements for optimum functinal characteristics and manufacturing practicability are considered, the number of ~~mxxx~~ suitablo designs narrows to a fow. Only two major systems are currently usod in a large number of applications for high powor ratings and high spoeds.

ROLLER CLUTCH:

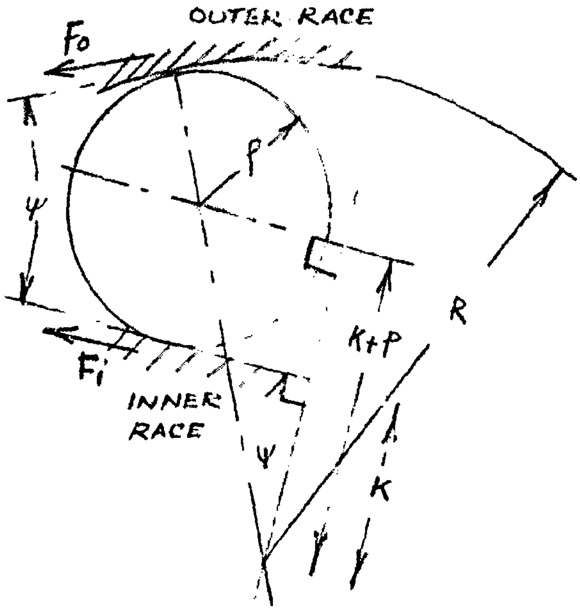
Until a few years ago, the most frequently used freewheeling lutch was based on the wedging roller priniplo. Tho inner member, frequently called the cam, is polygon-shaped. Its flat surfaces provide the tapered spaces for the rollors. Different design approached shapo this part either like a saw-toothed wheel, Fig.', or just like a symmetrical polygon, but oither caso tho effect is tho same. If the camlobes are symmetrical in contour, a rotainor

FIG. 1



FREE WHEELING CLUTCH

FIG. 2



or cage for the rollers must bo provided to limit their motion in the overrumning position and tokeep ~~xxxx~~ them equally spaced. In overrunning position, the rolers must always contact both inner and outer members, so that in case of torque reversal, all rollers respond simultaneously by moving into driving position toward the smaller ond of the tapered space. If tho clutch parts were completely inelastic and no sliding action existed, the smallest deviation from perfect uniformity of engagement would put all the load on one roller alone, However, the effect of inaccuracies is reduced by deflections of the material and the tendency of this clutch to increase the wedging angle with increasing load, as shown in the following anaylysis.

In Fig.2 rotation of the inner race in the direction of the arrow marked "driving" tends to wedge the roller between the outer race and the flat portion of the can, and thereby causes transmittance of torque. From Fgg. 2

$$\psi = \cos^{-1} \frac{k+f}{k-f}$$

Equilibrium conditions for the roller are readily established preparatory to rolating Po and Fo. For the sum of the moments equate to zero,

$$F_o \cdot f - F_i \cdot f = 0$$

$$F_o = F_i$$

Similarly, for verttical forces

$$(P_o - P_i)\cos\frac{\psi}{2} + (F_o - F_i)\cos\frac{\psi}{2} = 0$$

For horizontal forces

$$(P_o - P_i)\sin\frac{\psi}{2} - (F_o + F_i)\cos\frac{\psi}{2} = 0$$

Therefore, For horizontal forces

$$P_o = F_o \cot \frac{\psi}{2}$$

During positiv driving, the torque, Q, on the uter race is

$$Q = F_oR\, n \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where n = number of rollers With p = force per unit of roller length

$$p = \frac{P_o}{l} = \frac{F_o}{l} \cdot \cot \frac{\psi}{2} \quad - \quad - \quad - \quad (4)$$

The limit of the torque capacity is governed by the surface compressive stress of roller and cam. It is determined by the Hertz equation:

$$Sc = 0.591\sqrt{\frac{p.E}{2\,r}}$$

where Sc = surface compressive stress and E = modulus of elasticity.

Expressing the allowable tangential force Fo as a function of Hertz stress, by substitution of equation 5 in Equation 4, gives

$$F_o = \frac{2r}{E} \cdot l \cdot \left(\frac{Sc}{0.591}\right)^2 \cdot \tan \frac{\psi}{2}$$

By substitution of Equation 6 into Equation 3 and with E = 3',000,000 psi, the torque capacity is

$$Q = 1.9' \times 10^{-7}\, Sc^2 Rnl + \tan \frac{\psi}{2} \quad \_\_\_\_\_\_ \quad \ldots\ldots (7)$$

In practical design, the limits of these terms must be checked for all operating conditions. Factors l and n remain constant, but R, p and ψ change under the influence of applied load. Limiting conditions are set by several considerations.

The tangent of the initial angle $\psi/2$ must be less than the coefficient of sliding friction. Otherwise, engagement will not take place. Once the clutch has started to engage, the angle ψ will increase de to increase of R (expansion of the outer race) and due to decrease of p and K (compreession of the roller and cam). Friction then changes from sliding to static, raising the limit for tan $\psi/2$ to the coefficient of static friction. Also, the high compression at the surface will break the film of lubricant, further increasing the static friction.

The rigidity of the parts, particularly cam and outer race, must be adequate to prevent $\psi/2$ from increasing above the allowable limit. In actual design, the procedure is to calculate the expansion of the outer race and compression of inner race and cam, under load, and then to recalculate the new angle, ψ'. By modification of Equation 1,

$$\psi' = \cos^{-1}\left[\frac{(K-\Delta K)+(p-\Delta p)}{(R+\Delta R)-(p-\Delta p)}\right]$$

where ΔK, ΔP and ΔR are the dicnsional changes under load. This equation is best solved by ri l and error, since ΔP ΔK and ΔR are functions of Q and ψ'

The permissible limit for surface stress depends on metallurgical conditions of the surfaces and supporting sections of the parts. A high degree of surface finish is required to minimize wear. Typical design values are as follows:

1. Surface hardness of outer race, cam and rollers: 60 Rockwell 'C' minimum

2. Depth of hard case for cam and outer race: 0.080 inch minimum. Rollers should be through hardened

3. Surface finish: 5 rms desired, 10 rms maximum

4. Allowable Hertz surface compressive stress: 600,000 psi at maximum possible overload; 360,000 is a desired maximum for continuous load

5. Initial engagement angle 6 deg maximum

6. Final engagement angle (under full load): 8 deg maximum

7. Length-to-diameter ration of rollers from 2 to 3

Since designs of out race outer race and cam are governed by rigidity requirements, analysis for tensile and compressive stresses, respectively, is not required. Analysis for surface compressive stress is sufficient.

SPRAG CLUTCH :-

The sprag clutch is composed of circular inner and outer races and irregularly shaped wedging elements, or sprags.

Operation of the clutch can be deduced from Fig.3
As the outer race rotates in the driving direction, each clutch sprag tends to rotate clockwise with respect to itself. Because of the offset centers of curvature, wedging results and torque is transmitted.

Popularity of sprag clutches has grown in recent years for several reasons:

1. Since sprags, the locking elements, are commercially available as standard items, the clutch is economical to design and manufacture.

2. More sprags can be employed in a given space than can rollers; therefore, torque capacity tends to be greater for a given size.

3. Radius of curvature of the sprag can be larger than of the equivalent roller in the same space, further increasing torque capacity.

Geometry of the sprag clutch, detailed in Fig.3, can be defined by the following equations. Application of the cosine law gives for the triangle having the sides, r+p, R - P, and d,

$$\psi = \cos^{-1} \frac{(r + p)^2 + (R - P)^2 - d^2}{2(r + P)\,(R - P)} \quad \ldots\ldots\ldots(9)$$

In the same triangle, with the law for the tangent of half the difference of two angles,

$$\frac{U - Y}{2} = \tan^{-1} \frac{R - r}{R + r} \tan\left(90 - \frac{\psi}{2}\right) \quad \ldots.(10)$$

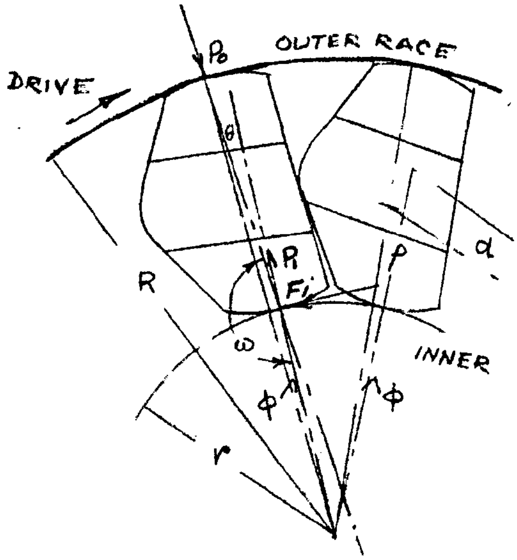
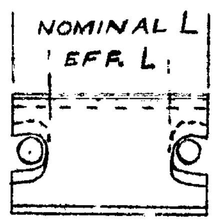


FIG.5.

SPRAG CLUTCH



END VIEW

Also from Fig.3

$$W = 180 - (\phi + \delta) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

and

$$= 180 - W = \phi + \delta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (12)$$

or if   is known

$$= \sin^{-1} \left(\frac{F}{R} \sin \theta \right) \ldots\ldots\ldots\ldots (13)$$

Forces under equilibrium conditions can be related by several simple equations. Since the resultant of F and P must be on the line of action,

$$P_i = F_i \cot \theta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

$$P_o = F_o \cot \delta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (15)$$

From equated moments,

$$F = F_i \frac{r}{R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (16)$$

For small angles, as characteristic for this application sn   is approximately equal to tan   . Therefore,

$$P_o \simeq P \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (17)$$

Torque transmitted by the clutch is simply

$$Q = F_i rn \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

where n = number of sprags. From Equation 14, the pressure per unit of effective length l is

$$p = \frac{F}{l.\tan \theta} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (19)$$

The surface compressive stress by the Hertz equationfor two cylindrical bodies is

$$Sc = 0.C9' \quad P^{\mathrm{s}} \frac{r + p}{2\ rp} \quad \ldots\ldots\ldots\ldots (2)$$

From Equations 10 and 20,

$$F_{i} = \frac{2\ r\ \rho l}{E\ (R + p)} \left(\frac{Sc}{0.59'} \right) \tan \quad \ldots\ldots\ldots\ldots (2')$$

Finally, by substitution of Equation 21 in Equation 18,

$$Q = 1.9' \times \text{XXXX}\ 10 - 7\ \frac{nl\ell r\ \tan\theta}{r + \rho} \quad \ldots\ldots\ldots (22)$$

The limits for the various factors in a practical design are as follows:

1. Surface hardness, case depth, surface finish and limits for surface compressive stress same as for roller clutch

2. Initial angle corresponding to $\psi/\circ$ on the roller clutch: 3 to 4 deg maximum

Comparison: Several significant points mark the functional difference in the roller and sprag clutches.

Under extreme overload, the roller clutch tends to slip. That is, thewedging angle becomes greater than the effective friction angle. When the torque is then reduced, the clutch re-engages and normal operation is restored .

The sprag clutch, under extreme overload, may have one or more sprags turned beyond the largest effective diameter. Then the clutch will lock in both driving and overrunning directions.

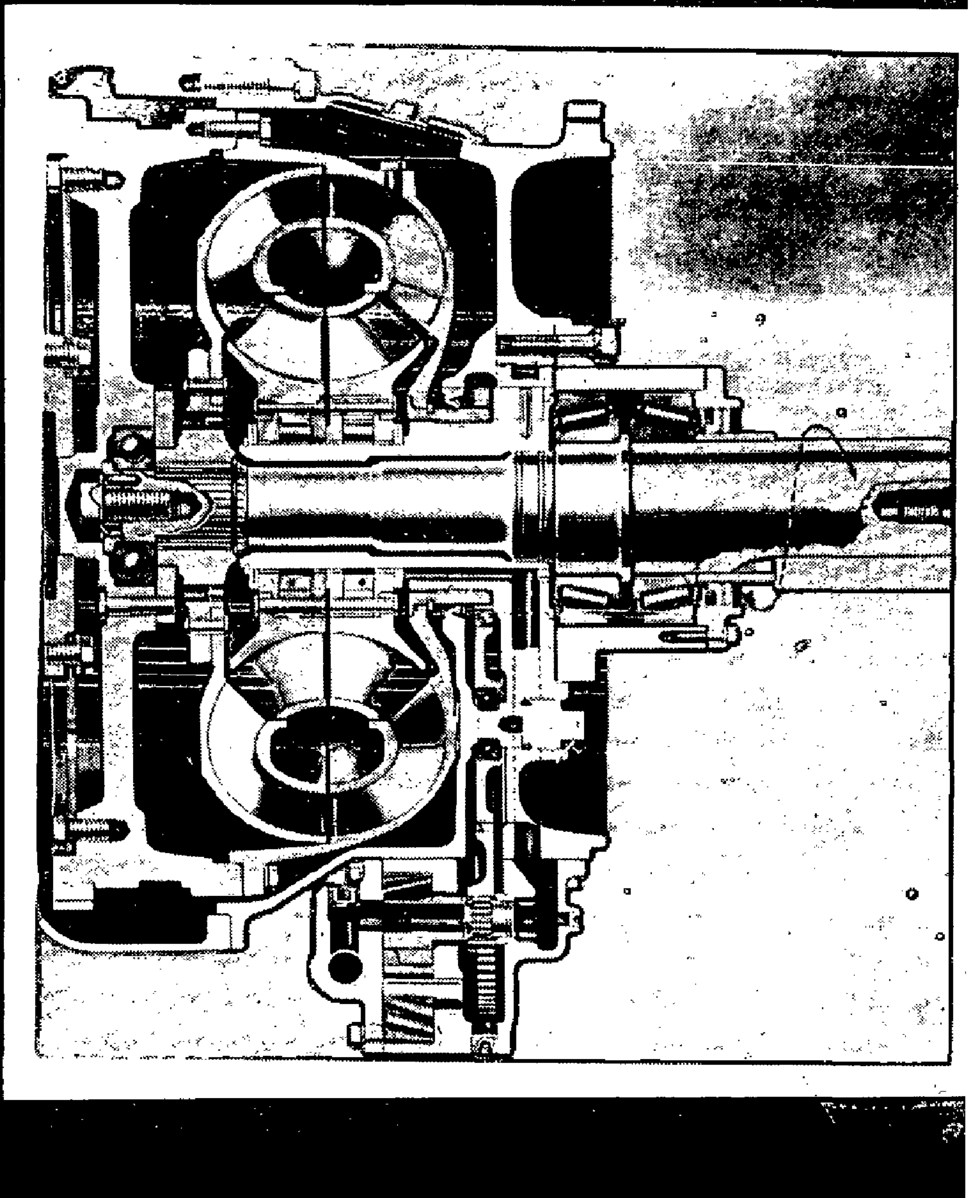
For high-speed freewheeling over extended periods, wear of the roller clutch leaves the rollers round, since they are free to rotate. Contact of the cam on the rollers during overrunning occurs on different points on the cam than when under load. The sprags are not free to rotate and may tend to be flattened by wear.

Since a sprag clutch is smaller in diameter for the same torque capacity, its surface speed in overrunning will be lower than that of the roller clutch, tending to give less wear.

On the roller clutch the outer race should be on the overrunning side, in order to avoid pressure of the rollers against the outer race because of centrifugal action during overrunning. On the sprag clutch, the inner race should overrun, since it is of smaller diameter and has the lower surface speed.

Applications: The field of application of freewheeling clutches has been extended with the development of more intricate automotive and aircraft propulsion drives. The Packard Ultramatic transmission uses a sprag freewheeling unit in the stator mounting of the torque converter. Thus, it provides the equivalent of a gearshift in the torque converter and provides for high efficiency over a wide range of operation. The design principle of the sprag clutch permits machining accuracy and high surface finish without hampering mass production methods. In this application, economy and simplicity of the sprag clutch are decisive.

Where the power ratings are high, from several hundred and up to thousands of horsepower, the number of sprags required would be too great to insure adequate distribution of load among them. For such applications the roller clutch is generally preferred.



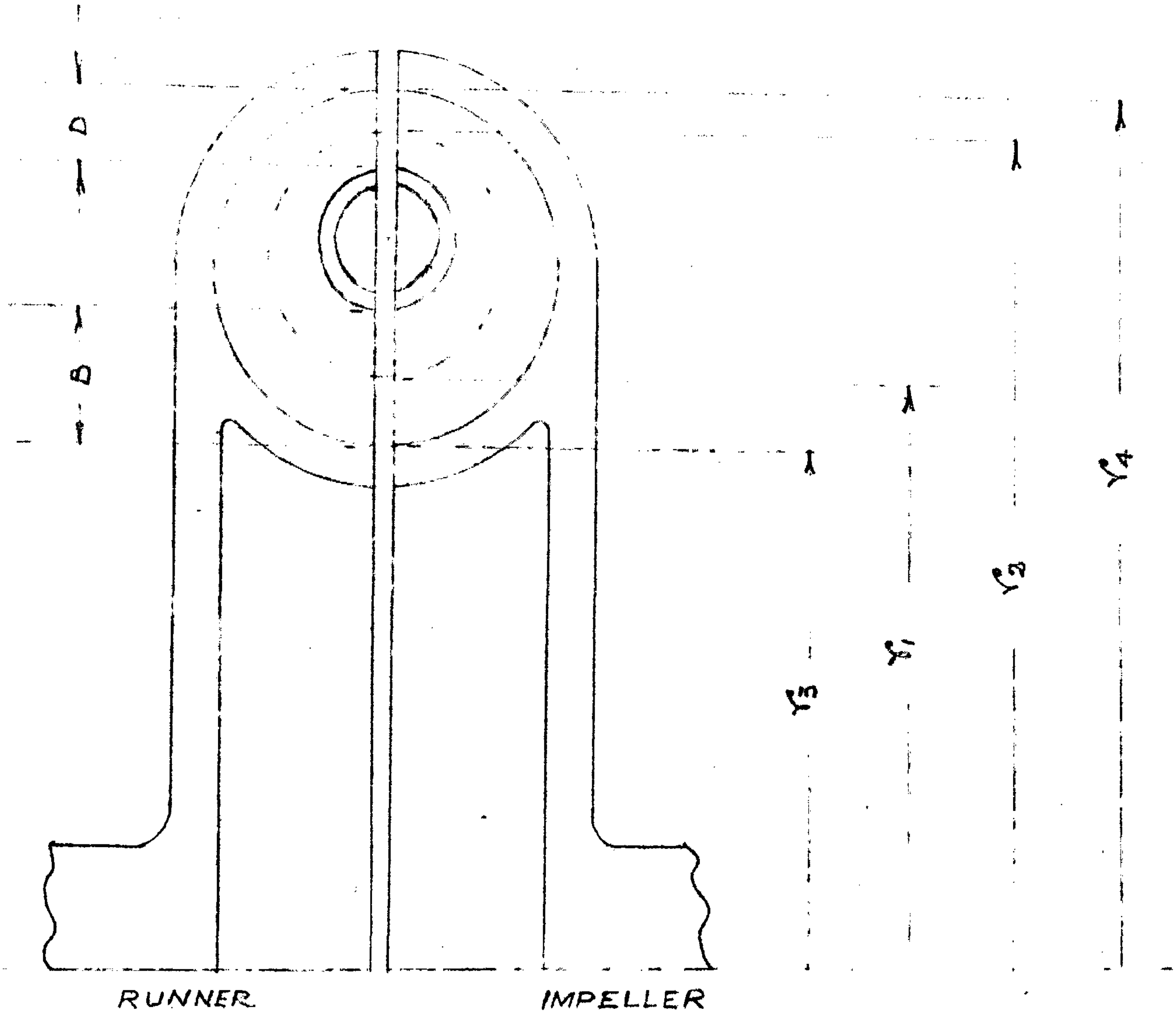
HYDRAULIC CLUTCH

This is used on various designs fitted with automatic transmission systems. It consists of a driving member on the engine shaft and a driven member on the gear box main shaft. The two are separated by a light oil of suitable viscosity. This fluid transmits the drive.

The driving member is integral with the outer casing of the flywheel, which is bolted to the engine shaft. There are a series of cup shaped pockets separated by radial webs on the inner surface of both the driving and driven member. The driven shaft runs on ball bearings within the flywheel casing. There also is an oil retaining ring and spring on the inner side of the bearing to prevent leakage.

~~Such~~ When the car is stationery and the engine is started the rotation of the driving member causes the oil its cells to flow towards their outer periphery. It then passes through the cells of the driven member and comes back to the cells of the driver. In this way the oil starts on a circulatory motion between the cells of the driving and the driven members. In passing from the wols of the driving member to those of the driven the oil loses momentum which is imparted to the driven member and so that also starts to rotate. If both the driver and the driven members rotate at same speed the circulation of oil would stop and hence no more power could be transmitted. This however is impossible since the load on the driven always cau it to lag behind the driver.

At ordinary spees the oil needs but little retardation



HYDRAULIC CLUTCH

to develop the required torque hence the lag or slip between the driving and driven members is insignificant. At low engine speeds, however, the slip can become 100 percent at full torque at that speed thus providing the cordition that the engine can develop full torque in gear without moving the car. This occurs at about 600 r.p.m. From the curve it can be shown that while at low speeds the slip is nearly 100% it falls off rapidly as the spped increases and at normal speed of a car the slip is of the order of 4% only.

The power is transmitted almost wholly through the loss of kinetic engergy caused by the difference in the linear tangential velocity of the liquid at the outlet D of the impeller and the lower tangential velocity at the outlet B of the runner. The impeller horse power is usually greater than that of the runner due to the difference in speed. The torque on both members are always equal.

The couplings are designed to have the cross-sectional area of fluid path constant.

$$2 r \pi B = 2 \quad r \pi D.$$

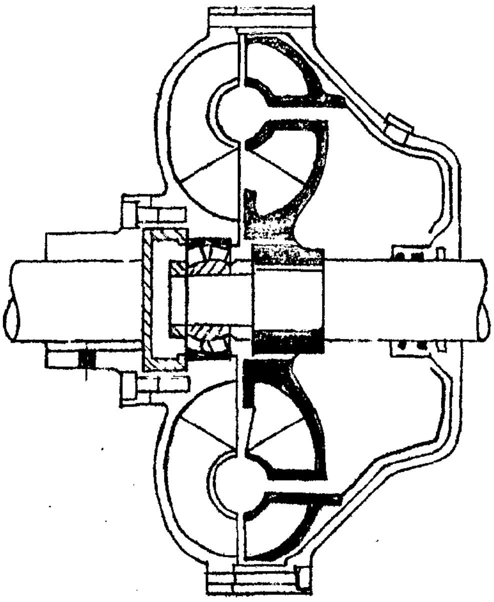
$$\therefore \quad B = D \frac{r}{r}$$

If Q be total volume of liquid circulating in cubic inches.

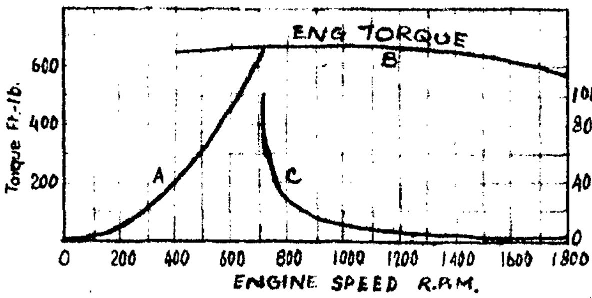
$$Q = a \cdot \frac{2 \pi (r_1 + r_2)}{2} \quad k$$

where a is area of liquid circulating

k is compensation factor for radial blades. Usually taken as 0.85.



HYDRAULIC CLUTCH



A- RUNNER STALLED TORQUE

B- MAX. ENG. TORQUE

C - SLIP AT MAX. TORQUE

The total weight of circulating fluid

$$W = Q.d$$

where d is density in lbs per cubic inches.

The centrifugal force, F(i), on the liquid in the impeller in lbs is

$$F(i) = (\text{Mass of fluid}) \times (\text{radius of rotation}) \times (\text{angular velocity})$$

$$= \frac{W}{4g} \times \frac{r_1 + r_2}{2 \times 24} \times \left(\frac{2 \pi N}{60} \right)^2$$

where N is speed of rotation of impeller in r.p.m.

$$\therefore \quad F(i) = \frac{W(r_1 + r_2) N^2}{282000}$$

Similarly

$$F(r) = \frac{W(r_1 + r_2)n^2}{282000}$$

where n is speed of rotation of runner in r.p.m.

The other formulae used in design are

$$F = F(i) - F(r)$$

$$L = \frac{(r_2 - r_1)}{12}$$

$$H = \frac{F}{A.d.12}$$

$$H = f. \left(\frac{12 L}{D} \right) \frac{V^2}{2g}$$

$$A = 2 \pi r_2 D$$

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$$P = 12VZAd$$

$$KE(i) = \frac{P\ v(i)^2}{2g}$$

$$KE(r) = \frac{PV_r^2}{2g}$$

$$KE = KE(i) - KE(r)$$

$$V(i) = \frac{2\pi r_\circ\ N}{12 \times 60}$$

$$V_r = \frac{2\pi r_\circ\ n}{12 \times 60}$$

$$HP(i) = \frac{KE}{550}$$

$$T = \frac{(HP) \times 5250}{N}$$

where F is net centrifugal force causing vortex circulation

S is specific gravity of liquid about 0.875 average

is absolute viscosity of liquid, poises

L is circuit length, feet

H is hydraulic head, ft.

V is velocity of liquid along path, ft/sec.

v(i) is linear velocity tangential to shaft rotation of liquid at outlet of runner or inlet to impeller ft/sec.

v_r is linear velocity tangential to shaft rotation of liquid at outlet of runner or inlet to impeller. ft./sec.

A is mean average area, transverse to vortex circuit at outlet of impeller, square inches.

f	is friction factor
D	is radial width of vortex path at impeller outlet, inches.
B	is radial width of vortex path at impeller inlet, inches.
Z	is turbulence friction factor
g	is acceleration due to gravity
P	is fluid circulating rate, lb/secs.
KE(i)	is K.E. delivered by impeller, ft.-lb/sec.
KE(r)	is K.E. received by impeller, ft-lb/sec.
HP(i)	is H.P. delivered by impeller
HP(r)	is H.P. delivered by runner to output shaft
T	is torque, lb-ft.

THE ADVANTAGES of hydraulic couplings:-

(1) Prevents the transmission of torsional vibrations.

(2) As there is no mechanical coupling between the driving and driven members, the coupling protects the engine from damage due to sudden shock loads.

(3) Permits rapid declutching

(4) Large clearances between rotating members provide a considerable degree of flexibility and make extremely accurate alignment unnecessary.

(5) The coupling operates with a fixed quantity of oil in the working circuit and does not require an external tank or pump. Heat generated is disipated by radiation.

(6) Wear reduced to minimum.

<u>DISCUSSION</u>:-

Hydraulic coupling has been the cause of the outstanding improvement in the American automobile in the last decade or two.

The automobile engine is fundamentally a high speed power source; at low speeds its power output is small and it is impossible to couple it directly to a stationery load. Even the conventional dry plate must slip until the automobile has attained a speed such that the corresponding input speed of the transmission is atleast as great as the stall speed of the engine; otherwise the engine will die. Probably the greatest advantage of the hydraulic coupling is that it will allow the engine to operate at idling speeds even though the connection between load and power source remains unbroken. Further more it will perform in this manner for long periods without any detrimental effect to itself.

The property of the hydraulic coupling to automatically clutch and declutch itself opens a field of driving which is enteirly new to car operators. In the first place it allows the car to be stopped and held stationery, as at a traffic signal, without either declutching or shifting to neutral. Then it makes it possibel for the engine to acquire ~~xx~~ a speed at which it can supply its maximum torque for the purpose of starting the car - at a time at which a high torque is most needed.

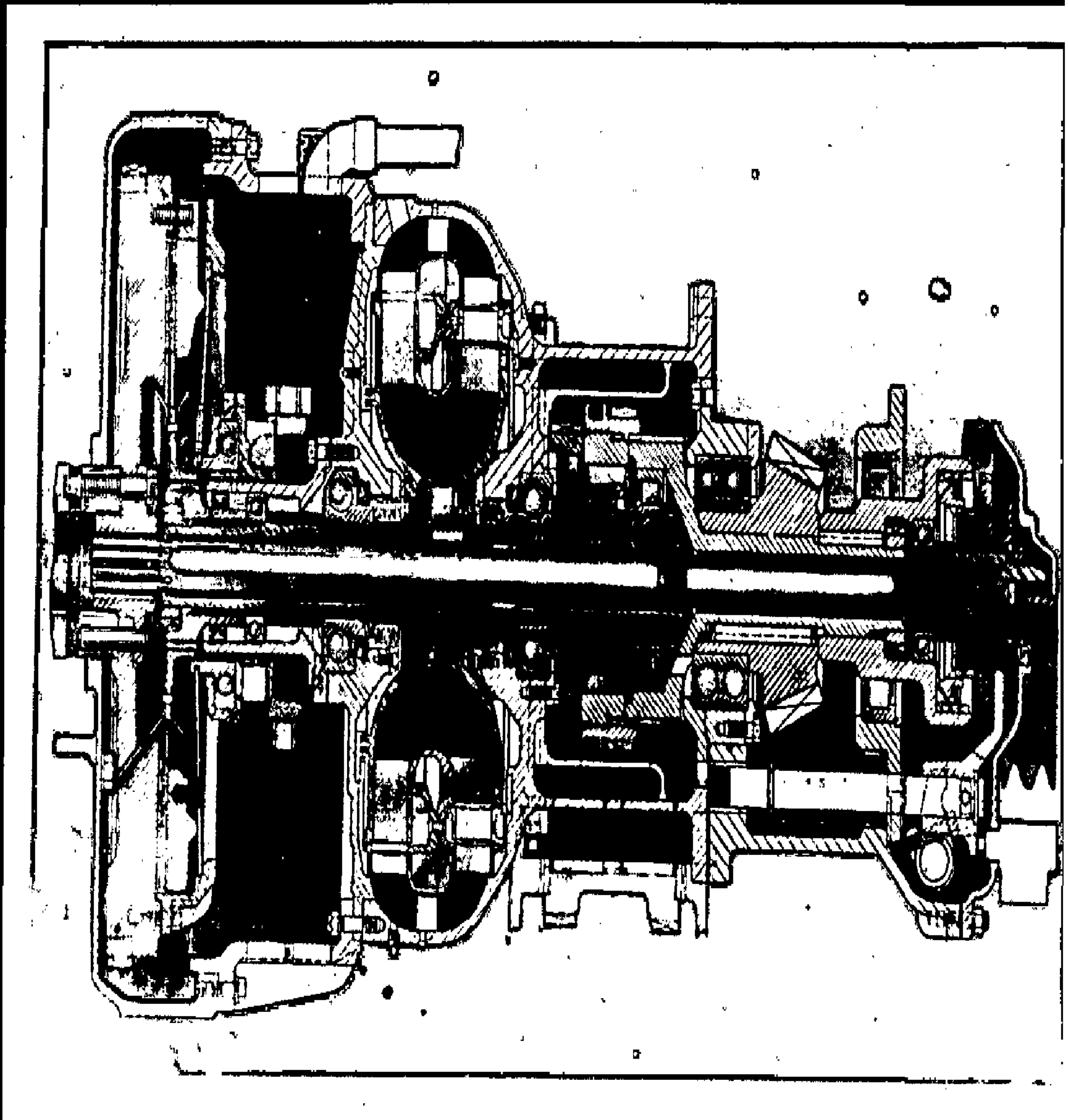
These two points are important as regards car operations specially since there seems to be tendency to depend more upon high-powered engines rather than upon multiple-speed transmission for acceleration and performance. Most of the driving is done in high gear; Specially in America where hydraulic clutches have gain ground most.

For all ordinary starts in a car equipped with a hydraulic coupling it is no longer necessary to have an overall gear ration of 10:1 since the engine is allowed to operate at a speed very near its peak torque output. Thus at the instant of break away the engine is delivering a great deal more torque than is the case with a plate clutches. Because of this fact, former breakaway performances can be equaled by using an overall ration of about 6 to 1. If now we use an combination ofreat axle and transmission which has an automatic shift from,lot us say, an overall ratio of 5 or 6 to 1 to one of 3 or 4 to 1 we shall have enough breakaway torque and yot maintain normal driving gear ration comparablo with present day ovordgfvo.

This transmission can be built so that below 10mrxp.m.p.h the transmission will stay in the top gear or shift into the lower or(kick-down' gear, depending upon throttle operation. In other words, at these low speeds, the driver can make the transmission shift to the lower ration either by releasing the accelerator entirely or depressing it to its extreme position. Between these two extreme throttle position, the car continuos to operate in topgear. Between 10 m.p.h. and 50 m.p.h. the shift the lower gear ratio can be obtained only by drpressing the accelerator as far as possible. Above 50 m.p.h. direct drive is maintained at all throttle position. This together with a manually operated emergency low gear and reverse gear is the new transmission made possible by the hydraulic coupling. With such an unit all normal forward driving can bo done by operating only the accelerator and the breake.

The advantage of the hydraulic coupling installation which is most noticeable to the driver, is that now he can oporate tho brake pedal with his left foot since he does not have to oporate the clutch pedal.

The old difficulty of starting, on a hill is gone since he can hold the car with his left foot and with th transmission in gear, operate the accolerator with hio



right foot; the clutch pedal need not be touched. In this way he always has definite control of the car since he need not release the breake until sufficient torque is applied to the rear wheels to more the car forward. This is a distinct advantage also when operating the car in close quarters, such as parking, where a fine control of speed is required. In heavy city traffic where the present driving method requires that the right foot be moved from breake to accelerator and back to breake again the new left-foot braking system is a difinite advantage. Due to elimination of this tiring operation and since fatigue is a major contributor to trafic accidents it is felt that the coupling is an important safety improvement.

The hydraulic coupling contributes further to safety by making skidding a rarity . The slipping charateristic of the coupling which allows the car wheels to rotate very slowly while the engine is continually applying driving effort, makes it possible to maintain a condition of rolling friction. With the present rigid coupling the rear wheel must rotate at about 40 r.p.m. even in lowest possible gear ratio. This means that, in starting on ice or slippery surface , very careful clutch 'feathering' is necessary to prevent the wheels from reaching this speed instantly, thereby slipping. Of course once the tires begin sliding instead of rolling, the coefficient of friction becomes lower and skidding continues.

C H A P T E R - III

UNIVERSAL JOINTS

1. Types
2. Design
3. Efficiency
4. Modern developments

UNIVERSAL JOINT

Since the gear box, in the type considered, is fixed to the frame directly or indirectly, whilst the rear wheels, their axles and axle casing must be allowed to move up and down under the spring deflection action as the car proceeds along irregular roads. it follows that the connecting member from the gear-box to the back - axle must also follow the movement whilst at the same time transmitting power. This necessitates the use of a device which is employed for driving two shafts inclined to one another at an small angle, and known as UNIVERSAL JOINT. There are various types of universal joints and it can in general be divided into the following groups.

(I) PLAIN UNIVERSAL JOINTS

(2) CONSTANT VELOCITY UNIVERSAL JOINT

(3) RUBBER TRUNNION UNIVERSAL JOINT

PLAIN UNIVERSAL JOINTS: The common cross pin type of joints was originated centuries ago by Cardan and Hooke, and it has been developed into many forms of high mechanical perfection and have been successfully applied where speed and velocity are not critical. Inherent in this design, however, is the undesirable feature of irregular action which cannot be elimminated without abandoning the basic design. Dynamic limitations of these simple joints are best illustrated by analysis of their mechanical performance characteristic, which are substantially alike for all such joints.

For each complete revolution of a Hookes joint operating at a specified angle, there are two positions in which the driven shaft is advanced in rotation relative to the driving shaft and two intermediate positions in which the driven shaft has lagged a similar amount. These advances and lags, alternating twice for each rotation, results in pulsating, variable speed of the driven shaft.

As the joint angle increases, the amplitudes of the pulsations increase at an even more rapid rate until they have a destructive effect upon the joint as well as the parts connected to it. The relation between the speeds of the driver and driven is given by

$$\frac{W_2}{W_1} = \frac{\cos A}{1 - \sin^2 A . \cos^2\theta}$$

where W is angular speed of driver

W is angular speed of driven

A is angle of intersection of shafts.

Q is angular position of driven shaft.

Hence we have

$$W_2 = W_1 \frac{\cos A}{1 - \sin^2 A . \cos^2\theta}$$

The maximum value of driven speed is

$$W_2(max) = W_1 . \frac{1}{\cos A} .$$

The minimum value of driven speed is

$W_2 (min) = W_1 \cos A.$

Honce it will bo seen that above a contain valuo of joint angle the ratio of minimum to maximum speed, which is given by Cos A, is so high as to make this joint inpracticable. In any event theso pulsations are objoctionable sourco of vibrations, noise and rapid wear, ospecially when the inortia of tho connected masses is considerablo.

The porformanco characteristic of Hooke's Joint vary only slightly depending upon their particular construction, but may bo summarised as given in tho figuro. Constant speed rotation of the driving shaft through 360 is roprosented by a circlo with a constant roctor, C, for tho radi s. Driven shaft variablo - spood rotation, on tho other hand is roprosented by tho superimposod concentric ollipso in which tho instantaneous speeds at any given angle of rotation is indicated by the variable length of voctor .

Thoro are four points at tho intorsection f tho ollipso and circlo at which tho speeds of both shafts are matched. Tho included area betwoon ollipse and circlo, compriso the total gain or loss of speed of thedrivon over the driving shaft in a tupical Hooko's Joint. Thoy beingaliko, but opposed, cancol out.

Values for speed 'V', acceleration 'A' and displacement 'd' can be determined by the following graphical method. As mentioned constant speed is represented in a polar vector diagram by the circle described with the unit of constant velo city C as radius. Applying joint angle A_0 the cos OD and Sec OE will be the short and long axis of the superimposed ellipse with vector 'V' representing the variable speed measured in the scale of 'C'. A tangent constructed at any point 'X' of the ellipse will include an angle B with A line through point 'X' perpendicular to ray OX. The value tan B gives the instantaneous acceleration or deceleration at 'X'. The values range between zero at the four terminals of the major and minor axis of the ellipse to a maximum at the points of intersection.

Angular displacement of the driven relative to the driving shaft is the difference between the angles of rotation of both shafts at the same instant expressed as d = F - T where

F is position of driven shaft and T is position of driving shaft. The relationship between F,T and A is given by

$$\tan F = \frac{\tan T}{\cos A}$$

In fifure 1. tanT is shown as length F G and the COSA as OD. Division of FG by OD gives the new lenth FH equal to tan F .

At this particular point of the ellipse the dis- placement angle 'd' is included between rays OG and OH.

In cartesain coordinates the above values of variable velocity 'V' acceleration and displacement 'd' are shown graphically. Angular shaft displacements grow rapidly with increased joint angles. Values at various joint angles may be roughly calculated as shown below:-

Total speed variation = $0.03A^2$% of constant speed.

CONSTANT - VELOCITY JOINTS: Those were designed primarily to overcome certain deficiencies. Such mechanisms produce true constant velocity much like a sot of bevel goars in which the teeth are replaced by an intermediary member adjustable to a ch nge of bevel angle. The characteristic. Feature of this joint is their symmotrical layout with respect to a third member such as the ball and ball cage shown in figure. In most cases tho value of a novel design can be immediately judgod by thepresence or lack of such a thirdmember and / or the symmetry of the adjoining driving and driven members with rospect to the intermediary mombor.

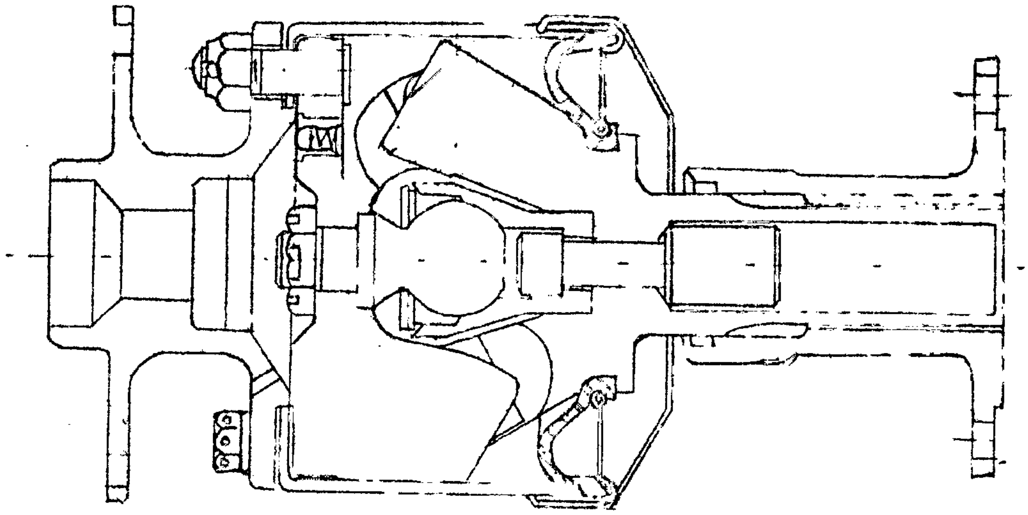
Another group of joint design can be classed as foundamentally of the variable - speed type but, include corrective or compensating parts to convert th variablo into constant speed. Theso joints are how ver complicatod and often sonsitive structuro and are not very popular.

Of the first group, the TRACTA, BENDIX - WEISS and RZEPPA universal joints conform to the mentioned features, namely a driving and a driven member symmotrically arranged about a third transmitting memboroperating in a plane which muchbisect the joint anglo, be perpendicular to the plane of the shafts and located by the true joint contre.

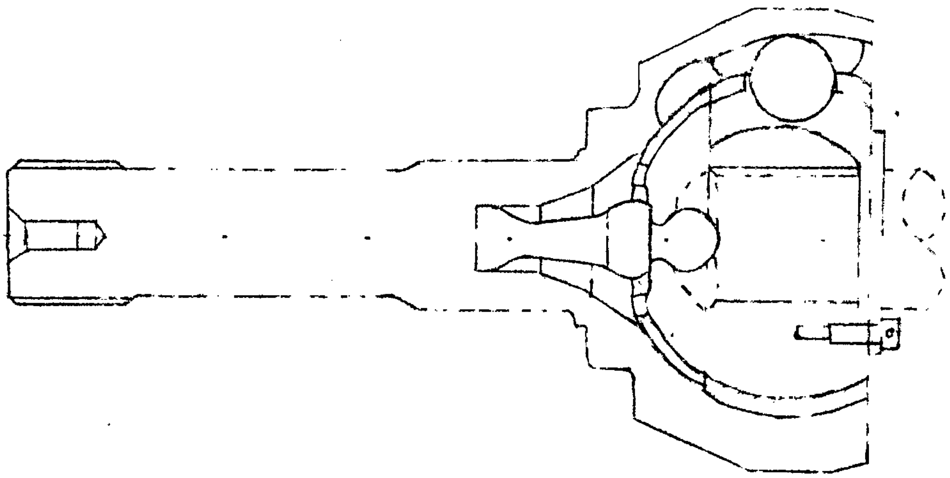
For the TRACTA joint shown in figure the third member is a hinge consisting of a slotted and a spigotted part joined in the manner of an Oldohan coupling, which can be deflected to the required joint angle and will also permit a limited amount teloscoping required for proper functioning. Two identical forks, attached to the respective shafts, engage this hinge as shown in figure.

For the BENDIX and RZEPPA joints the third member comrises a series of balls (4 or 6) respectively) which are retained in grooves on the driving and driven members. Both half of the TRACTA and BENDIX joints must be held in proper aligument by external radial and thrust bearings, mounted mounted in the supports which sourround the joint. with plain journals, the inward thrust is frequently taken by a ball located at the true joint centro, while thrust collars absorb separating thrust of considerable magnitude.

In the case of ball and groove joint developed by F.F. Miller as in figure, the central ball is held in any direction and so makes a strong housing with shaft supporting bearings unnecessary. Similarly, parts of the RZEPPA joint are interlocked in a ball and socket fashion. Therefore no radial or thrust bearings are needed to locate~~d~~/ both shafts with respect to each other. For high angle joints, as in cars with front drive, the half ball grooves of both driver and driven members are so



BALL AND GROOVE CONSTANT VELOCITY JOINT-MILLE



OLD TYPE UNIVERSAL JOINT

WITH PILOT PIN FOR LOCATING GAGE.

curved that the contact condition of the balls in their grooves remain practically constant. When the grooves are straight as in the case of the low angle BENDIX joint a limited axial movement of one end relative to the other is possible and shaft bearings for radial and trust loads are not required. Since the driving balls also act as a shaft support.

Balls are retained in the grooves of the BENDIX joints without any retainer. The half grooves are crossed more or less at all permissible shaft angles so that each individual ball will be theoretically located at the intersection of the groove centres without the help of a special ball cage. If a pair of half-ballgrooves should become parallel by increasing the joint angle excessively, a ball under such condition would lose its definite location.

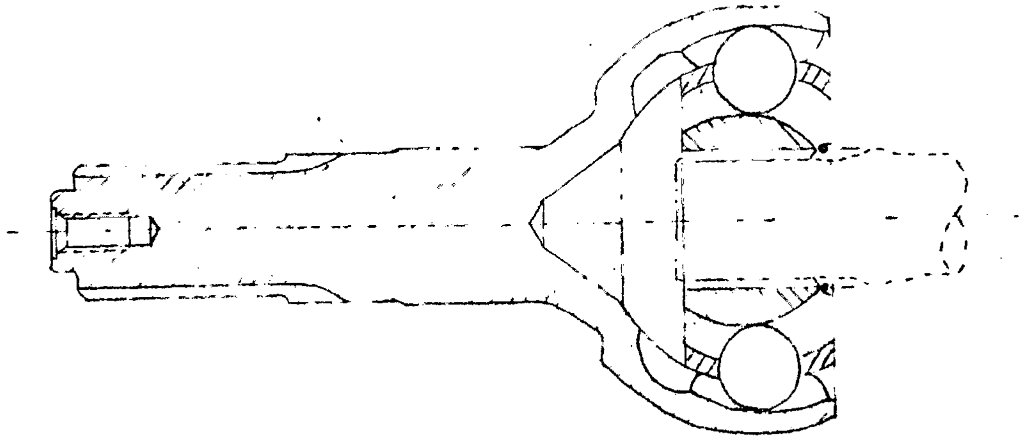
A similar condition existed for the original, earlier RZEPPA joints which were fitted with concentric inner and outer race grooves. However no ball could move independent of the other because of the common ball cage. In order to locate the balls in the proximity of zero joint angles a special piloting device was fitted, consisting of a pilot and a saucer - shaped spherical segment which formed the continuation of the ball cage. The required action of bringing the balls and cage into correct relation with the shaft angle was obtained by means of a pin shape lever having three bearings - in the inner race outer race and pilot . Beyond an operating angle of about 11 degrees this device was not necessary. the crossing of grooves being enough for positive ball and cage positioning.

To-day the matching half ball grooves are longer made from a common centre but from two centres symetrically located and off set an equal distance from the true joint centre so that the half grooves actually converge towards one side of the joint in a wedge fashion, compeling the six balls into the correct position by means of the ball cage. Sufficient off set makes self positioning of the balls so positive throughout the entire angular range of the joint that additionalxxx piloting parts are nolonger required. Currently, however, pilots are still fitted in the largest joint sizes with nominal shaft diameters of 2½ inches to 3 inches mainly for manufacturing reasons and as a safeguard againstxxx excessive stresses at operating angles from approximately 20 to 35 degrees

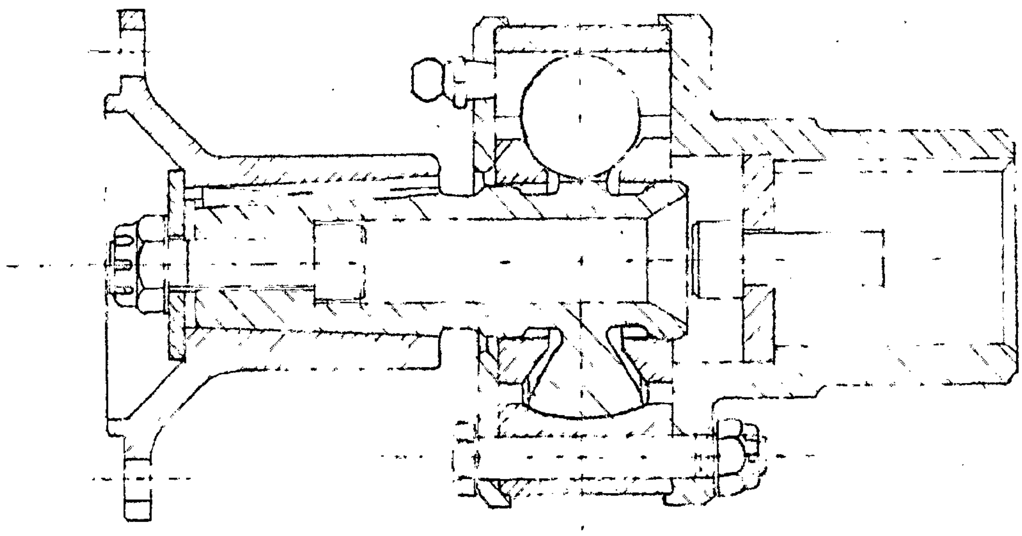
BASIC TYPES: RZEPPA joints are of two types

(1) DISC TYPE
(2) BELL TYPE

They come in nominal sizes corresponding with their drive - shaft diameters ranging from 15/16 to 3 inches DISC and Bell joints of the same nominal size have interchangeable internal parts. The distinguishing difference lies in the shae of the outer ball race and the shaft seal.



BELL TYPE JOINT



DISC joints are furnished with a disc like, short cylindrical outer race having six holes for bolting the joint to a suitable companion flange on one side and to a cover on the other side. The cover is fitted with a flexible diaphragm of oil-resistant synthetic rubber, through which the drive shaft extends, providing Freedom to securing 18 degrees and to slide in or out. These joints are especially suited for high speed propeller shaft drive where vibration is critical. The drive shaft rests, sliding or locked, in the splined inner ball race with its weight supported within the joint on spherical surfaces making external splined connections unnecessary and thereby eliminating misalignment, runout, and consequent vibrations.

Bell type joints are primrily designed as power steering drives for front drive axles or articulated driving axles with independant whellsusponsion. Here the outer driven is shaped spherically (see figure) open on one side with the other sidemerging into a driving shank, which is usually part of the bell shaped forging. These joints ~~are~~ are capabloof 37 degrees deflection. Lubricant seals are generally provided as part of the axle huts housing, but may be furnished separately as non revolving swivel housings surrounding the joint and having a flange for fastering to a suitable face of the wheel huts.

All joints have six driving balls fitted in hardenod andground groovos for transmitting torquo simultanoously in oithor diretion of rolation. A closo fifting ball cage holds the ball in corroct aligumont. The joint assembly has considorablo ond - thrust capacity in either direction and no external supports are required to presorve the aligument of the individual joint parts. Only one shaft oither at driven or driving ond, roquiros support. Tho complote unit can bo handled, mountod and removod from an ossombly without loss of interval alignment or possibility of accidontal disassombly.

JOINT CAPACITY:- Obviously every universal joint has limited capacity for load, spood, angle, and the combination of theso threo factors. Load is gonorally governod by tho pormissiblo prossure in the journals of the cardan's joint, or the flat surfaces of the TRACTA, in rolation to speed, angle and life expectancy.

For ball and groovo joints. loads aro dotorminod by tho sizo and number of balls and thoir distanco from the joint axis. In accordance with ball bearing practico, the pormissible crushing load of balls will vary appoximatoly with tho square of the diamotor. Sincotho ball distanco to tho axis of a joint is a multiplo of its diamotor. pormissible load in congruent design is proportionato to tho cubo of thoir linoar dimonsions:

Allowable ball pressures are experimentally determined with due regard to groove curvature, surface and contact conditions. With increasing joint angles contact conditions require a reduction of pressure in order to maintain the standard life expectancy under normal conditions. Standard life expectancy is based on a fixed number of cycles which the joint will endure without excessive wear and is generally expressed in work-hours. It is identical for all kinds of joints under standard condition of load, speed and angle. For any working condition different from standard the expected life can be predicted by the formula

$$L = n^2 \left(\frac{c.e.k}{p}\right)^3$$

where	n	is speed r.p.m
	c	is normal rated load capacity, h.p per 100 r.p.m.
	p	is load h.p
	e	is angle factor
	k	is speed factor

Centrifugal forces exerted by balls on the groove surface at speeds far above normal will some what reduce load ratings (factor K)

Overheating due to internal friction or vibration caused by inadequate supports limits the speed or angle at which a joint may be safely run. As a rule, the product of speed in r.p.m. and angle in degrees should not exceed a constant limiting figure set approximately at 16,000.

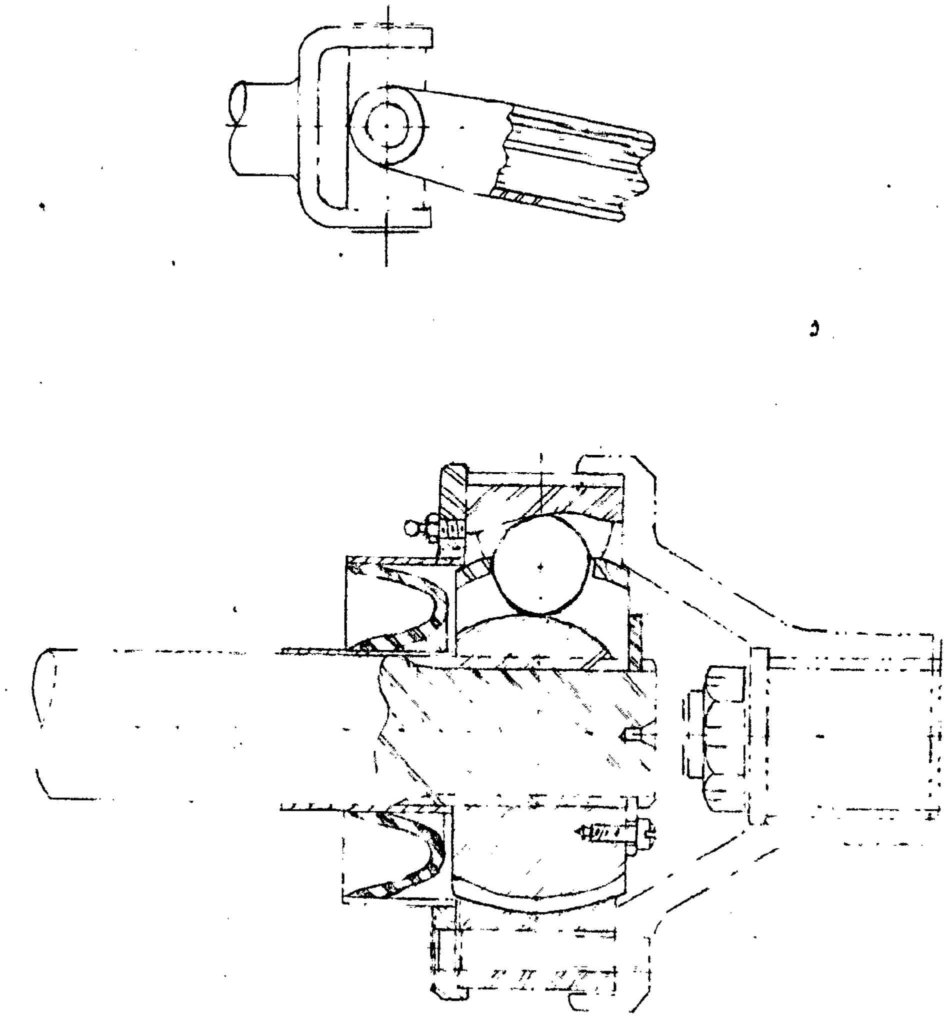
EFFICIENCY:- Universal joint efficiency is high; losses are chiefly due to internal friction and for some types are so slight that they are not easily measured. Load tests with ball type joints operating at angles larger than 25 degrees at full load and low speeds have shown losses of about 2%, diminishing with the angle. There may be additional losses for CARDAN joints operating under critical conditions where the variable speed will produce inertia effects detrimental to efficient power transmission. For instance, such variable speeds, necessarily resulting in variable torques, may produce stalling effects under critical conditions as may be encountered in front drive vehicles negotiating steep, sharp turns requring full steering lock. Strictly speaking, these cannot be called mechanical losses but have some effect. Nearly all losses are converted into heat and hence sufficient cooling must be provided to prevent seizure of parts. At low speeds this condition is installed not critical and joints may be installed in closed housings which also serve as reservoir for lubricants. At high speed however such housings usually hinder cool ng of the joint, thereby reducing its capacity unless proper ventilation is provided. At high speeds closed housings are usually not preferred.

MODERN DEVELOPMENT:- Recent inovations in ultra-high speed engines transmissions and shafts have brought newly developed types of universal joints capable of transmitting loads at smmall angles and higher speeds than was heretofore considered safe. A very successful joint of this type is shown in figure. This constant-relocity joint has a ~~xxx~~ number of balls placed in straight half grooves o the inner and outer race. Both races are interlocked by a spherical bearing which absorbs both radial and thrust loads. Here again the poinciple prevails that the driving balls must be located on the plane bisecting the shaft angles. This exact lo ation is controlled by two conical pilot rings seated spherically on the inner race ~~xxx~~ shaft and guided by end enclosures of the joint. With the shafts at an angle, both pilots are displaced in opposite direction and locate the balls occurat ly between the conical surfaces.

Due to reduced internal friction and perfect balance, these joints can safely operate at speeds much higher than permissible for the earlier types. Since mostly the angles required are small, the joint is designed for a shaft clearance angle of 7 to 9 degrees. When larger angles are required at lessor speeds, the clearance can be increased to approximately 12 degrees by certain modifications of the pilot guiding surfaces. In the combination of joint angle and speed the limits are set by vibrations, heating and sealing of lubricants but are much higher than the limits previously given.

Regardless of the mthod of suspension, the wheels of articulated driving axles are generally connected by jointed shafts to a differential housing contrally mounted on the spring supported frame or body of the vehicle. In order to avoid excessive angles at full spring deflections it is important to provide for the reatest length of connecting shafts. This is particularly important for steering joints because the up or down movement due to spring deflection will deduct a certain amount of available steering angle. The compound angle may be roughly calculated as the square root of the sum of the squqres of the component angles.

A solution for gaining maximum length for half axles is shown in Figure. In this design the inboard universals are developed as part of the differential side rears contained in the differential housing to form a very compact unit. For nonsteering rear axles wheel joints may be mounted on the outside of the wheel hub, giving additional shaft length and easy accessibility.



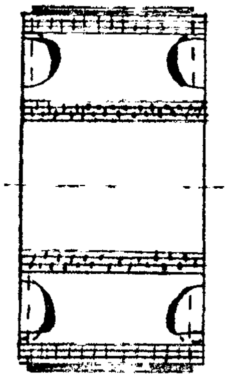
CONSTANT VELOCITY DISC JOINT

FOR SEVERE SERVICE.

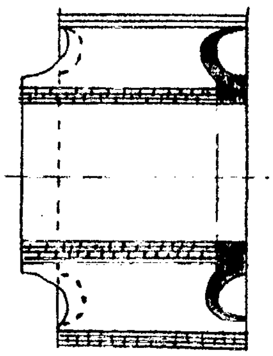
RUBBER TRUNNION UNIVERSAL JOINT:- Another type of universal coupling, the layrub one, provides for all relative movement at the two ends of the propeller shaft, by means of compressed rubber trunion units mounted in the end couplings The advantages of this coupling are that it has no wearing metal members, it is silent in action; requires no lubrication attention and if properly fitted has an exceedingly long useful life. Moreover it provides an elastic damper form of drive, insulating the engine and gear box against transmitted driving shocks via the rood wheels.

The principle of the coupling trunnion block is shown in figure. The bore of the block is reinforced by liners of high tensile wire cloth, formed by winding it tigertly on a mandrel and securing by cold soldering. The screen to then inserted into a mould and the rubber cylindrical member vulcanised to it. For attachment of the blocks to the driving flanges, shouldered sleeves with spigots for fitting into recesses on the flanges are pressed ino the screens, the whole being clamped together with a nut and bolt. The bolts ae subjected to tensile stress only, the shearing stress due to the drive being taken of the large-diameter spigots formed on the sleeves.

Usually, the resistance deflection in the axial, direction is about to of that across the axis of the rubber block, so that there is ample axial displacement to avoid fitting of the usual splined sliding joint. The four rubber



RUBBER TRUNNION BLOCK.



block, so that there is ample axial displacement to avoid fitting of the usual splined sliding joint. The four rubber block unit is commenly employed on motor-cars and all but the largest commercial vehicles; six block units are fitted in the latter case.

CHAPTER - IV

PROPELLER SHAFT

1. Description
2. Design

CHAPTER - VI

PROPELLER SHAFTS

VARIATION in angular velocity between two universally jointed shafts, thot is providing the joints are of the Hoke tupo and not constant-ratio joints, depends upon angularity between driver and driven members. Each of the joints has two rocking axes, each of which must maintain anglo of 90° one to the other at all times. Incorrect angular relation between the driving anddriven fo ks is the origin of many transmission streds es and vibrations. It is therefore of primary importance that each fork or pin at one end should be mounted in the same plane as its corresponding member at the other end; even then, if there is any angul rity in the shaft, uniform revolution will still not bo possible, as the fluctuation in speed varies proport- ~~nllg~~ ionally with the shaft anglo.

When a joint of the type in Fig 1 is in motion the pins describe wh t is termed two "gre t cicles" having common diameters,since they represent the boundaries of the sections formed by the two paths. As the distance between A and B is const n it may borepresented by a quadrant of the gre t circle. The two ares set out in Fig are desc ibod whilst pin AB travols between tho two points of intersection, and the point of maximum doviation from the vertical occurs when coicidencewith those intersections is attained by either pin A or B. ~~Hhhm~~ When A is coincident with those intersection points, angular volocity of the drivenshaft

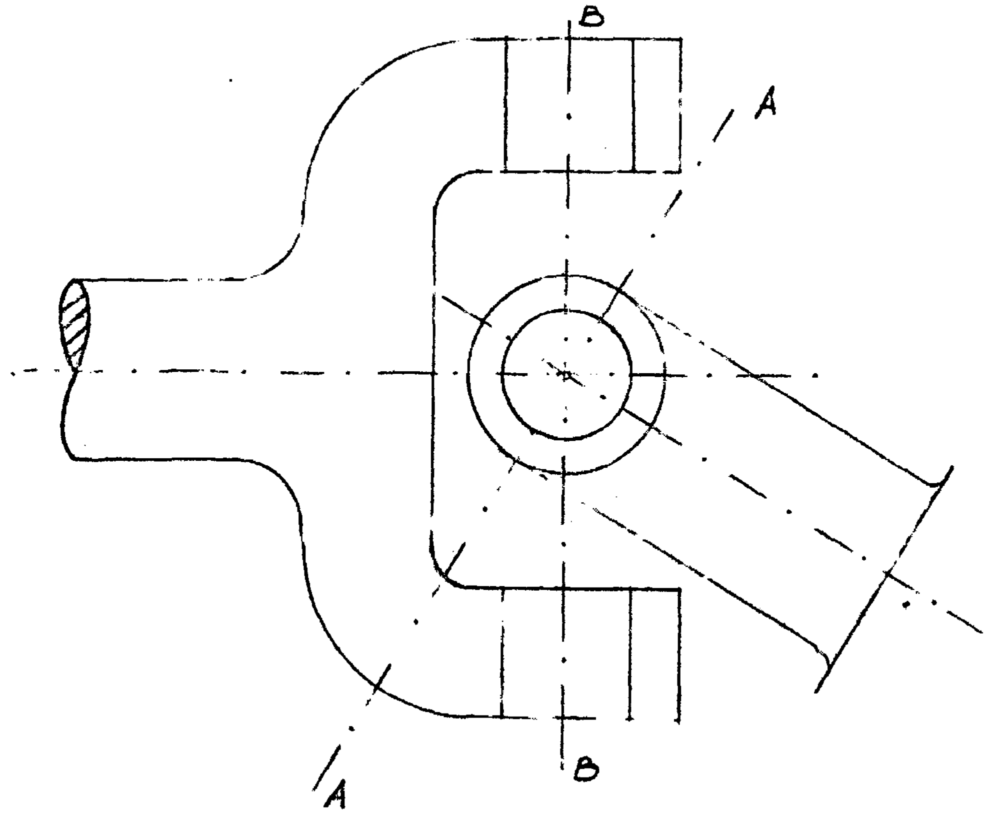


FIG. 1

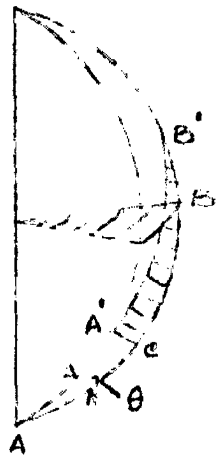


FIG. 2

is less than that of the driving shaft, and the opposite condition occurs when B is in coincidence. As the joint rotates there are four points at which the speeds of each shaft are equal.

In Fig point A has moved to A' and B to B', which point is found by describing an arc equal in length to AB. From B' point C is found at the point of intersection of a further great-circle quadrant $B'C$ being equal to $B'A'$. This geometrical representation may be interpreted thus:

θ = the angle between shafts CAA', the angular motion of driving shaft is equalled by AA', whilst AC = angular motion of the driven shaft, and finally $B'A'C = 90^\circ$ as does $B'CA'$

$\cos\theta = \tan a \cot b$, where $AA' = b$ and $AC = a$

by reciprocals $\dfrac{\tan a}{\tan b} = \cos\theta$

and for small angles $\dfrac{AC}{AA'} = \cos\theta$

Solving for the ratio of angular velocity

$$\tan b = \cos\theta \tan a$$

and by differentiating $\sec^2 b\,db = \cos\theta \sec^2 a\,da$.

$$\therefore \quad \frac{db}{da} = \cos\theta \frac{\sec^2 a}{\sec^2 b}$$

As AC or b is not known, by squaring the expression for angular velocity.

$$\tan^2 b = \cos^2\theta \cdot \tan^2\theta .$$

$$1 + \tan^2 b = 1 + \cos^2\theta \cdot \tan^2\theta .$$

$1 + \tan^2 b = \sec^2 b$ Substituting in the diferentiation of the angular velocity expression

$$\frac{db}{da} = \cos\theta \cdot \frac{\sec^2 a}{1 + \cos^2\theta \cdot \tan^2 a}$$

which is the final ratio of angular velocity between driving and driven shafts.

If the figures are worked out for various degrees of angular motion, and shaft angles and the speed fluctuations plotted, it will be noted that the speed differences between thetwo shafts is considerable. If the angle is increased from 8 to 24 the 2 per cent vatiation becomes 18 per cent and therefore it will be readily understood that lack of uniform velocity combined with change of direction of the rotating pins is a possible source of shaft vibration.

Vibration of the shaft may be excited through either end loads, centrifugal force or torsional vivration. Should the shaft be fitted ~~centrifugal force or torsional~~ without a sliding muff considerable end thrust is experienced and, further, if the joint is of flexible type, the texture of flexible material will damp the actual shock load, but reduced the critical speed of the shaft tube. It is desirable therefore not to neglect end loading when making propeller-sh ft calculations.

The maximum bending moment on the shaft: According to Prof. Greenhill this may be stated as

$$f = \frac{B.M}{Z} + \frac{F}{A}.$$

where F = end load

A = sectional area

W = weight of shaft (lb per unit length)

α = velocity (radians per sec)

B = flexural rigidity of the shaft or EI

y = deflection at X from the centre

and load due to centrifugal force = $\frac{W}{g} \cdot \alpha^2 \cdot y$. per unit length

Now if the effect of gravitation be taken into account there is noe position in each revolution at which centrifugal force and weight exert their greatest effect

and $$\frac{d^4y}{dx^4} + \frac{F}{B} \cdot \frac{d^2y}{dx^2} - \frac{W\alpha^2}{gB} \cdot y - \frac{W}{B} = 0$$

which solve into

$$y = A_1 \sin\beta x + A_2 \cdot \cos\beta . x + A_3 \cdot e^{\gamma x} + A_4 e^{-\gamma x} - \frac{g}{\alpha^2}.$$

and being the two roots of quadratic equation

$$\beta^2 = \frac{F}{2B} + \sqrt{\frac{F^2}{4B^2} + \frac{W.\alpha^2}{gB}}$$

$$\gamma^2 = -\frac{F}{2B} + \sqrt{\frac{F^2}{4B^2} + \frac{W.\alpha^2}{gB}}.$$

y has the same value for values of x whether positive or negative. $A_1 = 0$, $A_3 = A_4$ and assuming that Y = o and X = L, that is for no shaft deflection, and also that when X = L $\frac{d^2y}{dx^2} = 0$ then

$$A_1 = 0$$

$$A_2 = \gamma^2 g / \alpha^2 (\gamma^2 + \beta^2) \cos\gamma L$$

$$A_3 = A_4 = \beta^2 g / 2\alpha^2 (\gamma^2 + \beta^2) \cosh \gamma L.$$

$\therefore$ at any point along the shaft the bending moment is and the B.M. at the middle of the shaft is

$$EI\left(-\beta^2 A_2 + 2A_3 . \gamma^2\right)$$

TO FIND THE LOWEST CRITICALSPEED

$(\beta^2+\gamma^2)A_2 . \cos\beta L = 0$ and since $(\beta^2+\gamma^2)$ is not zero unles $A_2 = 0$, $\cos\beta L = 0$

Under stable conditions $\beta L = \pi, 2\pi, 3\pi$ - - - .
and the lowest critical speed $\beta = \frac{\pi}{L}$

$$\beta^2 = \frac{\pi^2}{L^2} = \sqrt{\frac{F^2}{4B^2} + \frac{W\alpha^2}{gB}} + \frac{F}{2B}$$

$$\therefore \quad \alpha = \sqrt{\frac{gB}{W}\left(\frac{\pi^4}{L^4} - \frac{\pi^2 F}{L^2 B}\right)}$$

~~from which~~

The above formulae do not take into consideration any inaccuracies in weigh t distribution but they assume that the weight acts as a radial force in the same direction as the centrifugal force. Should the shaft not be perfectly balanced, the shaft will deflect towards the heavier side and a point will be reached when for a period there will be an excessive bibration . It will, however, disappear with any alternation in shaft. speed.

At this critical speed bibration is caused by a change in the axis of rotation, andthe shaft, instead of jotating around its gommetrical centre, rotates about an axis through the centre of gravity and its geometrical centre. It therefore becomesdeflected in su h amanner that the geometrical centre traces a circular path around te centre of gravity of rotating mass at each rvolution.

The ~~shaft~~ whirling speed may be defined as that at which the stiffness is zero and its period infinite. The interval of time in which the shaft is passing through its period infinite. The interval of time in which the shaft is passing through its critical value is so small that it does not permit of large shaft deflections, hence the fact that repture does not always occur.

In vibrations tho shaft, starting from rest, increases its angular velocity up to the point where vibrations occur, the amplitdo of course being small. Further increas the vibration amplitudo until eventuallythey reach a ma ~~shaft~~ still further speed increase eliminates these tions, until at another shaft speed the gibration mmence and ~~shaftx~~ start the cycle once more. S increased until shaft distortion and fractu

Torsional resilience must be cons should bo designed to accommodato an

$1\frac{1}{2}^\circ$ per 10 ft of shaft, which figure should be a maximum at maximum stress. Previous calculations should be subjected to a re-check 16 ensure this condition, and also that of adequate elastic strain energy, as the material should not be stressed torsionally above its elastic limits.

The shear resilience of a tubular shaft is $\frac{x}{2\pi} . 2\pi . L . r . t$

where $x = \frac{r}{R} \cdot f_s$. and where f_s is the intensity of shear stress at outside radius R

t = thickness of material

r = inner radius of shaft.

The torsional resilience of a tubular shaft is:

$$\frac{\pi L}{N} \cdot \frac{f_s^2}{R_1^2} \cdot \int_{R_2}^{R_1} r^3 . t = \frac{\pi L}{4N} \cdot \frac{f_s^2}{R_1^2} \cdot (R_1^4 - R_2^4)$$

$$= \frac{R_1^2 + R_2^2}{R_1^2} \cdot \frac{f_s^2}{4N} \text{ . vol. of shaft.}$$

where N = modulus of torsional rigidity

R_1 and R_2 = inner and outer radius of tube respectively

End thrust to which the shaft is subjected arises from many sources. It has its origin partly from a combination of resulting loads from vertical displacement of the axle and gyroscopic movement of the joints. which tend to substantially reduce the natural hamonic vibration of the shaft - when the interval of time between blows from road obstructions or inequalities is equal to the periodic time of one or more road springs. These combined loads have application at the rear end of the shaft and all act through the rear axle end

$e = \frac{AE}{L}$, which is the elastic force per unit of deflection

where W = combined end loads

A = cross-sectional area of shaft

L = length between joints

E = modulus of elasticity

g = gravity

K = torsional rigidity of shaft

k = radius of gyration.

The frequency of torsional vibration, $n = \frac{1}{T}$, where T = time of torsional vibration, and is equal to $T = 2\pi\sqrt{\frac{I}{K}}$ I beig the ~~xxxxx~~ moment of inertia $\frac{Wk^2}{g}$ or $n = \frac{1}{2\pi} \cdot \sqrt{\frac{K}{I}}$ per second

If K = torsional rigidity of shaft and N = modulus of transverse elasticity, $K = \frac{N\pi \cdot d^4}{32L}$

substituting $n = \frac{1}{2\pi} \cdot \sqrt{\frac{N.\pi d^4 . g/32}{Wk^2 . L.}}$

or. $\frac{d^2}{20}\sqrt{\frac{N.g}{W.k^2L}}$ per second

which for steel tube at 12×10^6 approximately gives ~~whichxfoxxxxxxxx~~

$$n = 3400\, d^2 \sqrt{\frac{I}{Wk^2 L}} \text{ per sec.}$$

If the shaft is built up of varying sections of different diameters the freqoncy of vibrations is of course different as the twist produced by unit torsion moment is the sum of ~~the~~ twist in each system. An examplo of such a shaft is the swaged onds down to a small splined shaft diamotor.

Frequeency may be found by a modification of Morley's formula.

$$\frac{32}{\pi n}\left(\frac{l_1}{d_1^4}+\frac{l_2}{d_2^4}\right)$$

or written for stoel as

$$n = 3400\sqrt{\frac{I}{Wk^2 \Sigma\, l/d^4}}$$

above with note ion as Fig 3.

Incidentally, when determining the material safo stress it must not beoverlooked that most of the stresses are alternating and a e often reversed, whilst it is moro dangorous still to fail to appreciate tho fact that it is the stress range which has an even greater bea ing upon failure than does the actual magnittudo of the stress. Considering stress recersals, a safo material figure is 7½ tons compression. 7½ tonstension and 5½ tons in shear duo to torsion. This is based on an ordinary 30-45 carbon stoel of 15 to 22 tons yield and an approximato elongation of 10 to 18 per cont.

A graphical illustration of vibration phenomena is shown in Fig. Tho shaft starting from rest gradually increases its angular velocity to a point where bivration occurs.

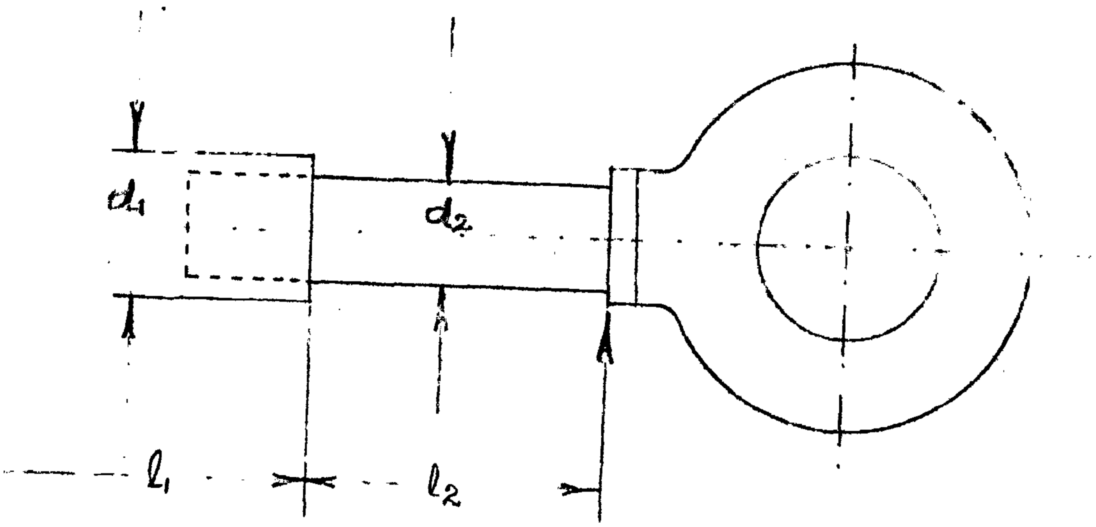
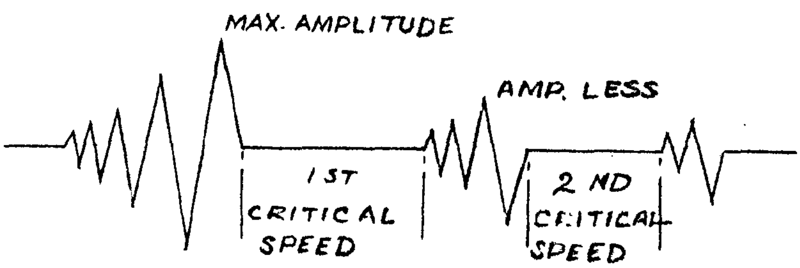


FIG. 3



VIBRATIONS

FIG. 4

The amplitude is at first small, but with an increase in speed the vibrations also build up until they eventually reach the maximum amplitude. If the shaft is on a balance machine the indicators will register the magnitude, when suddenly the indicator become at rest. It is at this speed that critical speed of the shaft has been reached. Any further increase in revolutions reproduces further vibration, but of smaller magnitude. Such phenomenon occurs with each succeeding speedincrease until eventually the shaft becomes distorted and probably fails.

Stresses such as these, if applied in magnitude exceeding the elastic limit of the chosen section, produce an effect on the material structure such that repture due to tension or compression resolves itself into ultimate failure through shear.

CHAPTER - V

VEHICLE MOVEMENT

1. Types of Resistance
2. Tractive Effort.
3. Geometrical Progression
4. Gradient Performance And Acceleration

VEHICLE MOVEMENT

The mechanics of a moving vehicle are confined primarily to simple calculations from accepted formulae. In considering a wheeled vehicle three main factors are concerned -

(a) Rolling and frictional resistance

(b) Gradient resistance.

(c) Air or wind resistance.

Rolling resistance varies considerably with the type of road surface as indicated in the following table -

Road surface	Resistance lb/ton
Railroad	10
Good Asphalt	15
Medium Asphalt	22
Poor Asphalt	29
Wood Paving	30
Granite Sets	35
Best Macadam	45
Ordinary Macadam	50-60
Soft Macadam	97
Well-rolled Gravel	57
Small Cobbles	60
Medium Cobbles	130
Large Cobbles	240
Hard Dry Clay	100

Sand Road	360
Loose sand	560

An average figure which appears to give a little in hand for all general purposes is 60 lb/ton. This, of course, does not apply to specialised military or agricultural vehicles. Rolling resistance for cord tyres is approximately 33 per cent less than that for fabric tyres, and the figures are practically constant for speeds of 20 to 60 m.p.h

Frictional resistance is another variable factor. It includes resistance to motion through transmission losses, such as gear efficiencies, oil churnge churning, tyre adhesion and many other influences. A useful general approximation is-

$$F = 30 + 0.012\, W$$

Where f= frictional resistance in lb

W = total weight of vehicle.

Transmission losses are usually estimated at 10 percent in direct gear and 15 to 20 per cent in low gear, so that in calculating performance, tractive effort should be estimated at efficiencies of 90 percent top gear and 85-80 percent in lower gears, as an average. This figure includes such losses as occur in oil churning in a closely designed gearbox and confined rear axle. It also takes into account temperature losses. For private cars these figures are low, 85 to 89 per cent being the usual figures assumed. The tryes is about two-thirds that of the total chassis loss.

GRADIENT RESISTANCE

Gradient-resistance figures must be added to those for rolling resistances in compling the summation of total resistance to motion. Gradient resistance is a function of vehicle weight and gradient, and it does not depend on vehicle

Speed It may be expressed as $P = \frac{W}{G}$

where G = gradient

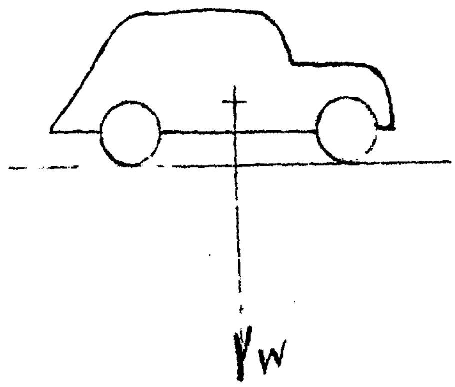
W = Weight of vehicle

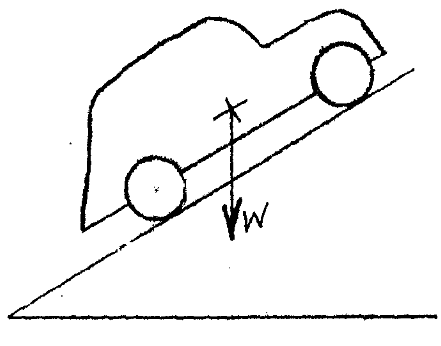
W sin = P if the gradient is expressed in angular dimensions.

Wind or air resistance is dependent upon speed and is calculated from the formula -

$$p(n) = p(h) \frac{2 \sin\theta}{1 + \sin\theta}$$

The value varies according to the amount of streamlining effected or angle of inclination of the surface to normal, for instance for a streamlined car a constant of 0.0017, or ~~forxinstancexforxaxstreamlined~~ for a single-deck streamlined coach body 0.0021, may be sufficiently accurate for all practical purposes: 0.0032 is used in connection with double-deck passenger vehicles. These constants form part of the expression ph = kG A, where V velocity in ft/sec and A the projected area in sq.ft.


W


W

TRACTIVE RESISTANCE

The sum of these three resistances to motion is termed the tractive resistance of the vehicle and may be expressed by the equation(TR) $=W(R + \frac{2,240}{G}) + KV^2A$ where (TR) tractive resistance

W = Vehicle weight in tons

R = rolling resistance lb/ton

G = radient

V = velocity in ft/sec

A = Projected area in s.ft.

Examination of the power curves of the engine reveals that whereas the torque curve rise to a pointand gradually falls with further increase in engine speed, the b.h.p curve continues to rise throughout the range of engine speeds b.h.p for the purpose of actual work calculation is of little value, and merely represents the rate at which engine is performing work. It becomes, however, a necessary figure for speed calculations. Torque represents the actualuseful work performed,and its value is consequently applicable to work calculations as exemplified in gradient climbing ability.

In explanation of this it should be noted that torque signifies a turning moment, that is a force acting at a lever arm and is expressed in lb/ft, whilst power is made up of torque and angular velocity. Thore for work/time = Power, which is the rate of doing work, hence the product of torque and angular velocity is power. It is some-times necessary to calculate the torque from the horse power and speed of engine revolutions, in which case

$$T = \text{ or } 6{,}250 \times \frac{(HP)}{n}$$

lb/ft, n being the number of engine revolutions per minute.

TRACTIVE EFFORT

To overcome the external forces, effort must be applied in the form of a torque at the road wheels, and the first step is to ascertain what overallgear ratio, that is low gearbox and axle ration, is necessary to convert the available engine power to useful work.

If e = transmission efficiency

r = overall gear ratio in low gear

D = running radius of type

T = engine torque in lb/ft

then tractive effort at wheels is found from the formula

$$T = \frac{t \times 2 \times e}{\frac{D}{12}}$$

As Te must at least equal Tr

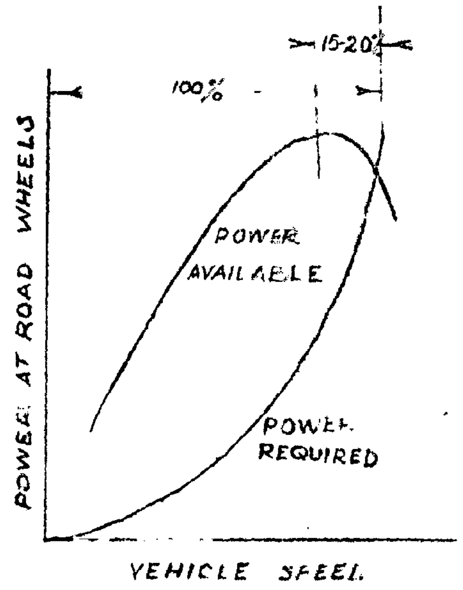
then $$\frac{T \times c \times r}{\frac{D}{12}} = W\frac{(R + 2{,}240)}{G} + K V^2 A$$

and $$r = \frac{W\left(R + \frac{2240)}{G}\right) + K V A}{\frac{\frac{D}{12}}{T \times e}}$$

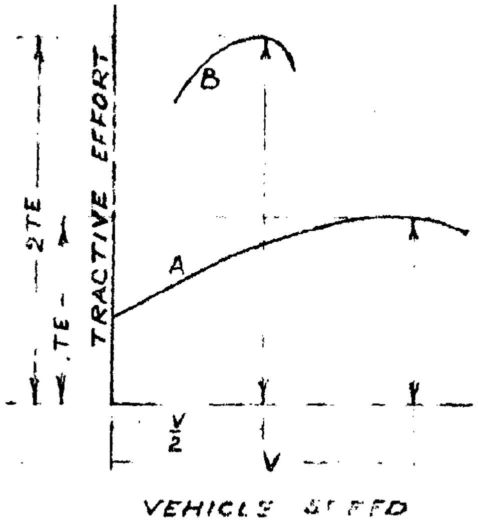
The extremes in conditions both of the forces opposing motion and those which are required under certain circumstances to overcome such resistances have been established. It now becomes necessary to ascertain requirements for the desired performance in intermediate stages.

Knowing the engine characteristics, we can arrange the intermediate gearbox speeds. It should be remembered that it is advisable that the peak of power at the road wheels should occur at say 15-20 per cent early on the engine power curve, that is to say the gear ratio is such that the peak occurs at a speed of 15-20 per cent lower than the ultimate speed which is given when the peak is projected on to thepower required curve. It is desired that the number of speed changes should ensure that thedrop in engine speed and consequent vehicle speed is not too great when making the gear shifts, whilst at the same time permitting ease of gear change by the driver.

The mainfunction of the gearbox is to maintain engine speed at the most economical value under all conditions of vehicle motion, so that the optinum valo of power output/fuel consumption is achieved. This however is not easy to accomplish with the ordinary gear-type reduction; the ideal would be an infinitely variable transmission. It is known, of course that the larger the step between two gears the more difficult is the change. The selection of cofrect ratios is therefore important.



15-20%
100%
POWER AVAILABLE
POWER REQUIRED
POWER AT ROAD WHEELS
VEHICLE SPEED



B
A
TRACTIVE EFFORT
2TE
TE
V/2
V

GEOMETRICAL PROGRESSION

Geometrical progression affords a selection which has many merits, since the vehicle is propelled in gear by a series of engine accelerations and decelerations. If the ratios are in geometrical progression, then the engine speed range is constant throughout all gears. This, of course, is the theoretical ideal.

In practice, however, many other features enter into the final gear-ratio selection; for instance, low-speed gear is often an emergency one, whilst the change down must frequently be quickly effected. Such conditions would be assisted by reducing the speed range of the engine between ratio and low gear in order that the loss of vehicle speed should be a minimum and the wating period for synchronisation of the two gear wheel speeds be reduced.

Geometric progression represents a series of quantities in which each term is obtained by multiplying the preceding term by some constant factor called the common ration, for example, 1.3, 9, 27, 81, etc. each term being three times that precedding it. For a series of terms n having common ratio r and the first term a the sum

$$S = \frac{a(r^n - 1)}{(r - 1)}$$

or, if r is less than 1, a more convenient expression is

$$\frac{a(1 - r^n)}{(1 - r)}$$

It is not advisable to allow engine revolutions to reach maximum value before changing gear, as power will be cut off suddenly and the vehicle will lose considerable road speed. Moreover, maximum engine torque is developed far below maximum engine revs, and in low gear it is desirable to utlise this power when climbing steep gradients. The engine-speed range permitted by the gear ratios should therefore reach its maximum just above the speed for maximum torque.

Incidentally, a certain amount of clutch slip is often experienced, when re-engaging takes place, more particularly in the lower gears, with higher engine speed and lower vehicle speed. The effect of this is that the vehicle speed has increased over that in the higher gear even before the drive has become positive.

The selection of intermediate gears permits the plotting of tractiveffort curves against road speeds, in each of the four gears selected. Previously road speeds have been related to engine speed on the level ground. If the ideal tractive effort curve be plotted on the same ordinates, that is through the point on each of the four curves corresponding to the selected engine speed, which indicates the most economical point of working on the engine torque curve, it will be seen that ~~thrus~~ there is a portion of each individual effort curve overlapping the ideal. This means that the gear ratios selected provide in each case for the maximum pulling power at any vehicle speed. The ideal tractive-effort curve (Fig) is a rectangular hyperbola based on ordinates of vehicle speed and tractive effort.

For example, if T_t = tractive force, and
V = vehicle speed ft/min

then from the formula $\frac{T_t \times V}{33,000}$ = b.h.p all the data for plotting such a curve can be ascertained

GRADIENT PERFORMANCE AND ACCELERATION

We can now consider the gradient performance, and investigate the accelerating properties of the vehicle. Incidentally, it is of course appreciated that if a gearbox ratio is modified so that the total overall ratio between road wheel and engine is double its original ratio, curve A becomes curve B (see Fig) all horizontal dimensions being halved and all vertical distances doubled, as for given engine speeds and doubled total ratio the vehicle speed is halved but the tractive effort is doubled.

Using the same scale as for the tractive effort, we can calculate the total tractive resistance at different gradients. The curves obtained cut the tractive efforts at various points.

The vehicle speed at which the best pulling power of the engine is exerted in each gear. It is read from each tractive effort curve at the point of intersection with the ideal tractive-effort curve.

The climbable gradient at each of these optimum points is easily estimated from the formula

$P = \frac{W}{G}$ or $G = \frac{W}{P}$, where G = gradient

W = vehicle weight

P = excess power.

For the purpose of ascertaining acceleration characteristics of the vehicle it must be borne in mind that the limiting factor is the adhesion between the type and road, for if excess tractive effort is put through the wheels, the latter will spin and adhesion will be lost. The coefficient of adhesion varies and depends upon the surface of the road, and to some extent upon the rear-axle weight. If we dividethe driving force at the wheels by the vehicle weight we arrive at a

factor $K = \frac{T_E}{W}$, T_E of course varies in each gea and is a maximum at the point of maximum torque on the engine-performance curve. This factor may be used in calcuting vehicle acceleration thus

If T_E = tractive effort in lb/in

W = mass of vehicle lb

f = acceleration (32.2 ft/sec/sec)

$T_E = mf$ or numarically $T = \frac{W}{32.2} f$

therefore $f = \frac{32.2\,T}{W}$, but $K = \frac{T_c}{W}$ and substituting we get

$$f = 32.2\,K \text{ ft/sec/sec.}$$

This expression, however, bears ~~hxwsvsrxxhx~~ no relation to the coefficient of adhesion between tyre and road, and deals only with vehicle gross weight. It has been stated that the maximum effort which can be exerted at the ground without wheel skid will depend upon the weight upon the rearaxle x coefficient of adhesion, hence T = where where axle weight gross and coefficient of adhesion (usually 0.7) for for normal surfaces,

by substitution $\omega\mu = \frac{Wf}{32.2}$ and $f = \frac{32.2}{W}\,\omega\mu$.

C H A P T E R - VI

THE GEAR BOX

1. Design Considerations
2. Internal Teeth
3. Lubrication
4. Gear Shafts
5. Planetary Transmission
6. Action and Reaction
7. Synchromesh Gears

....

THE GEARBOX

Tho maximum torquewhich a gear will transmit is known to be proportional to thesquare of the tooth thickness at tho base, and also to the face width and pitch diameter. It is inversoly proportionalto the height of the tooth. Similarly the torque capacity of a gear set varies as the cube of its linear dimonsions. Of those dimensions, (a) the shaft-contro distance and (b) dimensions between bearings, are two of tho mst important.

In the original gearbox layout a fairly accurate shart can be mado by assuming tho shaft-centre distance conforms to or (the former for private cars, and the latter for trucks) and the bearing centres at T being maximum engine torque in lb/ft. This latter dimonsion naturally depends finally upon tho gear-face widths and gear movements and, instead of tedious calculations in the initial stages of design, an approximation of face width may be taken as represented by the expression where

L = maximum permissible load on tooth at pitch circle

and P = normal diametral pitch.

K is a constant -

11,000 - 14,000 for the first reduction gear in four speed box.

14,000 - 16,000 for third speed

16,000-21,000 for second speed

26,000 - 30,000 for first speed

For a five-speed box these figures are slightly revised to

13,000 - 15,000 for fourth speed

15,000 - 17,000 for third speed

20,000 - 22,000 for second speed

26,000 - 30,000 for first speed

The diamentral pitch is determined by the centre distance, the ratios required, and the helix angle of the tooth, bearing in mind that none of the main wheels should have less than, say, fifteen teeth. The total number of teeth in any two pairs of mating gears is usually the same. There are three implications of this that the helix angle is the same for all pairs of gears; that the diamentral pitches are differentineach pair to compensate for any difference in helix angle, or, finally, that spur gears are adopted.

In the choice of helix agle two considerations should be borne in mind, (a) that it is desirable that the circular helical advance over the face width should be t least equal to the ircumferential pitchin order that mm tooth contact is maintained on the pitch line at some point, (b) that the thrust load emanating from the first reduction set should not be substantially exceeded by thrust from the other gear sets, as these thrusts are in the opposite direction and therefore approximately cancel out.

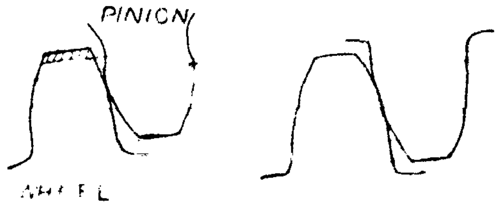
Since one of the major requirements of gearbox design concerns quientness of running and maximum efficiency. These conditions can be achieved if the selected gear teeth are "corrected" that is if the addendum is increased and the dedendum decreased in the pinion and vice versa in the wheel.

$$\text{Base pitch} = \frac{\text{Pitch circle diameter x x cos pressure angle}}{\text{Number of teeth}}$$

$$\text{Circular pitch} = \frac{\text{Pitch circle diameter x}}{\text{Number of teeth}}$$

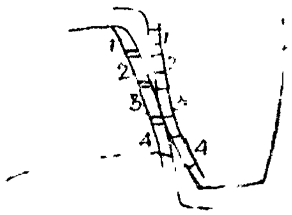
For this continuity of action, the line contact XY must be greater than the base pitch and the more teeth in action at one time the less the strees in the gear teeth; this is graphically represented by the ratio of the length of line XY to the base pitch.

Fig. shows the "corrected" profile, in which full advantage has been taken of the tooth cut to standard depth, so that no part of the wheel tooth works with any part of the pinion tooth other than where the profile is true involute, which now extends to the base-circle diameter. With the elimination of undercutting the strength has also been considerebly increased. In order to analyse the type of contact at various positions on the tooth flank, divide the working face of one tooth into a number of equal divisions and transfer to the mating tooth the portions with which each works.



UNCORRECTED CORRECTED

FIG



FIG

Reference to Fig shows that at the pitch circle only are the divisions equal and that only rolling therefore takes ~~pxk~~ place at this point. At the tip of the wheel tooth illustrating that sliding takes place at the lower part of the tooth flanks. Obviously wear is greatest at this point. Compromise will be necessary to obtain the desired amount of correction to suit individual requirements.

Correction coefficients for spur wheel and pinion and helical wheel and pinion, that is where the addendum is made equal to m (1 + K), m = module of cutter, K is correction coefficient, 1 = number of teeth in pinion and T = number of teeth in wheel, are given in Table.

Before passing from tooth considerations, and whilst appreciating that the reduction of gear noise is of primary importance, it would be well to analyse the type of noise, its category and the probable cause.

Gear	Number of Teeth	Sprial Angle	Virtual Number of Teeth	Cooreotion Factor
Spur Pinion	t	0	t	$K = 0.4(1 - \frac{t}{T})$
Spur Wheel	T	0	T	$K = 0.4(1 - \frac{t}{T})$
Helical Pinion	t	a	tsec a	$K = 0.4(1 - \frac{t}{T})$
Holical wheel	T	a	T sec a	$K = 0.4(1 - \frac{t}{T})$

Gear noises generally fall within three categories

(1) A ring of high-pitched whine

(2) A low-frequency growl

(3) Those omitting an irregular "hammer".

It is believed that "bounce" is primarily responsible for the majority of such noises and it is apporent under varying circumstances; by "bounce" is meant a robound of the teeth in mesh. It is accepted that the relative velocity and angular position of two wheels in mesh will be correct when contact takes place at he pitch line, but that any profile inaccuracy produces a variation in angularposition equal to the sum of such errors at the point of contact of the teeth. It follows that irregular angular velocity occur between the driven and driving gear.

Rotational speed and tooth load are controlling factors in the magnitude of the prodced effect. Immediately the the condition is passed, wherein the torque transmitted to the driver produces decleration coincident with that produced by profile inaccuracies, separation of teeth occurs, and contact is only restored through impact. The resulting rebound causes the hammering noise in category 3. This of course, depends somewhat on the allowable backlash, and such a pair of gears can quite probably run quietly up to the speed at which separation occurs.

Even when the growling noise appears separation may be in evidence, but noise in this case may be due to local pitch errors, whenthe frequency of the noise would be equal to the number of revolutions of the wheel. The difference in type of noise is explained by the fact that the bounce may be damped out entirely before there is further impact. Eccentricity of bore could produce this noise by causing a constant repetition of relative pitch and profile error. However, on the other hand, should the gear berunning in constant mesh, under no load or light conditions, bouncing mingh occur and a prolonged rattle with no definite period being the result.

One of the maincauses of "whine" is gear web or nave n weakness, or a binration of web brought into action by the general finish of the gears and rough spots on the teeth The presence of lubricant between tooth faces of course, damps out the natural vibration to some extent, and thus contributes to a ~~xt~~ quieter running gear.

Internal gears: Fundamentally, external-gear and internal gear systems differ in one respect only that of base circles, in which those for the external gears lie on opposite sides of the path of tooth contact whereas in the case of internal gears the base circle for both mating wheels lie on the same side of this contact line. It will be appreciated that the internal gears have a greater lentth of pressure line and consequently the arc and duration of contact is longer, whilst the amount of overlap between meshing teeth is increased. Moreover, the internal-gear tooth possesses a difference in tooth curvature between contacting surfaces which results in greater bearing area. This is due to elastic distortion across the tooth face when under load.

The sliding velocity of the internal gear is also less than that fo a similar external pair. In view of these qualities, it will be seen that a greater amount of correction is possible with the internal-gear tooth, which, whilst strengthening both teeth, permits contact to take place where the radius of tooth curvature is a minimum.

One of the points on the debit side of internal gears is that of interference which occurs at thetip of the pinion whilst pasing throught the internal addendum circle of the wheel. It is not advisable in ordinary application to mesh two internal gears of standardtooth form if the difference in tooth numbers is less than twelve.

In specialcircumstances, howdver, the difference may be educodto even one tooth, but this demands a degree of corr ction which removes the whole of the arc of contact for some distance beyond the pitch point.

INTERNAL TEETH

Before leaving the design of the gear teeth, a word should be said concerning the internal tooth of the gears used for dog or clutch purposes as exemplified in the constant-mesh pinion and elsewhere. It is seldom found that these teeth are generated as an internal gear profile; more often the method employed is that of drilling and cleaning with a cutter having the same number of teeth ~~as the clutch.~~

In common with other internal gears, the bore of the internal gear must be at least equal to the base-circle diameter, whilst without effecting the strength of the gear it may be opened out to the pitch circle diameter. The strength of such an internaldog or clutch depends primarily upon the strength of the teeth in shear; thus where

F = load at pitch line (lb)

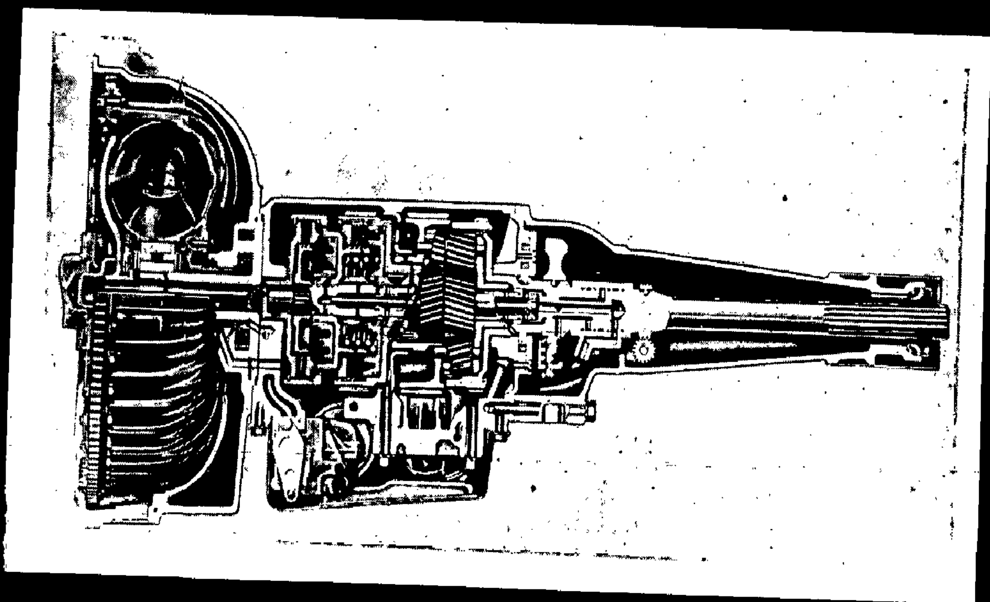
r = radius of dog (in)

W = width of the internal gear

f = safe shear stress (lb/in)

$F = r\,Wf$.

The diameter of drillhole is fairly standardised for vaious tooth forms, for instance with 20 stub or 20 full-depth tooth, the following table may be adopted, whilst the minimum width ofworking tooth has been found by experience as in Table.



Lubrication of the gears is a most important feature, and should be studied carefully. The main characteristics of the chosen lubricant should be:

(a) Must be capable of thorough distribution flow through all ball bearings and small holes of passages, and must have no corroding effect on parts with which contact is made.

(b) Should be of such character and body s to minimise power loss through churning, and should offer minimum resistance to gear change mechanism.

(c) Should be capable of exerting a washing action on gear teeth. etc. and posees properties for maximum heat absorption and dissipation.

(d) Must have a consistency such as to prevent leakage under normal conditions.

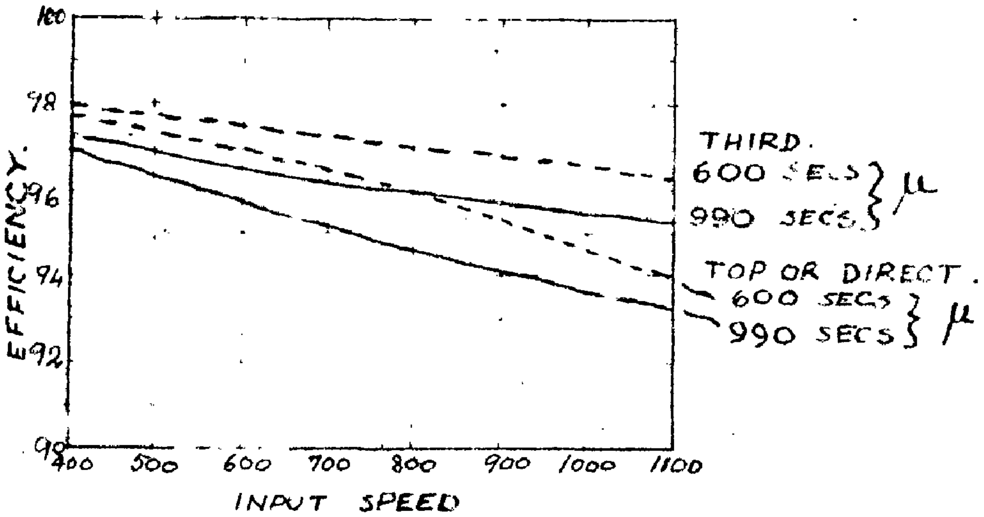
The effect of lubrication upon the output efficiency dependsupon several factors. It would appear that excessive quantity has considerable effect, as also does that of an oil having too high a viscosity. Connecerning the quanity desireable, it shoudl be appeciated that the greater power loss occurs (due to excessive amount)in the direct -drive position It o ccurs, however. In all gears and increase until the gears are completely covered. The loss is doubtless due to cavitation or oil churning, in other words reduced equivalent wheel immersion. The passages co cut by the wheel are filled either with air or lubricant at a higher temperature

20th stub or 20 full-dept tooth

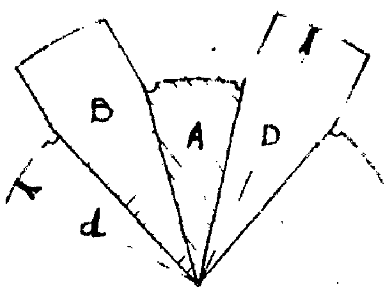
Diametral Pitch	Drill Diameter	Width of Tooth
5 DP	$\frac{9}{32}$	$\frac{9}{16}$
6 DP	$\frac{15}{64}$	$\frac{7}{16}$
7 DP	$\frac{13}{64}$	$\frac{3}{8}$
8 DP	$\frac{11}{64}$	$\frac{5}{16}$
9DP	$\frac{5}{32}$	$\frac{9}{32}$
10 DP	$\frac{9}{64}$	$\frac{1}{4}$

than the surrounding bulk. Adequate lubrication of the indirect gears is given by an oil level which immerses the teeth of one of each pair of gears, generally about one-fifth to two-fifths of the box volume.

The fall in efficiency on direct drive is considerable between quantities of lubricant filling the box one-fifth and three-fifths full, as test figures have indicated a fall of from 97.5 to 90.5 per cent, whilst under the same conditions third gear losses are from 97.2 to 96 per cent.



LUBRICATION AND
GEAR EFFICIENCY.



SPLINES

The effects of viscosity are also important as, for similar input speeds a drop in offioncy for both gears, throughtout a viscosity range varying from 200 to 1,000 Redwood seconds, has been observed as 95 to 93½ per eent for direct drict drive and third gears falls from 96.6 95.5 percent.

Tho relation between efficiency, input speed and viscosity is shown in Fig. from which it will be soon that, for normal oil filling at two-fifths full, direct drive is the gear affected most. The low viscosity lubricant shows decreassed loss of power, most probably on account of reduced cavit tion. Tooth friction varies only slightly with the viscosity of the oil and is unaffected by change in speed if torque is constant, but such friction varies with the load transmitted since it is dependent upon tho contact pressure between the teeth. It will therefore be appreciated that a low-viscosity lubricant is desirable, with the filler plug a ranged so that the tips of the gearsonly are immorsed in oil. A practice adopted during initial running in is to use tomporarily an excess pressure lubricant which has the effect of tooth polishing and in consequence the possibility of film breakdown is minimised when the reversion to standard lubricant is made. However, on the larger and heavier gear boxes there are consideror ations other than gear teeth lubrication, as exemplified in the oil requirements of the ball bearings tho cones of a synchromesh mechanism, and the lubrication through tho shaft centre of remote gears on sliding splines. In such cases it is desirable to fit a small auxiliary oil pump, preferably driven from the layshaft end, when all these and other points receive a positive and direct oil supply.

GEAR SHAFTS

The gear shafts are subject to stresses from combined torsion and bending moments and in extracting the shaft diameter from the formula $T = \& \sqrt{R + T}$, where T = torsional moment and R = bending moment, due consideration should be given to the effect of splines upon the strength of the shaft.

There is a stress concentration at the base corner of the splines and the consequent reduction in effective diameter combined with the lowering of the fatigue-stress limit, which is due to the continuous change of section and the broken periphery of the shaft, has been the subject of much study by photo-elasticity methods. It is thereby established that the diameter upon which to base calculation is one smaller than the base of splines. Further, there is the effect of torque on the spline and its transmitted effect upon the spline base, which renders it still more desirable to assume a decreased diameter.

The olastic limit for a splined shaft is less than that of a plain shaft of diameter equal to the base diameter of splines,whilst shear strength is reduced, depon-ding upon the number of splines, by 5 to $7\frac{1}{2}$ per cent. ~~Hnhm~~ When such a shaft is under torsion the sectors A and B become helices of which the lengths differ, although both are functions of the diameters A and B respectively. Measured round the helix, B is longer than A, the difference boing $\pi d_B - \pi d_A$ and thus a shearing force is set up along the radial lines.

Continou s reversals of strees eventually cause the shaft to fracture, the fractures forming perfect sectors. Obviously the period in which such fractures occur depends upon the difference between the respectiv e diameters D and d, and the greater the difference theearlier the fracture It is for this reason that for an increasednumber of smaller splines the shaft isstronger than with a small number of large splines.

A reliable practice is to use a base diameter plus 15 per cent increase over that found from the expression F= T/Z when it will be found unnecessary to make further add-itions for correctionduo to shaft deflection. Many differing methods of mounting the layshaf t gears are available, and, providing there is rigidity in tho assembly, there is is little more to add, except in the type which employs a rigid or fixed shaft and rotating gears.

In such a design the gear wheels are often formed in 'cluster' Apart from any productinn difficulties the problem of gear noise again arises. Tooth inaccuracies will produce noise on one gear which must inevitably be transmitted through the layshaft train and become operative on the other gears. The effect cannot be accurately computed, but it is highly probable that any whine or ring may eventually become resonant throughout the whole system. It is possible that a break is desirable in the continuity of the gear bosses as it h s beenfound in many cases that it splits up the medium thrugh which sound may be transmitted.

Such calculations as are required for the determination of bearing loads an shaft deflections are straightforward and need no explanation. They are, however, set out under their appropriate heading in the section devoted to formulae.

The gearbox contributes much to the general success of otherwise of the design. Rigidity is its first essential, it is subjectedto torsion from the reaction which it transmits to its point of attachment either to engine or frame. The value of this load is equal to

$T (E.r - 1)$ where T = engine torque

r = reduction ratio

E = gearbox efficiency

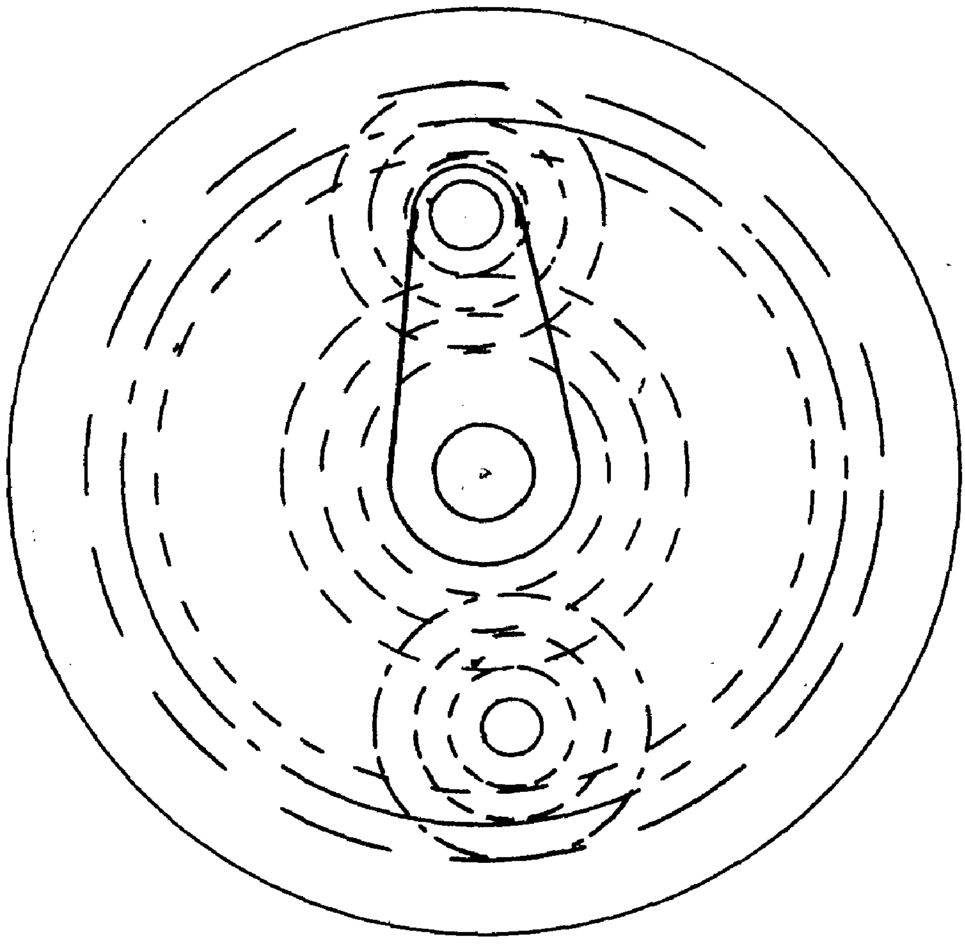
and the highest value is therefore in reverse gear. Should the casing become distorted under load, shaft misalignment will have considerable eff ct upon the meshing of the gears.

It is worthy of note that the material surrounding the bearing housings is subject to tensile stress in many directions, and the available material should therefore be apportioned accordingly. Precautions must be taken to avoid resonance, and it would appear desirable to break up any flat surfaces by the introduction of suitable ribs.

PLANETARY TRANSMISSIONS

Many of the problems which arise with the two-shaft gearbox areeliminated by the epicyclic or planetary-type arrangement of gears.There are two basic types of subh transmissions, that consisting of spur gears entirely and that which employs internal gears. The sum pinion A or the driving member being attached to the input shaft of the transmission. The planetary pinions C, of which there may be two, th ee or more, mesh with this gear, and also with the internal gear B, and are mounted on a planet carier, concentric with the input shaft. If ring B be rendered stationary by mechanical means and pinion A is rotating clockwise, the pinions C will roll on the internal gear in clockwise diretion. Whilst rotating in an anti-clockwise direbtion about their own axes. The planet carried D then possesses the same directional rotation (clockwise) as the driving member, but at a r duced speed. If it is desired to the drving shaft or gear A. This forms a ddirect drive.

The calculation of speed ratios, which is the ratio between the number of trns made by the arm D and the number of turns made by the sun wheel, is similar to that for ordinary gear trains. If the pinions make one complete revolution about the driving-shaft centre, they will also be re-volved about their own axes to the extent of $\frac{B}{C}$ By calculating the motion of the sun wheel A which is required to produce each of these planetary wheel motions and adding them



together, we obtain the number of revolutions of A to provide one revolution of carrier D, which represents the gear ratio for that particular train.

The first motion of the planetary gears, that around their own axes, is expressed as
For the second motion, that of one complete revolution in the same direction as that necessary to produce the first motion of the pinions, the sum of the two is

This is the reduction ratio of the gear set.

Now if it is presumed that pinion A is held against rotation and load taken off carrier D, the power being applied to ring B, the carrier will revolve in the same direction as B. If also the planetary gears with their carrier make one complete revolution, then by rolling around the sun wheel the pinions turn on their own axis $\frac{a}{c}$. To produce these two motions the angular motion of the internal gear for one revolution of the carrier D is

$$\frac{a}{b} + 1 = \frac{b + a}{b}$$

This assembly comprises three members, the sun gear, the ring gear, and the carrier, each one of which can be held against rotating, so that six combinations are possible as power can be transmitted between the two remaining members. Two of these combinations are reducing gears, namely sungear to planet carrier and ring gear to sun gear tosend planet carrier to ring gear, whilst the remaining two combinations are reverse gears, one step down and one set up.

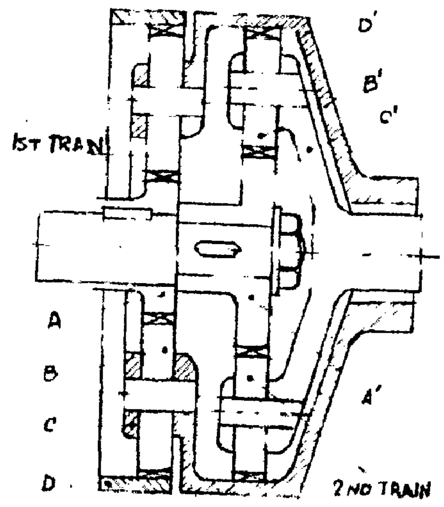
Compound planetary-gear Sets The fig. illustrates a manshaft or driving shaft upon which two sun sheels are attached rigidly, with two ring gears and two planet carriers to form a compound planetary set. The reduction ratio from A to C and thence to D' is $\frac{d+a}{a}$ and when the ring gear of the first train D is restrained from motion, the gear becomes a low forward gear. In the second train A', B', D', A' and D' are in motion and consequently transmit motion to carrier C, the total motion being the sum of those which would be conveyed to it if ring gear D were stationary and A' described $\frac{d+a}{a}$ turns, also if sun gear A were held stationary whilst ring gear D' made one turn.

The motion imparted to C by sun wheel A is

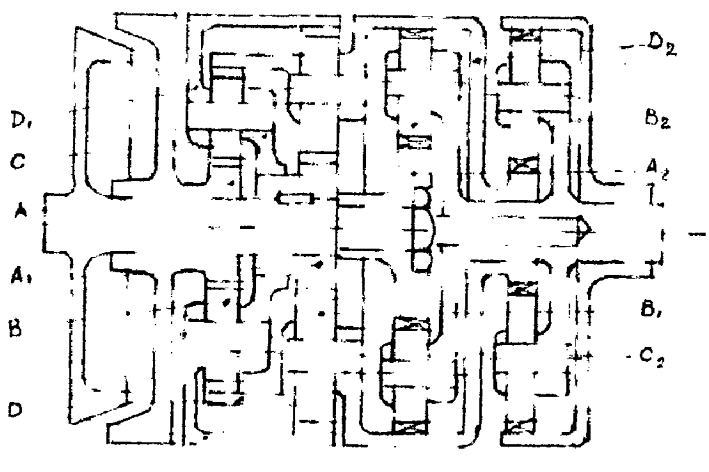
$$\frac{\frac{d+a}{a}}{\frac{d'+a'}{a'}} = \frac{a'(d+a)}{a(d'+a')}.$$

As the reduction from D' to C is $\frac{d'+a'}{d'}$ and D' makes one revolu- the motion imparted to C' by ring gear D' is

$$\frac{1}{\frac{d'+a'}{d'}} = \frac{d'}{(d'+a')}.$$



COMPOUND PLANETARY GEAR



FOUR-SPEED COMPOUND SET

Total motion equals the sum of these two expressions

$$\frac{a'(d+a) + ad'}{a(d'+a')}$$

and if A revolve $\frac{d+a}{a}$ times, then the overall ratio (that is, is between A' and C' is $\frac{(d+a)(d'+a')}{a'(d+a)+ad'}$.

~~and if A revolve~~ ~~times then the overall ratio~~

Four-speed Compound set: with this combination low forward speed is obtained by holding ring gear D in which case torque is transmittod by the last train of gears, whilst the gear ratio is $r_2 = \frac{d_2+a_2}{a_2}$ Release of ring gear D_2 and restraint of D_1 gives second speed, torque beig obtained through the 1 st two trains of gears, from which it has previously been seen the gear ratio is

$$r_1 = \frac{(d'+a')(d_2+a_2)}{a_2(d'+a') + a'd_2}.$$

For third speed sun wheel A is held and ring gear D released so that all three planetary trains are in action. The ratio in this case is therefore

$$(a_2+d_2)\left[\frac{(a_1+d_1)(a+d) - dd_1}{d_2 a_1 (a+d)}\right]$$

Fourth speed, which is direct, is obtained by locking the assembly together that is by ~~xx~~ release of drum brakes and bringing into operation the friction clutch (see fig.

There are several small points which require attention small points which require attention concerning the geometry of "tooth setting". If several planetary pinions are used in an internal-type ring gear, the relationship between the number of teeth on the driving pinion and the planetary prinions mustbe definite, other wise it will not be possible to assemble the train. According to the number of trains of gears, the number of teeth a a $-a_p a + 1$, must be divisible by that number. Henco if three trains are used, the divisor will be 3. Assuming that $a - 1$ is divisible by 3 then.

$$a = 3x + 1$$

$$c = a + 2b = 3x + 2b + 1$$

$$\frac{a}{3} = x + \frac{1}{3} \cdot \text{pitch}$$

$$\frac{c}{3} = x + \frac{2}{3} \cdot \text{pitch}.$$

If the axes of each of the three pinions are set at 120 one to the other, and the planetaries have an equal number of teeth, then two of their tooth centres will be opposite and a sun-pinion tooth centre will be one-third circular pitch from the line connectijg that axis of the right = hand pinion and the sun pinion. If this is so then obviously a tooth centre ~~touch~~ of the ring gear will be one-third pitch beyond this axis line produced. If the numbers of teeth are odd, then a space and not a tooth centre is opposite another tooth centre line. Therefore the number of teeth in the planetary gears plus one should be such that they are divisible by 3, and in the case of the two planetary gear sets both doiving gear and planetaries may have either an even or odd number of teeth.

If there are throo tains thon, if a is divisiblo by 3, b must also bo divisiblo by 3 and a - 1 b + 1, a + 1, b - 1 must all bo capable divisicn by th-t number, With four planetaries, both a and b may either be odd or oven, but both must bear tho same sign.

Consider tho all spur type of planotary-gear set, which diagrammaticelly is out in Fig. In principlo it comprises throo sets of indepen3ent gear trains and low forward spood is obtained by holding theplanet carrier against motion whon power is exotted by gear A through

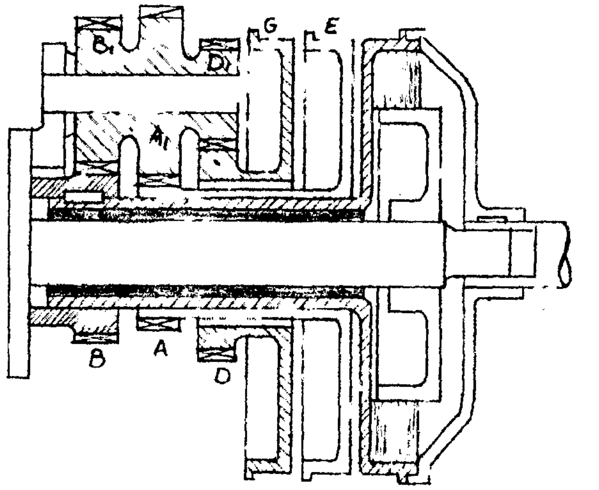
A_1 and D_1 to gear D. The reduction is, of course $\frac{a_1 d}{a d_1}$.

For high forward speed the drive is direct through tho clutch. For ~~xxxxxxx~~ roverse motion, the gear B isheld against rotation. If the carriorrotates anti-clockwise B_1 rolls on gear B and the gear cluster A_1, B_1, D_1 revolvos anti-clockwise about its own axis and makes $\frac{b}{b_1}$ revolutions in which case A makes revolutions - in which case A makes $\frac{ba_1}{ab_1}$ revolution, but in a clockwise direction. Ths, when related to the ~~mxx~~ motion ~~fox~~ of the carrior, givos a total motion for A us.

$$\frac{a_1 b}{a b_1} - 1 \qquad \text{CLOCKWISE}$$

Tho motion of tho planetary pinions cause goar D to make $\frac{bd_1}{b_1 d}$ clockwisc revs and again related to the motion of tho carrior, tho motion of D.

$$= 1 - \frac{bd_1}{b_1 d} \qquad \text{ANTI-CLOCKWISE}$$



ALL SPUR PLANETARY GEAR

Should d b D will revolve in the reverse direction and the expression has a positive value. To find the reduction ration use the expression.

$$\frac{d\,(\,a_{,}b - ab_{,})}{a(\,b_{,}d - bd_{,})}$$

It is understood that in any spur gears the sum of the tooth numbers ofmating gea s is the same. If this sum is noted as X then

$$x - a = a_{,}$$
$$x - b = b_{,}$$
$$x - d = d_{,}$$

which if substituted in the foregoing expressions, the reduction may be stated as $\frac{d\,(b - a)}{a\,(d - b)}$

In top gear B is connected to the driven shaft through the clutch, both brake drums being free, and the gear becomes an ordinary train since the planer pinion cluster B, A, D, does not rotate, but revolves solid with the whole gear. B is running at egine speed. Ratio 1 to 1

In top second gear brake drum E is fixed and the train is B, A A, It is convenient to tabulate wheel speeds for varous conditions of fixed members. X is fixed and B, and A, will have one positive revolution (see Table

The speed ratio for second gear is therefore

$$\frac{\frac{a_1}{a}-1}{\frac{a_1}{a}} : 1$$

Reverse gear: In this case the reverse brake drum G is fixed, and E is free, and the train is X B, B D, D. see table

Fixed Member X		Fixed Member A		
Revolutions of		Revolutions of		
X	0	X	$\frac{a}{a1}$	Driver
B A	+ 1	B A	$1+\frac{a}{a}1$	
B	-1	B	$\frac{a}{a}-1$	Follower
A	$-\frac{a}{a}$	A	0	Fixed wheel

Fixed Member X		Fixed Member D		
Revolutions of		Revolutions of		
X	0	X	$\frac{d}{d1}$	Driver
B D	+ 1	BD	$1+\frac{d}{d}1$	
B	-1	B	$-1+\frac{d1}{d}$	Follower
D	$\frac{d1}{d}$	D	0	Fixed wheel

The speed ratio for reverse gear is therefore

There are many propriotory makes of epicyclin gear box in which the necessity for a separate clutch is eliminated, since by virtue of their design, if the ultimate fixed member be made free and gradually speeded up and the clutch action through which engine and transmission is coupled is obtained in the epicyclic gearing itself.

ACTION AND REACTION

The principle that force or action and reaction are equal and opposite applied not only to loads and forces, but to moments and torques. Therefore if the transmission-output shaft develops a torque in one direction, the power unit tends to rotate in the opposite direction, except of couse if the transmission is in direct drive, as in that case both output torques are similar in magnitude and direction. The torque is then taken by the engine case. If the output is greater than that at input due to insertion of a gear box or similar mechanism a reaction member in the transmission is essential. The gear-set casing forms a convenient means of absorbing this reaction. The conventional automobile transmission usually comprises a number of gear trains, each of which effect torque conversion in different ratios. There are, of course, many types of mechnisms which are used to make this onversion and, in turn, some of the more widely accepted will be reviewed.

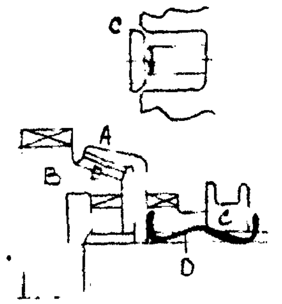
First, the gear type of reduction, both "clash" and synchromesh, involute spur teeth or helical gears The main function function of the gears is to transmit motion from one shaft to another, with uniformvelocity, with the minimum of noise, andwith a little shock loading as possible The practice of synchronising the moving parts to be engaged considerably relieves the last two points, whilst the use of constant-mesh helical gears assists the first condition. With both the clash andconstant-mesh types, the methods of bearing-load computations are somewhat similar, but there are one or two conditions to be satisfied in the synchronising type which do not apply to the straightforward sliding gear.

Synchromesh designs may be separated generally into two types, (a) the inertia lock and (b) the constant load. Cone clutches arethe mediums through which the synchronising is caried out in bothtypes and the main difference is that it is not possible to 'clash' the geas by too much pressure on the control mechanism with the inertia-lock designs whereas this is not so with the latter type. Very brief explanations of the two systems will make this clear.

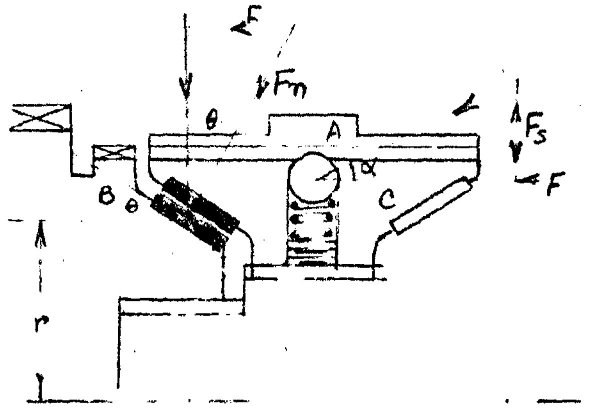
Inertia-lock Synchromesh: Any endwise movement of the synchronising cone clutch brings into operation the inertia lock, which prevents positive gear engagement until there is no relative slip between the two cones. When such a condition has been reached the two gears will mesh. It will be appreciated that the time taken to synchromise is inversely proportional to the load applied, since the greater the force exerted on the hand lever the greater the force between the two cones andin consequence, the greater the inertia lock load. Due to the difference in speeds at the moment of changebetween the two cones, that on the gear B and that on the drum A,both cone and drum rotate slightly until the projections F on the drum make contact with mainshaft spline sides.

The engaging dog C is, however, exerting pressure on the drum through the medium of the chamfered edges and it is impossible to move the engaging dog further during the perid in which the torque on the drum is greater than that caused through the chamfered faces. This drum torque decreases, however, as the speeds approach synchronism and, when it is just less than the torque between the two chamfered faces the drum moves forward and permits the dog to follow through to engagement.

Constant -load Synchromesh (Fig): In this type the gears can still be "crashed" or the gears be made to mesh before synchronisation in speed takes place by undue heavy load applied to the gearcontrol level, if such a load overcomes the pressure required to overcome the springs which in turn control the ball loading, whether the mating parts are synch-ronised in speed or not. Reference to the rotation in Fig will show that presure from the change-speed lever is applied to the outer ring A. This is transferred to inner member C through the groove and the spring loaded balls, bring the cones in contact. The speeds are thus synchronised between the shaft and engaging dog.



INERTIA SYNCHROMESH
LOCK



CONSTANT LOAD SYNCHROMESH.

Additional pressure on the change-speed lever depresses the ball springs, allowing the outer member to slide and positively engage the gear. The cone angles are of considerable importance, as upon the angle depends the loads required for synchronising. The angles are usually about
as this figure also permits sufficient longitudinal movement to maintain the cone face clearance which approximates 0.005 in.

It is essential if goo d synchronisation is to be maintanied that lubricant should be dispersed immediately the cone clutches engage: thus in consequence, the design of oilways requires careful consideration. The actual diameter and angle control the speed wth which synchronisationtakes place and the clutch must perform the function of changingthe inertia of the moving parts from their running speeds to that of the new gear velocity.

The fundamental formulae required in estimating the torque reqired for synchronism are of course force = mass X acceleration,

$$\text{or } \frac{M}{g} \times A$$

$$\text{Torque} = \frac{M}{g} \times K^2 \times \frac{12\pi \times n}{12t}$$

where K = radius of gyration (in)

M = Weight (b)

g = gravity accelaration (32.2 ft/sec/sec)

n = r/sec

t = time taken to synchronise in sec.

since mass x radius of gyration = I

$$\text{Torque} = \frac{I \bar{\pi} . n}{6t}$$

The maximum load applied to the cone when a change of gear from say to to third is made occures when the third gear road speed is such that the engine revs are maximum for that gear. Assuming the road speed to be constant, the difference in speeds of the affected parts in the gear box should be tabulated in r/sec. If this conditionis maintained the speed of the mainshaft, and other parts slidably splined to it, will remain constant, bt the layshaft and gears in mesh with those on the mainshaft must be speeded up together with the clutch disc and constant-mesh pinion. The layshaft and gears, clutch disc and constant-mesh pinion speeds are, of course, mainshft speed and third-speed ratio.

The moments of inertia for all rotating parts should be ascertained and the torque required to synchronise the speeds will thus be

$$T = \frac{\bar{\pi}}{6t} \left(I n + I_1 n_1 + \ldots \right)$$

From this torque, which is applied by the cone clutch, the normal cone load becomes $F_N = \frac{T}{r\mu}$ and the axial load $F = \frac{T . \sin\theta}{r\mu}$ = whilst the spring load $F_s = \frac{F . \tan\alpha}{N}$ where

N = number of balls.

C H A P T E R - VII

THE REAR AXLE

1. Casing
2. Design
3. Arrangements
4. Bearing Loads.

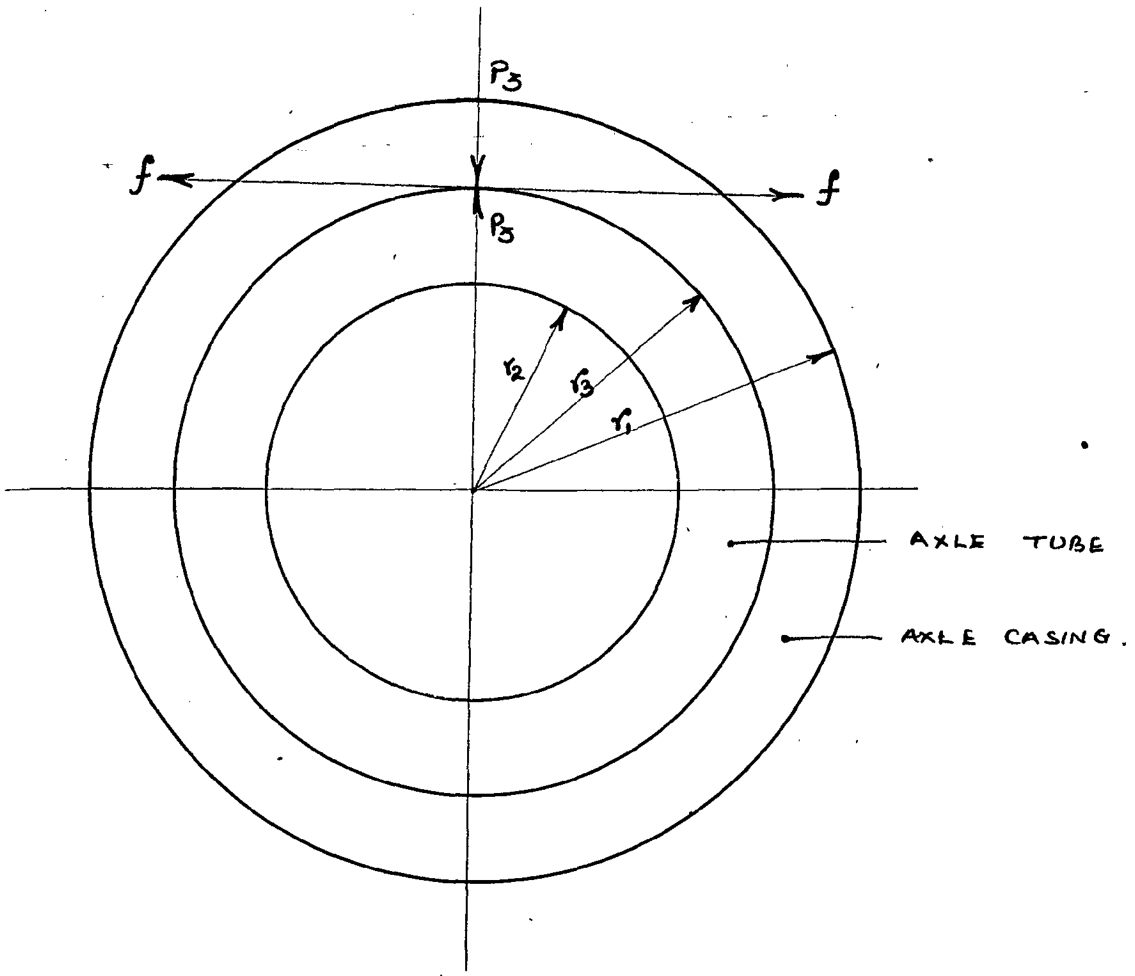
REAR AXLES

CASINGS

The type of axle which incorporates the three-piece fabricated design, the t is the c st centre and tubular arms, has many merits and if the design is carried through in accord nce with latest practice good results should accuse. The loder method of pressing the tubular arms into position required extreme ccuracy in order that any ke-ways, etc. should be maintained in line wth each other. Moreover, the possibility of slackness developing between arms and case could never be overlooked, and the bursting strees could never be correctly calculated.

The present method of 'freezing" the tubes overcomes many of the past ~~xxxxxxxxx~~ deficienies and affords means whereby correct hoop stresses may be found for different ranges of shrinkage fits. Briefly, the method adopted provides for the immersion of thetubeend into a liquid oxygen bath at a temperature of approximately 120 F belowzero which for steel gives a contraction of roughly 0.0045 in, whilst the casing is heated to a temperature of 140 F forapproximately 0.0025 in expansion.

When both tube and casing are mated the interference is approximately 0.003 to 0.0045 in and the preloading is in the order of 7 ton/in. The assembly is then securely dowelled as an additional precaustion against any movement. Alternative freezing agents may be used such as methylated spirit



P_3
f
f
P_3
r_2
r_3
r_1
AXLE TUBE
AXLE CASING.

or trichlorethyline in conjunction with solid CO.

It is interesting to anaylyse the hoop stresses, which may be derived from Lames basic theory, which states that:

$$P = C_1 + \frac{C_2}{r^2}$$

where p = internal pressure

r = internal radius

C_1 & C_2 are constants

where P_3 = initial pressure due to shrinkage at surface radius r_3 by derivation (see Fig)

$$C_1 = \frac{P_3 r_3^2}{r_3^2 - r_2^2}$$

and $$C_2 = -\frac{P_3 . r_3^2 . r_2^2}{r_3^2 - r_2^2}$$

since f also equals $\left(C_1 - \frac{C_2}{r^2}\right)$

by substitution:

f = compressive stress in the ring of axle tube

$$= P_3 \left(\frac{r_3^2}{r_3^2 - r_2^2} + \frac{r_2^2 . r_3^2}{r_3^2 - r_2^2} \cdot \frac{1}{r^2} \right) \text{ p.s.i.}$$

and f_1 = tensile stress in the axle casing

$$= -P_3 \left(\frac{r_3^2}{r_1^2 - r_3^2} + \frac{r_1^2 . r_3^2}{r_1^2 - r_3^2} \cdot \frac{1}{r^2} \right) \text{ p.s.i}$$

The maximum stress in the tube occurs where $r = r_2$ and again substituting we find f_2 = maximum stress $= \frac{2P_3 . r_3^2}{r_3^2 - r_2^2}$

whilst maximum stress in the casing occurs at r_3

f_3 = maximum stress = $-P_3 \left(\frac{r_1^2 + r_3^2}{r_1^2 - r_3^2} \right)$

Between this range of formulae the stress can be found at any point in the section of either the arm or the casing.

P_3 is, of course, dependent upon the amount of interference between the two machined diameters of the tube casing, and the relation may be expressed as:

$$\Pi d^3 = \text{interference} = \frac{4 P_3 \cdot r_3^3}{E} \cdot \frac{r_1^2 - r_2^2}{(r_1^2 - r_3^2)(r_3^2 - r_2^2)}$$

E representing elasticity 30,000,000.

Pressure P and stress f can thereforebe calculated for any desired amount of interference. A typical diagram is plotted which shows the stresses in the casing and tube for varying interferences. From these curves it will be seen that the maximum stresses occur in the tube or arm at its inner diameter and that these stresses are compressive, whilst those for the casing re tensile and are also maximum at the bore. The curves are based on a casing bore of approximately 4⅝ in diameter material, ⅝ in thick tube, internal bre 3⅛ in diamer Such a design also possesses the advantage that the track may be modified without the necessity for new parts. It is probably a little heavier than the one piece casing.

The three-piece axle c sing is not always manufactured in the same manner; the tubular arms may be pressed and dowelled into the cast centre. Pressing into position nearly ~~nnx~~ always produces some degree of deform tion in both inside and outside diameters in both parts. The extent of this condition can be calcul ted with accuracy providing the materials are not stressed beyond their proportional limits, whilst the assumption is made that the components are acted upon by uniformly distributed pressure acting radially both internal and external.

Where a = inner radius

b = outer radius

p = inner pressure

p = outer pressure

R = radius

E = modulus of elasticity

a = poeisson's ratio

D = deformation at R

the expression which holds good for bushes, or bearing is(from fig)

$$D = \frac{1-\mu}{E} \cdot \left(\frac{a^2 P_1 - b^2 P_2}{b^2 - a^2} \right) \left(R + \frac{1+\mu}{E} \right) \left[\frac{a^2 b^2 . (P_1 - P_2)}{(b^2 - a^2) R} \right]$$

Now assume two cylinders, the outer diameter of the inner cylinder exceeding the inner diameter of the outer cylinder by some tolerance, then with further notation:

E = modulus of elasticity, inner cylinder

E = modulus of elasticity, outer cylinder

μ_1 = poisson's ratio, inner cylinder

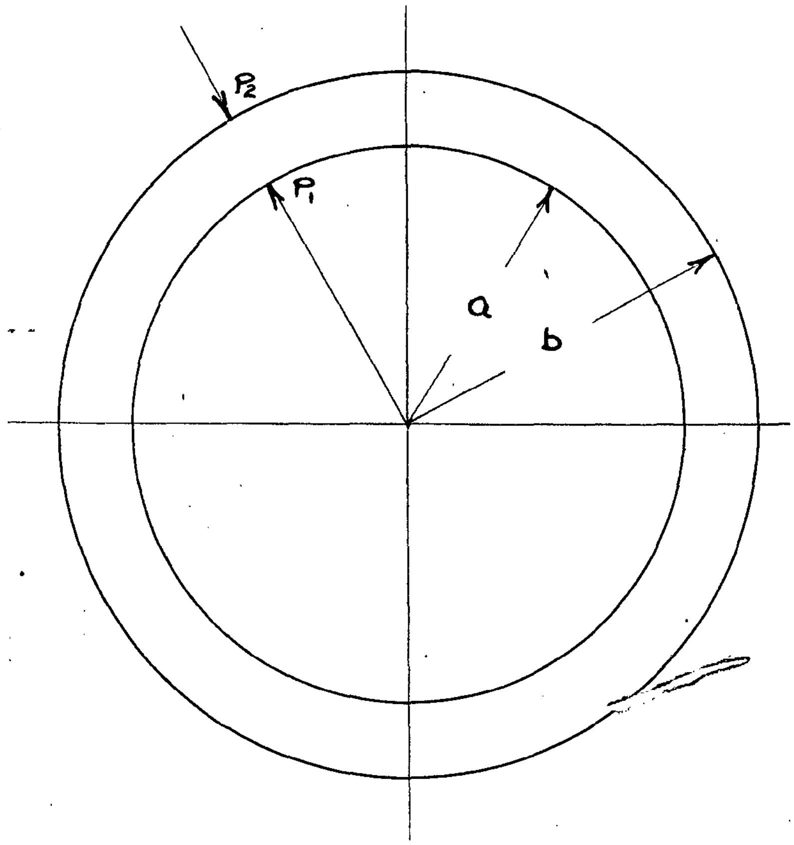
μ_2 = Poissón's ratio, outer cylinder
δ = fit between inner and outer cylinder
P = Pressure between cylinders

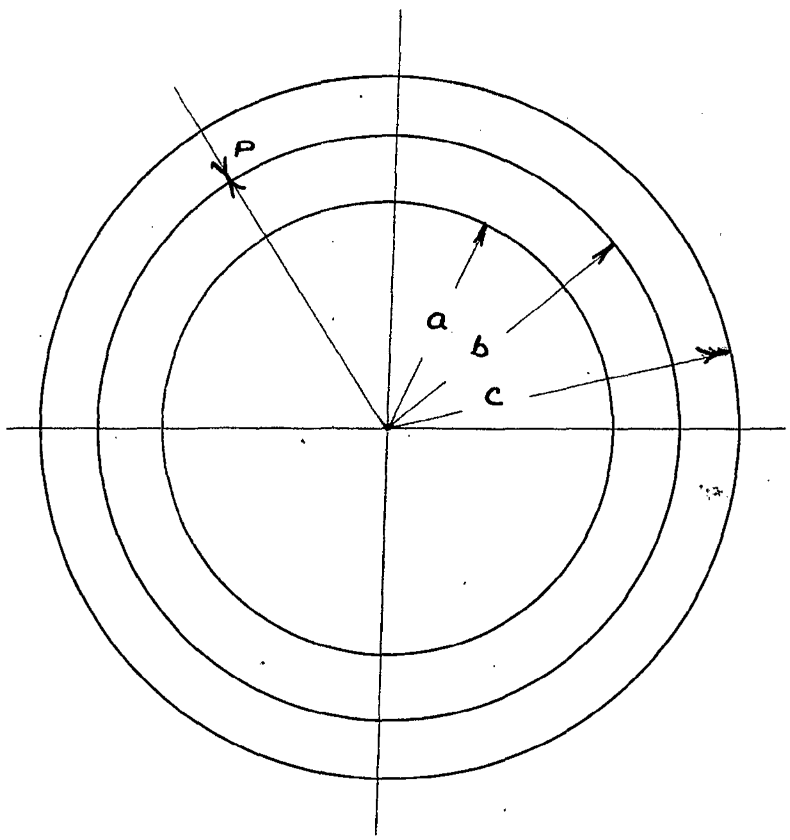
D_1 = increase in inner radius of outer cylinder

D_2 = decrease of outer radius of inner cylinder

D_3 = decre se of inner radius of inner cylinder

D_4 = increase of outer radius of outer cylinder





DEFORMATION DIAGRAM.

Obviously the sum of the amounts of deformation of both cylinders must be equal to

Therefore $D_1 - D_2 = \delta$

and for the first expression for D D and D are obtained.

$$D_1 = \frac{bP}{E_2} \cdot \left(\frac{b^2+c^2}{c^2-a^2} + \mu_2 \right)$$

$$D_2 = \frac{-bP}{E_1} \left(\frac{a^2+b^2}{b^2-a^2} - \mu_1 \right)$$

substitution in expression for δ

$$\delta = \frac{bP}{E_2} \left(\frac{b^2+c^2}{c^2-b^2} + \mu_2 \right) + \frac{bP}{E_1} \left(\frac{a^2+b^2}{b^2-a^2} - \mu_1 \right)$$

from which P will equal

$$P = \frac{\delta}{\frac{b}{E_2} \left(\frac{1 + b^2/c^2}{1 - b^2/c^2} + \mu_2 \right) + \frac{b}{E_1} \left(\frac{1 + a^2/b^2}{1 - a^2/b^2} - \mu_1 \right)}$$

If this value of P be used, then from expression D D and D are found to be:

$$D_3 = \frac{-2a}{E_1 (1 - a^2/b^2)} \cdot P$$

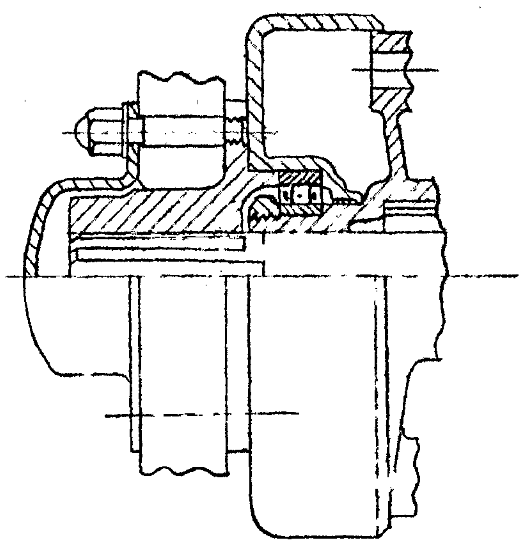
$$D_4 = \frac{2b \cdot b/c}{E_2 (1 - b^2/c^2)} \cdot P$$

Should the inner and outer cylinders be of the same material, the expression for P D and D can be simplified and since they would be independent of elastic constants, could be combined. Therefore, combining P and D and noting that both moduli of elasticity and ~~bth~~ both poisson's ratios are equal:

$$D_3 = \frac{-(a/b)\left(1 - \frac{b^2}{c^2}\right)}{\left(1 - \frac{a^2}{c^2}\right)} \cdot \delta$$

$$D_4 = \frac{(b/c)\left(1 - \frac{a^2}{b^2}\right)}{\left(1 - \frac{a^2}{c^2}\right)} \cdot \delta$$

$\frac{3}{4}$ FLOATING AXLE SHAFT.



REAR AXLE ARRANGEMENTS

There are three generally adopted hub-bearing and axle-shaft arr ngements, (a) fully floating, (b) semi-floating type, the axle shaft transmits driving torque only and it is not subjected to thrust loads emanating from vehicle wiggght, since the weight is supported by the axle casing. Such an arrangement is shown in Fig whèch is a type favoured that of the other arrangements.

Should an axle-shaft break, it may be withdrawn without dismanting the wheel or of jacking up the vehicle; moreover there is no danger of a wheel comàng adrift. The two bearing employed for hub mounting share the load from each wheel the load linousually being slightly nearer to the inner bearing- The maximum stree in the casing usually occurs at the change of section adjoining the inner bearing, its magnitude being expressed as:

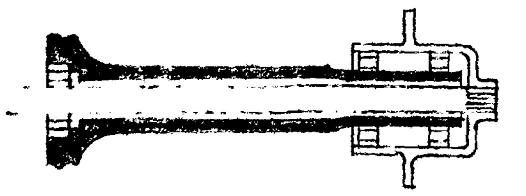
$$f = \frac{B.M}{\frac{\pi}{32}(D^3 - d^3)}$$

where B.M. = maximum bending moment (in/lb)

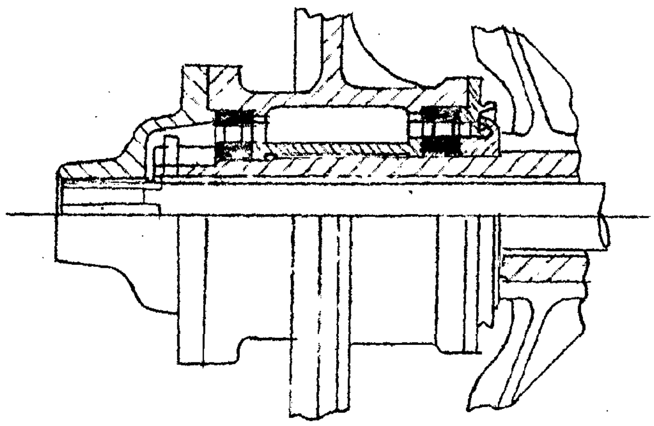
D = outside diameter of tube (in)

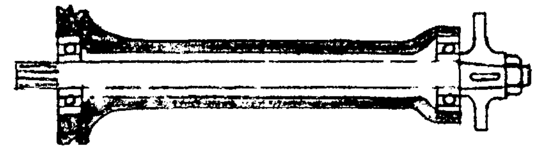
d = inside diameter of tube (in)

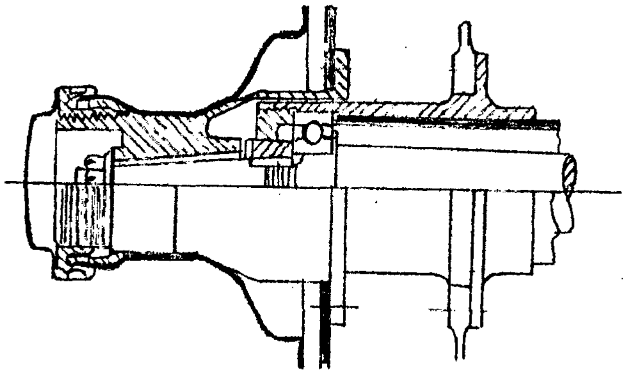
The bore of the bearing should therefore be such that the casing stress should not exceed this figure, which is usually arranged at approximately 10 tons/in . Axle-bearing loads emanate from two sources, (a) vehicle weight and (b) skid reaction. The maximum ground reaction is obtained when the vehicle is traversing a bend, when contrifugal force



FULL FLOATING AXLE-SHAFT







comes into effect and reduces the load on the inner wheel increasing that on the outer wheel by a similar amount.

If W = load on rear axle

E = track of the sheels

then the reaction on the outer wheel is R = $\frac{0 \cdot 5W}{2} + \left(\frac{\frac{Wv^2 \cdot h_1}{gr}}{E} \right)$

where g = acceleration due to gravity

r = radius of bend (ft)

h = height of centre of gravity of vehicle

v = velocity of the vehicle (ft/sec)

This load is divided between the two wheel bearings in the following proportions:

Inner bearing $R \cdot \frac{a}{c} = R_1$

Outer bearing $R \cdot \frac{b}{c} = R_o$

A further load due to tractive effort at the road T is distributed:

Inner bearing $T \cdot \frac{a}{c} = T_1$.

Outer bearing $T \cdot \frac{b}{c} = T_o$

The total radial load there fore on each bearing emanating from ground reaction is:

Inner bearing $\sqrt{\left(R \cdot \frac{a}{c}\right)^2 + \left(T \cdot \frac{a}{c}\right)^2}$.

Outer bearing $\sqrt{\left(R \cdot \frac{b}{c}\right)^2 + \left(T \cdot \frac{b}{c}\right)^2}$.

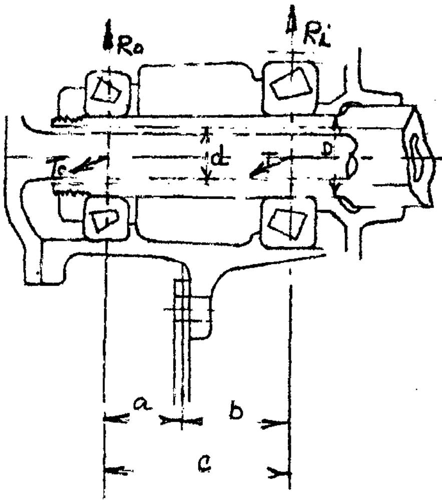


FIG. 1

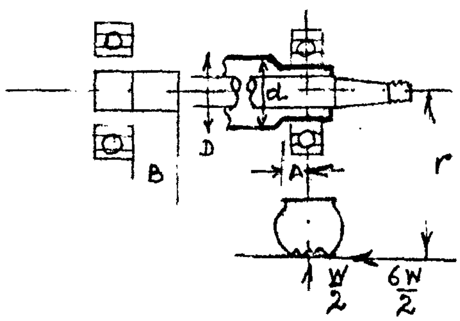


FIG. 2

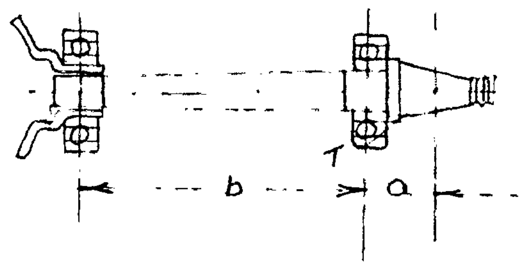


FIG. 3

The skid reaction S is the same for each bearing, but the direction is changed and the load due to $S = \frac{S \cdot r}{c}$ is in a downward direction on the outer bearing and upward at the inner bearing (r = running radius of the type)

The magnitude is $\frac{R_o \cdot W \cdot v^2}{W \cdot g \cdot r}$

A check upon bearing spacing should be made with loads due to S and adjustment made to the centres if the bearing is overloaded The axle -shaft diameter being subject to torsion load only may be assesed from the expression.

$$f = \frac{T/2}{\frac{\pi}{16} \cdot d^3}$$

Where T = low-gear torque (that is clutch slip torque x lowest gear ratio)

As a general rule, the three-quarter floating axle possesses one outer-wheel bearing and the load line is coinciedent with the centre line of this bearing. In consequence the axle shaft transmits torque only when the vehicle is running in a straight line. When however, the car is rounding a bend, a bending moment is imposed on the shaft through the skid reaction.

This moment may be expressed as B.M. = o.6Wr and the stress therefore may be found from the usual B.M = fZ(Z in shear) (see Fig).

The axle arm or tube is most highly streesed at the point A, from the centre line of bearing , where B.M. = WA (in/lb)

~~B.M.~~

and $f = \dfrac{B.M}{\frac{\pi}{32}(D^3 - d^3)}$

The axle-shaft stress at the driving end, ~~that~~ that is the differential end, may be calculated from torque considerations only and the expression is similar to that for the fully floating

and where the practice of tapering the shaft towards ~~the~~ its inner end is carried out, for reasons of permitting a certain amount of torsional resilience which is necessary to prevent fracture when sudden torque loads are applied the length of the reduced diameter should at least equal four diameters. In practice the permissible torsional deflection is in the order of ½ 10 1 and 1½ = angle of deflection

$$\phi = \frac{57.3\ T/2B}{\frac{\pi}{32} \cdot d^4 E} \qquad E = 12 \times 10^6\ p.s.i.$$

Any deflection of the shaft must be kept within the safe limits for the outer bearing for the wheel. The graphical solution is the most ~~th~~ practical and should be used in order to obtain deflections at any point along the length of the shaft. The same deflection-diagram construction may be applied to a similar shaft of the semi-floating type.

An axle of the semi-floating type is generally used on light vehicles. The axle shafts must withstand bending in xxxin addition to torsion, whilst, due to the overhang of the wheel bearing centre line, the inner bearings or differential-casing bearings are subjected to loading emanating at the wheelsIf the wheel bearings are chosen to withstand straight high speedrunning they will have sufficient capacity to withstand loading from cornering at lower speeds (see fig

If W = rear-axle load

R = ground reaction

then $\frac{W}{2}$ = R and the loads on the bearing due to R

will be: wheel bearing $R\left(\frac{a+b}{b}\right)$

and those due to T = $T\left(\frac{a+b}{b}\right)$

The total wheel-bearing loading can be expressed as

$$\sqrt{\left\{R\left(\frac{a+b}{b}\right)\right\}^2 + \left\{T\left(\frac{a+b}{b}\right)\right\}^2}$$

The loads on the differential bearing due to R will be:

$$R\frac{a}{b}$$

and those emanating from T = $T\frac{a}{b}$

To these loads, however, those from the pinion and ring gear must be added and of course the magnitude of load varies according to the type of drive, either spiral-bevel straight bevel, hypoid gear of worm gear.

The wheel bearings should be selected for size after the axle shaft has been stressed and a suitable diameter agreed.

The procedure in setting out the graphical solution of axle-shaft strenght and deflection a accurate and the stages are straightforward. An exact understanding is necessary of required conditions of the shaft to be stressed, since the maximum bending moment (which is at the centre line of wheel bearing) occurs when the vehicle is rounding bend ~~nttipping~~ at tipping speed, although te B.M. which occurs when turning a bend of say 120 ft. radius at 20 m/h is more valuable where shaft deflection is concerned, assupplying data for normal bearing performance, and since the skid reaction moment is additional to ground reaction moment on the axle shaft at the inside of the curve, the conditions are more severe for average driving.

It is therefore advised to stress for the lat-ter condition, in which case, if

W = rear-axle weight

h = height of centre of gravity

R = running radius of tyre

E = rear-wheel track

a = distance of wheel to bearing

r = radius of bend of road = 120 ft

the centrifugal force $= \frac{Wv^2}{gr}$ and if V = 20 m/h (that is 29 - 3 ft/se)

and R = (ground reaction) $= \frac{W}{2} - \frac{0{\cdot}221\,Wh}{E}$

S = (skid reaction) $\frac{R}{W} \times 0{\cdot}221 \cdot W$

(1) Bending moment therefore, which is the first value to calculate, is B.M. $= Ra + \frac{SR1}{a}$

Plot the B.M. diagram, and divide it into a series of figures shown shadedplot in Fig. . Take note that these divisions include any change of section in the shaft.

(3) From the point of eentre of gravity of each division draw verticel lines and letter each one A,B, C, etc.

(4) Next lay out the polygon of forces. In a vertical downward dimection from O, mark off segment A to scale equal to the area of division A in the bending-moment diagram. Proceed similarly with other divisions of the B.M. diagram.

(5) From the vertical scale line lay off the pole distance where E = 30,000,000 lb for steel

and $I = \frac{d^4}{64}$ where d = shaft diameter (in)

EI = flexural rigidity of division A of axle shaft.

(6) Draw OA to any convenient point A, at pole distance. Draw from the next point down the vertical scale representing the next division B line extending until it intersects pole distance of division B, and so on until all the divisions have been treated and the polygon of forces is complete.

(7) Now construct the funicular polygon from the diagram of forces, commencing with a line parallel to OA produced until it intersects vertical line A. Follow this procedure for each polygon of force line, in each case commending at

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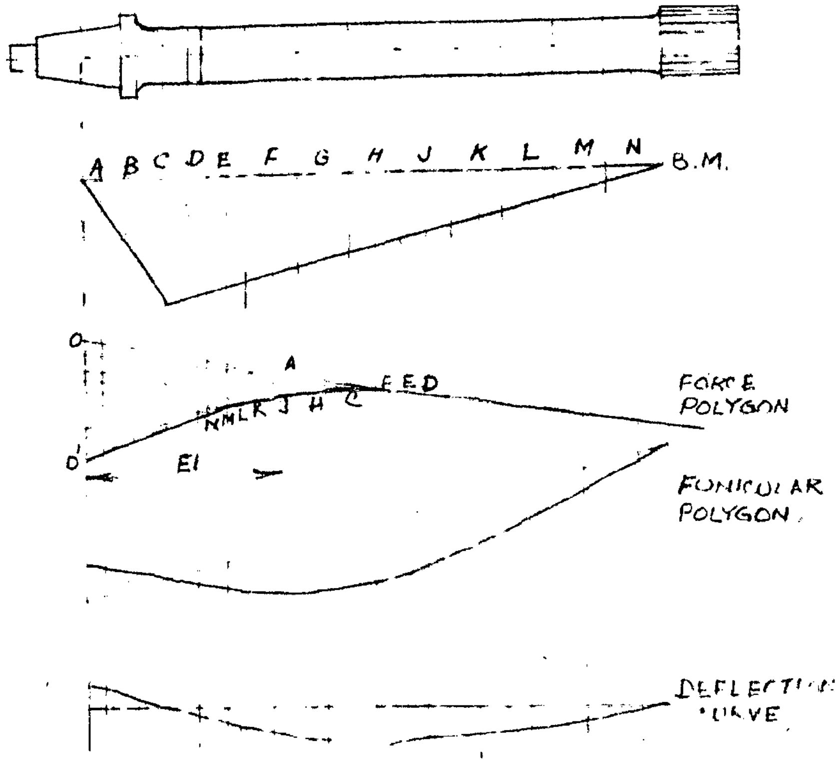


AXLE SHAFT STRESSES

the previous intersection point and parallel to its corresponding division on the vertical force line. At the point of intersection of the last line through the differential support bearing, draw the closing line of the polygon to intersect the centre line of the wheel bearing.

(8) The deflection curve may now be drawn as the vertical distance of any point on the funicular polygon base line and the curve is proportional to the axle deflection. In order to obtain the angle of deflection at the bearing centre line, draw a tangent to the deflection curve as it passes through this position and divide it by the ration of the two scales, that is, the vertical scale on the polygon of forces or B.M. scale, and the EI scale, or horizontal distance.

$\therefore$ if α = diagram angle of deflection to scale

$$\frac{\alpha}{EI} \times BM = \text{the true angle of deflection.}$$

As previously remarked, the loads from the final drive gear must be added to the bearing loads and for different types of drive the following calculations are utilised.

Bearing loads with spiral-bevel gears: Rear-axle pinions generally rotate clockwise for forward vehicle motion, and they possess a left hand spiral, that is, when viewed from the power input end. Such an arrangement is very desirable on account of the tendency in condition (·) and (4) to force the pinion out of mesh (a feature which can be overcome by the provision of an adequate thrust bearing), but in the case of conditions (2) and (3), the pinion is threst further into mesh with the possibility of tooth wedging. As the gear tooth can be cut with either left or right-hand spiral there are four combinations possible; the first one is the generally chosen arrangement (see Fig.   ):

(1) Left-hand spiral pinion = clockwise rotation.

(2) Right-hand spiral pinion= clockwise rotation

(3) Left-hand spiral pinion = anti-clockwise rotation.

(4) Right-hand spiral pinion= anti-clockwise rotation.

Bearing loads will be computed for condition (*)

where T_p = pinion thrust

T_g = gear thrust

F = tangential force

α = tooth pressure angle

$\beta = \dfrac{\text{Pitch cone angle (Pinion)}}{2}$

δ = spiral angle.

$$\text{Tangential force} = F = \frac{\text{Torque input (lb in)}}{\text{Mean pitch radius (in)}} = \frac{TQ}{rp}$$

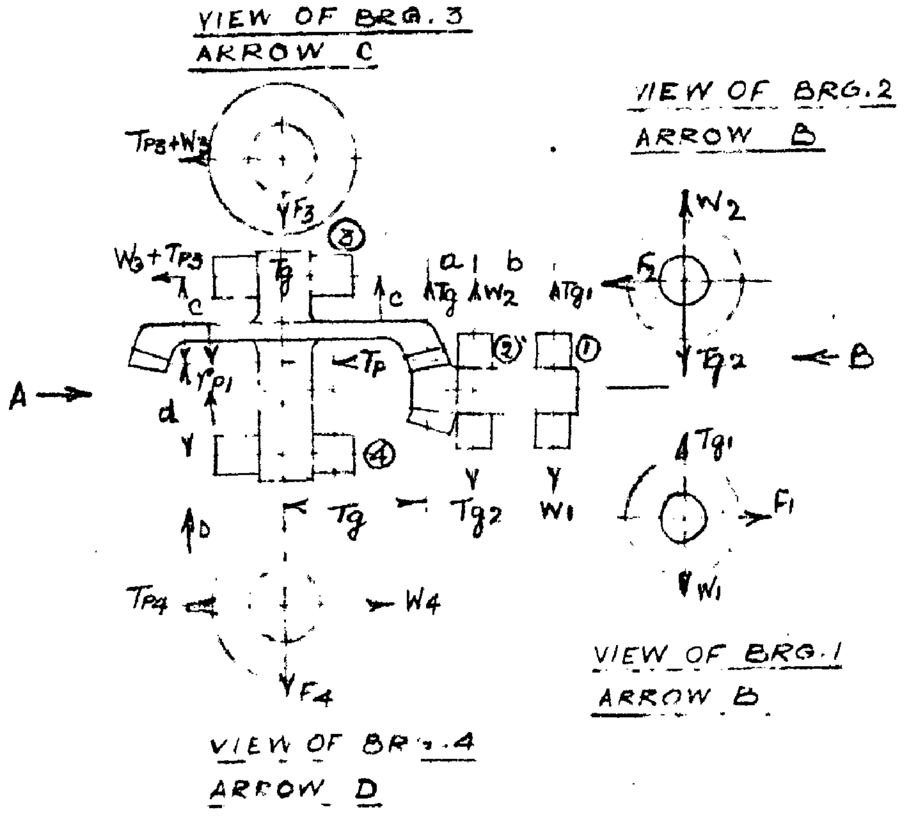
$$rp_i = \frac{\text{Pinion pitch diameter - Tooth face x sin}}{2}$$

$$r_g = rp\left(\frac{N_p}{N_g}\right) \quad \text{where } N = \text{number of teeth}$$

$$= \tan^{-1}\left(\frac{N_p}{N_g}\right)$$

The values of T_p are reversed in directions in condition (2) and (4), whilst for the four combinations the values of T_g and T_p are as follows:

(*) Left-hand spiral - clockwise rotation and (2) right-hand spiral



VIEW OF BRG. 3
ARROW C
VIEW OF BRG. 2
ARROW B
A
B
VIEW OF BRG. 1
ARROW B
VIEW OF BRG. 4
ARROW D

- anti-clockwise rotation

$$T_p = F \left(\frac{\tan \alpha \sin \beta}{\cos \delta} + \tan \delta \cos \beta \right)$$

$$T = F \left(\frac{\tan \alpha \sin \beta}{\cos \delta} - \tan \delta \cos \beta \right)$$

(3) Right-hand spiral - clockwise rotation and (4) left-hand spiral - anti-clockwise rotation.

$$T_p = F \left(\tan \delta \cos \beta - \frac{\tan \alpha \sin \beta}{\cos \delta} \right)$$

$$T_g = F \left(\tan \delta \cos \beta + \frac{\tan \alpha \sin \beta}{\cos \delta} \right)$$

Bearing loads with hypoid gearing : The loads on the hypoid-pinion bearings are of a similar nature to those of the spiral-bevel pinion, but those supporting the gear ring differ. There is a small amount of endways slide combined with the rolling action of the tooth in the hypoid gear, due to the tooth profile, which has been mentioned previously.

The following notation and expressions supply sufficient data to enable the bearing loads to be calculated:

T_p = pinion thrust

T_g = gear thrust

F = tangential force

α = tooth pressure angle (drive side)

β = $\frac{\text{pitch cone angle (pinion)}}{2}$

r_o = effective gear radius

δ_p = spiral angle (pinion)

δ_g = spiral angle (gear)

d = pinion offset

$$\text{Tangential force } F = \frac{\text{Input torque (lb/in)}}{\text{Mean pitch radius (in) (pinion)}} = \frac{TQ}{P}$$

$$= \frac{\text{Pinion pitch diometer} - \text{Tooth face} \times \sin\beta}{2}$$

$$r = \text{mean gear pitch radius} = rp\,\frac{N}{N} \times \frac{\cos\delta_p}{\cos\delta_g}$$

$$T_p = F\left(\frac{\tan\alpha\sin\beta}{\cos\delta_p} + \tan\delta_p\cos\beta\right)$$

$$T_g = F\left(\frac{\tan\alpha\sin\beta}{\cos\delta_p} - \tan\delta_p\cos\beta\right)$$

$$r_e = \sqrt{r_g^2 - d^2}$$

C H A P T E R - VIII

HYPOID AND WORM DRIVE

1. Worm Drive
2. Hypoid Drive

C H A P T E R - VIII

HYPOID AND WORM DRIVE

1. Worm Drive
2. Hypoid Drive

WORM DRIVE

The worm gear possesses several advantages over the other forms of drive. It has a high overload capacity, whilst the nature of tooth action is such that the effect of wear is less delecterious in comparison with other toothed gearing. On the other hand, the heat generatedthrough the relatively high velocity between worm and wheel necessitates higher working temperatures and consequent attention to adequate lubrication.

One of the reasons for the greater strength of the worm set as compared with the bevel gear is that the mean relative radius of curvature of the contacting surfaces is much greater. This also is true for the ordinary spur gear, where the reciprocal of the relative curvature radius is proportional to the sum of the reciprocals of pitch radii of the two gears, the value depending more upon the diameter f the small wheel than that of the larger gear.

In a bevel gear of high ratio the pinion is small and, whilst the surface stress is high, the torque capacity of the wheel is relativelylow.

With regard to efficiency, however, the worm gear is hardly comparable with the bevel or helical gear, the order being approximately 94 against 98 per cent respectively. This loss of efficiency is occasioned, it is believed partly because of the high ratio of mean velocity of sliding to the mean circumferential velocity of the wheel,whilst the power loss through tooth friction is another important factor. If tooth friction only is taken into conderation, and the worm regarded as a coiled wedge of effective angle equal to the lead angle of the worm at mid depth of thread, then the efficiency may be expressed by the formula:

$$E = \frac{\tan \lambda}{\tan (\lambda + \phi)} \quad \text{with worm driving}$$

$$\text{and } E_1 = \frac{\tan(\lambda - \phi)}{\tan \lambda} \quad \text{with wheel driving}$$

where λ = lead angle of the worm

ϕ = angle of friction between worm and wheel

A lead angle of about 45 gives the highest efficency, although considerations other than those of a theoretical nature may diecte to an appreciably smaller angle, whilst it should be appreciated that the greater the lead angle the smaller must be the diameter of the worm. This consideration is important as naturally with the decre se in worm diameter the bending stresses increase, as does also the deflection of the worm shaft under the influence of tooth load. The high local stress thus produced lead to a higher

coefficient of friction, so that actually a smaller lead angle that $45 - \phi/2$ may possibly give better results in practice. The deflection when transmitting maximum torque is, of course, greater than when operating in top gear, and it is possible to escortain the amount allowing for this in the initial setting.

However carefully the worm assembly is designed a certain amount of flexibility in the axle case, worm housing, and in the bearings in inevitable, and conditions of contact between worm and wheel are therefore disturbed.

To counteract this effect the axial position of the worm wheel may be adjusted so that contact in the unloaded condition is towards the leaving side of the wheel tooth. The amount of initial offset is ,of course, found by trial and marking. Slight axial movement of the worm wheel in one direction causes movembent of the contact bearing in the opposite direction. In this respect a worm having a large lead angle is more sensitive than the smaller angle .

Since it is not possible to check the face bearing when the set is under load it is generally satisfactory if, for a light load, contact bearing is concmtrated towards the leaving side and covering about 75 per cent of the available area of tooth flank. Rigidity of mounting is therefore of primary importance and it is desirable to maintain the axial position of the worm wheel rim to within ± 0.0015, a figure which in practice has been foundsatisfactory.

To accomplish this the journal bearing must allow only a little radial play and the thrust bearing must be accurate, for it will be appr eciated that non-axial end thrust tends to cause the wheel to tilt in the plane containing its own axis and the common perpendicular to the axes, in which cse lateral movement of the rim relative to the worm axis will occur. It is for this reason that the bearing ratio to worm-wheel diameter must be as small as possible , whilst in order to minimise the effect of any relative displacement between worm and worm wheel which may occur in a direction parallel to the wormwheel axis, the b arings must be given adequate support in the form of rigid caps and well-designed bolts. Theg worm-gear carrier should also ~~derrixex~~ derive some support against distortion, by means of a steady spigot machined in the banjo or main-axle casing.

Finally the question of lubrication must be considered. It woul be preferable to use a castor base oil, if replacement at short regular intervals could be maintained. An oil with such a base has a low coefficient of friction, but it also has a tendency towards rapid deterioration. A mineral oil is therefore usually recommended which can retain its lubricating properties at working temperaturesof 200 F, a figure which is often found in heavy duty drives.

Adequate sump capacity is vital and the casing dimensions should permit dipping of one or other of the gears under all circumstances. If lubrication is unsatisfactory, which may beproduced by reason of oil-film failure of faulty gear alignment and consequent imperfoct both contact wear and abrasive action producting bronze dust will ne noticeable. The oil film maxy fail due to many causes, although the most comon reason is from inadequate entry gap, which is usually accompanied by high temperatures. The remedy is to form a clearance by careful filling, between the entering edge of the teeth and by resetting the worm wheel. The entry of lubricant between the thread and tooth will thereby be facilitated.

Whould the tooth stresses be of high over"pitting " of of the worm-wheel teeth is likely to occur. Such a phenomenon is not unduly alarming, as failures are rate. Pitting occurs early in the gear life and after the initial development it would appear that the r te at which the pitting extends is considerably reduced. For the purpose of bearing load calculations, the normal tooth force is assumed to comprise (a) a tangential for ce at the pitch radius of worm F, (b) a force tending to separate the worm from the gear S, and (c) a thrust T produced by the lead angle of the worm (see fig)

Torque input

The tangential force = r_ω

where r_ω = pitch radius of worm (in)

and r_g = pitch radius of worm gear (in)

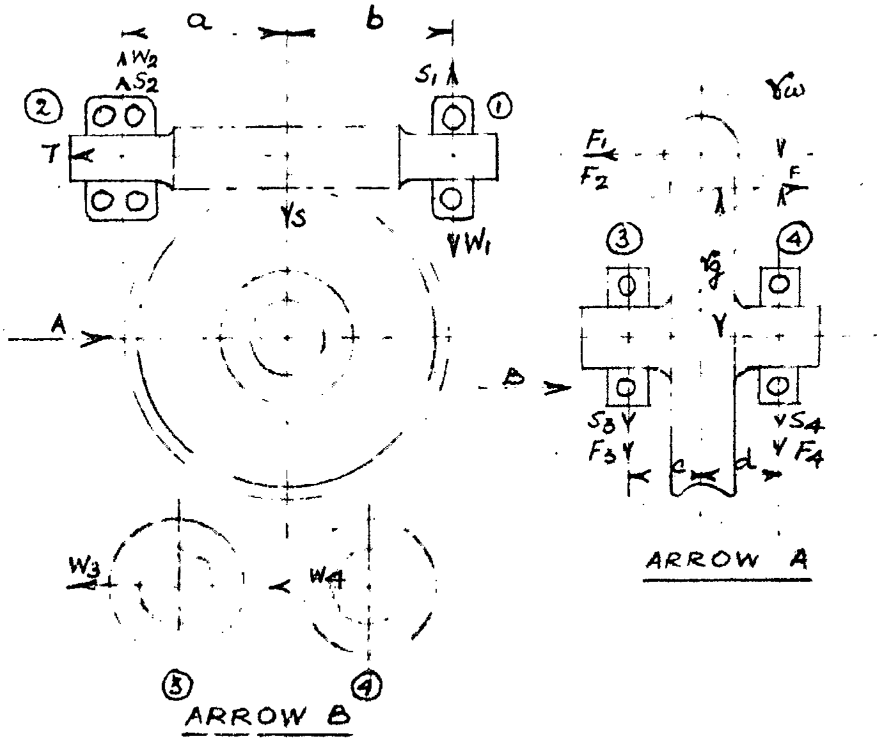
$r_g = \frac{1}{2\pi} \cdot N_g \cdot \beta$. if N_g = Number of teeth in gear

and β = axial worm pitch

The separating force $S = \frac{F \tan\alpha}{\tan\gamma}$ α = tooth pressure ang

γ = lead angle of worm

The worm thrust $T = \frac{F}{\tan\gamma} = \tan^{-1} \frac{\text{Lead}}{2\pi \cdot r_\omega}$.



a
b
W_2
S_2
S_1
2
1
T
S
W_1
A
B
W_3
W_4
3
4
ARROW B
$\gamma\omega$
F_1
F_2
F
λ
3
4
γg
S_3
F_3
S_4
F_4
c
d
ARROW A

HYPOID GEARS

Due to the offset position of the hypoid pinion, the profiles of the opposite sides of the teeth are not symmetrical the concave side being flatter than the convex side. It is therefore necessary, in order to obtain approximately the same conditions of tooth contact on both sides, to make the pressure angles unequal. These angles have been found to be most satisfactory at 17½ on the driving side ~~xkm~~ and 25 on the coasting side Incidentally, for equal conditions of tooth contact on both sides, the following conditions must be satisfied:

(a) Equal arcs of action

(b) Equal duration of contact

(c) Similar relative radii of curvature of profile

(d) Equal freedom from undercutting.

An increase in the spiral angle increases the pinion diameter, thus improving the number of teeth in contact, and in turnthis increases the axial thrust in the normal tooth load. However, it is generally understood that the load-carrying capacity increases more rapidly than the axial thrust and the best compromise appears to be effected with a pinion-spiral angle of 50 and a gear-spiral angle of 25. Due to the unsymmetrical relationship between gear and pinion, these figures mean an approximate offset of 1" in. Compare this with a normal set of spiral bevels.

For equal smoothness, a spiral angle of at least 45 would be ne essary and say a 16 pressure angle for both whilst the tooth load for the hypoid is 15 per cent higher approximately than the tangential load, the spiral bevel normal tooth load being 36 per cent greater than its tangential loads. The maximum tooth loads recommended for automobile rear-axle sets are 1,600 lb/in of gear face width direct drive, and 4,200 lb/in of face width in low gear.

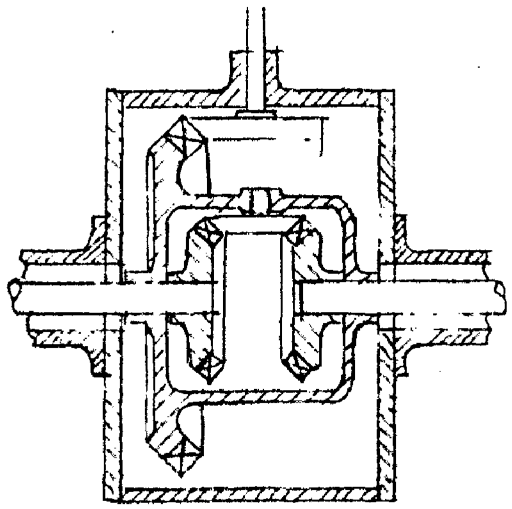
It will be seen from the foregoing that for the increased diameterof a hypoid pinion, as compared with the plain-spiral pinion, a cosiderable increse in thetooth strength is obtained. It is recommended that the offset for hypoid should ~~gxxx~~ not exceed one-eighth of the gear diameter.

THE DIFFERENTIAL

The differential is interposed between the two drive shaft of the rear axle in order than the torque or effort applied to each may be equal although the speeds of revolution may differ. This is necessary when the vehicle is turning in a circle when the outer wheels are required to rotate at greater speed than the inner wheels due to the greater distance they have to travel. but when the vehicle is travelling in a straight line there is no relative motion between any of the differntial gears.

In pinciple the two bevel gears C and D are fixed to the two axle shafts, whilst the bevel pinions A and B are in constant mesh with both gears and are attached to the differential cage E through a pin or two pins in the case of four-pinion set. This cage is bolted to the final-drive gear. If thecage E is held fixed it will be clear that if the gear C be rot ted in a forward diretion gear D will be rotated backwards at the same speed (see Fig)

If, on the other hand, the cage be rotated in a forward direction and the wheel C is still rotating forwards in relation to the cage, ~~then D will~~
Assume the speed of the cage to be 250 r/min forward and the gear C turning forwards at say 5 r/mm then gear D still turns backwards at equal speed of 5 r/min. So the actual speed of C is 255 r/min since itsforward motion is added to that of the cage, whilst the speed of gear D will be 245 r/min as its



DIFFERENTIAL

backward motion relative to tho cago is subtracted from tho spe d of the cage. The torques transmitted to each of tho two axle shafts are ocul, and each is oqual to half tho torque applied to the cage throgh the final driver gear.

The total torque is the load on the differential pinion pin, whilst half the torque is the load applied at the pinion teeth, from which dimensions of the pin and gear teeth may be calculated.

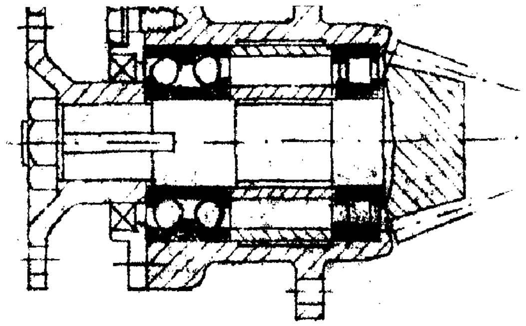
The bevel - and spur gear types of differential are very efficient parts of the transmission, due in the main to the small degroo of friction developed and it is becauso of this efficiency as a copensating device that serval underiablo features occur which after both performance and whool action at tho ground. Should one wheel momentarily slip, or leave the ground for any reason when the vehicle is riding over an uneven surface, that wheol rapidly increases its speed of revolution owing to tho ~~mxgxmx~~ slight internal resisttance offored by tho differential gear.The following roterdation of the whoel speed,as it strikes the ground upon roturn, imparts by way of the differential an impulsive or jorky accoleration to the other wheel,involving a difference in velocity botwoon tho two revolving wheols. This is one of the cases of vehicle skid when there appears to be no apparent reason.

Moreover, should the vehicle be stationarh with one wheel resting upon slipoery or soft ground, th-t wheel will spinupon the application of torque as the differential affords no material resistance. In consequence, instead of propelling the vehicle, it merely sinks lower into the ground

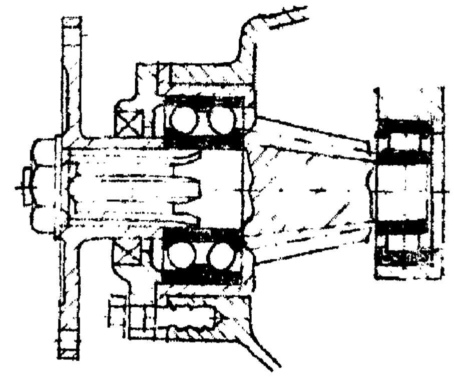
The bevel gear tooth profiles ame partly responsible for the equal division of ~~t~~torque between the driving wheels, as it will be understood th t with involute form of tooth the normal lie that is a line norm-l to the tooth curvature at the point of contact, will pass through thepoint of contact o' the pitch circles and therefore the ratio of tooth leverage between the two axle-shaft gears will be constant with a value of unity.

The tooth design is such that a considerable variation of leverage is allowed between the dodendum and addendum portion of the teeth, and it is apparent that the lever arms very in each position. This being the case, the ratio of axle shaft gear. varies uniformly to that of gear ~~Furthermore~~ when pinion rotates about its axis, thus 1 ◦ 1: 1 : 1 + ~~1:1:1:1~~ Further rotation of pinion changos the ratio again uniformly from 1: :+, 1+: 1.

This cycle of torque r tio distribution may of course occur as many times during therevolution of one driving wheel



BEVEL PINION GEAR SHAFT BEARING



STRADDLE MOUNTING FOR PINION GEAR SHAFT

(assuming th-ttho opposite wheel is provented from rotation) as thoro are teeth in tho axlo-shaft gear. It is claimed that with this poriodic transfer one wheel cannot spin although, for conditions of equal traction of the driving whool and straight ahead driving, the prinion and gears assumo tho position when torque distribution is equalised.

It is therefore desirable, in a gear having plain involute teeth, to provido some degree of friction in the compensating gear, and numborous devices have been designed to this ond. Attempts have also boen made to lock the differential as conditions render such action necessary.

ZF SELF- LOCING DIFFERENTIAL

One of the friction devices which found grot favour on on the Continent is the ZF slf-locking differential, and its basis is that of very low transmission efficiency, or in ort in other words, it possesses considerablo friction. This friction is in turn ocasioned by relatively hgh pressure botwoen the carrier dogs and tho curved paths. ~~XXX~~ whon the vehicle is starting, having resistance at one rear whool only, tho firction is increased by tho back pressuro set up by acceleration of the slipping whool. This high internal fri tion virtually transforms the differential into a self - retarding transmission gear which does not permit wheel-speed difference. It does not, however, dispense with its compensating qualities.

When driving round a bend the differential is controlled by the wheels, the parts transmitting the torque , sch as the axle shaft, being subjected to an angular change in their respective position, whilst the driving parts are relatively stationary. The construction of the differential is simple constituting a cam drive in which one of the axle shaft is connected to the inner and the other shaft to the outer cam body the bevel or worm final drive being effected by the sligind shoes guided in the central cage.

Another form of differential gear employs spur gears instead of bevel gears. It performs similar duties and is widely popular overseas. Wheels are fixed by keys or splines to the two axle shafts. A long spur pinion meshes with wheel the teeth of pinion not quite reaching those of wheel. Another spur pinion meshing with also meshes with wheel . The pinions are carried on pins supported by the differential case , the case in turn forming the mounting for the specific type of final drive bevel worm, or spur gear.

Tor ue is conveyed to gear through pinion and the tooth pressures tend to make pinion revolve upon its pin. This pressure is opposed by pressure between the teeth of the two pinions at the centre which in turn makes pinion revolve. Obviously in view of this a similar condition arises between pinion and wheel should pinion not be rotating on its pin, then the two pressures acting upon it must be eqal and therefore equality exists between thepressures on the teeth of gears and pinion. Torques therefore, are also equal.

Several sets of spr pinions are usually employed, each pair being equally spaced around the periphery of gears. If the differential case is held to prevent rotation and one wheel is rotated in anti-clockwise direction, then pinion will rotate in clockwise direction, and pinion in ~~xxxx~~ and other pinion in anti-clockwise rotation other wheel will obviously rotate clockwise. If, therefore, oneof the differential wheels rotates faster than the case, the other wheel rotates exactive the same amount slower than the case; this is a similar action to that which occurs in tho beveltype differential.

The question of friction has been dicussed and some of the reasons for its desirable presence and in this connection, in order to prevent or to control the degree of wheel slip, resort is sometimes made to a differential lock. Such mechanisms provide for the locking of one or both differential wheels to their causing or by the prevention of rotation of the pinions on their pins or "stars", in which case the differential ceases to function and the whole of the torque is then applied to the crown wheel and transmitted to whichever road wheel retains its adhesion with the road surface. The importance of such a mechanism is more apparent in the corss-country vehicle or the tractor than in a light truck or passenger car, when the ground surface may vary between wheel and wheel. The absence of such a lock may under such circumstances render the vehicle totally inoperative, since any applied torque to a slipping wheel merely aggravates that condition.

The splines for the driving of the differential wheels will be assumed to be of the involute type, the basic formulae for tooth proportions being well known as follows:

$$\text{Circular pitch} = \frac{\quad}{\text{Diametral pitch}}$$

$$\text{Addendum and dedendum} = \frac{0.50}{\text{Diametral pitch}}$$

$$\text{Circular tooth thickness} = \frac{1.570}{\text{Diametral pitch}}$$

$$\text{Pitch diameter} = \frac{\text{Number of teeth}}{\text{Diametral pitch}}$$

$$\text{Major diameter} = \frac{\text{Number of teeth plus one}}{\text{Diametral pitch}}$$

$$\text{Major diameter} = \frac{\text{Number of teeth minus one}}{\text{Diametral pitch}}$$

whilst they apply equally to internal and external shafts.

It have previously been seen the weakness of the straight spline with its undercut at the spline base, but with the involute type torsional tests have shown that the strength is equivalent to a shaft larger than the minor diameter, which indicates that with the fillet the spline teeth actually add strength under fatigue loading

Moreover the involute provides greater bearing surface and tooth contact and this permits closer fits with decreased noise and greater possibilities of interchangeability. From these aspects alone it is highly probable that the straight sided spline will eventually be superseded in favour of the involute type.