

**DESIGN AND CONSTRUCTION OF A STARTER MECHANISE
FOR APPLICATION IN A TWO-STROKE ENGINE**

BY

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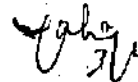
A Thesis submitted to the postgraduate
School Ahmadu Bello University, Zaria, In partial
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the Degree of Master of Science (M.Sc.) In Production
Engineering.

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OCTOBER, 1990

DECLARATION

I declare that this thesis has been composed by me and that it is a record of my own investigation. It has not been submitted in any previous application for a degree. All quotations are distinguished by quotation marks and the sources of information are acknowledged by means of references.


Y. Y. YUSUF

October, 1990

CERTIFICATION

This thesis entitled, "Design and Construction of a Starter Mechanism for Application in a Two Stroke Engine", by Yahaya Ya'akub YUSUF, meets the regulations governing the award of the degree of Master of Science (Production Engineering) of Ahmadu Bello University, and is approved for its contribution to knowledge and literary presentation.



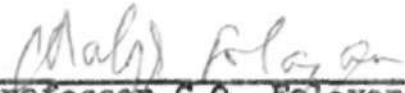
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In His Name everything is what it is. There is no doubt all praises are due to Him alone.

Oh my God and the God of everything, the only one God. Muhammed Abdullah Ahmed Hamid, Al-Musthapha, Your blessings be upon him, by whose light you animate every heart.

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ABSTRACT

In accordance with the objective of the Federal government of Nigeria to support the development of low-cost vehicles based on locally available technology, the Department of Mechanical Engineering of Ahmadu Bello University was given the task of developing the engine. The engine, for a start, is a 2-stroke, 2-cylinder petrol engine

Part of the engine is an overrunning starter Mechanism which was designed, constructed and tested in this project.

The overrunning starter Mechanism is in the form of a freewheeling clutch which transmits torque in only one direction and overruns in the other direction.

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NOMENCLATURE

- A - addendum, mm
- $\bar{A}$ - tensile stress area of the bolt, mm²
- B - factor to take care of the inertia of parts
other than the rim of the clutch
- BCD - base circle diameter, mm
- b - width of the tooth, mm
- D, d - diameter, m (mm)
- E - kinetic energy of the flywheel, Nm
- $\bar{E}$ - modulus of elasticity, N/m²
- F - force N
- FL - free length of the spring, mm
- G - modulus of rigidity, N/m²
- H - dynamic capacity
- h - life of the roller bearings in hours
- h^- - height of the tooth profile, m
- I - moment of inertia, kgm²
- i - current, Amperes.
- $\bar{I}$ - moment of inertia about the bending axis, m⁴
- J - polar moment of inertia, m⁴
- k - radius of gyration, m
- $\bar{k}$ - spring constant, N/m
- L - length, m
- M - mass, kg
- $\bar{M}$ - moment, Nm
- N - angular velocity, rev/min
- n - number

x

OD - outside diameter of the ring gear, mm
 PC - circular pitch, mm
 PD - pitch diameter, mm
 Pd - diametral pitch, mm
 PO - power, Watts
 R - rim radius, mm
 $\bar{R}$ - coefficient of velocity of variation of the flywheel.
 RC - root circle, mm
 r - radius, mm
 S - stress along line of contact of rollers and racing members of the clutch, N/m^2
 SF - service factor
 SL - solid length of the spring, mm
 T - torque, Nm
 t - time, sec.
 $\bar{t}$ - number of teeth
 $\underline{t}$ - thickness of the tooth profile, mm
 V - velocity, m/s
 V_0 - voltage, volts
 X - spring deflection, mm
 y - form factor
 Z - section modulus, mm
 α - angular acceleration, rev/sec^2
 σ - allowable stress, N/m^2
 ϕ - pressure angle
 ω - angular velocity, rad/sec.

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- θ - angle formed by a radius passing through the centre of the roller and a perpendicular from roller centre to its point of contact with the outer racing member.

Subscripts

- 1 - refers to initial
2 - refers to final
av - refers to average
bo - refers to bolt
c - refers to clutch
co - refers to coils
cr - refers to crank
d - refers to driving
ex - refers to external
f - refers to flywheel
ge - refers to gear
in - refers to internal
j - refers to Journal
l - refers to Lever
no - refers to normal
pi - refers to pinion
R - refers to resultant
r - refers to rollers
rtd - refers to rated
ta - refers to tangential
w - refers to Webs
wi - refers to Wire.

CHAPTER 1INTRODUCTION1.1 Previous Work in Auto-technology in Nigeria

Very little progress has been achieved in the field of Automobile in Nigeria. This is because very little deliberate efforts have been made to make a break through in auto-technology.

Work on automobiles in Nigeria may be either of the followings or a combination of any of them. They are as follows:-

- (i) First investigation
- (ii) Design and development
- (iii) Adaptation
- (iv) Copy, and
- (v) Assembling

There is no set up in Nigeria that is fully engaged in first investigation in the field of automobiles. The aim of first investigation is to come up with new ideas to replace the old ones, and these ideas must be supported with data or performance chart to determine the superiority of the new idea. However, a new idea may not be necessarily superior to the old one, but the proponent of the idea must prove that it is adequately efficient and adaptable.

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An example is the "Testing of Locally Made Air Filters".⁽¹⁾ The filters are specially made plastic foam by the Royal Foam Factory, Shereda Industrial Estate, Kano. A standard test rig was built, then the efficiency of the filter and the pressure drop across it were determined. As soon as it was proved that it is efficient and adaptable, it was tried on a Peugeot 505 Saloon for more than a year and there was no problem. It even proved more renewable than the paper element filter. However, its commercial production is yet to be carried out.

An example of adaptation is the work of the late Professor Awojobi on a military vehicle. He introduced a second gearbox and steering wheel, so that the vehicle need not make a turn the way conventional vehicles do in order to move backward, and yet not in a reverse mode. Hence when it is moving backwards, its movement is as good and convenient as when it is moving forward. During the war if such a vehicle encounters the enemy, it will return to its source quickly and at such a pace as it is moving forward. There is ease of manoeuvring. This vehicle is known as Autonov.⁽²⁾

Assembling is the oldest form of auto-technology to be introduced in this country. The assembly plant is set up here and fed with Completely Knocked Down (CKD) parts from overseas. Examples are Peugeot

Automobiles of Nigeria (PAN) Ltd., and Volkswagen of Nigeria.

The CKD parts are imported from France and West Germany respectively. The original intention is transfer of technology through gradual increment of local contents. For a very long time this objective was not achieved except recently that few of the parts are procured locally.

1.2 Low-Cost Vehicle Project

Around October 1986 when the Federal Government of Nigeria introduced the Second tier foreign Exchange Market (SFEM), prices of automobiles went up abruptly due to the depression of the value of naira.

The price of the most costly Peugeot Automobile of Nigeria (PAN) product just before this period was not more than ₦20,000.00. The price of Volkswagen Beetle was around ₦5,000.00. As soon as SFEM was introduced, a Peugeot product costs about ₦60,000.00 and Volkswagen for about ₦15,000.00. As the value of naira continues to depreciate, prices of vehicles and spare parts continue to increase.

In May 1988, 504 GR/base sold for ₦45,980.00 while 504 GR/AC cost ₦55,000.00⁽³⁾. Today the price of 505 Peugeot saloon is about ₦120,000.00.

Though the government has tried to cushion the situation by increasing transport allowance for Civil servants over the years, the impact of this measure is not felt due to simultaneous increase in transportation cost.

To alleviate the problems the government had, at one time, allowed importation of spare parts with very little import duties. In addition the state governments have involved themselves directly in commercial transportation by mass importation of buses for mass transit exercise. All the measures depicted above cannot be lasting solutions to the problems of transportation and mass mobility in this country. Hence the Federal Government decided to address itself to the real issue.

In 1987, the Federal Ministry of Industries invited the Ahmadu Bello University "and at the meeting spelt out its intention to develop a vehicle using available local technology. A home grown vehicle in terms of materials and technology".⁽⁴⁾ Consequently in November the same year the Department of Mechanical Engineering of A.B.U. and Kwara College of Technology were commissioned to design the engine, and the body and the transmission system, respectively.

Different alternatives were considered as regards where to start from. The options available are designs of four stroke engines or two stroke engines.

Four stroke implies that the cycle of operation is completed in two revolutions of the crankshaft. There are four operations within a cycle of operation and each stroke is used to effect one. Assuming that the piston is at the top dead centre. If it is made to move down the cylinder, with the inlet port opened and the exhaust port closed, a mixture of air and fuel is admitted into the cylinder. This is the induction stroke. During the second half of the first revolution of the crankshaft, the piston moves and the mixture is compressed into the top of the cylinder while both ports remain closed. This is known as the compression stroke. A sparking plug is provided at the end of the cylinder which induces an electric spark resulting in the burning of the compressed mixture. The high pressure created due to the burning of the petrol vapour forces the piston down the cylinder, with the two ports closed. This is called the power stroke, and it represents the first half of the second revolution of the crankshaft. The exhaust stroke takes place during the second half of the second revolution of the crankshaft. The piston moves

up to drive the burnt gases out of the combustion chamber. (see fig. 1.1).

This gives a complete cycle of operation and the inlet port reopens while the exhaust port closes for the next cycle of operation to commence.

Two stroke implies that the cycle of operation is completed in a single revolution of the crankshaft. In other words, two operations take place during one stroke of the piston, fig. 1.2 is a schematic representation of the operating principle.

In (1) all the ports are closed, the piston is rising and therefore the mixture is compressed. At the top dead centre (t.d.c) the fuel vapor is ignited while the inlet port is opened to admit fresh mixture.

Compression and induction have occurred in a single stroke. The compressed petrol vapor burns rapidly resulting in elevated temperature, and expansion of the gases. The piston is forced down, the inlet port closes, the exhaust port opens to allow gases to escape while fresh mixture is drawn from the crank-casing into the combustion chamber through a transfer port. The piston rises up again for another cycle of operation.

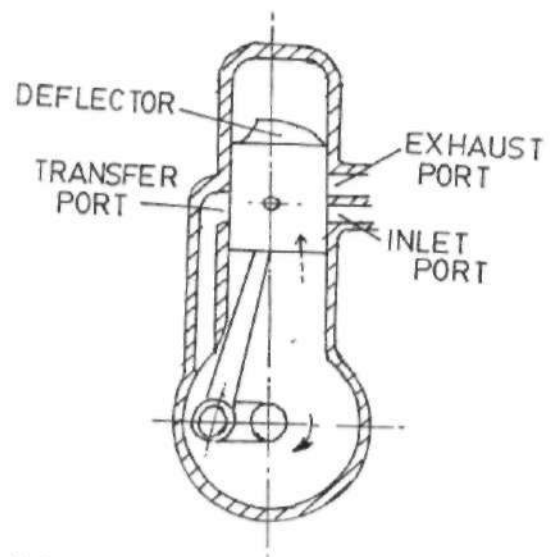
There is only one 'working stroke' for three 'idle strokes' in a four stroke cycle. In a two

stroke cycle there is only one 'idle stroke' for every 'working stroke'. However, the operation is not efficiently performed due to the fact that the burnt gases are not expelled completely before fresh charge is allowed into the chamber. Despite this, the two-stroke cycle is sometimes adapted because of its simplicity, fewer parts, easy manufacture and smoothness.

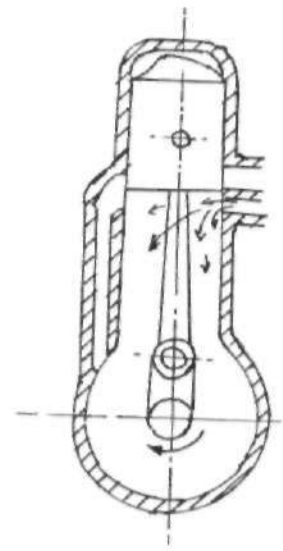
Even the principle of operation depicted above manifests the simplicity of a two-stroke engine when compared to four stroke engine.

Hence it was decided that the project should start with the development of a two stroke engine.

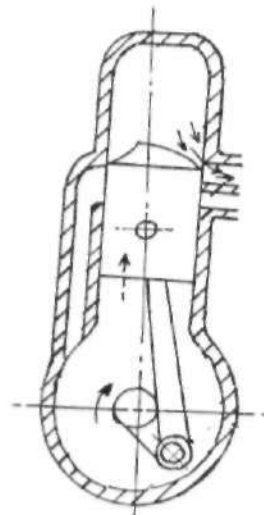
The engine under development is a 2-cylinder, petrol engine with a rated power of 9Kw. In this project a mechanism for starting the engine is designed, constructed and its functionality tested.



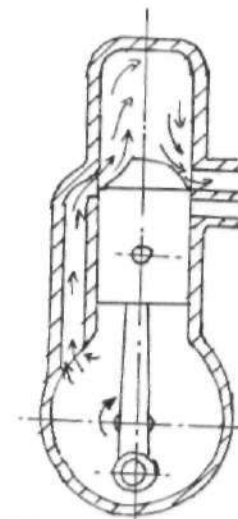
(a) MIXTURE COMPRESSED ABOVE,
DECREASING PRESSURE BELOW,
ALL PORTS CLOSED



(b) PISTON PASSING T.D.C.,
MIXTURE ABOVE IGNITED,
FRESH MIXTURE ENTERING
CRANKCASE



(c) PISTON MOVES DOWN
COMPRESSING THE
CRANKCASE MIXTURE.
EXHAUST PORT JUST
UNCOVERED



(d) PISTON
PASSING B.D.C. EXHAUST AND
TRANSFER PORTS OPEN. MIXTURE
ENTERING CYLINDER DIRECTED
TO THE TOP BY THE DEFLECTOR, DRIVING
OUT BURNT GAS THROUGH EXHAUST
PORT

FIG. 1.2: Operation of an engine on a two-stroke cycle

CHAPTER 2LITERATURE REVIEW

2.1

CLUTCHES

A clutch is a mechanism used to engage and disengage shaft at will. Clutch is employed to perform several functions such as transmission of torque, maintain constant speed and power, quick starts and stops, gradual starts, non-reversing (back stops) and overrunning functions. Various criteria are used to classify clutches. They may be grouped into Mechanical, Electrical and pneumatic or hydraulic by virtue of their mode of control. Clutches are also classified on the basis of the principle of operation.

Below is a review of the major types of clutches.

2.1.1 Positive Engagement Clutches

Positive-contact clutches are of three types namely square-jaw clutch, spiral-jaw clutch and tooth clutch. While in operation they interlock to produce a mechanical joint.

Square-jaw and spiral-jaw types consist of two halves, one of which is made to travel axially on splines to effect connection and disconnection of the clutch. The former transmits power in both directions whereas the

later transmits power in only one direction of rotation thus exhibiting overrunning clutch characteristic.

Positive clutches are simple, inexpensive and generate no heat. However, they are limited to small torque and low speed applications. Square-jaw clutches transmit torque with virtually no axial component of force.

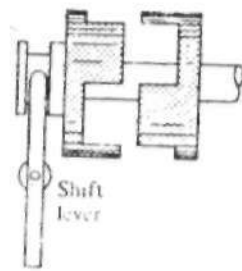
Tooth clutch, also referred to as slip clutch, has teeth on the two faces of the two halves. Various modifications of the teeth are available including rhomboidal and triangular shapes. These contacting surfaces give a component of the transmitted torque in the axial direction.

Sketches of these three types of positive engagement clutches are shown in fig.2.0.

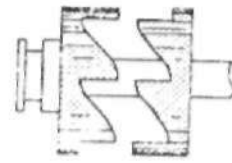
2.1.2 Friction Clutches

When it is desirable to engage or disengage a power source and a driven machine with the power source in operation, friction clutch is employed. Friction clutch functions as an automatic overload protection device. When the torque reaches a pre-determined maximum, the surfaces slip to protect the machine.

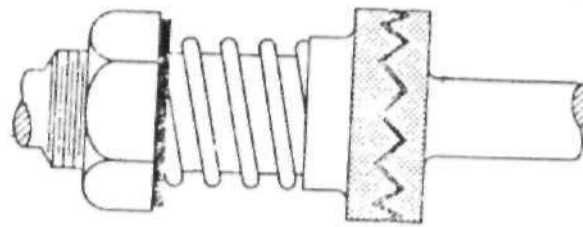
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Square-jaw clutch.



Split-jaw clutch.



Spring-loaded slip clutch.

Fig. 2.0: Positive engagement clutches

Friction clutch could be used to disengage the driver from the load to effect starting the driver under no-load condition. It is also employed where it is necessary to start or stop the load without stopping the power source. The former application is found in internal combustion Engines as clutch power take-off and the latter in punch press where the clutch is "engaged for each stroke of the press without stopping the motor" .

Friction clutch is the most common of all clutch types and the most versatile in application . Either one or both of the mating faces of the two halves are made of friction materials. The surfaces may be either flat or cylindrical. When the two faces are properly engaged, the clutch members rotate as one unit. Slippage occurs when the power transmitted is greater than a certain maximum. This feature averts shocks and overloads. However this is an unavoidable disadvantage in that it creates heat dissipation, and thus a limitation.

Depending on the design, friction clutch may slip continuously for overload protection, or used as an "on-off" device where slippage occurs intermittently.

Types of friction clutches

Axial types: These clutches derive their name from the fact that they rotate in an axis perpendicular to the face. The faces employed is either of the disk or plate form. A very simple example is shown as in fig.2.1. It consists of two disks held at the end of two coaxial shafts, with a device to effect engagement of the surfaces.

Axial clutches that are designed with sleeves, in which wheels are mounted, are known as sleeve clutches. Alternatively the sleeve could be used as the wheel itself. Friction clutch with sleeve is as shown in Fig.2.2. The friction surfaces are brought into grip in a vise-like arrangement. The internal pressure created is extremely high. A typical value for "utility clutch rated for 40hp at 100 rpm is something over 10,000 lb⁽⁵⁾).

More than one disk could be employed and the more the number of disks the greater the torque capacity of the clutch. Again there is attendant problem of excessive heat dissipation and thus a limitation. Overheating creates difficulty in separating the disks. Multiple disks and thus more friction surfaces, are

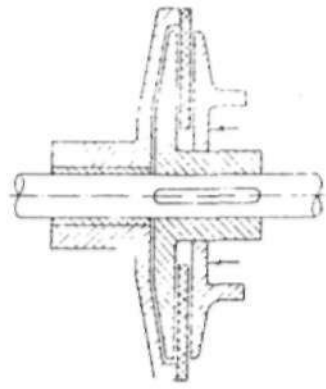


Fig.2.1: Axial friction clutch

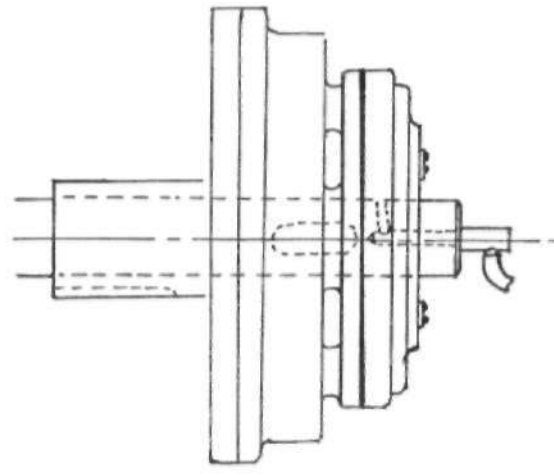


FIG 2.2: Air clutch with sleeve

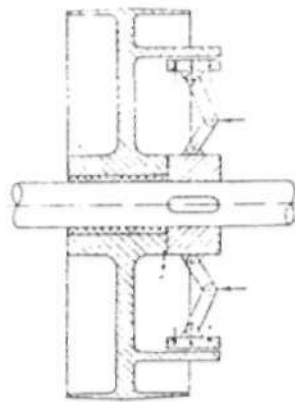
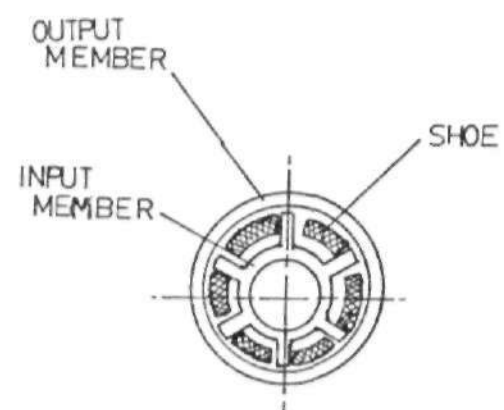
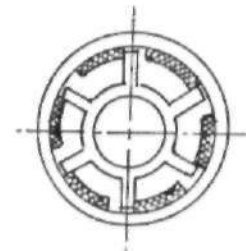


Fig.2.3: Radial friction clutch



(a) Free



(b) Engaged

FIG 2.4 Centrifugal clutch

advantageous in static or low-speed use. In high-speed dynamic application less disks are employed so as to reduce the danger of heat dissipation.

Radial types: The axis of operation of this type of clutches is the same as that of the shaft. The designs of these types of clutches are well diversified. The variations are in the design of the mechanism otherwise the friction surfaces or the contacting surfaces are practically of the same form. They are cylindrical.

The different versions include Ring, Shoe, Band and Rim clutches.

Fig.2.3 and Fig.2.4 are illustrative examples of radial type of friction clutches. Fig.2.4 is also referred to as centrifugal clutch because the shoes are caused to move out radially for the clutch to engage by means of centrifugal force. The other force that comes into play is the friction force. The engagement is enhanced with increasing magnitude of the centrifugal force and thus the friction force. It operates under no-load starting condition. The prime mover starts running untill when it reaches a certain speed, the speed at which the centrifugal force is enough to effect engagement of the clutch. Obviously this guarantees smooth pick-up of load

with minimum shock. Prime movers of low starting torque characteristic find suitable application with centrifugal clutches.

Free and delayed engagement characteristic could be achieved if shoe springs are provided. Depending on their stiffness the springs work against the centrifugal force and return the shoes to the inner member when it is necessary to disengage the clutch.

Dry-fluid clutch is another form of centrifugal clutch. A schematic representative of the clutch is shown in fig.2.5. It consists of a housing containing heat-treated steel shot, known as the "fluid" (and often referred to as the flow charge), a rotor suspended in the housing and attached to the driven shaft. When the prime mover is started the housing rotates with the motor shaft thus throws the flow charge to its perimeter. The rotor then tends to cut through the charge, encounters some resistance and finally rotates with the housing. Eventually they both "achieve full-load speed, operating without slip and with 100 percent efficiency" (6).

Fig.2.6 are examples of Rim or Drum type clutches. It is an external contracting clutch supplied with a hollow tube round the periphery

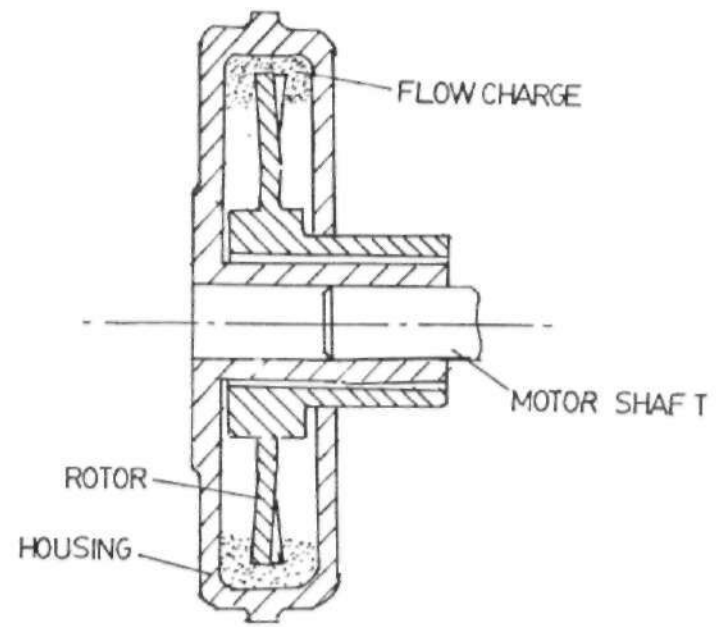


FIG2.5: Dry fluid clutch

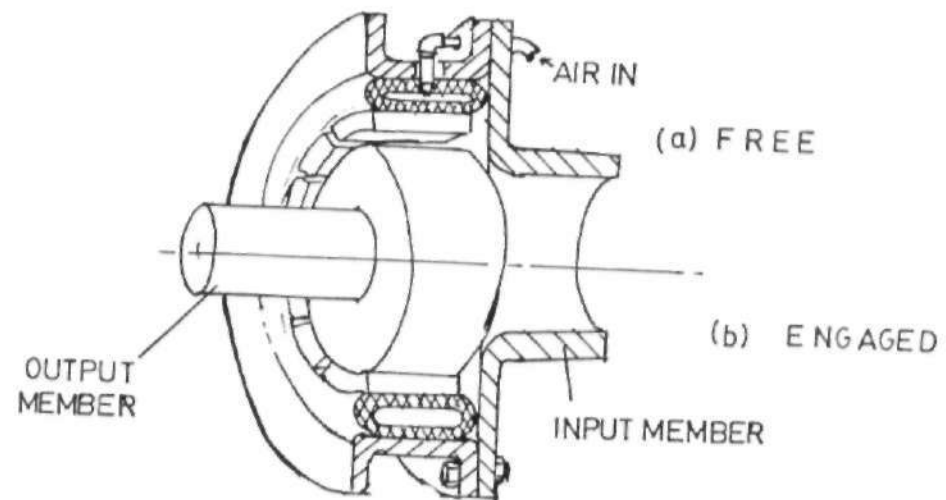


FIG 2.6: Drum type clutch

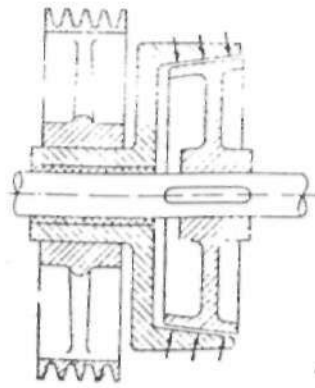


Fig.2.7: Cone friction clutch

of the contacting friction surface. When the tube is inflated it expands, contracts the circular shoes which grip the inner member. As the pressure increases, 360° contact surface is established. This allows even distribution of pressure and load, and averts uneven wear of friction surface. The rubber damps out vibration and shocks. Centrifugal force tends to pull the friction shoes out radially but this "is overcome by increasing the air pressure more than it is required for engagement". Centrifugal force is, however, advantageous during disengagement. Contracting shoes clutch applications include cyclic and continuous operations.

Another modification of radial-type friction clutch is as shown in fig.2.3. What appears like a lever is pushed inward and the circular friction surface moves out radially to ensure engagement of the clutch.

In fig.2.7 the friction surface is modified to a cone-shaped form. It functions by simply setting up a wedging action between the two cones. The contacting surfaces grip themselves the way a tapered cork is pushed into a bottleneck. This is coined as a cone clutch. It is suitable for heavy-power-transmitting torque. The grip occur

very easily but it is a bit difficult to disengage the clutch. It is rarely used nowadays because other clutches have overtaken it.

Hitherto, clutches that seek advantage of two surfaces that lock-up at some speed to effect operation because both or one of the surfaces is provided with friction materials have been discussed.

What are these friction surfaces like and what are they made of?

They are made of bare iron or steel, lined with metallic or non-metallic materials.

There are varieties of non-metallic linings. "Woven" and "molded" linings are the common ones. Woven type is an asbestos in the form of a woven fabric which is bound together with resinous compounds. Molded type is also made of asbestos, which has been softened by soaking in water (or caustic potash). In both types reinforcement could be ensured by using brass.

Cork filler bonded together with resinous compound is also employed as a non-metallic lining. Thermosetting adhesives is used to provide bonding between the linings and a metal backing plate.

Protein and synthetic type of binder could be used with asbestos filler and cork filler in all the instances outlined above in their respective applications.

The coefficient of friction of asbestos lining ranges from 0.2 to 0.4 and that of cork ranges from 0.10 to 0.50. The exact value strictly depends on the operating conditions such as speeds, pressure, design and lubrication.

Steel and bronze made with sufficient coefficient of friction are employed as metallic friction surfaces. The coefficient of friction ranges from 0.20 to 0.25. Sintered bronze is also employed in which case it is bonded to steel backing. Compact designs, good wear, heat resistance, high rate of heat transmission, increased allowable pressures, smooth starting, are all the advantages of this type of linings.

The exact performance characteristics of any lining material cannot be stated in this review. It depends on factors such as the composition and the operating condition. Only the manufacturer can explicitly state them.

Friction clutches surpasses other clutches in the scope of application. This is mostly because it can slip when required. This slippage

characteristic poses the problem of heat dissipation and wear. Hence a friction clutch is often regarded as a benefactor as well as a matyr. In order to minimize the adversity inherent in this conflicting behaviours of friction clutches, proper selection, installation and maintenance must be carried out.

In selecting a friction clutch just like any other type, there are two most important things to be considered. First, the suitability of the clutch and then its size.

Perhaps the best way to go about determining the most suitable type is by reflecting on the various advantages and disadvantages of every known friction clutch. Having known the job the clutch is required to do, therefore, the next thing is to consider the various features all the clutches have.

The merits and demerits of some friction clutch types are outlined in table 2.0 below. The classification is based on the mode of control.

Table 2.0: MERITS AND DEMERITS OF VARIOUS CLUTCH TYPES

Clutches	Advantages	Disadvantages
Mechanical clutches	- Cost of obtaining them and operation is low	- Poor sensitivity

Clutches	Advantages	Disadvantages
Air clutches	- Can be located any where on a shaft - Variable torque capacity	- Require the operator to expend a lot of energy - Cannot be adapted to remote control - Precise and accurate adjustment cannot be obtained.
	- No problem of adjustment - Reasonably good sensitivity - Can be adapted to remote control - Wide spectrum of torque capacity	- Located near the free end of a shaft only. - Require specially drilled shafts - Require costly auxilliary equipments, though not in all cases

Clutches	Advantages	Disadvantages
Magnetic clutches	- Can be located anywhere on a shaft - Easily adaptable to remote control - Reasonably good level of sensitivity can be obtained - Gradual starting is possible - Possibility of achieving variable torque	- They require constant d-c supply - Current is consumed throughout the engagement cycle - Elaborate control apparatus is required.

Capacity is offered top priority in determining clutch size. A clutch with rating torque slightly more than the torque peak for the operation in which it is used experiences less wear, tear, reduced stresses and the rate of heat absorption is enhanced. However, where heat dissipation is a necessity, such as frequent starting of heavy-inertia load an oversize clutch may not necessarily be the panacea. The heat problem must be considered or taken care of.

There is no direct way of dealing with this problem. Different manufacturers have different ways of approaching the problem, which is based on

their experience. In all, they only lessen the danger imposed on the clutch but it is not completely eliminated.

The necessity for proper installation of clutches cannot be over emphasized. Wrong installation adversely affects its performance. Installation manuals from manufacturers are available but the best installation comes with experience.

Very often, friction clutches are regarded as trouble-makers. But if the right clutch for a job is selected, properly installed and given adequate maintenance it will run very smoothly for a very long time. Record even shows that some clutches have been employed for about 30 years .

2.1.3 Overrunning clutches

Freewheeling clutches have some features which make them more suitable for certain operations. They transmit torque in only one direction and overruns in the other direction. Engagement and disengagement occur rapidly and automatically.

There are several variations of overrunning clutches. Few examples are shown in fig.2.8 and 2.9

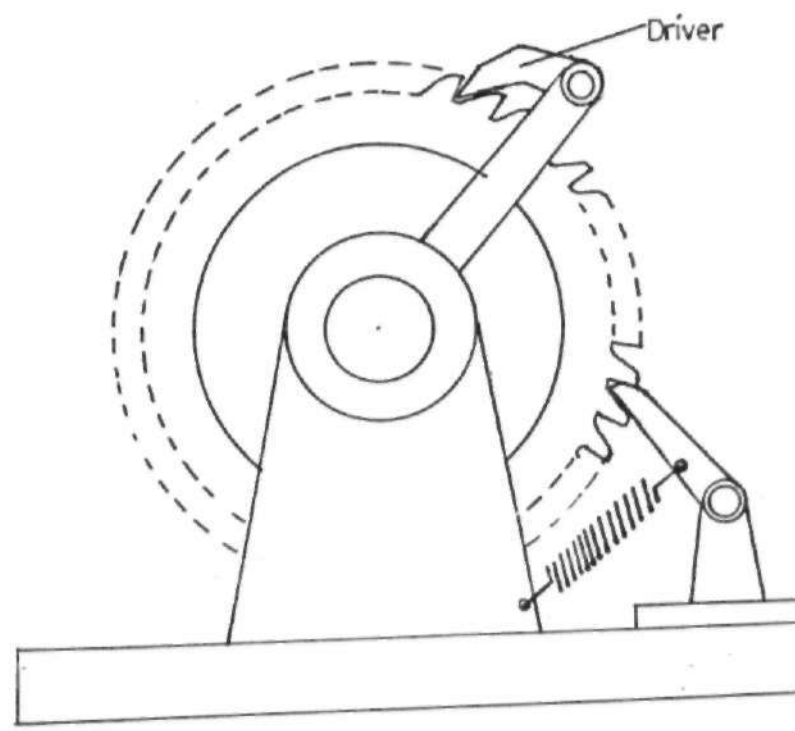


Fig.2.8 : Ratchet and pawl mechanism

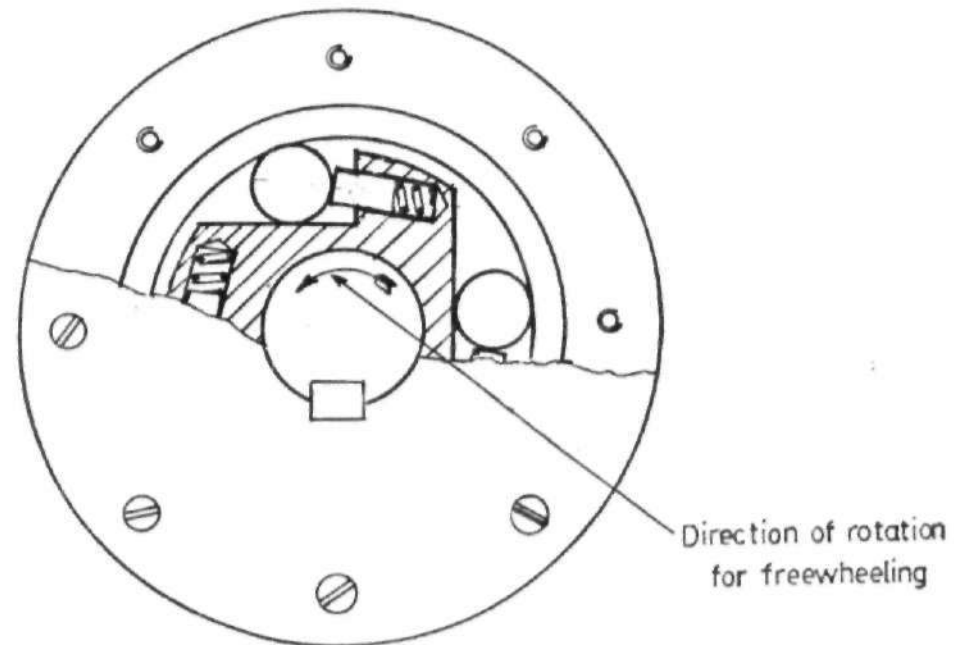


Fig. 2.9: Overrunning clutch roller type

Fig.2.8 is essentially a Ratchet and Pawl mechanism. Gravitational force engages the driving pawl, which is at the top. While the driving pawl shifts back for another turn, the down pawl keeps the ratchet in position. When in operation the driving shaft is made to undergo either reciprocating or oscillating movement. This movement is converted to an intermittent rotary motion. With proper design the motion is positive. This diagram illustrates an elementary overrunning clutch. It is applied in feed mechanism.

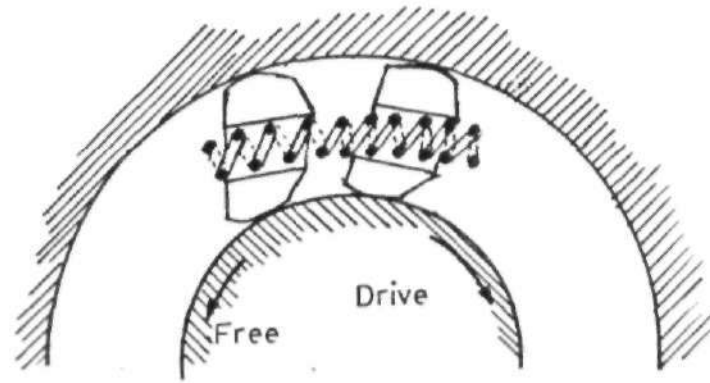
Fig.2.9 consist of two races - the inner and the outer. Any of these members can be the driver and the other the driven. When the driver rotates, torque is transmitted to the driven under the elastic tension of the rollers. There are springs acting against the rollers via push rods, so when the clutch is in operation the rollers are wedged continuously between the cam surfaces and the outer race, hence the engagement is positive. As soon as the prime mover is started, this clutch acts instantly and almost automatically. In addition, the operation is quiet. When the inner member rotates in anticlockwise direction the members freewheel. This particular design of overrunning

clutch finds applications in internal combustion engines and bicycle coaster brakes.

Fig.2.10 is also a commercial overriding clutch with two concentric circular races having sprags between them instead of rollers as in the foregoing description. The rockers are used to transmit torque. Energizing springs help to keep them in contact with the two races. This design accommodates greater number of sprags hence it transmits more torque than roller clutch of the same diameter. Fig.2.11 illustrates the principle of operation. When the inner race rotates anticlockwise the clutch freewheels. On the other hand when it rotates clockwise the sprag is tightly wedged between the races and rolling action occurs. The motion is not completely positive, smaller wedging angles results in greater backlash or lost motion.

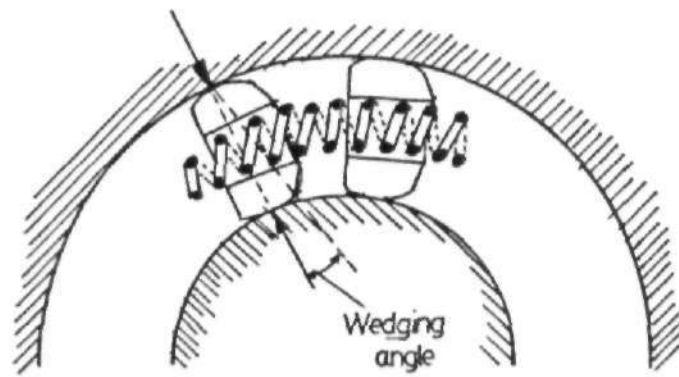
Where rapid operation is preferred to space limitation the roller type is considered.

Overrunning clutches are widely employed. Applications include rotor freewheeling in helicopter drives, intermittent motion in feed mechanism,



Sprag in overrunning position

Fig.2.10: Overrunning clutch sprag type



Sprag under wedge torque

Fig.2.11 Principle of operation of a sprag clutch

bicycle coaster brakes, starter clutches in automobiles, precision devices, fan freewheeling and punch press feeds.

2.1.4 Magnetic clutches

Magnetic clutches operate with the aid of magnetic forces that are set up between the clutch members. There is no mechanical contact between the faces of the rotating members.

"Magnetic flux couplings" or "magnetic drives" are other names used to denote magnetic clutches. These names clearly differentiate them from electrically actuated friction clutches in which magnets are primarily used for control.

There are three types of magnetic clutches namely magnetic-particle clutches, Hysteresis clutches and Eddy-current clutches.

In magnetic-particle clutches the space between the members is filled with ferromagnetic powder. The ferromagnetic powder is a mixture of iron particles and lubricant. A magnetic field is set up in the working gap. A direct-current control coil creates a magnetic circuit in the gap, as a result of which the magnetic medium tends to shear with great difficulty.

The difficulty in shear encountered by the medium, which is a function of the strength of the field, causes the clutch to drive.

Dry lubricant such as flake graphite is employed in this type of clutch. However, when an oil slurry is employed, the clutch is referred to as a magnetic-fluid clutch. Fig. 2.12 is a schematic representative of magnetic particle (or fluid) clutch depicting the rotating members, the working gap, excitation coil and the stator.

Hysteresis clutch consist of two rotors. One is magnetic and houses a rotating magnet coil, the other is made of high-hysteresis hard-steel. The former is placed inside the later with a very small air gap between them. Very strong intrinsic poles on the magnetic rotor induce magnetic poles on the steel rotor, which is equally very powerful. Once these poles are set up they cannot be displaced. Hence when the driver which is the steel rotor is rotated, it rotates with the driven member as if they were a single unit.

Unlike the magnetic-particle clutches, hysteresis clutches are not designed for zero slip. They slip when the load exceeds the torque rating. Fig. 2.13 illustrates the arrangement of the components of hysteresis clutches.

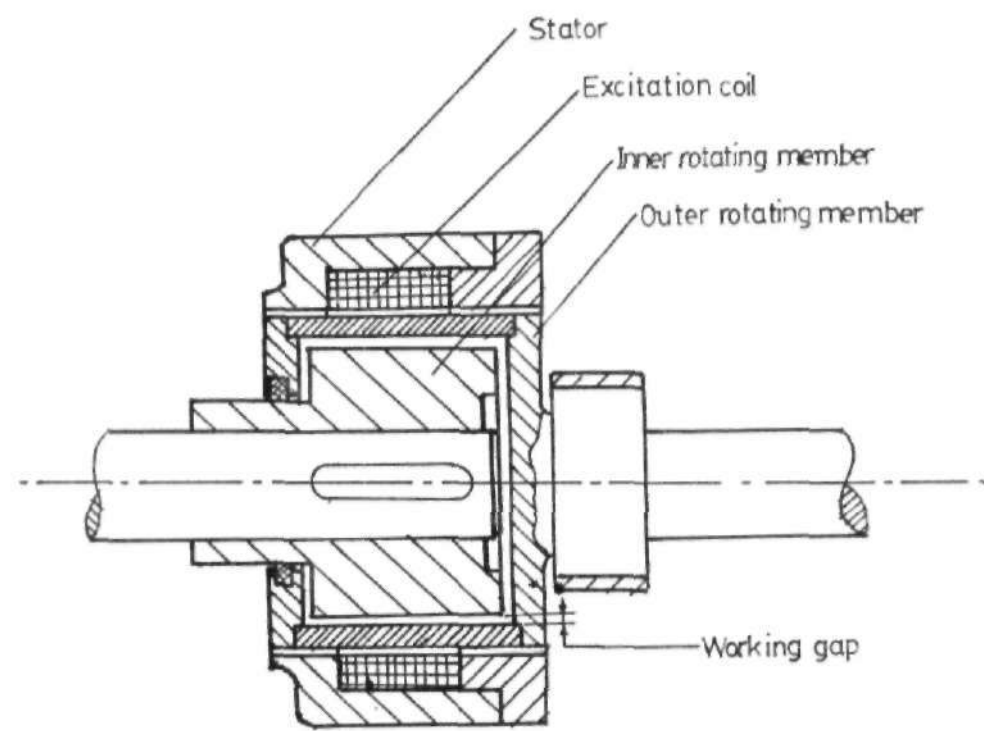


Fig.2.12 Magnetic-particle clutch

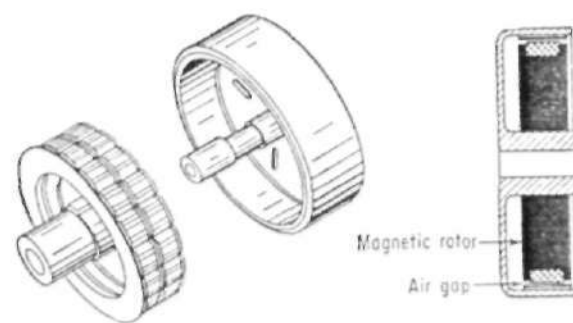


Fig.2.13: Hysteresis clutch

Eddy-current clutches are somewhat similar to the hysteresis clutches in construction. The difference is that the high-hysteresis hard-steel is replaced with soft-iron or special mild steel "and the magnetic segments are interconnected to induce eddy current and inhibit hysteresis". So, the characteristic operation is different.

2.2 Flywheel

It has been mentioned in section 1.4 of chapter one that there is either a power stroke for three idle strokes in a ~~four-stroke~~ engine or one idle stroke in a two-stroke engine. During the other strokes in which power is not developed, the piston is still required to move up and down the cylinder. This implies that the energy requirement of a motor vehicle engine varies during its cycle of operation, and the rate of power supply is constant. To assure smooth operation of the engine a flywheel is attached to the crankshaft. During the power stroke, it stores the excess energy as kinetic energy of rotation. Some of this energy is supplied to the machine during the other strokes to maintain its speed. This is the essence of a flywheel.

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2.3 STARTER MECHANISM

Above, various clutches and the importance of a flywheel have been discussed. The starter mechanism intended to be designed consist of a starter clutch and a flywheel. The starterclutch is a freewheeling type. It has two racing members, one of which is the driver and the other is the driven. A starter motor drives the whole unit. Once the motor is started, the driver engages the driven through a wedging action of three rollers and torque is transmitted under the elastic tension of the rollers. The rollers maintain contact with the racing members with the help of springs.

The clutch is bolted to the flywheel. When the clutch is actuated, the flywheel accelerates to a speed higher than that of the driving member of the clutch and the clutch freewheels.

CHAPTER 3

DESIGN AND SELECTION OF MATERIALS3.1 FLYWHEEL DESIGN

Variation in stored kinetic energy of the flywheel is given by

$$E_2 - E_1 = \frac{1}{2} I_f (\omega_2^2 - \omega_1^2) \text{ ----- (1)}$$

where $I_f = MK^2$.

Therefore,

$$E_2 - E_1 = \frac{1}{2} MK^2 (\omega_2^2 - \omega_1^2)$$

Assuming the radius of gyration is very, nearly equal to the average rim radius.

Therefore,

$$E_2 - E_1 = \frac{1}{2} M(R_{av}^2 \omega_2^2 - R_{av}^2 \omega_1^2) \text{ ----- (2)}$$

$$\therefore E_2 - E_1 = \frac{1}{2} M(V_2^2 - V_1^2) \text{ ----- (3)}$$

$$\text{Let } \frac{V_2 - V_1}{V_{rtd}} = \bar{R} \text{ ----- (4)}$$

$$\text{Assume } V_2 + V_1 = 2V_{rtd} \text{ ----- (5)}$$

Combining equations (4) and (5)

$$V_2^2 - V_1^2 = 2\bar{R}V_{rtd}^2 \text{ ----- (6)}$$

Substituting equation (6) in (3) gives

$$E_2 - E_1 = M\bar{R}V_{rtd}^2 \text{ ----- (7)}$$

If the part of the flywheel other than the rim gives a flywheel effect of 5 to 15 percent of the rim and a factor B is introduced such that:-

$$E_2 - E_1 = BMRV_{rtd}^2 \text{ ----- (8)}$$

B ranges from 1.05 to 1.15⁽⁷⁾

Empirically the safe working stress of the material will not be exceeded if the following rated velocity is adhered to⁽⁵⁾.

Cast	iron	36.6m/s
Cast	steel	91.5 "
Forged	steel	152.5 "
Welded	rolled	152.5 "

Equation (8) is used to determine the mass of the flywheel if the energy and speed variation of the engine are selected or known.

The followings are the estimated values of some of the performance indicators of the engine.

Speed variation	=	4,950 - 5050 rpm
Energy variation	=	350Nm
The power	=	9Kw

The flywheel is to be designed to satisfy the above conditions.

Coefficient of velocity variation

$$\bar{R} = \frac{V_2 - V_1}{V_{rtd}} = \frac{5,050 - 4,950}{5,000} = 0.02$$

Taking $B = 1.15$, $\bar{R} = 0.02$ and substituting in equation (8) gives

$$350 = 1.15 \times M \times 0.02 \times (91.5)^2$$

$$M = 1.82 \text{ kg}$$

The mass of the flywheel = 1.82kg

The drawing of the flywheel is among the set of drawings at the inside of the cover page.

3.2 CLUTCH DESIGN

If direct measurement with torque lever is possible then:

$$\text{Clutch torque } T_c = P \times L_1 \times SF \text{ ----- (9)}$$

Where direct measurement is not possible, approximate torque rating may be determined analytically, based on the masses of the flywheel, crankshaft, connecting rod, piston, attached masses, and the pressure of the gas in the cylinder.

For a rotating body, torque is given by

$$T = I \alpha \text{ ----- (10)}$$

$$\text{But } \alpha = \frac{\omega}{t} = \frac{2\pi N}{60t} \text{ ----- (11)}$$

$$\text{The average rim radius } R_{av} = \frac{R_{ex} + R_{in}}{2}$$

$$\text{But } R_{ext} = 0.06\text{m and } R_{in} = 0.05\text{m}$$

Recall that K was assumed nearly equal to the average rim radius,

$$\text{Therefore, } K = 0.055\text{m}$$

Moment of inertia of the flywheel

$$I_f = MK^2$$

Substituting the values for $M = 1.82 \text{ kg}$ and $K = 0.055\text{m}$, gives:-

$$I_f = 5.45 \times 10^{-3} \text{ kg/m}^2$$

From the design of the crankshaft of the engine:-

$$\text{Mass of each web} = 0.77\text{kg}$$

$$\text{Mass of the four webs} = 3.08\text{kg}$$

∴ Moment of inertia of the webs

$$I_w = \frac{1}{2} Mr_w^2$$

Substituting the values for $M = 3.08 \text{ kg}$ and $r = 0.044\text{m}$, gives $I_w = 2.97 \times 10^{-3} \text{ kgm}^2$

Mass of the three journals of the crankshaft = 0.924kg.

Assuming it is of uniform thickness, the moment of inertia

$$I_j = \left(\frac{L_j^2}{12} + \frac{r_j^2}{4} \right)$$

Substituting the values for $M = 0.924 \text{ kg}$,
 $L_j = 0.22 \text{ m}$, $r_j = 0.012 \text{ m}$ gives $I_j = 3.76 \times 10^{-3} \text{ kgm}^2$

Mass of the connecting rods = 0.55 kg

Mass of the crankpins = 0.036 kg

For the purpose of analysis it is assumed that some of the mass of the connecting rod is concentrated at the crank and the rest at the gudgeon pins.⁽⁸⁾ Let approximately half of its mass to be concentrated at the crank.

∴ Total mass of the body being rotated at the crank = 0.311 kg .

Moment of inertia due to the crank

$$\text{pins } I_{cr} = M \left(\frac{L_{cr}^2}{12} + \frac{r_{cr}^2}{4} \right)$$

Substituting the values for $M = 0.311 \text{ kg}$,
 $L_{cr} = 0.008 \text{ m}$, $r_{cr} = 0.01 \text{ m}$ gives $I_{cr} = 0.009 \times 10^{-3} \text{ kgm}^2$.

$$I = I_f + I_w + I_j + I_{cr} = 12.19 \times 10^{-3} \text{ kgm}^2.$$

The clutch is required to accelerate the bodies to the required speed within a very short time, say one second. Hence let t in equation (11) be = 1 sec .

The speed of the engine = 5,000 rpm. Substituting the values of N , I and t in equation (11) gives torque,

$$T = 6.38 \text{ Nm.}$$

This is the torque necessary to overcome the inertia of the rotating mass and also to bring them to the required speed in a second.

Assuming the piston is at the bottom dead centre (b.d.c.). The pressure in the cylinder, at the beginning of a stroke is 1 atmosphere; at the end of the stroke, just before burning of the fuel commences, the pressure rises up to 7 atmospheres (the designed compression ratio is 7:1) and this is when the force exerted on the piston crown by the compressed fuel mixture is highest. Torque transmitted by the starter clutch must be able to overcome this load. Let the load be P_1 .

If the load exerted on the piston crown when the fuel starts burning is P_2 , then $P_2 > P_1$ for the crankshaft to turn at a faster speed for the clutch to freewheel. The power developed due to P_2 is equal to 9kw at 5,000 rpm.

$$\begin{aligned}
 \therefore \text{Torque Produced} &= \frac{\text{Power}}{\omega} \\
 &= \frac{9,000}{5,000} \\
 &= 1.8 \text{kgm} = 18 \text{Nm}.
 \end{aligned}$$

Hence maximum torque rating of the clutch

$$\text{is} = 6.38 + 18 = 24.38 \text{ Nm}$$

$$\text{From fig.3.0, } F_R = F_d / \sin \frac{\theta}{2}$$

$$\text{But } F_d = \frac{T}{n_r r}$$

$$F_R = T / n_r r \sin(\frac{\theta}{2}) \text{ ----- (12)}$$

Substituting the values for $T = 24.38 \text{ Nm}$, $n_r = 3$,
 $r = 21.5 \times 10^{-3} \text{ m}$ and $\theta = 10^\circ$ in equation (12) gives
 $F_R = 4233.7 \text{ N}$.

Stress along line of contact is given⁽⁹⁾ as

$$S = 0.591 / F_R \bar{E} \left(\frac{dr}{dr} + \frac{dex}{dex} \right) \text{ ----- (13)}$$

But for mild steel $\bar{E} = 0.21 \times 10^{12} \text{ Nm}^{-2}$ and substituting the values for $\bar{E} = 0.21 \times 10^{12} \text{ Nm}^{-2}$, $d_r = 0.015 \text{ m}$, $d_{ex} = 0.043 \text{ m}$ in equation (13), gives $S = 167 \times 10^6 \text{ Nm}^{-2}$.

The yield stress of mild steel is approximately equal to $316 \times 10^6 \text{ Nm}^{-2}$. A factor of safety of 2 has been introduced. Therefore no local yielding will take place in the clutch members.

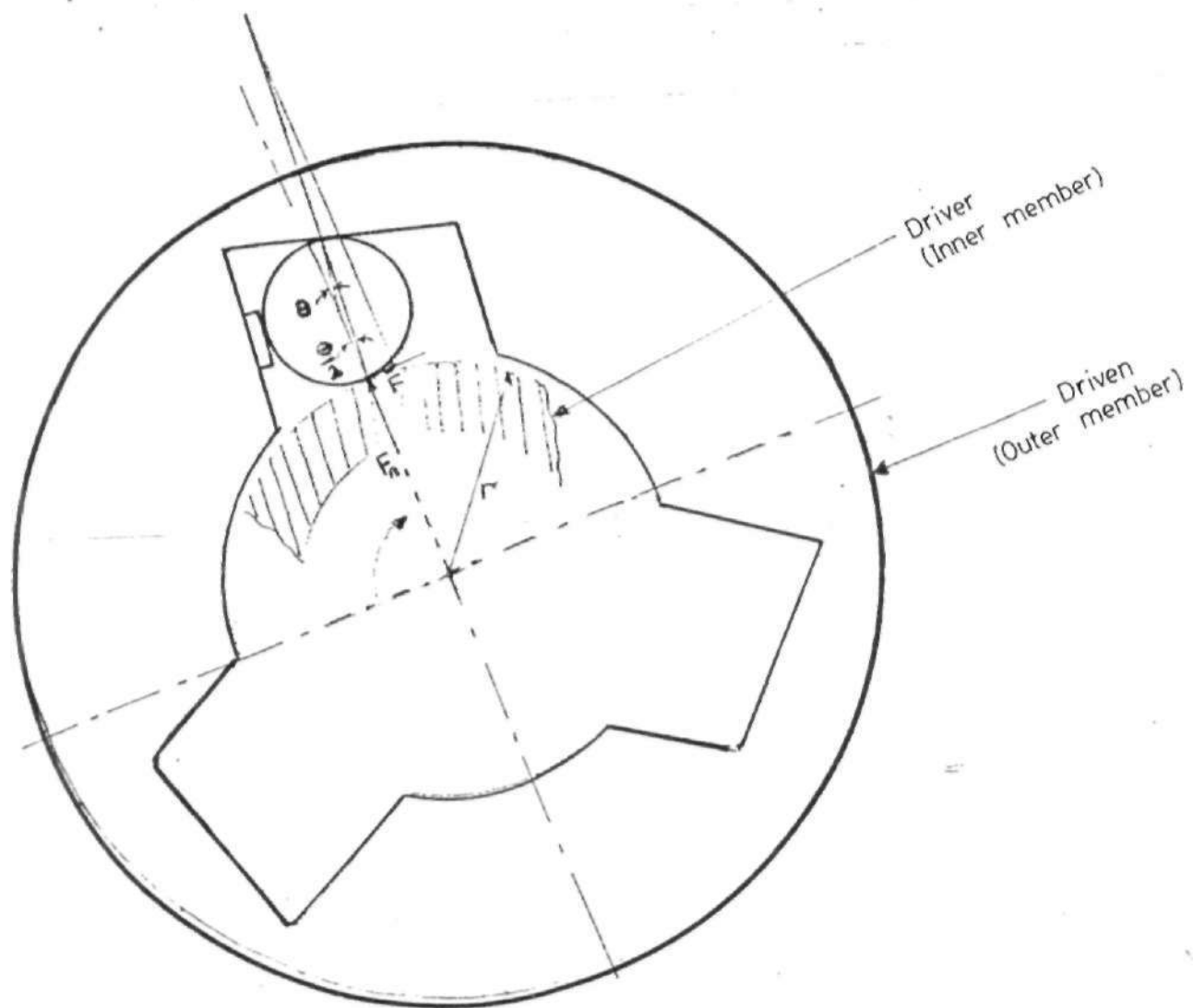


Fig.3.0: The driving force between the roller and the racing members.

3.3 SELECTION OF THE ROLLER BEARINGS

Cylindrical roller bearings make line contact with the raceways whereas ball bearings make point contact. This enables them to carry more radial load than ball bearings.

Cylindrical roller bearings have low-friction, excellent characteristic under starting and shock conditions, and require small lubrication. In addition, they are less expensive⁽¹⁰⁾.

In order to select the correct bearing for a given service condition, it is also necessary to know its dynamic capacity.

The load carried by a roller bearing is given by⁽¹⁰⁾

$$F_R = H \left(\frac{10^6}{60N_h} \right)^{3/10} \text{-----} (14)$$

Using the relationship $\frac{N_{ge}}{N_{pi}} = \frac{\bar{t}_{pi}}{\bar{t}_{ge}}$ and

substituting the values for $\bar{t}_{pi} = 9$, $\bar{t}_{ge} = 59$ and $N_{pi} = 8,800$ rev/min gives $N_{ge} = 1342$ rpm. The load carried by a roller bearing F_R is equal to the force at contact line of rollers F_R . From 3.2 the calculated value of $F_R = 4233.7N$. Substituting the values for $F_R = 4233.7N$, $N_r = 1342$ rpm and $h = 8000$ hrs (10) in equation (14) gives $H = 29473$.

The dynamic capacity of each bearing as well as design features are given in maker's catalogs, and based on the value of the dynamic capacity the correct bearing is selected.

3.4 SPRING SELECTION

The mass of the push rod and the roller is 0.02kg. Since the spring is just to maintain contact between the roller and racing members, the maximum load it will carry is $0.02 \times 9.81 = 0.2\text{N}$.

The force exerted on the spring is given by $F = kX$ ----- (15)

From fig.3.0 it is assumed that the maximum distance the rollers will travel is 1mm before engagement occurs.

Therefore,

$$\bar{k} = \frac{0.2}{1 \times 10^{-2}} = 0.2 \times 10^3 \text{N/m}$$

$$\text{But } \bar{k} = \frac{G d_{wl}^4}{8 n_{co} D_{co}^3} \text{ ----- (16)}$$

Let the solid length be equal to the depth of the push rod hole, that is 9mm. For a plain end spring, the solid length is given by:-

$$SL = d_{wl} (n_{co} + 1) \text{ ----- (17)}$$

For $SL = 9\text{mm}$ and $d_{wi} = 0.425\text{mm}$ (selected standard value⁽¹¹⁾) gives:-

$$n_{co} = 20$$

Substituting the values for $n_{co} = 20$,
 $G = 79323.6 \times 10^6 \text{ Nm}^{-2}$, $d_{wi} = 0.425 \times 10^{-3}$ and
 $\bar{k} = 200 \text{ N/m}$ gives $D_{co} = 4.3\text{mm}$.

The coil diameter is 4.3mm . Therefore a hole diameter of 4.5mm is chosen.

The free length of the wire is given as⁽¹¹⁾

$$FL = (5n_{co} - 2SL)/3 \text{ ----- (18)}$$

For $n_{co} = 20$ and $SL = 9\text{mm}$;

Therefore $FL = 27.33\text{mm}$.

Hence a working frelength of 28mm is chosen.

The selected spring is therefore defined by the following parameters:

$F = 0.2\text{N}$, $d_{wi} = 0.425\text{mm}$, $d_{co} = 4.3\text{mm}$ and
 $FL = 28\text{mm}$.

3.5 DESIGN OF THE RING GEAR ON THE DRIVING MEMBER OF THE CLUTCH

It is necessary for a gear designer to know all the dimensions of the gear, which are necessary for its production, before the loads and stresses

can be determined. "This means that a preliminary estimate must be made of the gear size, and this estimate must be checked out and then altered as the calculations indicate".⁽¹²⁾

Diametral pitch of half is one of the commonly used and preferred diametral pitches.⁽¹²⁾ This value will be used for the following calculations.
 $Pd = \frac{1}{2}$.

The pitch diameter is given by:-

$$PD = \frac{T_{ge}}{Pd}$$

For $T_{ge} = 59$ and $Pd = \frac{1}{2}$ gives $PD = 118\text{mm}$

The base circle diameter is given by

$$BCD = PD \cdot \cos\phi$$

For $\phi = 14\frac{1}{2}$ and $PD = 118\text{mm}$ gives $BCD = 114\text{mm}$.

The circular pitch is defined as:-

$$PC = \frac{\pi PD}{T_{ge}}$$

For $PD = 118\text{mm}$ and $T_{ge} = 59$ gives:-

$$PC = 6.3\text{mm}$$

The addendum A is defined as:-

$$A = \frac{1}{Pd}$$

For $Pd = 0.5$ gives:-

$$A = 2$$

The outside diameter $OD = PD + 2A$

For $PD = 118\text{mm}$ and $A = 2$, gives:-

$$OD = 122\text{mm}.$$

Let the root circle RC = 112mm

The above values give the features of the gear. The next thing is to determine the strength of the tooth, if the load is known, to see whether the stress is less or more than the allowable stress.

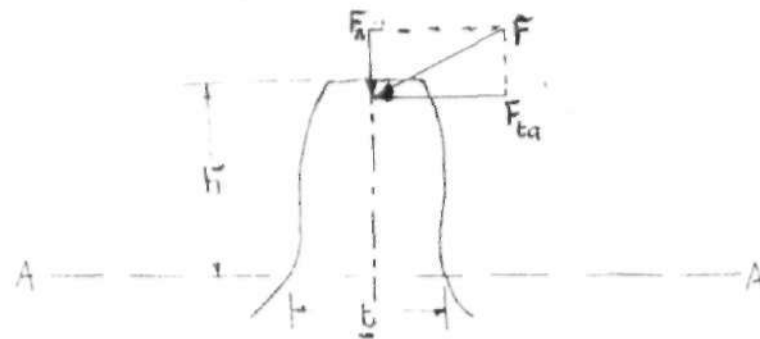


Fig. 3.2: Forces on tooth

Assuming that the load is carried by one tooth, that it behaves like a cantilever, and applying the simple bending equation to fig.3.2 gives stress,

$$\sigma = \bar{M}Z/\bar{I} \text{ ----- (19)}$$

where $\bar{M} = F_{td} \cdot h$

and

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

Assuming section A - A to be rectangular, then:-

$$\bar{I} = \frac{bt^3}{12}$$

Substituting the expressions for $\bar{M}$, Z and $\bar{I}$ in equation (19) gives:-

$$\sigma = 6 F_{td} h / bt^2 \quad \text{or}$$

$$F_{ta} = \sigma \cdot b \cdot t^2 / 6h = \frac{\sigma \cdot b \cdot t^2 \cdot PC}{6h \cdot PC}$$

$$F_{ta} = \sigma \cdot b \cdot PC \cdot y \text{ ----- (20)}$$

$$\text{where } y = t^2 / 6h \cdot PC$$

Equation (20) is known as Lewis formula.

$$\text{The allowable stress } \sigma = F_{ta} / b \cdot PC \cdot y \text{ --- (21)}$$

The tangential component of the force acting on the tooth $F_{ta} = T / (PD/2)$

Substituting the values for $T = 24.38 \text{ Nm}$
and $PD = 0.118 \text{ m}$

$$F_{ta} = 413.2 \text{ N}$$

Substituting the values for $b = 0.006 \text{ m}$,
 $PC = 0.0063 \text{ m}$, $y = 0.113$ (given in⁽¹²⁾) and
 $F_{ta} = 413.2 \text{ N}$ gives:-

$$\sigma = 96.74 \times 10^6 \text{ N/m}^2$$

The allowable stress for machinery steel with no heat treatment is $1.41 \times 10^8 \text{ N/m}^2$. This value is more than the stress in which the designed gear will be subjected to. The factor of safety is 1.5, which is necessary since the Lewis formula is based on approximations.

3.6 BOLT DESIGN

Recall from section 3.2, that the maximum torque rating of the clutch was calculated to be 24.38Nm.

The stress due to the torque is given by:-

$$\bar{\sigma} = TD_{bo}/2n_{bo}.J \text{ ----- (22)}$$

But:-

$$J = \pi D_{bo}^4/32 \text{ ----- (23)}$$

Substituting equation (23) in equation (22) gives:-

$$D_{bo}^3 = 32T/2\pi.n_{bo}.\bar{\sigma} \text{ ----- (24)}$$

The tensile strength of mild steel is $840 \times 10^6 \text{ N/m}^2$. Using a factor of safety of 5, the allowable stress in the bolts is $168 \times 10^6 \text{ N/m}^2$. Assuming that the stress due to the tightening torque is 5% that due to the torque being transmitted. Therefore the maximum allowable stress = $176.4 \times 10^6 \text{ N/m}^2$.

Substituting the values for $T = 24.38 \text{ Nm}$, $n_{bo} = 3$, $\bar{\sigma} = 176.4 \times 10^6 \text{ N/m}^2$ into equation (24), gives:-

$$D_{bo} = 6.17 \text{ mm}$$

Therefore the tensile stress area is:-

$$\bar{a} = \pi D_{bo}^2/4$$

Substituting the value for $D_{bo} = 6.17$
gives:-

$$\bar{a} = 29.9\text{mm}^2.$$

The nearest tensile stress area for metric bolts, coarse series is 36.6mm^2 (13). The bolt with this stress area is M8. Hence M8 bolt is selected. Pitch of the bolt is = 1.25mm.

3.7 SELECTION OF THE STARTER MOTOR

The force along the bigger circumference of the driving member of the clutch (along the ring gear) is F_{ta} .

From 3.5 the calculated value of $F_{ta} = 413.2\text{N}$.

This force must be equal and opposite to that produced by the shaft of the starter motor.

For a starter motor of shaft radius = 0.0058m, the torque developed due to F_{ta} is given by:-

$$\begin{aligned} T &= 413.2 \times 0.0058 \\ &= 2.4\text{Nm} \end{aligned}$$

The selected motor has the following specifications given in the manual:

Voltage, = 4.5 volts, locked rotor

Current, = 200 Amperes, locked rotor

Speed difference = 3,800 rpm

Therefore power, $P_0 = i \times V_0$

= 900 watts.

$$\text{Torque } T = P_0/\omega \text{ ----- (25)}$$

Substituting the values for $P_0 = 900\text{watts.}$ and

$\omega = 3,800 \text{ rpm}$ in equation (25) gives:-

$$T = 2.73\text{Nm.}$$

This implies that the motor can drive the clutch satisfactorily.

CHAPTER 4CONSTRUCTION AND TESTING4.1 CONSTRUCTION4.1.1 Production of Drawing No.1

A scrap of $150 \times 100\text{mm}^2$ was obtained. One end was faced, then the 35mm deep cavity was made, leaving a 29mm x 30mm extension within it. The workpiece was held on the jaws of the lathe machine by the cavity. The second end was faced leaving it with 50mm length. It was then turned to 120mm diameter. A length of 5mm was turned further to 78mm diameter on the opposite side of the cavity. A taper of 18mm and 14.5mm diameters was made, this was followed by chamfering the larger end to 26mm diameter as shown on the drawing. Equally spaced three points were located at 32mm from the centre. 8mm holes were drilled at each of the marked points. M8 tap was then used to thread the holes.

4.1.2 Production of Drawing No.2

Drawing No.2 was produced from a cylindrical workpiece of dimensions 150mm x 30mm. The workpiece turned to 122mm diameter and faced to 26mm.

A hole of 29 mm was drilled at the centre. A length of 20 mm was then turned to 44 mm diameter and chamfered as shown in the drawing. A ring involute teeth of the dimensions shown on the drawing was then produced.

4.1.3 Production of Drawing no.3

A length of 25 mm was cut out of 90 mm bar. 78 mm x 5 mm bore was made at one end and 86 mm x 13 mm bore at the second end. The first end of the workpiece was then chamfered as shown on the drawing.

4.1.4 Production of Drawing no.4

The outer race of the clutch was made out of 90 mm bar. A length of 20 mm was cut out of the bar stock. It was then turned to 86 mm and faced on both sides to a thickness of 13.5 mm. The profile as shown on the drawing was marked out on the workpiece. A 44 mm bore was made at the centre, and a milling cutter was used to follow the profile (roughly). It was then filed with hand to the required dimensions. Equally spaced three points 32 mm away from the centre was marked and 8.5 mm holes were drilled at each of this points. A counter sinking of 16 mm diameter and 9 mm deep was made at each of these holes. The push rod hole positions were determined, milled to flatness and finally drilled with 6.5 mm drill.

4.1.5 Production of Drawing no.5 & 6

Plates of the dimensions shown on the drawing were cut out of mild steel sheet of 1.5mm thickness and the holes were drilled to the specified dimensions.

The bolts and push rod were produced in the Departmental workshop and the rollers were bought from outside.

4.2 ASSEMBLY

The main clutch is inserted into the clutch housing and covering plate 1 is welded on to the housing to prevent the main clutch from moving out. Covering plate 2 is placed on the other side of the clutch when the push rod, springs and rollers are placed in position. The whole of this assembly is bolted to the flywheel. The ring gear is then pushed into the hole of the clutch assembly.

The exploded view of the above assembly is shown in fig. 4.0.

4.3 TESTING

The test for any prototype should determine whether it serves the intended purpose or not. The designed and constructed starter clutch is a freewheeling clutch. A freewheeling clutch should

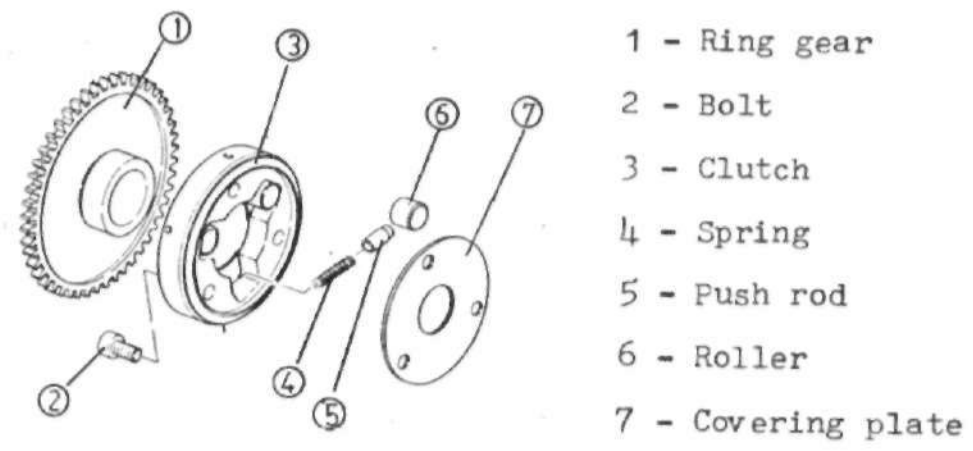


Fig. 4.0: Exploded view of the clutch

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transmit torque in only one direction at any speed and then freewheels in the opposite direction.

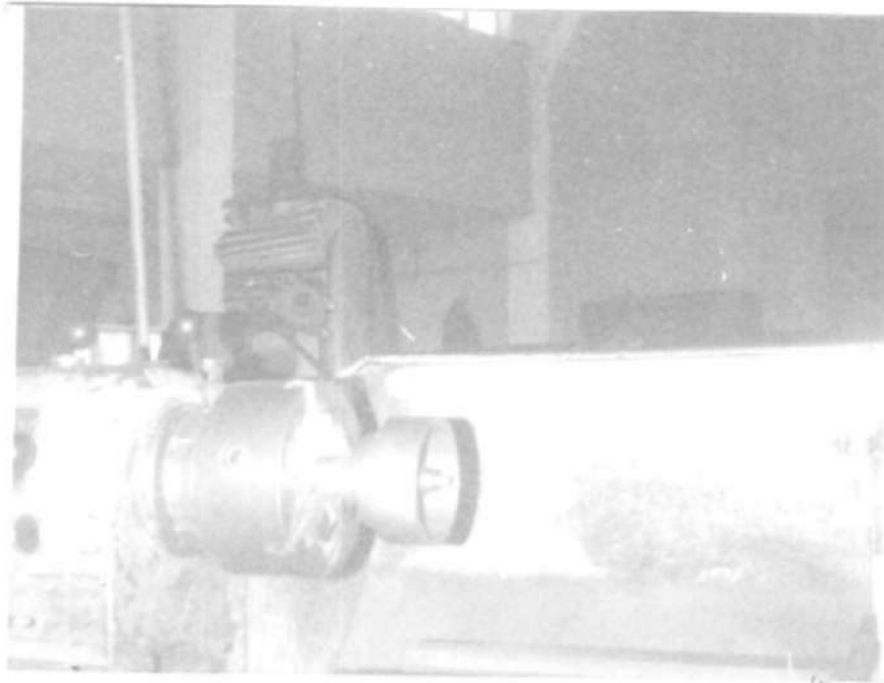
The set-up of the test rig is as shown in plate no.2.

The starter clutch and the flywheel were mounted on the lathe machine. When the machine rotates in a clockwise direction, torque is transmitted. And when it rotates in an anticlockwise direction, it freewheels, that is no torque is being transmitted.

This demonstration shows that the starter mechanism will freewheel while in operation.



Photograph of the parts of the starter mechanism



Photograph of the starter clutch mounted on the lathe machine for testing

CHAPTER 5DISCUSSIONS, CONCLUSIONS AND RECOMMENDATIONS5.1 DISCUSSIONS

The starter mechanism adopted in this design has some advantages over the inertia type. It is compact, it acts as a magnet, as a starter and also it stores energy.

The problem encountered in this project was in the production of drawing number 4. A depth of 9mm only is required for the hole containing the Grub screw. However it is not possible to drill the hole from inside, so it was drilled from outside, and then the Grub screw was inserted.

5.2 CONCLUSIONS

1. A starter mechanism for the 9kW two-stroke engine under development in the Department of mechanical Engineering, Ahmadu Bello University, has been designed, produced locally and tested.
2. Test shows that the starter clutch is a freewheeling clutch type. It transmits torque in one direction and freewheels in the other direction

5.3 RECOMMENDATIONS

1. The flywheel cavity should be lined with suitable magnetic material to help induce the electric current required for the spark plug operation.
2. The possibility of casting drawing number 4 for convenience and greater uniformity should be looked into.

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