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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

SAU1404 VEHICLE DESIGN CHARACTERISTICS

UNIT I INTRODUCTION

UNIT-I INTRODUCTION

Heat Engines :- Heat engines is a device which transforms the chemical energy of a fuel into thermal energy and uses this energy to produce mechanical work. Heat engines are divided into two broad classes.

a) External combustion engines
b) Internal combustion engines.

In an external combustion engine the products of combustion of air and fuel transfer heat to a second fluid which is the working fluid of the cycle, as in the case of steam engine or a stem turbine plant where the heat of combustion is employed to generate steam which is used in the piston engine or turbine .Sterling engine is also an external combustion engine.

In an internal combustion engine the product of the combustion are directly the motive fluid. Petrol, gas & diesel engines, Wankel engine, and open cycle gas turbine are example of internal combustion engine. Jet engine and rockets are also internal combustion engine.

The main advantages of internal combustion engines over external combustion engines are greater mechanical simplicity, lower ratio of weight and bulk to output due to absence of auxiliary apparatus like boiler and condenser and hence lower first cost, higher overall efficiency, and lesser requirement of water for dissipation of energy through cooling system.

Historical development.

Huygens Gunpowder Engine :- The earliest internal combustion can be credited to famous Dutch physicist Christian Huygens (1629-1695) in the year 1680. Huygens engine employing gunpowder.

Four Stroke Cycle :- All the engines developed uptil 1860 provided combustion of the charge at about atmospheric pressure. In 1862, Beau de Rochas, a Frenchmen, wrote a paper describing the fundamental principals for efficient operation of piston combustion engine, which were demonstrated in a practical engine by Otto, a German engineer. This laid the foundation of four stroke cycle engine which is used till today in all four stroke spark-ignition engines. This method of operation was explained in the four operations as follows.

1st Stroke :- Induction of charge during the outward stroke of piston.
2nd Stroke :- Compression of the charge during inward stroke of the piston.
3rd Stroke :- Ignition of the air fuel mixture during inward dead centre, followed by
expansion during the next outward stroke of the piston.
4th Stroke :- Exhaust during the next inward stroke of the piston.

The Diesel Engine (1892) :-

The term diesel engine is used throughout the world to denote compression-ignition oil engines, two stroke or four stroke, with air less fuel injection. This very important concept of compression-ignition can be credited to Rudolf Diesel (1858-1913), a German engineer born in Paris. In 1892 he proposed compression of air alone until a sufficient high temperature was attained to ignite the fuel which was to be injected at the end of the compression stroke. In his first experiments he tries to injects coal dust into a cylinder containing air that had already been highly compressed.

MODERN DEVELOPMENT

Wankel Engine (1957)Dr Felix Wankel was born on August 13th 1902 in Swabia, Germany. He invented basic design that led to the eventual development of the first successful rotary engine. Wankels first rotary engine was tested at NSU, Germany in 1957.

The standard terminology used in I.C Engine.

1. Cylinder Bore :- (B) The nominal inner diameter of the working cylinder

2. Piston Area :- (A) The area of the circle of diameter equal to the cylinder bore.

3. Stroke:- (L) A nominal distance through which a working piston moves between two successive reversals of its direction of motion.

4. Dead centre :- The position of the working piston and that moving parts which are mechanically connected to it at the momentum when the direction of piston motion is reversed.

5. Bottom Dead centre :- (BDC) Dead centre when the piston is nearest to the crankshaft.

6. Top Dead centre :- (TDC) Dead centre when the piston is farthest from the crankshaft.

7. Displacement volume :- (Vs) :- Vs=AxL.

   The nominal volume generated by the working piston when traveling from one dead centre to the next one.

8. Compression Ratio :- (CR or r) The numerical volume of the cylinder volume divided by the numerical value of the combustion space volume.

IC ENGINES CLASSIFICATIONS

1. Otto Cycle Engines or Spark Ignition Engines
2. Diesel Cycle Engines or Compression Ignition Engines.
3. Four Stroke Engines (One power stroke in two revolution of crankshaft)
4. Two Stroke Engines. (One power stroke in one revolution of crankshaft)

COMPARISION OF S.I. & C.I. ENGINES.

Description	**S.I. Engines**	**C.I. Engines**
1. Basic Cycle	Based on Otto Cycle	Based on Diesel Cycle
2. Fuel	Petrol, gasoline, High self ignition temp desirable.	Diesel oil, low ignition temp. desirable.
3.Introduction of fuel	Carburetor is used to mix fuel & air in proper proportion in suction stroke.	Fuel pump is used to inject fuel through injector at the end of compression stroke.
4. Ignition	Ignites with the help of spark plug.	Ignition due to high temp. caused by high compression of air & fuel.
5. Compression ration	6 to 10.5	14 to 22
6. Speed	High RPM	Lower RPM.
7. Weight	Lighter	Heavier
8. Starting	Low cranking effort	High cranking effort
9. Noise	Less	More

APPLICATION OF S.I. & C.I. ENGINES.

S.I. Engines :-

Small 2 stroke petrol engines is used where low cost of prime mover is main consideration. Ex- moped.

4 Stroke S.I. engines are used in Automobiles & Mobile gen. Set.

C.I. Engines :-

Two stroke C.I. engine is used where very high power diesel engines for ship propulsion.

Four stroke C.I. engine is used for all the HEMM’s

COMBUSTION PROCESS IN C.I. ENGINES.

The ideal sequence of operation for the four stroke C.I. engine is as follows.

1. Suction Stroke :- Only air in injected during the suction stroke. During this stroke intake valve is opened & exhaust valve is closed.

2. Compression Stroke :- Both valve remains closed during compression stroke.

3. Power or Expansion Stroke :- Fuel is injected at the beginning of expansion stroke. The rate of injection is such that the combustion maintain the pressure constant. After the injection of fuel is over the product of the combustion expands. Both valve remains closed during the expansion stroke.

4. Exhaust Stroke ;- Exhaust valve is opened & the intake valve remains closed in the exhaust stroke.

The typical valve timing for 4 stroke C.I. engine is as follows.

1. IVO upto 30 deg before TDC
2. IVC upto 50 deg. After BDC
3. EVO about 45 deg. Before BDC.
4. EVO about 30 deg. After TDC.
5. Injection about 15 deg. Before TDC.

CYLINDER ARRANGEMENT OF I.C. ENGINES.

There are several type cylinder arrangement in I.C. engines, commonly used in HEMM's are of two types.

1. In-line Engines :- Inline engine is an engine with one cylinder bank i.e. all cylinders are arranged linearly and transmit power to a single crank shaft.. This type is very popular with automobiles having 4 to 6 cylinders.
   Eg- NTA 855, KT1150.

2. V- Engines ;- An engine with 2 cylinder banks inclined at an angle to each other and with one crank shaft. Most of the bigger automobiles used this type of arrangement.
   Eg- CAT 3412 HEUI, CAT3408.

FIRING ORDER

Every engine cylinder must fire once in every cycle. This requires for a 4 stroke 4 cylinder engine, the ignition system must fire spark plug for every 180 deg. Of crank rotation. For a 6 cylinder engine the time available is still less.

The order in which various cylinders of a multi cylinder engines fire is called firing order. There are three factors which must be consider before deciding the firing order of an engine. These are a) Engine vibration b) Engine cooling c) Development of back pressure.

Following are the firing order of muliti cylinder engines

Sl.no.	Engines Cylindres	Firing Order
1.	3 Cylinder Engine	1-3-2
2.	4 Cylinder Engine	1-3-4-2
3.	6 Cylinder Engine	1-5-3-6-2-4
4.	8 Cylinder V shape Engine	1-8-4-3-6-5-7-2
5.	12 Cylinder V shape Engine	1-4-9-8-5-2-11-10-3-6-7-12

FUEL INJECTION

There are two types of injection system. They are

1. Air injection :- Fuel is metered & pump to the fuekl valve by a cam shaft driven fuel pump. The fuel valve is opened by mechanical linkage operated by a crank shaft which controls the timing of injection.

2. Solid Injection :- Injection of fuel directly into the combustion chamber without primary atomization is termed as solid injection. Every solid injection system must have a pressuring unit (Pump) & an atomizing unit(Injector). The different type of solid injection system are a) Individual pump & injector b) Common rail system C) distributor system.

FUEL CONSUMPTION

For a diesel fuel smooth spontaneous ignition at relatively low temp. is essential.

Cetane number :- The cetane rating of a diesel fuel is a measure of its ability to auto ignite quickly when it is injected into a compressed and heated air in the engine.

ENGINE PERFORMANCE :-

Engine performance is indicated by the term efficiency.
Various type of efficiencies are

1. Indicated thermal efficiency :- It is a ratio of energy in the indicated horse power to the fuel energy.

2. Mechanical efficiency ;- It is ration of brake horse power to the indicated horse power.

3. Brake thermal efficiency :- It is the ration of energy in the brake horse power to the fuel energy.

4. <u>Volumetric efficiency</u> :- Volumetric efficiency is defined as the ration of air actually induced at ambient conditions to the swept volume of the engine.

5. <u>Specific fuel consumption</u> :- It is a ratio of fuel consumption per hour to the horse power.

6. <u>Indicated Horse power</u> :- is the power produced inside the cylinder.

7. <u>Brake Horse Power</u> :- is the power available at the crankshaft.

<u>SUPER CHARGING</u> :-

The method of increasing the inlet air density, called super charging. This is done by supplying air at the pressure higher than the pressure at which the engine naturally aspirates air from the atmosphere by using the pressure boosting device called a super charger.

Objective of supercharging - To increase the power output for a given weight and bulk of the engine, to compensate for the loss of power due to altitude & to obtain more power from an existing engine.

<u>TURBOCHARGING</u>:-

Of the total heat input to an engine about 27 to 38 percent goes into exhaust. Whole of the energy cannot be utilized, however a part of it can be used to run a gas turbine which in turn will supply more air to the engine by driving a compressor. Such utilization of the exhaust energy boosts engine power and results in better thermal efficiency and fuel consumption.

Turbocharger are centrifugal compressors driven by the exhaust gas turbines. They are now a days extensively used for supercharging almost all types of 2 stroke & 4 strokes engines. By utilizing the exhaust energy of the engine it recovers a substantial parts of energy which could otherwise goes waste, thus turbocharger will not draw upon the engine power.

COOLING SYSTEM

All the heat rejected from the engine ultimately goes to air. Nevertheless, two basic systems are used to cool the engine. They are.

1. Air cooling.
2. Water cooling or direct air cooling using water as a transfer medium.

Application of air cooling :- Air cooling are usually used for small engines & for engines whose applications gives extreme importance to weight such as aircraft. For air cooling the cylinder head heat transfer area is increased by finning and air is passed over these fins.

Application of Water cooling :- In case of water cooled engines the cylinder and the cylinder head are enclosed in a water gacket. These water jacket is connected to a radiator(Heat exchanger). Water is caused to flow in the jacket where it cools the engine, then it gives up this heat to air in the radiator and is again circulated in the water jacket.

LUBRICATION SYSTEM OF C.I. ENGINES.

Function of lubrication system :- The following are the important function of lubricating system

1. Lubrication.
2. Cooling
3. Cleaning
4. Sealing
5. Reducing noise

Properties of Lubricating oil :-

1. Viscosity
2. Flash point
3. Carbon residue
4. Oiliness
5. Cleanliness
6. Colour
7. Acidity & neutralization number.

Types of Lubricating system

Various lubricating systems used for internal combustion engines may be classified as

1. Mist Lubricatinf system
2. Wet sump Lubricating system

Mist Lubricating System :- This system is used for 2 stroke cycle engines. Most of the engine are crank charged. i.e they employ crankcase compression and thus are not suotable for crank case lubricatiuon.

Wet sump Lubricating System :- In wet sump lubricating ststem the bottom part of the crankcase, called sump, contains the lubricating oil from which the oil is supplied to various parts. There are three types of wet sump lubricating system.

a. Splash system
b. Modified splash system
c. Full pressure system.

Model - Application Chart of Engines working at WCL

SL NO	MODEL/ APPLICATION	Application	HP
1	NTA855- LW35	LW35 Dumper	380
2	NTA855- R35	R35 Dumper	380
3	KT1150- D155	D155 Dozer	320
4	KTA1150- D355	D355 Dozer	450
5	KT1150-PC650	BE650 Excavator	439
6	KT1150-7271	7271 PayLoader	430
7	KTTA19C- 210M	210M Dumper	686
8	NTA855 BIG CAM	BH35-2 Dumper	400
9	NT855Big Cam-CK300	CK300 Excavator	320
10	NTA855Big Cam-EX600	EX300 Excavator	420
11	NT855- ReCp650	ReCp650 Drill	280
12	NT855-ROTACOAL	Rotacoal Drill	290
13	NT855-CK300	CK300 Excavator	320
14	NT855FFC-PC300	BE300 Excavator	235
15	NT855FFC-DEMAG	Demag Excavator	307
16	NT855FFC-LMP DRILL	LMP Drill	335
17	N743- MG605	BG605 Motorgrader	165
18	N743TC- CK170	CK170 Excavator	240
19	NT743- EX300	EX300 Excavator	246
20	NT743- CK180	CK180 Excavator	286
21	NT743-Water Pump	Dewatering Pump	240
22	6CTA8.3- EX300	EX300 Excavator	230
23	3412DITA-773B	773B-I Dumper	653
24	3412 HEUI	773B-II/ 773D Dumper	650
25	3408HEUI- 834B	834B Wheel Dozer	410
26	3406DITA-1035	HM1035 Dumper	380
27	3456EUI	834G Wheel Dozer	410
28	NT495-TYRE HAND	Tyre Hand	145
29	SA6D110-WA400	WA400 Pay Loader	240
30	SA6D140-BE650	BE650 Excavator	416
31	S6D125-BE300	BE300 Excavator	264
32	S6D140-D155X	D155X Dozer	320
33	S6D140-BG825	BG825 Motor Grader	280
34	S6D170-D155	D155 Dozer	320
35	SA6D170-210M	210 Dumper	648

SI Engine Variables and Emissions

Any engine variable that affects oxygen availability during combustion would influence CO emissions. The factors which influence flame quenching, quench layer thickness and post flame oxidation control engine out HC emissions. The burned gas temperature-time history and oxygen concentration control NO formation and emission. Hence the engine variables that influence burned gas temperature and oxygen concentration would affect the NO emissions. Principal design and operating variables affecting engine emissions are:

Design Variables:

- Compression Ratio
- Combustion chamber surface to volume ratio
- Ignition timing
- Valve timings and valve overlap
- Air motion, swirl tumble etc
- Charge stratification

Operating Variables:

- Air-fuel Ratio
- Charge dilution and exhaust gas recirculation (EGR)
- Speed
- Load
- Coolant temperature
- Transient engine operation: acceleration, deceleration etc.

The effect of some variables discussed below is typical in nature and variations in the trends with specific engine design change are observed.

Compression Ratio

The effect of compression ratio on engine emissions is shown on Fig. 3.1. The typical effect observed when the engine CR was reduced from 10:1 (CR used on high performance engines during pre emission control period) to 8.5 and 7.0:1 are given on this figure.

Use of high CR results in

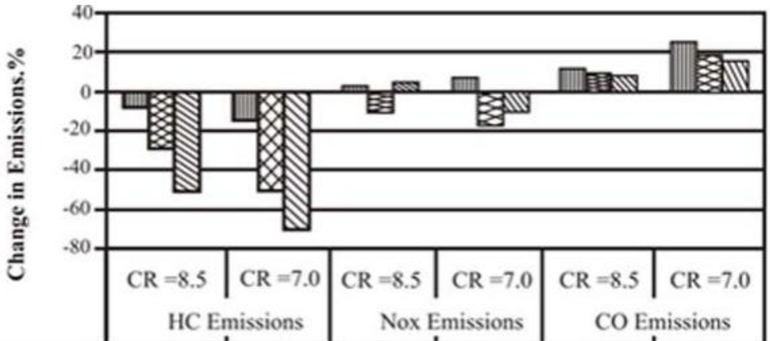
(i) Higher burned gas temperature

(ii) Lower residual gas content

These lead to higher NO emissions on volume basis. However, as engine efficiency increases with increase in compression ratio, brake specific NO emissions decrease. High CR combustion chambers result in

(i) High surface to volume ratio and
(ii) (ii) A proportionately higher crevice volume.
(iii) (iii) Lower exhaust gas temperatures

Thus the volume of flame quenching regions increases resulting in higher HC emissions. The problem is further enhanced as due to lower exhaust gas temperatures oxidation of the unburned HC is reduced during exhaust process. These factors result in an increase in HC emissions with increase in engine CR. At lower CR% fuel efficiency is also reduced thus increasing specific CO emissions.



	HC Emissions CR =8.5	HC Emissions CR =7.0	Nox Emissions CR =8.5	Nox Emissions CR =7.0	CO Emissions CR =8.5	CO Emissions CR =7.0
1500 rpm	-7	-14	2	-7	12	25
2000 rpm	-28	-50	-10	-17	9	18
2500 rpm	-50	-70	3	-10	8	16

Figure 3.1	Effect of reduction in compression ratio from 10:1 to 8.5:1 and 7.0:1 on SI engine emissions

Ignition Timing

The effect of ignition timing on NO and HC emissions is shown on Fig 3.2 When ignition occurs earlier in the cycle more heat is released before and around the top dead center. Thus, with advanced ignition timings higher peak cylinder pressures and temperatures result. As has been discussed lecture 5 with increase in combution temperatures NO formation increases. Hence , higher NO emissions are obtained as the ignition timing is advanced. As the ignition timing is retarded more burning takes place during expansion stroke resulting in lower peak combustion pressures and a lower of mass of charge is pushed into crevice volume. Also, at the retarded ignition timings exhaust gas temperature increases as the engine thermal efficiency is reduced. In the hotter exhaust gas with the retarded ignition timing higher oxidation rates of the HC and CO in the exhaust system are obtained.. Due to these reasons, lower HC emissions are obtained with retarded ignition timings. The disadvantage of the retarded ignition timing is lower engine efficiency, lower power and a poorer fuel economy. When the emission control legislation was introduced for the first time around 1970 in the USA and Europe, ignition timing versus speed and manifold vacuum curves were among the first engine parameters that were modified for control of NOx emissions due to ease of their adjustment.

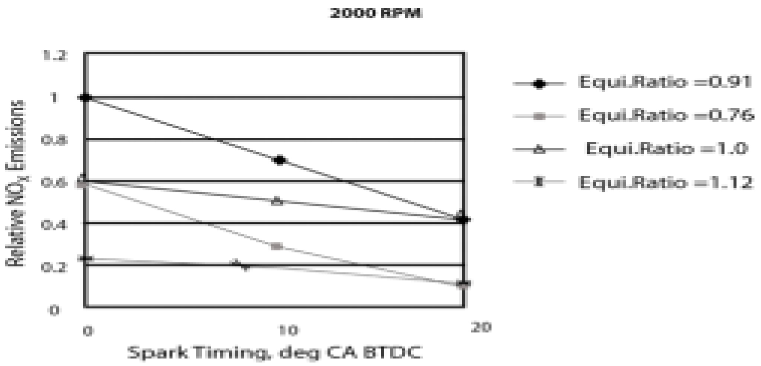


Fig 3.2(a) Effect of spark timing on NOx emissions.

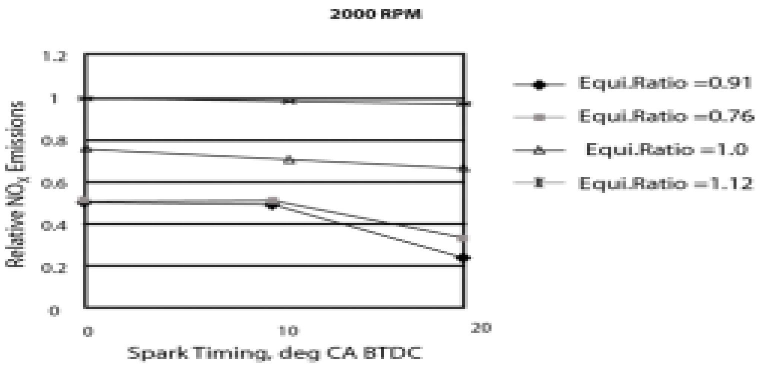


Fig Effect of ignition timing on HC

Air –Fuel Ratio

The effect of air-fuel ratio on engine emissions has already been discussed in Lectures 3 , 5 and 7. Carbon monoxide results due to deficiency of oxygen during combustion and is reduced as the mixture is leaned. CO emissions are reduced to very low values as the mixture is leaned to = 0.90 – 0.95 i.e. air-fuel ratio is increased above the stoichiometric value by 5 to 10%. Further leaning of mixture shows very little additional reduction in the CO emissions. With increase in air fuel ratio, the initial concentration of hydrocarbons in the mixture is reduced and more oxygen is available for oxidation. Hydrocarbon emissions therefore, decrease with increase in air-fuel ratio until mixture becomes too lean when partial or complete engine misfire results which cause a sharp increase in HC emissions For < 0.8 engine may misfire more frequently thereby increasing HC emissions sharply. The highest burned gas temperatures are obtained for mixtures that are slightly (5 to 10 percent) richer than stoichiometric. On the other hand, there is little excess oxygen available under rich mixture conditions. As the mixture becomes lean, concentration of free oxygen increases but combustion temperature start decreasing. The interaction between these two parameters results in peak NO being obtained at about = 0.9 –0.95.

Residual Gas and EGR

Burned residual gases left from the previous cycle or part of the exhaust gas recirculated back to engine act as charge diluents. The charge dilution by recirculation of part of the exhaust gas back to the engine is called exhaust gas recirculation (EGR). The combustion temperatures decrease due to charge dilution caused by the residual burned gases or EGR , the decrease in combution temperatures is nearly proportional to the heat capacity of the diluents as discusses earlier. The lower combustion temperatures resulting from the residual gas dilution/EGR reduces NO formation and emissions as shown on Fig 3.3.

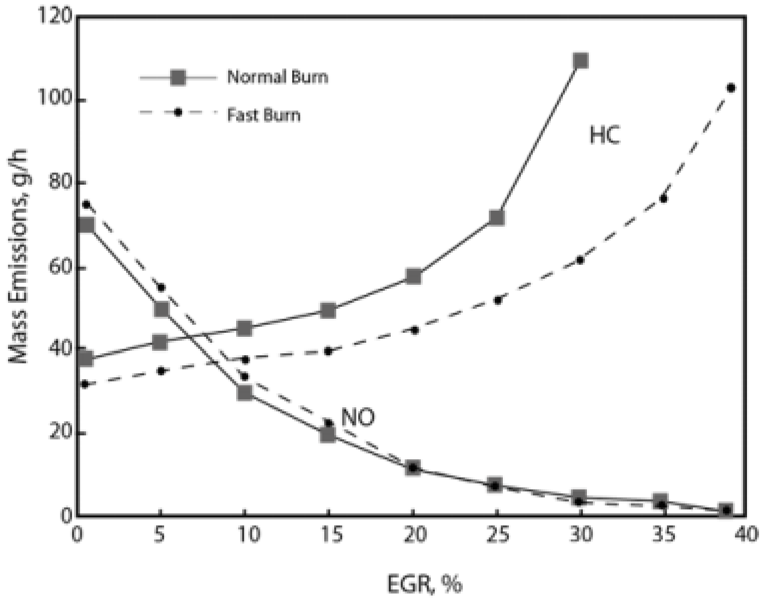


Figure 3.3	Effect of EGR on NO and HC emissions in normal burn and fast burn SI engines.

As the EGR is increased the combustion rates become more and more slow, and combustion becomes unstable. With increase in EGR cycle-to-cycle combustion variations increase and, more and more engine cycles having only partial combustion are observed . The frequency of partial burn cycles increases and these turn into misfired cycles at EGR rates of about more than 20%. . In the partial burn and misfire cycles, combustion remains incomplete and results in high HC emissions. Moreover with EGR the burned gas temperatures are reduced and post-flame oxidation also reduces. Increase in HC becomes sharp as EGR increases beyond about 20 percent for a normal combustion engine. With EGR rates of 20 percent or higher Fast burn engines due to higher flame speeds have higher burned gas temperatures and tolerate higher EGR rates before the combustion becomes very unstable and loss in fuel efficiency becomes unacceptably high. Fast burn rates are usually obtained by use of high air swirl and increasing turbulence in the charge through use of suitable designs of intake valve port and the combustion chamber. The amount of charge dilution or EGR is usually limited to below 15% due to its adverse impact on engine performance causing power loss, high specific fuel consumption and high unburned fuel emissions.

Engine Speed :

Volumetric efficiency of the engine changes with speed, it being highest in the mid-speed range. At high engine speeds the volumetric efficiency generally decreases resulting in high residual gas dilution. Although heat transfer rates increase with increase in engine speed as a result of higher turbulence, but total amount of heat transfer is lower due to shorter cycle time. This gives higher gas temperatures at higher speeds. However, at high speeds a shorter time is available for NO formation kinetics. The net result is a moderate effect of speed on NO although this is specific to the engine design and operating conditions. Increase in exhaust gas temperatures at higher speeds enhances post flame oxidation of unburned hydrocarbons. A reduction of 20 to 50 percent in HC emissions has been observed with increase in speed from 1000 to 2000 rpm.

Cold Start and Warm-up Phase :

Engine cold start and warm-up phase contribute significantly to unburned hydrocarbons. One of the main sources of HC emissions during cold start and engine warm-up period is very rich fuel-air ratio needed for ignition and combustion for several seconds after engine start. During cold start, the engine has to be over-fuelled 5 to10 times the stoichiometric amount of gasoline. To obtain robust ignition on the first cycle on cold start, a fuel vapour- air equivalence ratio above lean threshold limit (f = 0.7-0.9) is required. This threshold is independent of the engine coolant temperature. The fuel-air equivalence ratio supplied to the engine during cold start is in the range, f = 4 to 7. For the first few engine cycles, a large fraction of inducted fuel is stored as liquid film in the intake port and cylinder as only the most volatile fractions evaporate when the engine is cold. The liquid fuel films do not participate in combustion and is emitted as unburned fuel emissions.

Coolant Temperature

As the coolant temperature is increased, the contribution of piston ring zone crevice becomes lower due to decrease in gas density within this crevice. Secondly, the top piston-land side clearance is also reduced due to higher thermal expansion of the piston. A thinner oil film and reduced fuel vapour solubility would result in reduced absorption of fuel vapours in engine oil. Increased postflame oxidation at high temperatures also contributes to reduction in HC emissions. Increase in coolant temperatures has been observed to reduce HC emissions by about 0.4 to 1.0 % per K increase in temperature. An increase in the coolant temperature from 20 to 90° C, roughly results in 25% lower HC emissions and hence, the need of a rapid engine warm up is obvious. For reduction of the cold start and warm up HC emissions, an important area of development is to improve the fuel injection and delivery to the cylinder with minimum wall wetting. Over-fuelling during cold start and warm-up is to be kept at a minimum, while still forming the combustible charge.

Cooling system:

There are mainly two types of cooling systems :

(a) Air cooled system, and

(b) Water cooled system.

Air Cooled System Air cooled system is generally used in small engines say up to 15-20 kW and in aero plane engines. In this system fins or extended surfaces are provided on the cylinder walls, cylinder head, etc. Heat generated due to combustion in the engine cylinder will be conducted to the fins and when the air flows over the fins, heat will be dissipated to air. The amount of heat dissipated to air depends upon :

(a) Amount of air flowing through the fins.

(b) Fin surface area.

(c) Thermal conductivity of metal used for fins.

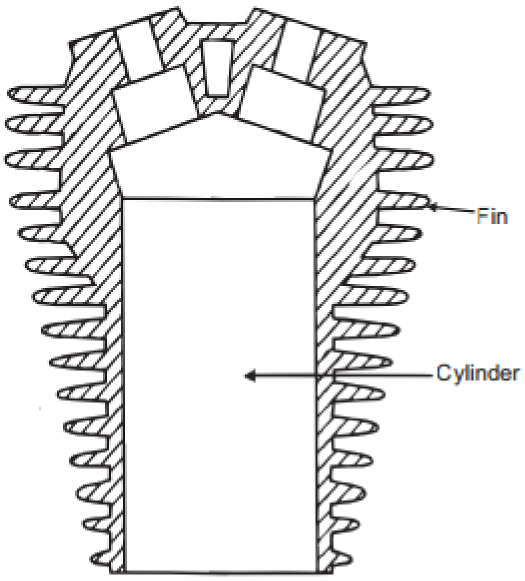


Figure 5.1 : Cylinder with Fins

Advantages of Air Cooled System:

Following are the advantages of air cooled system : (a) Radiator/pump is absent hence the system is light. (b) In case of water cooling system there are leakages, but in this case there are no leakages. (c) Coolant and antifreeze solutions are not required. (d) This system can be used in cold climates, where if water is used it may freeze. Disadvantages of Air Cooled System (a) Comparatively it is less efficient. (b) It is used in aero planes and motorcycle engines where the engines are exposed to air directly.

WATER COOLING SYSTEM

In this method, cooling water jackets are provided around the cylinder, cylinder head, valve seats etc. The water when circulated through the jackets, it absorbs heat of combustion. This hot water will then be cooling in the radiator partially by a fan and partially by the flow developed by the forward motion of the vehicle. The cooled water is again recirculated through the water jackets.

Types of Water Cooling System

There are two types of water cooling system :

Thermo Siphon System

In this system the circulation of water is due to difference in temperature (i.e. difference in densities) of water. So in this system pump is not required but water is circulated because of density difference only.

Pump Circulation System

In this system circulation of water is obtained by a pump. This pump is driven by means of engine output shaft through V-belts.

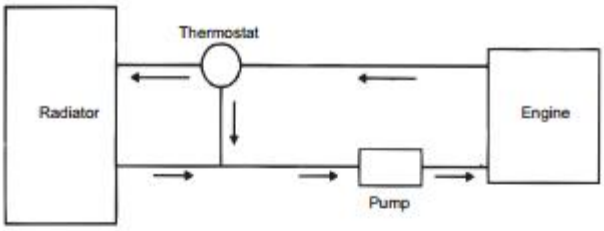


Figure 5.4 : Water Cooling System using Thermostat Valve

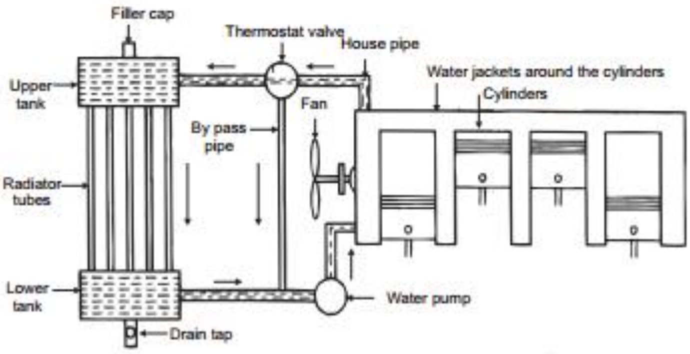


Figure 5.5 : Water Cooling System of a 4-cylinder Engine

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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

SAU1404 VEHICLE DESIGN CHARACTERISTICS

UNIT 2 PERFORMANCE CURVES

UNIT -II PERFORMANCE CURVES

ROAD PERFORMANCE CURVES: ACCELERATION, GRADEABILITY AND DRAWBAR PULL

The passenger car performance is based on acceleration, ability to go up a slope, top speed, fuel economy, noise level and durability. as wheel spin occurs, the acceleration decreases from the maximum. also the gear is designed for maximum fuel economy when the engine is developing 80% of its maximum torque as the automobile is moving at a constant speed. this gives 20% additional torque for acceleration. the power required to drive an automobile increases as the cube of the speed. when the power available matches the power required to push the vehicle, the speed becomes constant. excess power is required for acceleration and hill climbing. maximum speed is reached when there is no excess power remaining. The fig.1 illustrates the variation of full throttle power available at the wheels, for four gear ratios, with road speed. a curves are showing the power required by vehicle at various road speeds are also presented.

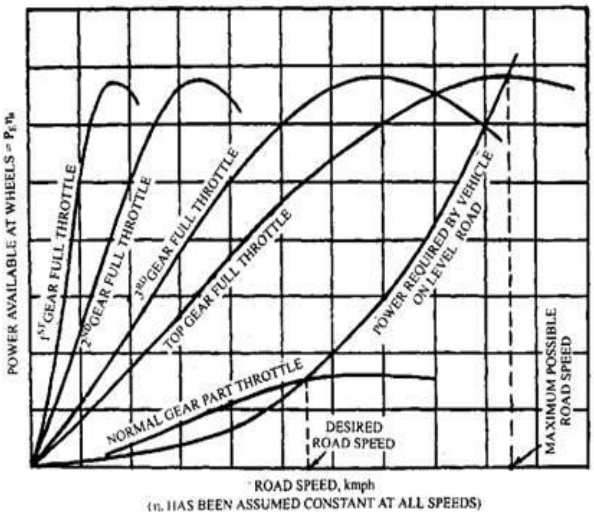


FIG . 1 ROAD SPEED Vs POWER AVILABLE AT WHEELS

At any speed, the difference of ordinates of power available and power required by vehicle gives the surplus power, which can be utilized either for acceleration or for drawbar pull or for hill climbing. by using the formula given in traction and tractive effort , the power available as indicated in Fig.1 can be converted into tractive effort. Hence tractive effort performance curves for four gear ratios can be plotted against road speed as shown in fig .2, the fig shows a road resistance curve is also presented. the difference between the ordinates of tractive effort and road resistance at any road speed gives the surplus tractive effort, which is utilized for acceleration, drawbar pull and hill climbing

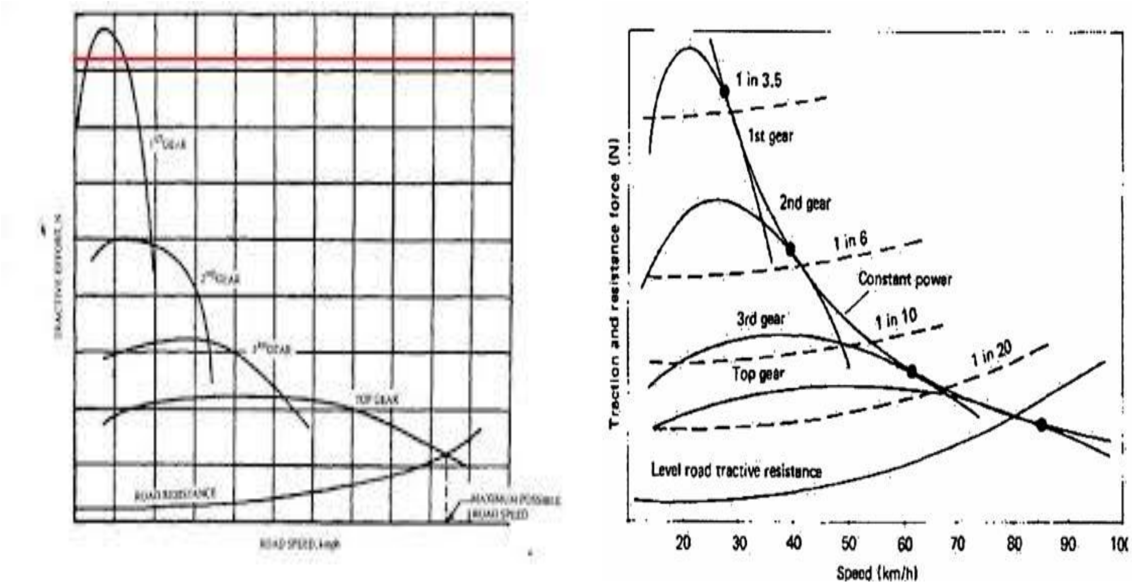


FIG.2 ROAD SPEED Vs TRACTIVE EFFORT

the fig.3 illustrate the relationship between the tractive effort and resistance and road speed

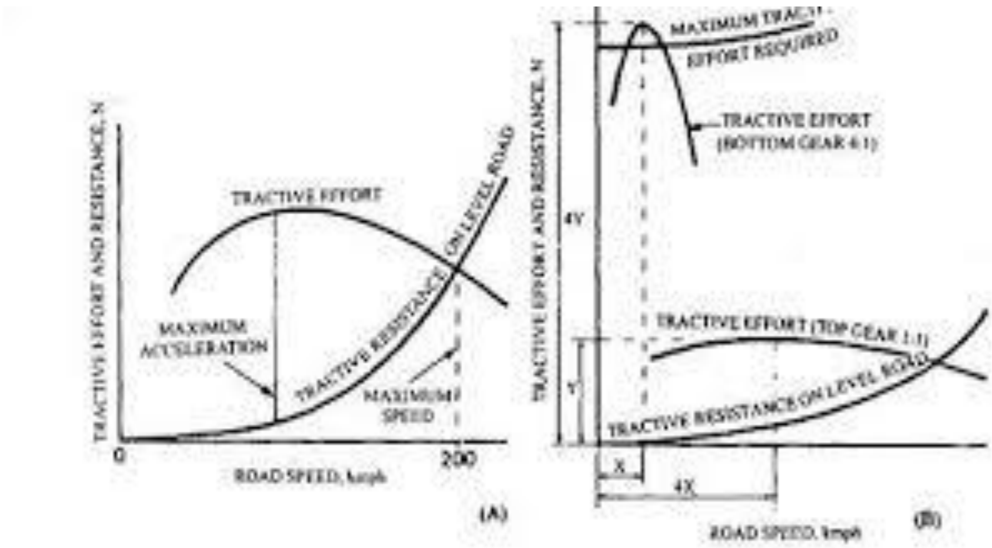


FIG.3 TRACTIVE EFFORT AND RESISTANCE Vs ROAD SPEED

ACCELERATION

when the vehicle is accelerated, its rotating parts are also accelerated depending upon their moments of inertia and the gear ratio in the drive line. due to this, weight of the vehicle is increased from W to W_E. this increased weight, W_E is called the effective or equivalent weight of the vehicle. when surplus power, i.e. surplus tractive effort is fully utilized for acceleration, then

$$\text{surplus or excess power} = W_E\, f\, \frac{V}{3600}\ kW$$

$$\text{maximum acceleration, } f = \frac{1}{W_E}\ (surplus\ power)\frac{3600}{V}$$

$$= \frac{1}{W_E}(P_E - P_R)\eta_t\ \frac{3600}{V} = \frac{1}{W_E}\ (P_E\eta_t - P_V)\frac{3600}{V}$$

$$= \frac{1}{W_E}\ (\,tractive\ effort - road\ resistance) = \frac{1}{W_E}\ (F - R)$$

GRADABILITY

the maximum percentage grade, which a vehicle can negotiate with full rated condition, is known as gradability. hence

$$surplus\ power = \frac{W\ X\ GradabilityX\quad V}{100X3600}$$

$$Gradability = \frac{100}{W}\ (P_E\ \eta_t - P_V)\frac{3600}{V}$$

$$= \frac{100}{W}\ (\,tractive\ effort - road\ resistance) = \frac{100}{W}\ (\,F - R)$$

Drawbar pull

when the excess power is fully utilized for pulling extra load attached to vehicle then

$$maximum\ drawbar\ pull\ = tractive\ effort\ -\ road\ resistance\ = (F - R)$$

road resistance in this case is made up of rolling resistance and air resistance. the Fig. 1 & 2 shows that maximum surplus power and hence maximum surplus tractive effort is provided at very low speeds of the vehicle. therefore for acceleration from start, for climbing steeper gradients and for large drawbar pull, first gear is best suited.

The maximum road speed is achieved in the gear when power available equals to power required and tractive effort becomes equal to level road resistance. if the vehicle is desired to run at a lower speed, the throttle is adjusted accordingly so that the part throttle power available curve intersects the power required curve at the desired road speed.

PROBLEMS:

1. The coefficient of rolling resistance of a truck weighing 62293.5N is 0.018 and the coefficient of air resistance is 0.0276 in the formula R = KW + KaAV2, N, where A in m2 of frontal area and V the speed in kmph. The transmission efficiency in top gear of 6:2:1 is 90%and that in the second gear of 15:1 is 80%. The frontal area is 5.574 m2. If the truck has to have a maximum speed of 88kmph in top gear, Calculate:

(a) the engine BP required.

(b) the engine speed if the driving wheels have an effective diameter of 0.8125m

(c) the maximum grade the truck can negotiate at the above engine speed in second gear.

(d) the maximum draw bar pull available on level at the above engine speed in second gear

Solution:

In top gear:

(i) $R = 0.018\, W + 0.0276\, AV^2$

$$= 0.018 \times 62293.5 + 0.0276 \times 5.574(88)^2$$

$$= 1120.3 + 1191.4 = 2312.7\ N$$

Engine $BP = \frac{RV}{1000\ \eta_t} = \frac{2312.7 \times 88}{1000 \times 0.9 \times 3.6} = 62.8\ N$

(ii) $V = \frac{2\pi N r}{G} m/min$

Hence

$$N = \frac{VG}{2\pi r} = \frac{88 \times 1000}{60} \times \frac{6.2}{2\pi \times 0.40625} = \frac{88 \times 1000 \times 6.2}{60 \times 2\pi \times 0.40625} = 3564 rpm$$

In second gear:

(iii) $V = \frac{88}{15} \times 6.2 = 36.4\ km/h = \frac{36.4}{3.6} m/s$

$$R = 0.018 \times 62293.5 + 0.0276 \times 5.574(36.4)^2$$

$$= 1121.3 + 203.8 = 1325.1\ N$$

Assuming that vehicle can climb the maximum grade of 1 in X, then $R = 1325.1\ + (62293.5/X)$

Now,

$$F = \frac{BP \times \eta_t \times 1000}{V} = 62.8 \times 0.8 \times 1000 \times (3.6/36.4)$$

=4968.8 N

Hence, 1325.1+$\frac{62293.5}{x} = 4968.8$

62293.5/x=4968.8$-1325.1 = 3643.7$

X=62293.5/3643.7=17.1

Maximum grade is 1 in 17.1

(iv) Maximum drawbar pull on level

=tractive effort available −tractive effort for resistance on level

=4968.8−1325.1=3643.7 N.

2. An automotive gear box gives three forward speeds and one reverse with a top gear of unity and bottom and reverse gear ratio of approximately 3:3:1. The centre distance between the shafts is to be 110mm approximately. Gear teeth of module 3.25 mm are to be employed.

Sketch the layout of a typical synchromesh gear box for these conditions giving the number of teeth for the various gear wheels and showing closely how the different ratios are obtained.

Solution:

Since the pitch is same for all wheel and the centre distance is the same for all pair of mating wheel, the total number of teeth must be same for each pair.

Thus,

$$T_A + T_B = T_C + T_D = T_E + T_F = \frac{110 \times 2}{3.25} = 68$$

In general practice, for better results the gear ratios are kept in a geometric progression or approaching to it.

If G_1, G_2, G_3 are $1^{st}, 2^{nd}$ and 3^{rd} or top gear ratios respectively, then

$G_2 = \sqrt{G_1 G_3}$=$\sqrt{1 \times 3.3} = 1.817.$

First gear ratio, $G_1 = \frac{T_B}{T_A}\frac{T_D}{T_C} = 3.3.$

Adopting the relation, $\frac{T_B}{T_A} = \frac{T_D}{T_C} = \sqrt{3.3} = 1.817.$

So that the speed ratios $\frac{T_B}{T_B}$ and $\frac{T_D}{T_C}$ will be as nearly equal as possible.

Hence, $T_A + T_B = 2.817\ T_A = 68, T_A = \frac{68}{2.187} = 24$

Therefore, $T_B = 68 - 24$ = 44 and also $T_C = 24 and\ T_D = 44$

Exact speed reduction $G_1 = (44/24)^2 = 3.36{:}\,1$

Second gear ratio, $G_2 = \frac{T_B}{T_A}\frac{T_F}{T_E} = 1.817$

$$\frac{T_F}{T_E} = 1.817\ \frac{T_A}{T_B} = 1.817\ \times \frac{24}{44} = 0.991$$

$$T_E = \frac{68}{1.991} = 34.05\ adopted\ and\ T_F = 68 - 34 = 34$$

Actual ratio, $G_2 = \frac{34}{34} \times \frac{44}{24} = 1.835{:}\,1$

Top gear ratio, $G_3 = 1{:}\,1.$

Reverse gear ratio:

The presence of an idler gives$T_I + T_J < 68$.

Speed ratio = $\frac{T_B}{T_A}\frac{T_J}{T_I} = 3.3$ approximately.

Adopting $T_I = 22\ and\ T_J = 40$

$$which\ gives\ the\ nearest\ approaching\ value\ of\ 3.3,$$

The exact reduction = $\frac{44}{24} \times \frac{40}{22} = 3.33{:}\,1.$

3. A motor vehicle weighs 7975.5 N and its engine develops 14.7 kW at 2500 rpm. At this engine speed the road speed of the car on the top gear is 64.37 km/h. Bottom gear reduction is 3.5:1 and the efficiency of transmission is 88% on the top and 80% on bottom gear. The diameter of tyres is 0.762 m and the projected front area of the vehicle is 1.116 m2. The coefficient of air resistance is 0.0314 N-h2/km2-m2. R=KAV2, where R is resistance in N, K is coefficient of resistance. A is the front are in m2. V is speed in km/h. Road resistance is 0.023W, N calculate

(a) Speed of car on bottom gear;

b) Tractive effort available at the wheels on top and bottom gear;

(c) Gradient at which car can climb on bottom gear.

(d) The tractive force at the wheels required to start up the car on the level and attain a speed of 48.28 km/hr in 10s. (Average air resistance may be taken as half the maximum and acceleration force to vanish at 48.28 km/h speed).

(a) On bottom gear, V=$\frac{64.37}{3.5} = 18.4$ km/h.

(b) On top gear, tractive effort, F= $\frac{P_E \times \eta_t \times 3600}{V}$

$$= \frac{14.7 \times 0.88 \times 3600}{64.37} = 723.5\ N.$$

On bottom gear, tractive effort, F =$\frac{14.7 \times 0.8 \times 3600}{18.4} = 2300.9\ N$

(c) Total resistance in negotiating the grade in bottom gear,

$$R = (0.023 \times 7975.5 + 0.0314) \times$$

$$(1.116\,(18.4)^2 + 7975.5 \sin\theta)$$

= 183.4 + 11.9 +7975.5 sin θ=195.3 +7975.5 sin θ.

Since available tractive effort is totally utilized in grade climbing, then

2300.9 = 195.3 + 7975.5 sin θ.

Sin θ = $\frac{2105.6}{7975.5} = 0.264.$

Hence, tan θ = 0.264 = 1/ 3.648

The grade which the car can negotiate on bottom gear is 1 in 3.648.

(d) The acceleration required to attain a speed of 48.28 km/h in 10s,

$$f = \frac{V}{t} = \frac{48.28}{3.6 \times 10} = 1.34\ m/s^2.$$

Total resistance on level,

R= $R_r + R_a = (0.023 \times 7975.5)$ +

$$(0.5\times 0.0314 \times 1.116\,(48.28)^2$$

= 183.4 +40.4 =223.8 N.

(As per the statement in the problem, the air resistance is taken as half the maximum)

Hence tractive effort required

$$= \text{R} + \frac{W}{g} V = 223.8 + \frac{7975.5}{9.81} \times 1.34$$

= 223.8 +1089.4 =1313.1N.

4. An engine is required to power a truck having a gross weight of 40937 N. the maximum grade which the truck will have to negotiate at 32 km/h in second gear is expected to be 15% (percentage grade equals tan θ $\times 100$). The rolling resistance coefficient is 0.017 and the air resistance coefficient 0.0324 in the formula, total resistance= $K_f W + K_a A V^2 kgf$, where A is in $m^2$and V in km/h.

the frontal area in $5.2\,m^2$. The transmission efficiency in second gear is estimated to be 80%. Calculate the minimum power which should be available from the engine and the gear ratio in second gear if this power is available at 2400 rpm and the effective radius of the wheels is 0.419 m. also calculate the minimum speed of this vehicle in top gear on level road at the same engine speed assuming a transmission efficiency of 90% in top gear. What is the gear ratio in top gear? The differential has a reduction of 3.92.

Solution:

Total resistance,

$$R = R_r + R_a + R_g = K_r W + K_a A V^2 + W sin\theta$$

$=0.017 \times 40937 + 0.0324 \times 5.2 \times 32 \times 32$

$$+49037 \times 0.1481$$

=40937 (0.017+0.1484) +172.5

=6771+172.5=6943.5 N.

Minimum power which should be available from the engine in speed gear with $\eta_t = 80\%$

$= \frac{RV}{\eta_t 1000} = \frac{6943.5}{0.8 \times 1000} \times \frac{32}{3.6} = 77.15\ kW.$

We have $V = \frac{2\pi N r}{G} m/min$

$$G = \frac{2\pi N r}{V} = \frac{2\pi \times 2400 \times 0.419 \times 60}{32 \times 1000} = 11.85.$$

Differential has a reduction of 3.92.

Hence second gear ratio is 11.85/3.92:1, i.e. 3.02:1

In top gear with $\eta_t = 0.9\ and$ with same engine speed, the total resistance on level

$= 0.017 \times 40937 + 0.0324 \times 5.2V^2, N$

$= \frac{BP \times \eta_t \times 3600}{V}\ where\ V\ in\ km/h.$

Hence (696+0.1685V^2) v=77.15$\times$ $0.9 \times 3600 = 249966$.

By trail, V=102.1km/h.

Maximum speed of the vehicle on level in top gear=102.1km/h.

Also as before, $G = \frac{2\pi Nr}{V} = \frac{2\pi \times 2400 \times 0.419 \times 60}{102.1 \times 1000} = 3.72$.

Hence top gear is 3.72/3.92:1, i.e. 0.95:1.

5. For typical motor car, the road resistance is given by 23N per 1000 N, the air resistance by the expression $0.0827V^2$, transmission efficiency 88% in top speed, car weights 19934N when fully loaded. Calculate

a) The Bp kW required for a top speed of 144 km/hr.
b) The acceleration in m/s^2at 48 km/hr, assuming the torque at 48 km/hr in the top gear 25% more than at 144km/hr.
c) The Bp kW required to drive the car up a gradient of 1 in 5 at 48 km/hr, transmission efficiency 80% in bottom gear. The resistance being in N and V the speed in km/h and g=9.81 m/s^2.

Solution:

(a) Total resistance at speed 144km/h

$$R = 23 \times 19.934 + 0.0827 \times 144 \times 144 = 458.5 + 1715$$
$$= 2173.5$$

The Bp kw = $\frac{RV}{3600\ \eta_t} = \frac{2173.5 \times 144}{0.88 \times 3600} = 98.8\ KW.$

(b) T_{E1} *and* T_{E2} *are the engine torque at the speed of* 14 Km/h respectively, and F_1 *and* F_2 are the corresponding tractive effort, then as given in the problem,

$$T_{E2} = 1.25\ T_{E1}$$

Hence,

$$F_2 = 1.25 F_1\ as\ radius, r\ is\ same = 1.25\ \times 2173.5$$
= 2717 N.

Total resistance at the speed 48 km/h,

R=458.5+0.0827 $\times 48\ \times 48 = 458.5 + 190.5 = 649N$.

We have, $F = \frac{w}{g} f + R$.

$2717 = \frac{19934}{9.81} f + 649.$

$$f = \frac{2068 \times 9.81}{19934} = 1.02\ m/s^2.$$

(c) For the gradient 1 in 5, tan θ = 0.2 and sin θ =0.196.
$= 649 + 19934 \times 0.196 = 649 + 3907 = 4556\ N.$
The BP required $= \frac{RV}{\eta\, 3600} = \frac{4556 \times 48}{0.8 \times 3600} = 76\ kW.$

6. Determine the gear ratios of a four speed gear box for a vehicle of weight 13341.6 N powered by an engine giving 20.6 kW at 1800 rpm. The vehicle has a frontal area of $2.23m^2$ and has a wheel diameter 0.71 m. the maximum gradient that the car has to negotiate is 1 in 4. The tractive resistance may be taken as 50 N per 2240 N of the car. The wind resistance is given by 0.03679 $AV^2, where\ A\ is\ the\ frontal\ area$ in m^2 and V is the vehicle speed in km/h. assume that the transmission efficiency is 0.75 and that at top gear, the car is expected to go over a grade of 1 in 40. State any other assumption you make.

Solution: $V = \frac{2\pi N r}{G}$

$$= \frac{2\pi \times 1800 \times 0.355 \times 60}{1000\ G} = \frac{240.775}{G} \text{ Km/h.}$$

In the top gear:

$$R_a = 0.03679\ A\ V^2$$

$$= 0.03679 \times 2.23 \left(\frac{240.775}{G}\right)^2 = \frac{4756.2}{G^2}\ N$$

$$R_r = \frac{50 \times 13341.6}{2240} = 297.8\ N$$

$$R_g = 13341.6/40$$

$$R = \frac{4756.2}{G^2} + 297.8 + \frac{13341.6}{40} = \frac{4756.2}{G^2} + 631.3.$$

Now,

$$BP \times \eta_t = \frac{RV}{1000}$$

$20.6 \times 0.75 = \left(\frac{4756.2}{G^2} + 631.3\right)\left(\frac{240.775}{G}\right)\left(\frac{1}{1000}\right).$

$$G^3 = 2.73\,G^2 + 20.6$$

By trial, the value of G = 4.
In the first gear:
Total resistance, R = $R_r + R_a + R_g$
$=297.8 + \frac{4756.2}{G^2} + \frac{13341.6}{4} = \frac{4756.2}{G^2} + 3633.2$
Hence as before,
$20.6 \times 0.75 = \left(\frac{4756.2}{G^2} + 3633.2\right)\left(\frac{240.775}{G}\right)\frac{1}{1000}$

$$G^2 = 1573G^2 + 20.6.$$

By trial, the value of G = 15.8.
If G_1, G_2, G_3, G_4 are1st, 2nd,3rd and top gear ratios respectively, then $G_4 = 1$.
$G_1 = \frac{15.8}{4}$= 3.95.

$$generally\ the\ gear\ ratios\ are\ in\ geometric\ progression\ then$$

$$G_3^2 = G_4\,G_2 = G_{2\ as}\,G_{4=1}$$

$$G_2^2 = G_1 G_3 = G_1 G_2^{1/2}$$

$$G_2 = G_1^{1/1.5} = (3.95)^{0.666} = 2.5$$

$$G_3 = G_1^{1/2} = (2.5)^{0.5} = 1.581.$$

The required gear ratios are 1: 1; 1.581:1; 2.5: 1; 3.95:1.

7. The maximum gear box ratio of an engine 75 mm bore and 100 mm stroke is 4. The pitch diameter of the constantly meshing is 75% of the piston stroke. If the module is 4.25 mm, calculate the size and number of teeth of gear for a three speed gear box. Calculate the face width of the constantly meshing gear using modified lewis formula. The engine torque is 910kgf-cm value of constant in the lewis formula is 0.07 and the allowable stress is 900kgf/cm^2. Draw the neat sketch of three speed gear layout.

If $G_1, G_2, G_3, are\ 1^{st}, 2^{nd}\ and\ top\ gear\ ratios$ respectively then, $G_1 = 4\ and\ G_3 = 1$.

Taking gear ratios in geometrical progression $G_2 = \sqrt{G_1 G_3} = \sqrt{4 \times 1} = 2$

First gear ratio $G_1 = \frac{T_B}{T_A}\frac{T_D}{T_C} = 4, giving\ \frac{T_B}{T_A} = \frac{T_D}{T_C} = \sqrt{4} = 2$

Adopting $T_A = T_c$ =16 to avoid interference, then $T_B = T_D$ = 32 adopted.

Then $T_A + T_B = T_c + T_D = T_E + T_F = 48$.

Pitch diameter of constantly meshing gear, gear A = 0.75$\times 100 = 75mm$.

Pitch diameter of pinion C = module $\times no.of\ teeth = 4.25 \times 16 = 68mm$.and that gear D and pinion B = 4.25 $\times 32 = 136mm$.

Second gear ratio, $G_2 = \frac{T_B}{T_A}\frac{T_F}{T_E} = 2$

$$\frac{T_F}{T_E} = 2\ \frac{T_A}{T_B} = 2\ \frac{15}{30} = 1$$

$$T_E = T_F = 24\ adopted.$$

Pitch diameter of pinion E and gear F = 4.25 $\times 24 = 102\ mm$.

Top gear ratio, G_2 = 1: 1.

$$T_e = \frac{D}{2000}\ F$$

$$89.27 = F\ \frac{75}{2000}, so\ that\ F = \frac{89.27 \times 2000}{75} = 2380.5\ N$$

Modified Lewis formula gives, F = $\frac{cfb}{1000}\ \frac{m}{1000}$

Substituting the values,

2380.5 = 0.07$\times 8829 \times 10^4 \times \frac{b}{1000} \times \frac{4.25}{1000}$

$$b = \frac{2380.5 \times 10^6}{0.07 \times 8829 \times 10^4 \times 4.25} = 90.6\ mm.$$

POWER REQUIRED BY THE VEHICLE:

A driving horse power at the road wheel is proportional to the total resistance and the EDF to give the required acceleration

$$vehicle\ speed = \frac{VX1000}{3600}\ m/sec$$

$$watt\ output = \frac{DFXVX1000}{3600}\ kg-m/sec$$

$$Driving\ HP = \frac{DFXVX1000}{3600} X \frac{1}{75}\ HP$$

$$DHP = \frac{DFXV}{270}\ HP$$

$$Brake\ HP = 1.1XDHP$$

$$Indicated\ HP = 1.1Xbhp$$

$$since\ IHP > BHP > DHP$$

RELATION BETWEEN ENGINE SPEED AND VEHICLE SPEED:

Let N- engine speed in rpm

speed of road wheel = $\frac{N}{Gr_A}\ rpm$

where G- gear ratio

$r_a -\ axle\ reduction$

1HP= 735.5 Watts

distance travelled by wheel/min = $\frac{N}{Gr_a}\ X2\pi R_W\ m/min$

$R_w = Effective\ wheel\ radius$

vehicle speed, $V = \left(\frac{N}{G\ r_a} 2\pi R_W\right) \frac{60}{1000} rpm/h$

V= $\left(\frac{N.R_W}{G.r_a}\right) (\frac{2\pi X\,60}{1000})$

$$\frac{N}{V} = \left(\frac{1000}{2\pi X60}\right) \frac{G.r_a}{R_W}$$

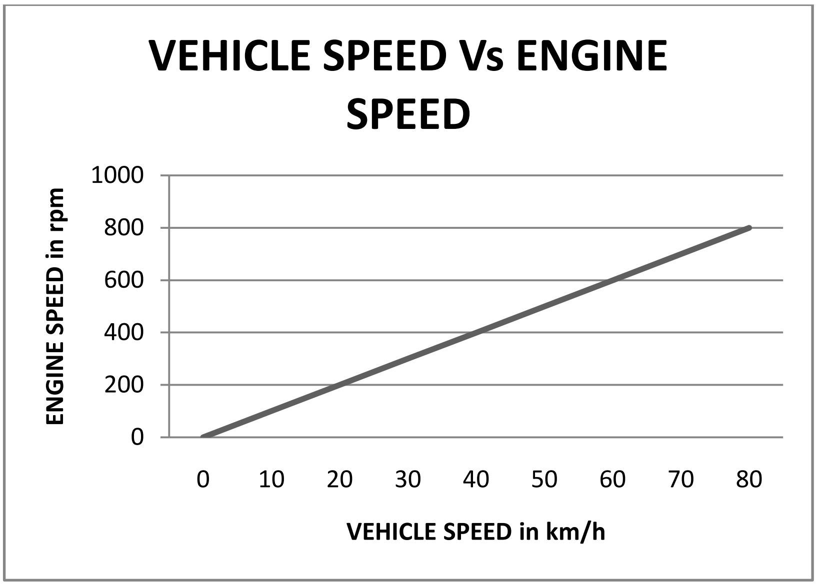
$$\frac{N}{V} = 2.655\ \frac{G.r_a}{R_W}$$

Assume, gear ratio, G=1:1

axle reduction, $r_a = 3\ or\ 4$

wheel radius, $R_W = 0.2844\ \cong 0.3m$

Graphs:



CALCULATION PROCEDURE:

vehicle speed, $V = 50\ km/h$

1. Excess Driving Force, EDF = $\left(\frac{W}{g}\right)\left(\frac{a}{K_a}\right)$ in kg f
2. Driving Force, $DF = EDF + \ R_T\ in\ kgf$
3. Driving horse power, $DHP = \frac{DFXV}{270}\ in\ HP$
4. Brake Horse Power, $BHP = 1.1\ X\ DHP\ in\ HP$
5. Indicated Horse Power,$IHP = 1.1\ X\ BHP\ in\ HP$
6. Engine Speed, $n = \frac{2.65XVXGX\,r_a}{R_W}\ in\ rpm$

Notes:

1. The power available at engine cylinder is called as Indicated Horse Power.
2. The power available at output shaft of an engine is called as Brake Power.
3. The power available at road and road wheels is called Driving Horse Power.

FORMULAS FOR CALCULATING EFFICIENCIES AT DIFFERENT VEHICLE SPEED

1. $FMEP = a + b\left(\frac{N}{1000}\right) + \; C(N/1000)^2$

   where, $FMEP = Frictional\ Mean\ Effective\ Pressure\ in\ bar$

   $N = Engine\ rpm\ at\ maximum\ BHP$

   $a = 0.5622$

   $b = 0.2811$

   $c = 0.0527$

2. $FHP = \frac{FMEP\ X\ L\ AX\ n}{4500X100}$

   $n = \frac{N}{2}\ for\ 4-5\ engines.$

   $LA = 1200 - 1300\ cc$

3. $IHP = BHP + FHP$

   where $IHP = Indicated\ Horse\ Power$

4. To find LA

   $IHP = \frac{IMEPXLAXn}{4500X100}$

$$LA = \frac{IHPX4500X100}{IMEPXn}$$

after calculating LA, try to check it up the variation between the assumed value of LA, calculated value of LA. It must be within 5%. If the variation is more change the assumed value of LA suitably and recalculate the calculated value of LA until the variation betwee the LA within 5%.After finding LA fix the number of cylinders.

If LA>1600cc, Assume number of cylinders as 6.

If LA< 1600cc, Assume number of cylinders as 4.

5. To find Bore(B) and Stroke(L):

   If B=L , Square engine

   If B>L, under square engine

   If B<L, Over square engine

   For under square engine and over square engine the maximum ratio between bore and stroke is

   $BXL = 1.2$

   using this find B and L

$$\frac{LA}{4} = \frac{\pi}{4}\ B^2\ X\ L$$

For square engine L=B

FHP from FMEP

BHP from graph

$IHP = BHP + FHP$

$\eta_{mech} = \frac{BHP}{IHP}$

$$BMEP = \frac{BHPX4500X100X2}{LAXN}$$

$$TORQUE = \frac{BHPX4500}{2\pi N}$$

Graphs:

$$V\ vs\ \eta_{mech}$$

$$V\ vs\ T$$

$$V\ vs\ BMEP$$

$$V\ vs\ DF$$

$$V\ vs\ BHP, IHP, FHP$$

Notes:

1. The maximum Torque must be available at 50% of V_{max}
2. Driving Force and Torque curves must be exactly in similar shape.
3. Starting and ending value of Torque is same

VEHICLE ACCELARATION :

Assume maximum acceleration from vehicle data is 0.9 to 1.05m/s^2 for the vehicle which is having maximum vehicle speed 120 km/h. the maximum acceleration always occurs at 1/3 of maximum vehicle speed. the acceleration of the vehicle at maximum speed is zero. now, using the concept draw a smooth curve for vehicle speed vs. acceleration. now find the acceleration for different vehicle speed from the above curve

DRIVING FORCE:

$DF = EDF + R_T \ldots\ldots\ldots\ldots . kgf$

where EDF= Excess Driving Force in kgf

$EDF = \frac{W}{g} X \frac{a}{K_a} \ldots\ldots\ldots\ldots .. kgf$

where W= weight in kg

a= acceleration

$K_a = acceleration\ constant$ (0.9)

g= acceleration due to gravity

R_T = total resistance in kgf

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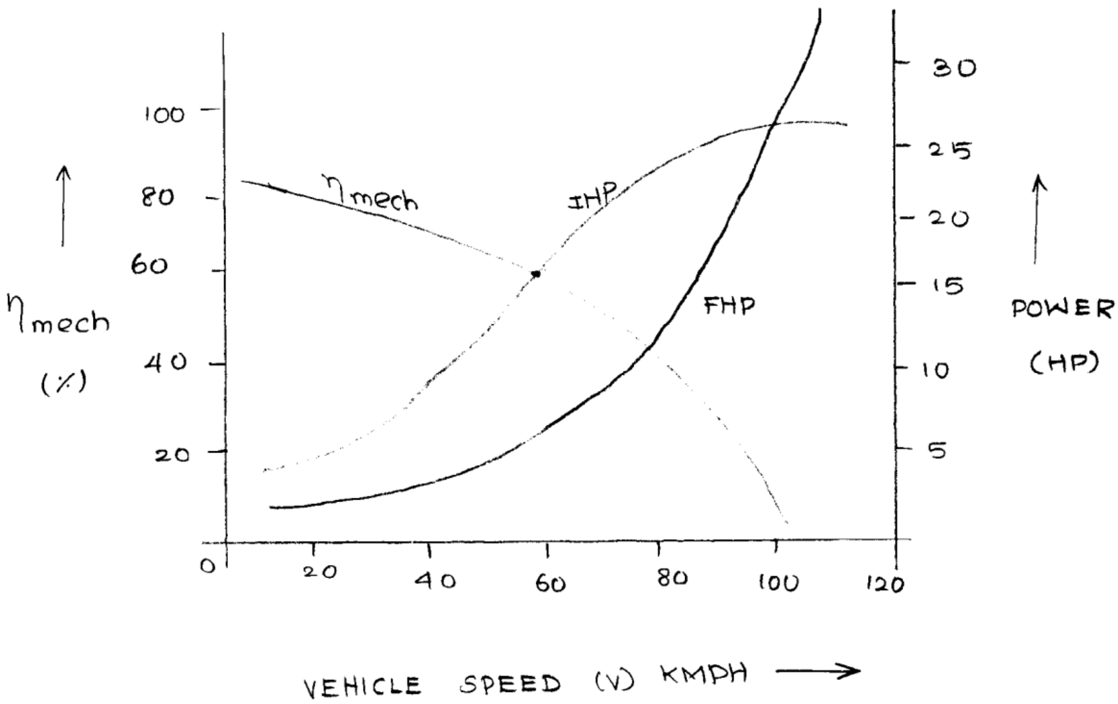

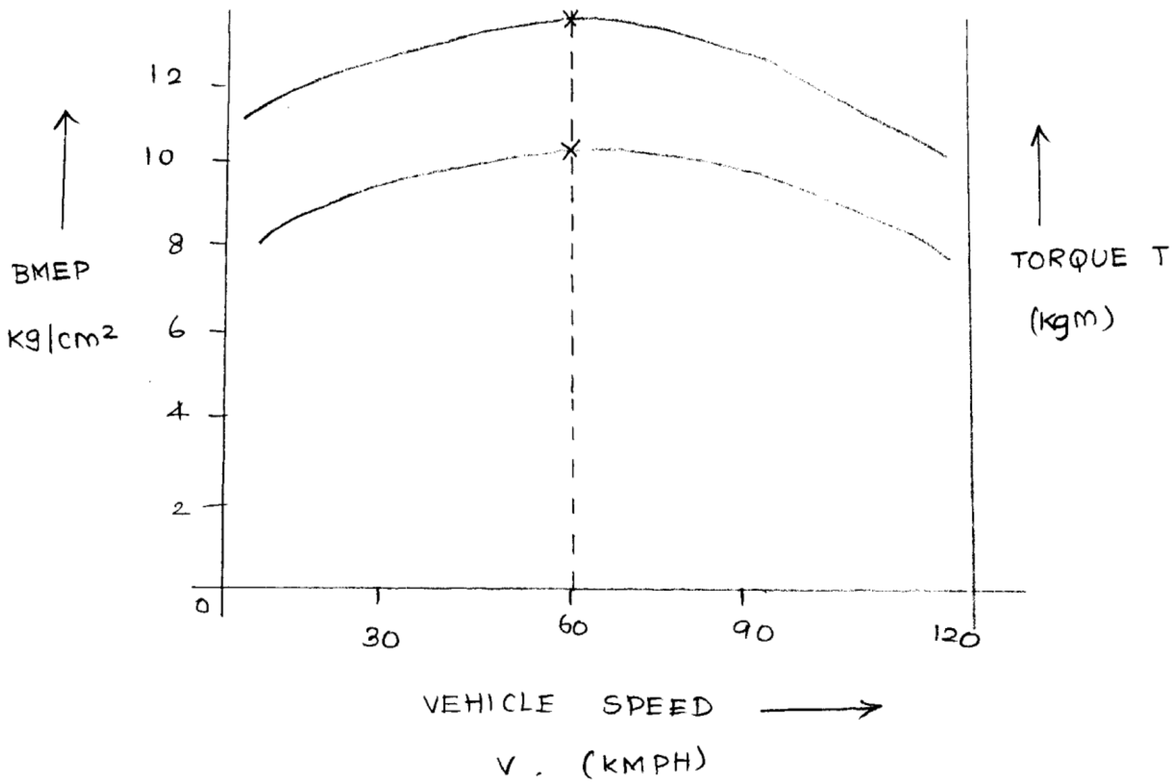

BMEP
Kg/cm²
TORQUE T
(kgm)
12
10
8
6
4
2
0
30
60
90
120
VEHICLE SPEED
V. (KMPH)

TEXT / REFERENCE BOOKS

1. Giri. N.K. “Automobile Mechanics” Khanna Publishers – New Delhi – 2002.

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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

SAU1404 VEHICLE DESIGN CHARACTERISTICS

UNIT 3 RESISTANCE TO VEHICLE MOTION

UNIT-III RESISTANCE TO VEHICLE MOTION

1. POWER OF PROPULSION

the motion of a vehicle moving on a road is resisted by aerodynamic forces, known as wind or air resistance, and road resistance, which is generally termed as rolling resistance. in addition to these two types of resistance, the vehicle has to overcome grade resistance when it moves up on a gradient, because the weight of the vehicle is to be lifted through a vertical distance. hence the power required to propel a vehicle is proportional to the total resistance to its motion and the speed.

Let, P_V = power required by the vehicle, kW,

P_R = engine power required, kW,

V = speed of the vehicle, km/h,

η_t = transmission or drive line efficiency,

R = total resistance, N,

R_a = air resistance, N

R_r = rolling resistance, N

R_g = grade resistance, N

Power required to propel a vehicle is given

$$P_V = \frac{RV}{3600} kW$$

where $R = R_a + R_r$ When vehicle moves along a level road.

$R = R_a + R_r + R_g$ when vehicle moves up a gradient

Calculation of engine power takes into account the losses in transmission. hence required engine power,

$$P_R = \frac{P_V}{\eta_t} = \frac{RV}{3600\ \eta_t} \text{ kW}$$

1.1 Air Resistance

this is the resistance offered by air to the movement of a vehicle. the air resistance has an influence on the performance, ride and stability of the vehicle and depends upon the sizes and shape of the body of the vehicle,its speed and the wind velocity. hence air resistance

$$R_a = K_a \, A \, V^2$$

where A= projected frontal area, m^2

V= speed of the vehicle, km/h,

K_a= coefficient of air resistance , N-h^2/m^2-km^2

= 0.023 for best streamlined cars

= 0.031 for average cars

=0.045 for trucks and lorries.

1.2 ROLLING RESISTANCE

the magnitude of rolling resistance depends mainly on

a) the nature of road surface
b) the types of tyre
c) the weight of the vehicle
d) the speed of the vehicle

the rolling resistance is expressed as

$$R_r = KW$$

where W= total weight of the vehicle, N

K= constant of rolling resistance and depends on the nature of road surface and types of tyre

=0.0059 for good roads

=0.18 for loose sand roads

= 0.015 a representative value

rolling resistance is given by

$$R_r = (\, a + bV)W$$

V=speed of the vehicle, km/h

a= 0.015 and b=0.00016

1.3 GRADE RESISTANCE

the component of the weight of the vehicle parallel to the gradient or the slope on which it moves is termed as GRADE RESISTANCES. Hence grade resistance is expressed as

$$R_g = W \sin\theta$$

where W= total weight of the vehicle,N

θ = inclination of the slope to the horizontal

$$Percentage\ Grade = tan\theta\ X\ 100, but\ for\ small\ values\ of\ \theta, tan\theta = sin\theta$$

TRACTION AND TRACTIVE EFFORT

the force available at the contact between the drive wheel tyres and road is known as tractive effort. the ability of the drive wheels to transmit this effort without slipping is known as traction. the tractive effort relate to engine power as follows

engine torque , $T_E = \frac{60000\ P_E}{2\ \pi\ N}$

torque at drive wheels, $T_W = (g.r\ X\ a.r)\ \eta_t\ T_E = G\ \eta_t\ T_E$ Nm,

Tractive effort, $F = \frac{T_W}{r} = \frac{T_E\ G\ \eta_t}{r}\ , N$

where $P_E = engine\ BP, kW$

$T_E = mean\ engine\ torque, Nm$

$\eta_t = overall\ transmission\ efficiency$

$g.r$ =gear box gear ratio,

$a.r$ = axle ratio,

G= overall gear ratio = = $(g.r\ X\ a.r)$

r= radius of tyre, m

N= rpm of crankshaft

RELATIONSHIP BETWEEN ENGINE REVOLUTION (N) AND VEHICLE SPEED (V)

using the relation, $\frac{2\pi r N}{G} = \frac{1000\ V}{60}$

the ratio between engine revolution, N and vehicle speed V can be obtained

$$\frac{N}{V} = \frac{1000\ G}{2\pi r\ X\ 60} = 2.65\ \frac{G}{r}$$

where V = Vehicle speed in km/hr

r = radius of tyre in m

thus $\frac{N}{V}$ ratio depends upon the overall gear ratio and wheel diameter. a vehicle with four different gears has four different values of $\frac{N}{V}$ ratio. the $\frac{N}{V}$ ratio increases as the wheel diameter increases, the overall gear ratio remaining constant.

Problems

#1. The engine of a jeep is known to be able to provide 40.5 bkW for propulsion purpose. in a certain application, the jeep weighing 12549N is required to pull a trailer of gross weight 10673N at a speed of 57.75km/h in top gear on level. the resistance to motion is given by the equation $R = aW + b\,V^2$, where a=0.016 and b=0.055, W is in N and V in km/h. find out the jeep is adequate for the job, if transmission efficiency is 90%. What is the pull in the coupling at this speed ? If the available power is just utilised in top gear by suitably loading the trailer, what is the pull in the coupling at 57.75km/h?

SOLUTION:

Total weight, W=12459+10673= 23132N.

resistance to motion, $R = aW + b\,V^2 = 0.016W + 0.055\,V^2$

$= 0.016X23132 + 0.055(\,57.75^2)$

= $370.11 + 183.20 = 553.32N$

Tractive effort, $F = \frac{BP\ X\ 1000X\ \eta_t}{V}$, where V in m/s

$$=\frac{40.5X1000X0.9X60X60}{57.75X1000} = 2272.2N$$

since $F < R$, the jeep is adequate for the job

the pull in the coupling at this speed=$0.016X10673 = 2272.2N$

extra pull on the coupling =$2272.2 - 553.3 = 1718.9N$

total pull on the coupling at same speed with extra loading

$$=1718.9 + 170.8 = 1889.7N$$

#2. An engine is required to power a truck having a gross weight of 40937N. the maximum grade which the truck will have to negotiate at 32km/h in second gear is expected to be 15% (percentage grade equals $tan\theta\, X100$). the rolling resistance coefficient is 0.017and the air resistance coefficient 0.0324 in the formula, total resistance= $K_f W + K_a AV^2 kgf$,where A is in m^2 and V in km/h. the frontal area is $5.2m^2$. the transmission efficiency in second gear is estimated to be 80%. calculate the minimum power which should be available from the engine and the gear ratio in second gear if this power is available at 2400rpm and the effective radius of the whels is 0.419m. also calculate the minimum speed of the vehicle in top gear on level the gear ratio in top gear? the differential has a reduction of 3.92?

SOLUTION:

total resistance, $R = R_r + R_a + R_g = K_r W + K_a AV^2 + Wsin\theta$

$$=(0.017X40937) + (0.0324X5.2X32X32) + (49037X0.1481)$$

$$=6943.5N$$

minimum power which should be available from the engine in second gear with $\eta_t = 80\%$

$$=\frac{RV}{\eta_t X1000} = \frac{6943.5}{0.8X1000} X\frac{32}{3.6} = 77.15kW$$

$$V = \frac{2\pi Nr}{G}\ m/min$$

$$G = \frac{2\pi Nr}{V} = \frac{2\pi X2400X0.419X60}{32X1000} = 11.85$$

Differential has a reduction of 3.92

hence second gear ratio is $11.85/3.92 : 1$ ie $3.02 : 1$

In top gear with $\eta_t = 0.9$ and with same engine speed, the total resistance on level

$$= (\,0.017X40937) + (0.0324X\,5.2\,V^2), N$$

$$= \frac{BP\ X\ \eta_t\ X3600}{V}\ where\ V\ in\ km/h$$

hence $(696 + 0.1685V^2)V = 77.15X0.9X3600 = 249966$

By trial $V = 102.1km/h$

maximum speed of the vehicle on level in top gear= $102.1\ km/h$

Also as before $G = \frac{2\pi Nr}{V} = \frac{2\pi X\,2400\,X0.419X60}{102.1X1000} = 3.72$

hence top gear ratio is $3.72/3.92\ : 1$ ie $0.95 : 1$

#3. A truck weighs 100111N and the engine develops 97bkW at 2400rpm. the transmission efficiency is 90% in top gear of 3.4:1 and 85% in third gear of 8.4:1. the performance of the vehicle is such that it will just reach a speed of 86.8km/h at 2400rpm at wide open throttle when running on the level in still air, and at the same engine speed in third gear it will just climb a gradient of 1 in 14. if the total resistance in N is given by the formula

$$R = KW + K_a AV^2 + Wsin\theta,$$

where A is in m^2, V in km/h and W in N, calculate K and K_a and hence the engine power required for climbing a grade of 1in 40 at 48km/h in top gear. how much more weight can be added to the vehicle to use the engine power fully under the above condition. Frontal area of truck = 5.575m^2

SOLUTION:

in the top gear on the level road with $V = 86.8\ km/h$,

tractive effort, $F = \frac{BP\eta_t X3600}{V} = \frac{97X0.9X3600}{86.8} = 3621N,$

tractive effort has been utilised to overcome resistance, then

$$3621 = KX100111 + K_a X5.575\ (86.8^2)$$

$$3621 = 100111K + 42000K_a \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.i$$

in third gear with same engine speed, $V = \frac{86.8X3.4}{8.4} = 35.1km/h$,

tractive effort, $F = \frac{97X0.85X3600}{35140} = 8456.4\ N$

this tractive effort has been utilised to overcome the grade of 1 in 14 at 35.1km/h

$$8456.4 = 100111K + K_a X5.575(35.1^2) + (100111/14$$

$$1305.6 = 100111K + 6860\ K_a \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.ii$$

subtracting equation i&ii gives

$$K_a = \frac{2315.4}{35140} = 0.06589$$

substituting the value of $K_a\ in\ equation\ i$,

$$K = \frac{3621 - 2767.4}{100111} = \frac{853.6}{100111} = 0.00852,$$

hence $R = 0.00852W + 0.06589AV^2 + Wsin\theta$

total resistance in climbing the grade of 1 in 40 at 48km/h in top gear,

$$R = (0.00852X100111) + \left(0.06589X5.575(48^2)\right) + (100111/40)$$

$$= 852.9 + 846.3 + 2502.8 = 4202N$$

engine power= $\frac{RV}{3600\eta_t} = \frac{4202X48}{3600X0.9} = 62.25kW$

tractive effort, $F = \frac{97X0.9X3600}{48} = 6547.5N$

hence tractive effort for drawbar pull= $6547.5 - 4202 = 2345.5\ N$

if W is the weight added to utilise this extra tractive effort, then

$$2345.5 = 0.00852W + (W/40) = (0.00852 + 0.025)W$$

$$W = 2345.5/0.03352 = 69973N$$

POWER REQUIRED BY THE VEHICLE:

A driving horse power at the road wheel is proportional to the total resistance and the EDF to give the required acceleration

$$vehicle\ speed = \frac{VX1000}{3600}\ m/sec$$

$$watt\ output = \frac{DFXVX1000}{3600}\ kg - m/sec$$

$$Driving\ HP = \frac{DFXVX1000}{3600} X \frac{1}{75}\ HP$$

$$DHP = \frac{DFXV}{270}\ HP$$

$$Brake\ HP = 1.1XDHP$$

$$Indicated\ HP = 1.1Xbhp$$

$$since\ IHP > BHP > DHP$$

RELATION BETWEEN ENGINE SPEED AND VEHICLE SPEED:

Let N- engine speed in rpm

speed of road wheel = $\frac{N}{Gr_A}\ rpm$

where G- gear ratio

$r_a -\ axle\ reduction$

1HP= 735.5 Watts

distance travelled by wheel/min = $\frac{N}{Gr_a}\ X2\pi R_W\ m/min$

$R_w = Effective\ wheel\ radius$

vehicle speed, $V = \left(\frac{N}{G\ r_a} 2\pi R_W\right) \frac{60}{1000} rpm/h$

V= $\left(\frac{N.R_W}{G.r_a}\right) \left(\frac{2\pi X\,60}{1000}\right)$

$$\frac{N}{V} = \left(\frac{1000}{2\pi X60}\right) \frac{G.r_a}{R_W}$$

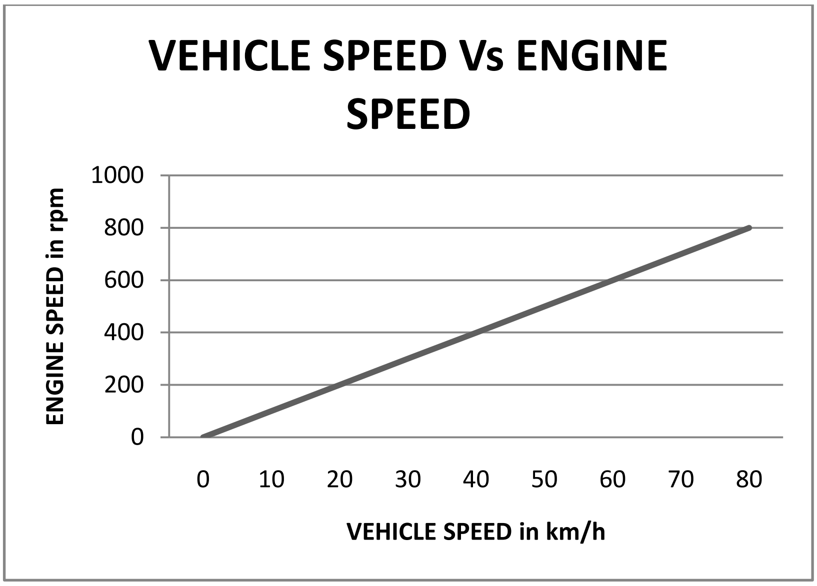
$$\frac{N}{V} = 2.655\ \frac{G.r_a}{R_W}$$

Assume, gear ratio, G=1:1

axle reduction, $r_a = 3\ or\ 4$

wheel radius, $R_W = 0.2844\ \cong 0.3m$

Graphs:



CALCULATION PROCEDURE:

vehicle speed, $V = 50\ km/h$

1. Excess Driving Force, EDF = $\left(\frac{W}{g}\right)(\frac{a}{K_a})$ in kg f
2. Driving Force, $DF = EDF +\ R_T\ in\ kgf$
3. Driving horse power, $DHP = \frac{DFXV}{270}\ in\ HP$
4. Brake Horse Power, $BHP = 1.1\ X\ DHP\ in\ HP$
5. Indicated Horse Power,$IHP = 1.1\ X\ BHP\ in\ HP$
6. Engine Speed, $n = \frac{2.65XVXGX\,r_a}{R_W}\ in\ rpm$

Notes:

1. The power available at engine cylinder is called as Indicated Horse Power.
2. The power available at output shaft of an engine is called as Brake Power.
3. The power available at road and road wheels is called Driving Horse Power.

FORMULAS FOR CALCULATING EFFICIENCIES AT DIFFERENT VEHICLE SPEED

1. $FMEP = a + b\left(\frac{N}{1000}\right) + C(N/1000)^2$

   where, $FMEP = Frictional\ Mean\ Effective\ Pressure\ in\ bar$

   $N = Engine\ rpm\ at\ maximum\ BHP$

   $a = 0.5622$

   $b = 0.2811$

   $c = 0.0527$

2. $FHP = \frac{FMEP\ X\ L\ AX\ n}{4500X100}$

   $n = \frac{N}{2}\ for\ 4 - 5\ engines.$

   $LA = 1200 - 1300\ cc$

3. $IHP = BHP + FHP$

   where $IHP = Indicated\ Horse\ Power$

4. To find LA

   $IHP = \frac{IMEPXLAXn}{4500X100}$

   $$LA = \frac{IHPX4500X100}{IMEPXn}$$

   after calculating LA, try to check it up the variation between the assumed value of LA, calculated value of LA. It must be within 5%. If the variation is more change the assumed value of LA suitably and recalculate the calculated value of LA until the variation betwee the LA within 5%.After finding LA fix the number of cylinders.

   If LA>1600cc, Assume number of cylinders as 6.

   If LA< 1600cc, Assume number of cylinders as 4.

5. To find Bore(B) and Stroke(L):

   If B=L , Square engine

   If B>L, under square engine

   If B<L, Over square engine

   For under square engine and over square engine the maximum ratio between bore and stroke is

   $BXL = 1.2$

   using this find B and L

$$\frac{LA}{4} = \frac{\pi}{4} B^2 X L$$

For square engine L=B

FHP from FMEP

BHP from graph

$IHP = BHP + FHP$

$\eta_{mec\,h} = \frac{BHP}{IHP}$

$$BMEP = \frac{BHPX4500X100X2}{LAXN}$$

$$TORQUE = \frac{BHPX4500}{2\pi N}$$

Graphs:

$$V\ vs\ \eta_{mec\,h}$$

$$V\ vs\ T$$

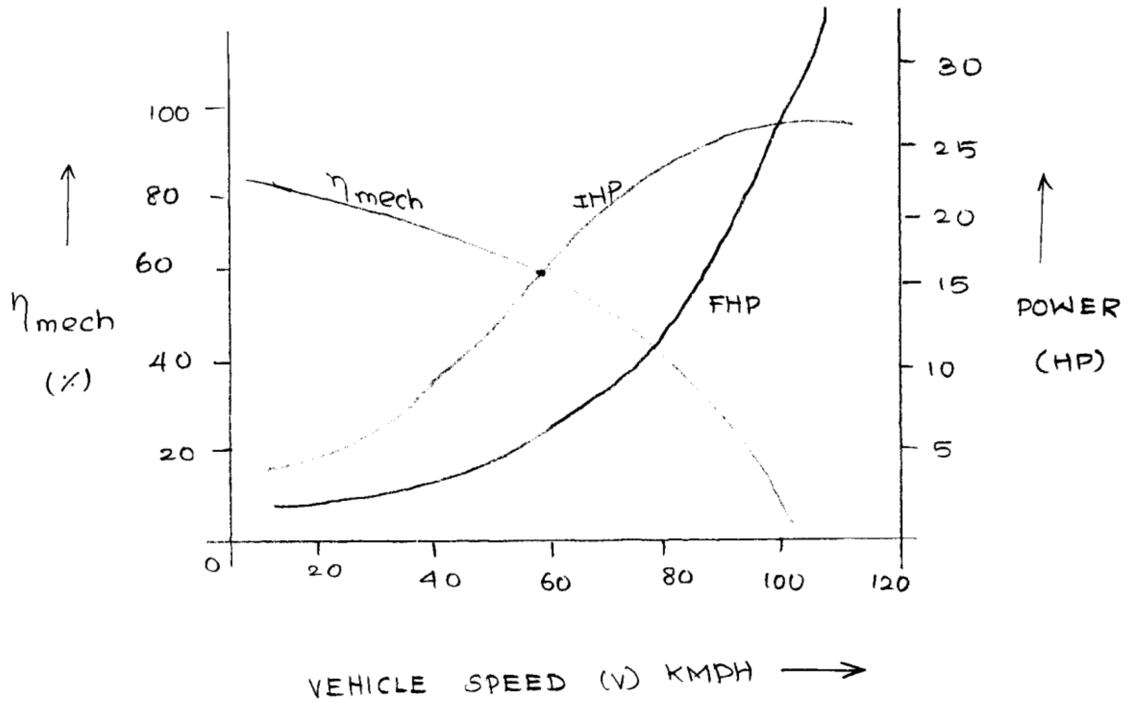
$$V\ vs\ BMEP$$

$$V\ vs\ DF$$

$$V\ vs\ BHP, IHP, FHP$$

Notes:

1. The maximum Torque must be available at 50% of V_{max}
2. Driving Force and Torque curves must be exactly in similar shape.
3. Starting and ending value of Torque is same.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10
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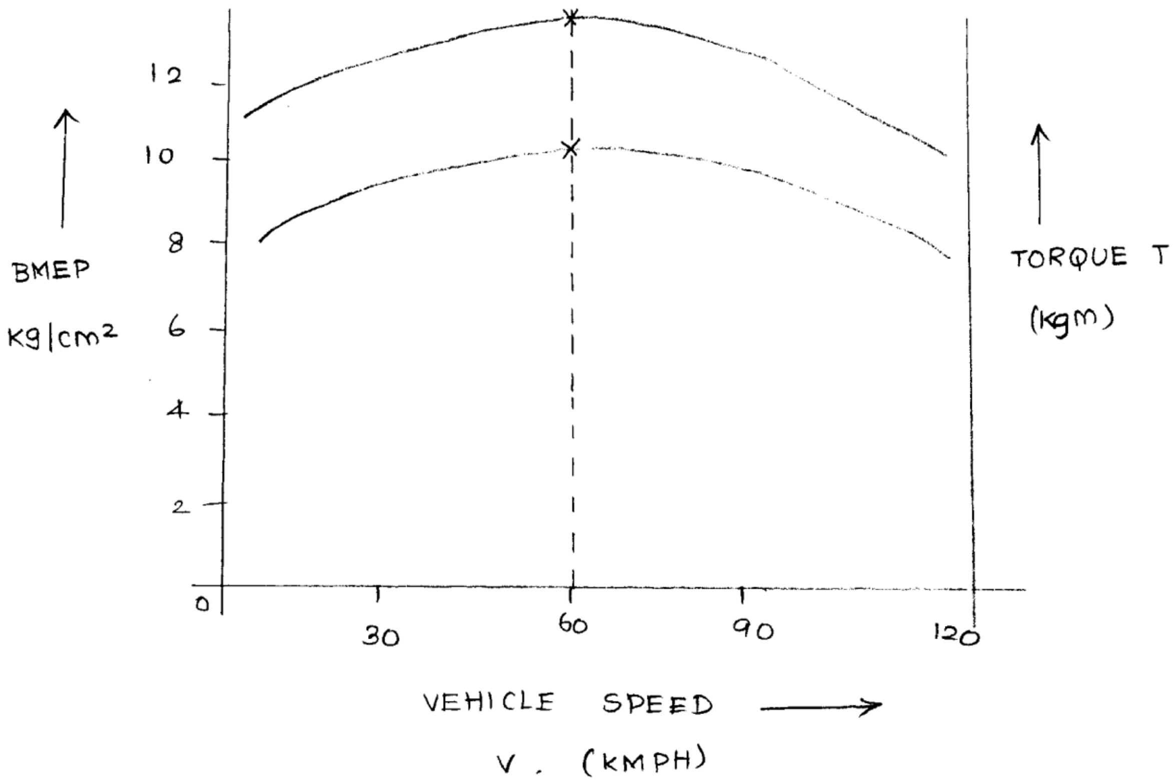
ηmech
IHP
FHP
ηmech
(%)
POWER
(HP)
VEHICLE SPEED (V) KMPH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BMEP
Kg/cm²
TORQUE T
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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

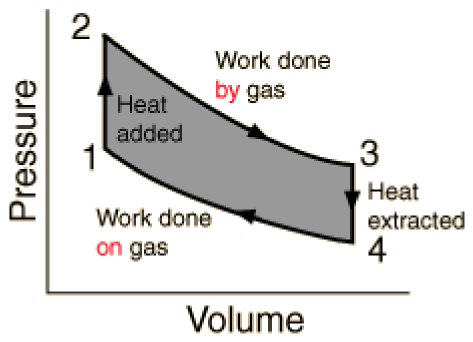
SAU1404 VEHICLE DESIGN CHARACTERISTICS

UNIT 4 ENGINE DESIGN

UNIT- IV ENGINE DESIGN

PV DIAGRAMS

Pressure-Volume (PV) diagrams are a primary visualization tool for the study of heat engines. Since the engines usually involve a gas as a working substance, the ideal gas law relates the PV diagram to the temperature so that the three essential state variables for the gas can be tracked through the engine cycle. Since work is done only when the volume of the gas changes, the diagram gives a visual interpretation of work done. Since the internal energy of an ideal gas depends upon its temperature, the PV diagram along with the temperatures calculated from the ideal gas law determine the changes in the internal energy of the gas so that the amount of heat added can be evaluated from the first law of thermodynamics. In summary, the PV diagram provides the framework for the analysis of any heat engine which uses a gas as a working substance.



For a cyclic heat engine process, the PV diagram will be closed loop. The area inside the loop is a representation of the amount of work done during a cycle. Some idea of the relative efficiency of an engine cycle can be obtained by comparing its PV diagram with that of a Carnot cycle, the most efficient kind of heat engine cycle.

FRICTIONAL MEAN EFFECTIVE PRESSURE

Experimentally, the power taken by the friction between the engine's components (P_f) is found by substracting the measured power at the crankshaft (P) from the potential power that we can obtained from the measured gas pressure acting inside the cylinder(s) (P_i).

$$P_f = P_i - P \quad (1)$$

Dividing equation (1) by the measured volumetric flow rate (Q_m), we get an equation of pressure:

$$FMEP_d = IMEP_d - BMEP_d \quad (2)$$

Where:

$FMEP_d$ = friction mean effective pressure differential

$IMEP_d$ = indicated mean effective pressure differential

$BMEP_d$ = brake mean effective pressure differential

So, the friction mean effective pressure represents the equivalent pressure «taken away» by the internal losses.

For reciprocating engines, we usually divide equation (1) by the theoretical volumetric flow rate, such that we obtain the average values instead of the differential values (see the BMEP page for more info).

Estimation

A relationship has been well established between the mean piston speed (v_{mps}) and the friction mean effective pressure to make a good estimation. This site goes further by an attempt to relate it as well to the indicated mean effective pressure:

$$FMEP_d = IMEP_d\ (0.09\ K_{sum} + 0.008\ v_{mps}) \quad (3)$$

Where K_{sum} is the sum of all values (K_x) for the engine components in use, as presented in the next table:

Engine component	K_x
camshaft	0.20
lifter	0.20
rocker	0.10
crankshaft	0.20
oil pump	0.10
water pump	0.10
fuel pump	0.05

distributor	0.05

Note: If a component is not used or has roller bearings (except for pumps, which will always require power), K_x = 0 for that component. Add them all to obtain K_{sum}.

Engine Displacement or capacity:

Engine displacement is the volume swept by all the pistons inside the cylinders of a reciprocating engine in a single movement from top dead centre (TDC) to bottom dead centre (BDC). It is commonly specified in cubic centimetres (cc or cm^3).

Engine displacement = bore X bore X stroke X 0.7854 X number of cylinders

Example: Cylinder bore diameter = 4.000"

Stroke length = 3.480"

Number of cylinders = 8

Engine displacement = bore X bore X 0.7854 X number of cylinders

Engine displacement = 4.000 X 4.000 X 3.480 X 0.7854 X 8

Engine displacement = 349.8586 cubic inches (round up to 350 cubic inches)

Stroke Length

Stroke Length = engine displacement / (bore X bore X 0.7854 X number of cylinders)

Example: Engine Displacement = 350 cubic inches

Cylinder bore diameter = 4.000"

Number of cylinders = 8

Stroke Length = engine displacement / (bore X bore X 0.7854 X number of cylinders)

Stroke Length = 349.8486 / (4.000 X 4.000 X 0.7854 X 8)

Stroke Length = 3.480"

Cylinder Bore Diameter

Cylinder bore diameter = square root of [engine displacement/(stroke X 0.7854 X number of cylinders)]

Example 1: Engine Displacement = 350 cubic inches

Stroke Length = 3.480"

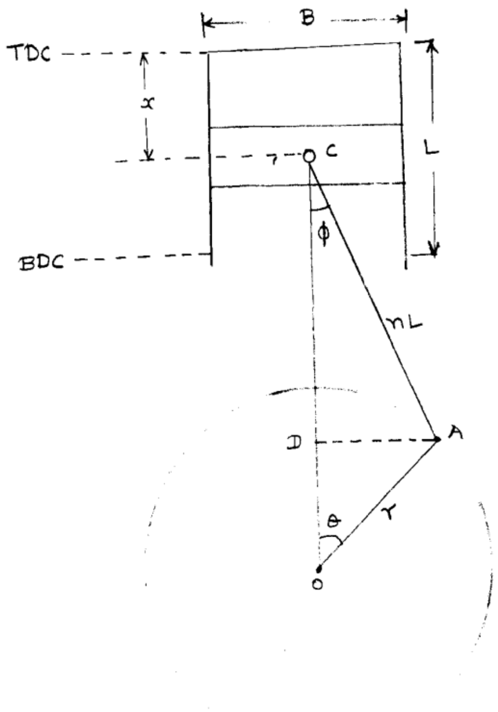
Number of cylinders = 8

Cylinder bore diameter = square root of [engine displacement/(stroke X 0.7854 X number of cylinders)]

Cylinder bore diameter = √[349.8486 / (3.480 X 0.7854 X 8)]

Cylinder bore diameter = 4.000"

PISTON VELOCITY AND ACCELERATION:



Let,

B = bore diameter

L = stroke length

n= length of connecting rod / stroke length (L)

r= crank radius = L/2

In the fig, the piston moved from TDC at a distance x

Now, $x = (nL + L/2) - CO$

From the fig. $CO = CD + DO$

$$CO = (L/2)\ cos\,\theta + nL\ cos\,\emptyset$$

$$x = (nL + \frac{L}{2}) - (\frac{L}{2}\ cos\,\theta + nL\ cos\,\emptyset) = \frac{L}{2}(1 - cos\,\theta) + nL\,(1 - cos\,\emptyset)$$

$$x = \frac{L}{2}(1 - cos\,\theta) + nL\,(1 - (1 - \frac{sin^2\theta}{8n^2}))$$

where, $\cos\emptyset = 1 - \frac{\sin^2\theta}{8n^2}$

$$x = \frac{L}{2}(1 - cos\,\theta) + \frac{L}{8n} sin^2\theta$$

Velocity of piston, $V_p = \frac{dx}{dy}$

$$V_p = \frac{d\theta}{dt}\left[\frac{L}{2} sin\,\theta + \frac{L}{8n}\; sin\,2\theta\right]$$

Now $\frac{d\theta}{dt} = \frac{2\pi N}{60}$

$$V_p = \frac{\pi N L}{6000}\left[sin\,\theta + \frac{sin\;2\theta}{4n}\right]$$

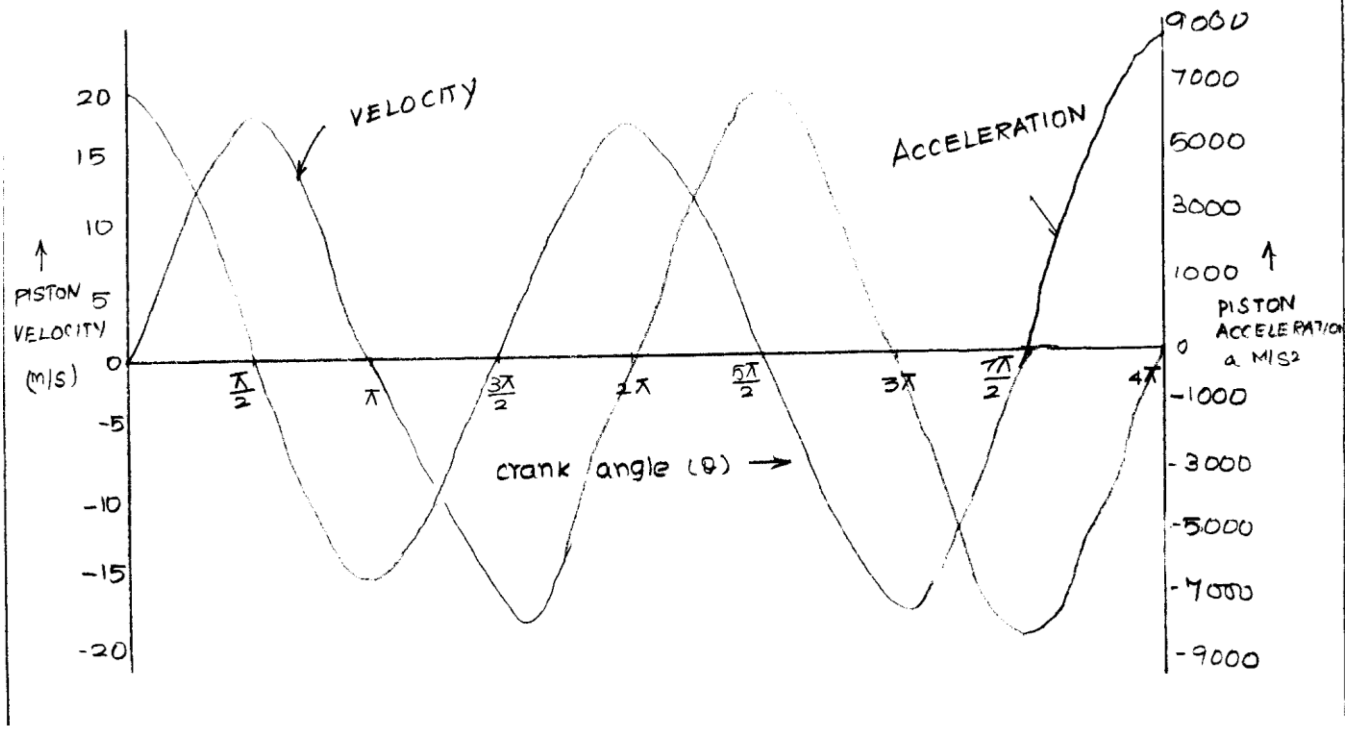
Acceleration of piston, $a_p = \frac{dV_p}{dt}$

$$a_p = \frac{\pi^2 N^2 L}{6000 \times 30}\left(cos\,\theta + \frac{cos\;2\theta}{2n}\right)$$

crank angle θ. Degree		$sin\,\theta$	$\frac{sin 2\theta}{4n}$	$sin\theta + \frac{sin 2\theta}{4n}$	V_p m/s	$cos\theta$	$\frac{cos 2\theta}{2n}$	$cos\theta + \frac{cos 2\theta}{2n}$	a_p m/s²
0	360	0.0	0.0	0.0	0.0	1.0	0.25	1.25	7853.75
15	375	0.2588	0.0625	0.3213	5.05	0.966	0.2165	1.1825	7429.6
30	390	0.50	0.1083	0.6083	9.564	0.866	0.125	0.991	6226.4
45	405	0.7071	0.125	0.8321	13.083	0.707	0.0	0.7071	4442.04
60	420	0.866	0.1083	0.9743	15.32	0.50	−0.125	0.375	2356.73
75	435	0.9659	0.0625	1.0284	16.17	0.259	−0.2165	0.0425	267.03
90	450	1.00	0.00	1.00	15.723	0.00	−0.25	−0.25	−1570.45
105	465	0.9659	−0.0625	0.9034	14.204	−0.259	−0.2165	−0.4755	−2987.6
120	480	0.866	−0.1083	0.7577	11.913	−0.50	−0.125	−0.625	−3926.9
135	495	0.7071	−0.125	0.5821	9.1522	−0.707	0.00	−0.707	−442.04
150	510	0.50	−0.1083	0.3917	6.1586	−0.866	0.125	−0.741	−4655.7
165	525	0.26	−0.0625	0.1963	3.086	−0.966	0.2165	−0.7495	−4709.1
180	540	0.00	0.00	0.00	0.00	−1.00	0.25	−0.75	−4712.3

Crank angle θ degree		$\sin\theta$	$\frac{\sin 2\theta}{4n}$	$\sin\theta + \frac{\sin 2\theta}{4n}$	V_p m/sec	$\cos\theta$	$\frac{\cos 2\theta}{2n}$	$\cos\theta + \frac{\cos 2\theta}{2n}$	a_p m/sec²
195	555	-0.259	0.0625	-0.1963	-3.086	-0.966	0.2165	-0.7495	-4709.11
210	570	-0.50	0.1083	-0.3917	-6.1586	-0.866	0.125	-0.741	-4665.7
225	585	-0.7071	0.125	-0.5821	-9.1522	-0.707	0.00	-7071	-4442.71
240	600	-0.866	0.1083	-0.7577	-11.113	-0.50	-0.125	-0.665	-3926.9
255	615	-0.966	0.0625	-0.9034	-14.204	-0.259	-0.2165	-0.4755	-2987.6
270	630	-1.00	0.00	-1.00	-15.723	0.00	-0.25	-0.25	-1570.75
285	645	-0.966	-0.0625	-1.0284	-16.17	0.259	-0.2165	-0.0425	-2987.6
300	660	-0.866	-0.1083	-0.9743	-15.32	0.50	-0.125	0.375	2356.125
315	675	-0.7071	-0.125	-0.8321	-13.083	0.707	0.00	0.707	4442.04
330	690	-0.5	-0.1083	-0.608	-9.564	0.866	0.125	0.991	6226.40
345	705	-0.26	-0.063	-0.3213	-5.05	0.966	0.2165	1.1825	7429.65
360	720	0.0	0.0	0.0	0.0	1.00	0.25	1.25	7853.8

GRAPH :



INERTIA FORCE, GAS FORCE & RESULTANT FORCE:

Inertia force:

Due to the acceleration of reciprocating mass an opposite force is created and it called as inertia force.

$$Inertia\,force = m \times a = \frac{W}{g} \times a$$

$$I.F = -\frac{W}{g} \times a_p$$

Since, inertia force is acting vertically up add a negative sign. Here a_p is acceleration of the piston in m/s^2.

g – acceleration due to gravity

W – weight of reciprocating parts in kgf.

Here, w is the weight of piston, piston pin, piston rings, circlips and one third of the connecting rod weight.

$$I.F = \frac{W}{g} \cdot \frac{\pi^2 N^2 L}{18 \times 10^4} \left(\cos\theta + \frac{\cos 2\theta}{2n} \right) \qquad \text{in kgf.}$$

Where, $W = (\,weight\ of\ reciprocating\ parts/cm^2) \times Bore\ area\ cm^2$

$$= (\,wt/\,cm^2\,) \times \frac{\pi}{4} B2$$

W = ------------- kgf.

From the following table we can find the weight of the inertia parts.

Cylinder Bore B mm	weight of reciprocating parts per sq.cm of bore area	
	Aluminium alloy pistons in kgf	C.I pistons in kgf
60	0.011	0.016
70	0.012	0.018
80	0.014	0.02
90	0.015	0.022
100	0.017	0.024
110	0.018	0.026

120	0.019	0.028
130	0.02	0.03

Gas forces :

Due to the pressure of the gas in the cylinder and combustion chamber a certain force is exerted on the piston. The force due to gas pressure acting on the piston crown is known as gas force.

The gas pressure is acting on te piston crown and combustion chamber and the atmospheric pressure is acting on the bottom side of the piston. The net gas force is equal to the bottom difference between the pressure on the piston crown and the atmospheric pressure at bottom side of the piston.

$$Cylinder\ pressure\ =\ gas\ pressure - \ back\ pressure$$

$$Back\ pressure\ =\ 1.03\ kgf/cm^2$$

Gas force = cylinder pressure x bore area

If we know, gas pressure from zero to 720^o crank angle we can easily determine gas force. Since piston starts from TDC and that time $\theta = 0$.

0^o – 180^o – power stroke

180^0 – 360^o – exhaust stroke

360^o – 540^o – suction stroke

540^o – 720^o – compression stroke.

1. During power stroke, pressure changes frm max. To min.
2. During exhaust and suction strokes the pressure is almost constant.
3. During comp. Stroke the pressure changes from min. To max.

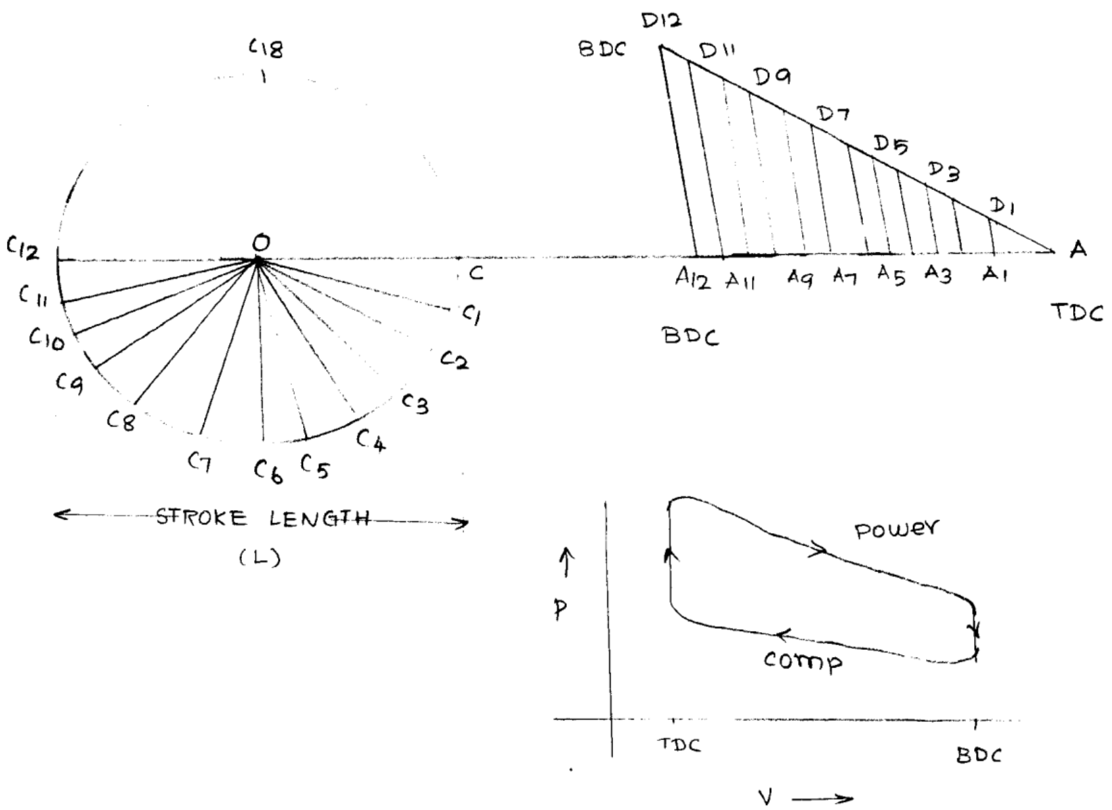
Gas pressure may be assumed as,

For exhaust stroke – 1.12 to 1.16 kgf/cm^2

For suction stroke – 0.95 to 0.98 kgf/cm^2

For power and compression stroke gas pressure can be determined from the PV diagram.

The absolute gas pressure existing in the cylinder can be obtained from the PV diagram. For this it is necessary to find the distance through which the piston has travelled from the top end of the stroke (for power stroke). Where the crank has turned through different angles from 0^o – 180^o. This can be done by constructing the diagram as described below.



CA = length of connecting rod = nL

AD = base length of PV diagram.

AA_{12} = L

1. Draw line CA as shown in figure. From C draw a circle with stroke length as a diameter at point o.
2. Divide this circle into 24 equal parts, i.e. each segment is 15^o with C_1, C_2, C_3,...as centres an CA as a radius cut arcs at A_1, A_2, A_3,...... A_{12}.
3. Draw a line AD conveniently at any angle from A. Such that AB is the base length of the PV diagram. Now join A_{12} with D.
4. Draw lines parallel to $A_{12}D$ through A_{11}, A_{10},....

5. Now we have points D_{11},D10,...on AD.
6. Transforms these point in PV diagram.
7. Corresponding to these points gas pressure can be determined from PV – diagram in both power and compression stroke.

Resultant force:

The algebraic sum of the gas force and inertia force gives the resultant force.

R.F = G.F + I.F

Calculation:

I.F = - $\frac{W}{g} \cdot \frac{\pi^2 N^2 L}{18\times10^4}\left(\cos\theta + \frac{\cos 2\theta}{2n}\right)$

W= (wt/ cm$^{2)}$ × $\frac{\pi}{4}B^2$

= 0.0139 x $\frac{\pi}{4}$ $(7.869)^2$

W= 0.68 kgf.

I.F = $-\frac{0.68}{9.81}\frac{\pi^2\times 3816^2\times7.689}{18\times10^4}\times 1.25$

I.F = - 544.4 kgf.

G.F = cylinder pressure x bore area

= (gas pr – back pr) x $\frac{\pi}{4}B^2$

= (74 – 1.03)x $\frac{\pi}{4}$ $(7.869)^2$

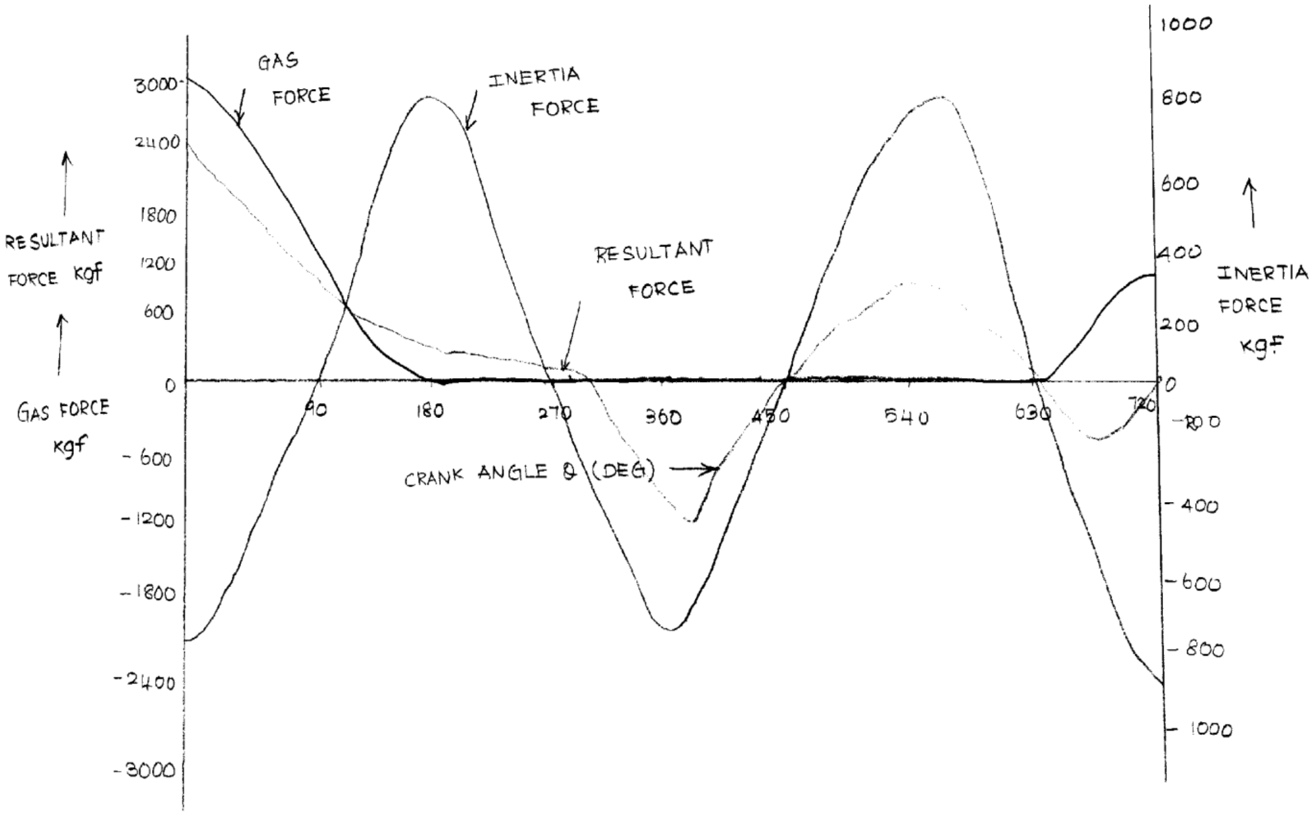
= 3548.733 kgf.

STROKE	Crank angle θ Deg	Gas Pr. kg/cm²	Back Pr. kg/cm²	Gas Force kgf	Inertia Force kgf	Resultant Force kgf	Turning Moment kgf cm
	0	74.0	1.03	3548.733	-544.394	3004.34	0.0
	15	67.5	1.03	3232.62	-514.997	2717.623	3435.5
	30	42.5	1.03	2016.801	-431.596	1585.205	3793.96
	45	24.0	1.03	1117.095	-307.953	809.142	2649.05
POWER STROKE	60	15.0	1.03	679.4	-163.32	516.08	1978.33
	75	10.0	1.03	436.236	-18.509	417.727	1690.224
	90	7.5	1.03	314.654	108.88	423.534	1666.4
	105	6.0	1.03	241.705	207.088	448.793	1595.2
	120	5.0	1.03	191.0721	272.197	463.2691	1381.1
	135	4.75	1.03	180.914	307.93	488.844	1119.6
	150	4.50	1.03	168.76	322.72	491.48	2980.15
	165	4.25	1.03	156.598	326.42	483.02	373.06
	180	4.0	1.03	144.44	326.64	471.08	0.00

STROKE	Crank angle θ (Deg)	Gas Pr. kg/cm²	Back Pr. kg/cm²	Gas Force kgf	Inertia Force kgf	Resultant Force kgf	Turning Moment kg-cm
	195	1.14	1.03	5.35	326.4215	331.7711	-256.241
	210	1.14	1.03	5.35	322.72	328.07	-479.8
	225	1.14	1.03	5.35	307.955	313.305	-717.554
	240	1.14	1.03	5.35	272.201	277.551	-827.43
	255	1.14	1.03	5.35	207.089	212.44	-755.1
Exhaust Stroke	270	1.14	1.03	5.35	108.88	114.23	-449.44
	285	1.14	1.03	5.35	-18.51	-13.1604	53.25
	300	1.14	1.03	5.35	-163.32	-157.97	602.11
	315	1.14	1.03	5.35	-307.91	-302.56	990.55
	330	1.14	1.03	5.35	-431.6	-426.25	1020.2
	345	1.14	1.03	5.35	-515.001	-509.65	644.28
	360	1.14	1.03	5.35	-544.4	-539.494	0.00

STROKE	Crank angle θ (deg)	Gas Pr. kg/cm²	Back Pr. kg/cm²	Gas Force kgf	Resultant Force kgf	Resultant Force kgf	Turning Moment kgf-m.
SUCTION STROKE	375	0·96	1·03	-3·4043	-514·997	-518·4013	-655·34
	390	0·96	1·03	-3·4043	-431·596	-435·0	-1041·1
	405	0·96	1·03	-3·4043	-307·953	-311·36	-1019·4
	420	0·96	1·03	-3·4043	-163·32	-166·72	-639·1
	435	0·96	1·03	-3·4043	-18·509	-21·9133	-88·7
	450	0·96	1·03	-3·4043	108·88	105·4757	414·99
	465	0·96	1·03	-3·4043	207·088	203·7	724·04
	680	0·96	1·03	-3·4043	272·197	268·8	801·34
	495	0·96	1·03	-3·4043	307·93	304·526	697·45
	510	0·96	1·03	-3·4043	322·72	319·316	492·112
	525	0·96	1·03	-3·4043	326·42	323·016	249·48
	540	0·96	1·03	-3·4043	326·64	323·236	0·00

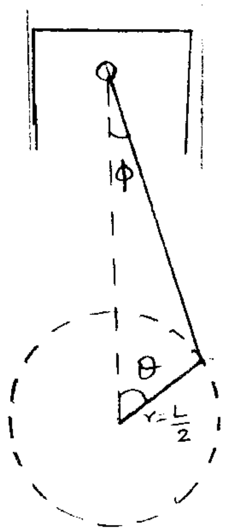
STROKE	Crank angle θ (deg)	Gas Pr. kg/cm²	Back Pr kg/cm²	Gas Force kgf	Inertia Force kgf	Resultant Force kgf	Turning Moment kgf-cm
COMPRESSION STROKE	555	1·20	1·03	8·268	326·42	334·7	-258·5
	570	1·25	1·03	10·7	322·72	333·42	-513·85
	585	1·30	1·03	13·131	307·93	321·061	-735·32
	600	1·35	1·03	15·563	272·20	273·763	-816·13
	615	1·40	1·03	17·994	207·09	225·083	-800·04
	630	1·75	1·03	35·016	108·8996	143·896	-566·17
	645	2·50	1·03	71·49	18·51	89·99	-364·2
	660	3·0	1·03	82·676	-163·32	-80·644	309·14
	675	5·5	1·03	217·39	-307·91	-90·522	296·36
	690	9·8	1·03	426·51	-431·6	-5·09	12·18
	705	14·5	1·03	655·08	-515·0	140·1	-177·1
	720	17·0	1·03	776·7	-544·4	232·3	0·00



SIDE THRUST:

Side thrust is defined as the force or thrust acting on the cylinder walls due to angularity of the connecting rod.

Since side thrust is maximum only in the power stroke, it's enough to find side thrust only for power stroke.i.e between 0^o to 180^o.



Side thrust = P_s= P.tan∅

P = Resultant force.

On resolving forces,

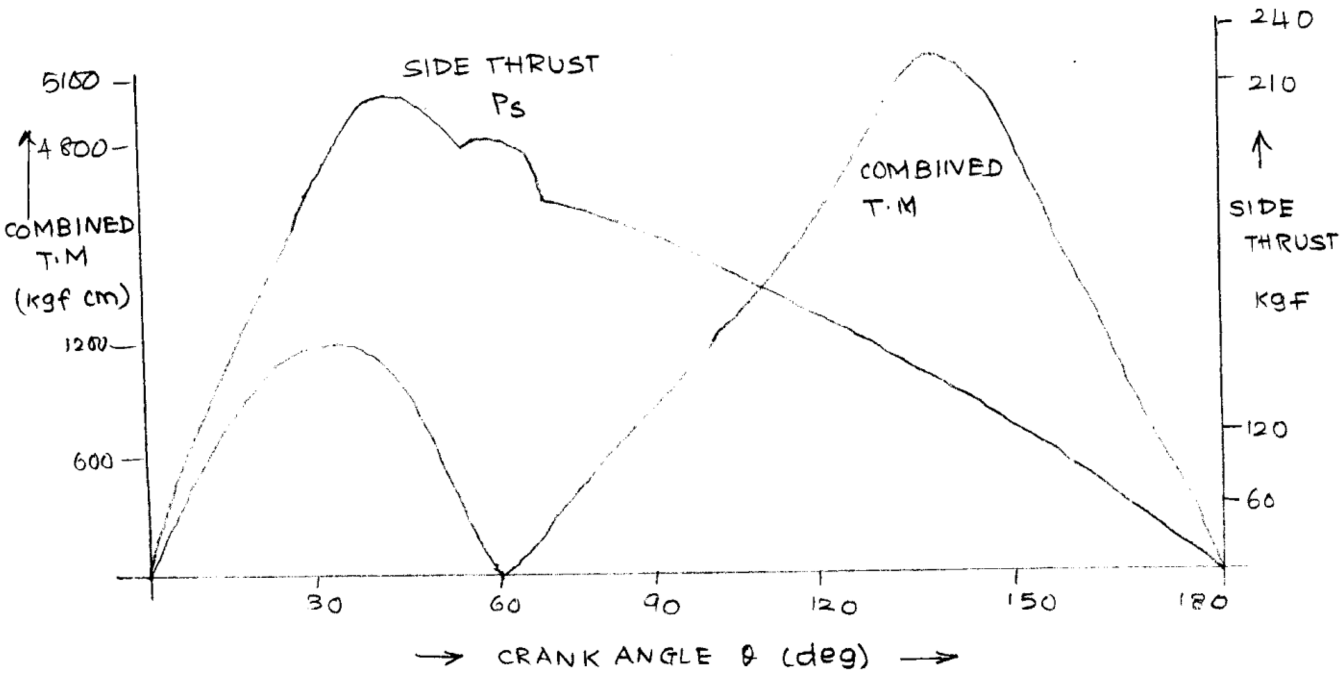
$$nL\ sin\emptyset\ =\ \frac{L}{2}\ sin\theta$$

$$sin\emptyset\ =\ \frac{sin\theta}{2n}$$

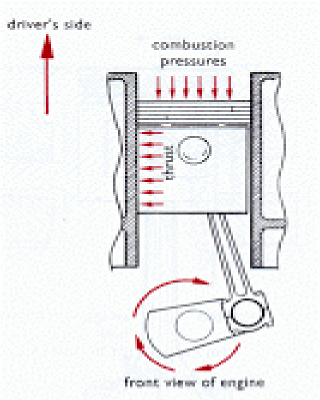
$\emptyset = \sin^{-1}(\frac{\sin\,\theta}{2n})$

crank angle θ (deg)	$\frac{\sin\theta}{2n}$	$\phi = \sin^{-1}\left(\frac{\sin\theta}{2n}\right)$	$\tan\phi$	side thrust $P_s = P\tan\phi$ kgf
0	0	0	0	0
15	0.0625	3.71	0.065	176.65
30	0.125	7.2	0.126	199.72
45	0.177	10.18	0.18	145.33
60	0.2165	12.5	0.222	114.45
75	0.241	13.97	0.25	103.95
90	0.25	14.5	0.26	109.36
105	0.2415	13.974	0.25	111.68
120	0.2165	12.504	0.22	102.74
135	0.18	10.18	0.18	87.8
150	0.125	7.181	0.126	61.9
165	0.0625	3.71	0.065	31.32
180	0.00	0.00	0.00	0.00

GRAPH:



Piston thrusts



During the power strokes, combustion pressures force the piston downwards. However, the piston does not bear evenly against the walls of the cylinder, but is thrust against the sides of the cylinder. This is caused by the angularity of the connecting rod (Figure 1). The combustion pressures force the piston downwards, and the connecting rod offers resistance, but it does this at an angle. The result is a side thrust of the piston against the cylinder wall, as shown. The piston also has a side thrust during the compression stroke, but this is on the opposite side of the cylinder. Also, this is a lesser thrust because the downward force from compression is much less than the downward force of combustion. The thrusts are sometimes referred to as the *major* and *minor* thrusts. Because the thrust during the power stroke (major thrust) is most important, this side of the engine is often referred to as the *thrust side* of the engine. It is necessary to know about the thrust side of an engine because the pistons in most engines have to be installed in a particular way. Pistons are often provided with a mark to show how they should be fitted in relation to the front of the engine.

- *Thrusts occur on both the upstrokes and the down-strokes, but the major thrust is during the power* stroke.

Major Thrust Side:

As a piston is pushed down the cylinder on its power stroke, the piston will meet resistance as it tries to turn the crankshaft.

The greater the load on an engine, the greater this resistance will be. The resistance will cause the piston to be pushed to one side with considerable force, as it is also pushed down the bore by the hot expanding air/fuel gasses above it.

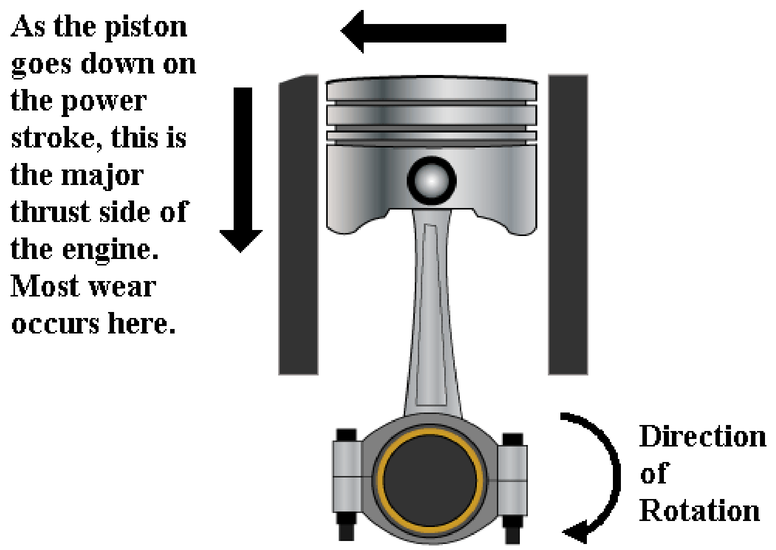
This side force is known as thrust, and it causes one side of the cylinder walls to wear at a faster rate than the rest of the cylinder wall. The side of the engine where this thrust acts is known as the thrust side of the engine.

Because there is extra wear on this side of the engine, the connecting rod will often have an oil squirt hole in it, so that oil under pressure from the crankshaft can be squirted on to the cylinder wall on the thrust side, in order to minimise wear and remove excess heat.

That is why it is so important to note any mark on the crown of the piston. Such marks will generally point toward the front of the engine, or one particular side of the engine. For example, in the case of the Lean 5 HP engine, the mark on the crown of the piston points toward the camshaft side of the engine.

It is also important to note that as well as a major thrust side, each engine also has a minor thrust side, which is opposite to the major thrust side. As the piston moves up on its compression stroke, and it meets the resistance of the fresh air/fuel mixture, this resistance will tend to force the piston off too one side as it moves upward on its compression stroke.

See also the glossary explanation* for minor thrust side *for more information.



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UNIT 5 GEAR RATIOS

UNIT-5 GEAR RATIOS

1.1 HISTORY OF GEARS

Indian history as per our mythological stories is more than 12,000 years old. Since then people living here have been striving to improve the living conditions. We also know that earlier people were living in the caves and the doors of the caves were made of granite. How were these heavy doors opened and closed? They were opened and closed by none other than a system with gear mechanism, wheel, lever and rope drives. However, the documented evidence has been lost due to destruction by the invaders and improper storing of palm leaf literature. The guru Kula method of teaching and passing of the information from mouth to ear procedure and keeping some of the advances as closely guarded secret have resulted in poor dissemination of the knowledge and documentation. But, the knowledge of gears has gone from India to east through some of the globe trotters from China as back as 2600 years BC. They have used the gears then ingeniously in chariots for measuring the speed and other mechanisms. Primitive gears shown in Fig. 1 were first used in door drive mechanism in temples and caves, and water lifting mechanisms 2600 B.C. in India and elsewhere.

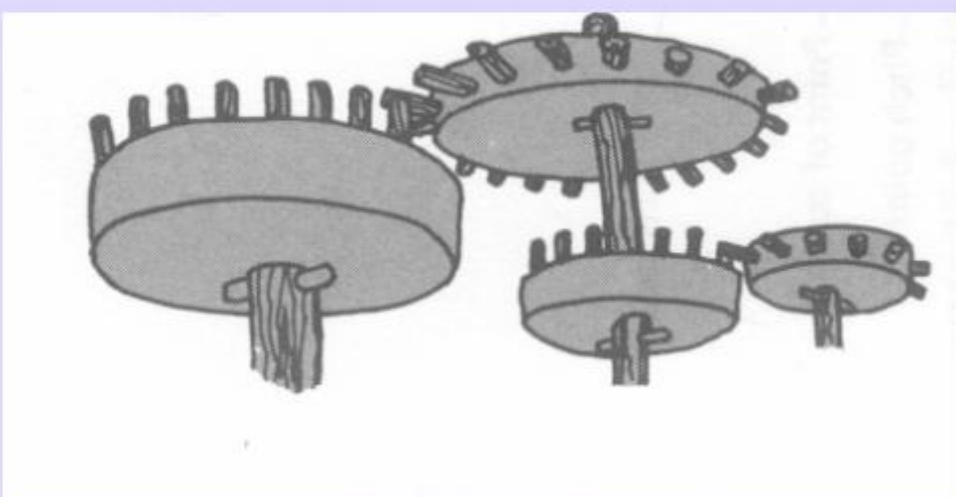


Fig.1.1 Primitive gears made of wood

1.11 Introduction

We have discussed earlier that the slipping of a belt or rope is a common phenomenon, in the transmission of motion or power between two shafts. The effect of slipping is to reduce the velocity ratio of the system. In precision machines, in which a definite velocity ratio is of importance (as in watch mechanism), the only positive drive is by ***gears*** or ***toothed wheels*.** A gear drive is also provided, when the distance between the driver and the follower is very small.

1.2 DEFINITION OF GEARS

Gears are toothed members which transmit power / motion between two shafts by meshing without any slip. Hence, gear drives are also called positive drives. In any pair of gears, the smaller one is called pinion and the larger one is called gear immaterial of which is driving the other. When pinion is the driver, it results in step down drive in which the output speed decreases and the torque increases. On the other hand, when the gear is the driver, it results in step up drive in which the output speed increases and the torque decreases.

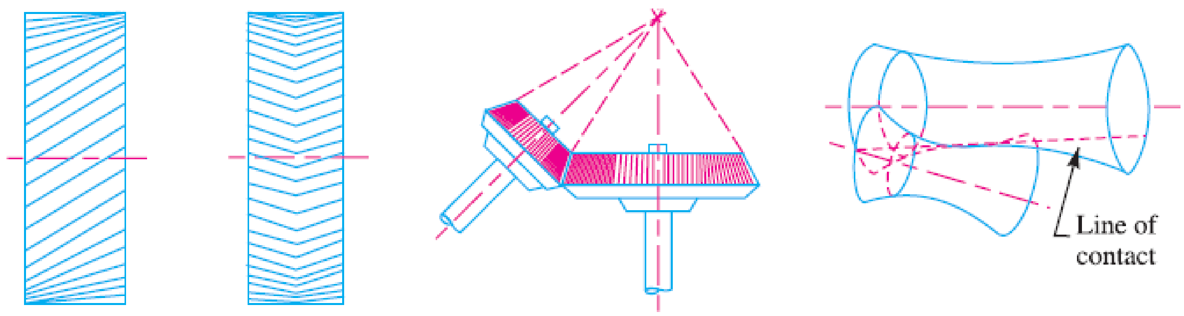
Classification of Gears

The gears or toothed wheels may be classified as follows:

1. According to the position of axes of the shafts. The axes of the two shafts between which the motion is to be transmitted, may be

(a) Parallel, (b) Intersecting, and (c) Non-intersecting and non-parallel.

The two parallel and co-planar shafts connected by the gears. These gears are called spur gears and the arrangement is known as spur gearing. These gears have teeth parallel to the axis of the wheel .Another name given to the spur gearing is helical gearing, in which the teeth are inclined to the axis. The single and double helical gears connecting parallel shafts are shown in Fig. respectively. The object of the double helical gear is to balance out the end thrusts that are induced in single helical gears when transmitting load. The double helical gears are known as herringbone gears. A pair of spur gears is kinematic ally equivalent to a pair of cylindrical discs, keyed to a parallel shaft having line contact. The two non-parallel or intersecting, but coplanar shafts connected by gears is shown in Fig. These gears are called bevel gears and the arrangement is known as bevel gearing. The bevel gears, like spur gears may also have their teeth inclined to the face of the bevel, in which case they are known as helical bevel gears.



(a) Single helical gear. (b) Double helical gear. (c) Bevel gear. (d) Spiral gear.

The two non-intersecting and non-parallel *i.e.* non-coplanar shafts connected by gears .These gears are called ***skew bevel gears*** or ***spiral gears*** and the arrangement is known as ***skew bevel gearing*** or ***spiral gearing*.** This type of gearing also has a line contact, the rotation of which about the axes generates the two pitch surfaces known as ***hyperboloids***.

2. According to the peripheral velocity of the gears. The gears, according to the peripheral velocity of the gears, may be classified as:

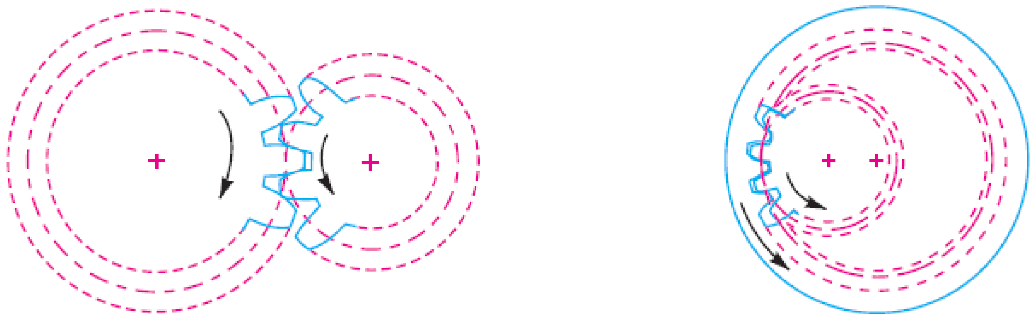
(a) Low velocity, (b) Medium velocity, and (c) High velocity

The gears having velocity less than 3 m/s are termed as low velocity gears and gears having velocity between 3 and 15 m / s are known as medium velocity gears. If the velocity of gears is more than 15 m / s, then these are called high speed gears.

3. According to the type of gearing. The gears, according to the type of gearing, may be

Classified as:

(a) External gearing, (b) internal gearing, and (c) Rack and pinion.



(*a*) External gearing. (*b*) Internal gearing.

In ***external gearing,*** the gears of the two shafts mesh externally with each other. The larger of these two wheels is called ***spur wheel*** or ***gear*** and the smaller wheel is called ***pinion.*** In an external gearing, the motion of the two wheels is always unlike, as shown in Fig. In ***internal gearing,*** the gears of the two shafts mesh internally with each other as shown in Fig. The larger of these two wheels is called ***annular wheel*** and the smaller wheel is called ***pinion***. In an internal gearing, the motion of the wheels is always like as shown in Fig. Sometimes, the gear of a shaft meshes externally and internally with the gears in a *straight line, as shown in Fig. Such a type of gear is called ***rack*** and ***pinion.*** The straight line gear is called ***rack*** and the circular wheel is called ***pinion.*** A little consideration will show that with the help of a rack and pinion, we can convert linear motion into rotary motion and *vice-versa* as shown in Fig.

4. *According to the position of teeth on the gear surface*. The teeth on the gear surface may be

(*a*) Straight, **(*b*)** Inclined, and **(*c*)** Curved.

We have discussed earlier that the spur gears have straight teeth whereas helical gears have their teeth inclined to the wheel rim. In case of spiral gears, the teeth are curved over the rim surface.

Terms used in Gears

The following terms, which will be mostly used in this chapter, should be clearly understood at this stage. These terms are illustrated in Fig. 28.6.

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

2. Pitch circle diameter. It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also called as pitch diameter.

3. Pitch point. It is a common point of contact between two pitch circles.

4. Pitch surface. It is the surface of the rolling discs which the meshing gears have replaced at the pitch circle.

5. Pressure angle or angle of obliquity. It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. It is usually denoted by φ. The standard pressure angles are 1 14 /2° and 20°.

6. Addendum. It is the radial distance of a tooth from the pitch circle to the top of the tooth.

7. Dedendum. It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.

8. Addendum circle. It is the circle drawn through the top of the teeth and is concentric with the pitch circle.

9. Dedendum circle. It is the circle drawn through the bottom of the teeth. It is also called root circle.

10. Circular pitch. It is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth. It is usually denoted by pc. Mathematically,

$$\text{Circular pitch, } p_c = \pi D/T$$

Where D = Diameter of the pitch circle, and

T = Number of teeth on the wheel.

A little consideration will show that the two gears will mesh together correctly, if the two wheels have the same circular pitch.

12. Module. It is the ratio of the pitch circle diameter in millimeters to the number of teeth. It is usually denoted by m. mathematically,

$$\text{Module, } m = D / T$$

13. Clearance. It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top of the meshing gear is known as clearance circle.

14. Total depth. It is the radial distance between the addendum and the dedendum circle of a gear. It is equal to the sum of the addendum and dedendum.

15. Working depth. It is radial distance from the addendum circle to the clearance circle. It is equal to the sum of the addendum of the two meshing gears.

16. Tooth thickness. It is the width of the tooth measured along the pitch circle.

17. Tooth space. It is the width of space between the two adjacent teeth measured along the pitch circle.

18. Backlash. It is the difference between the tooth space and the tooth thickness, as measured on the pitch circle.

NECESSITY OF GEAR BOX IN AN AUTOMOBILE

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

FUNCTION OF A GEAR BOX

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

TYPES OF GEAR BOX

PROGRESSIVE TYPE GEAR BOX

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

EPICYCLIC (OR) PLANETARY TYPE GEAR BOX

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

SELECTIVE TYPE GEAR BOX

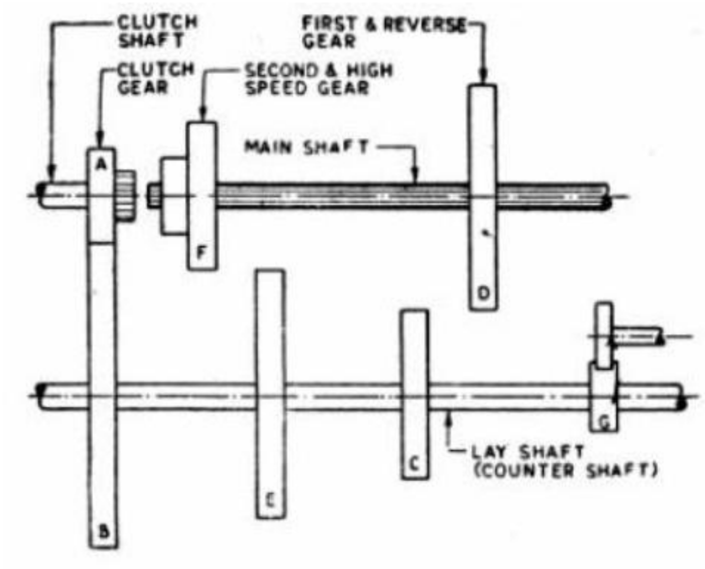
It is the transmission in which any speed may be selected from the neutral position. In this type of transmission neutral position has to be obtained before selecting any forward or reverse gear. Some selective type gear boxes are,

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

SLIDING MESH GEAR BOX

It is the simplest and oldest type of gear box.

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



FIRST GEAR

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

SECOND GEAR

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

TOP GEAR

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

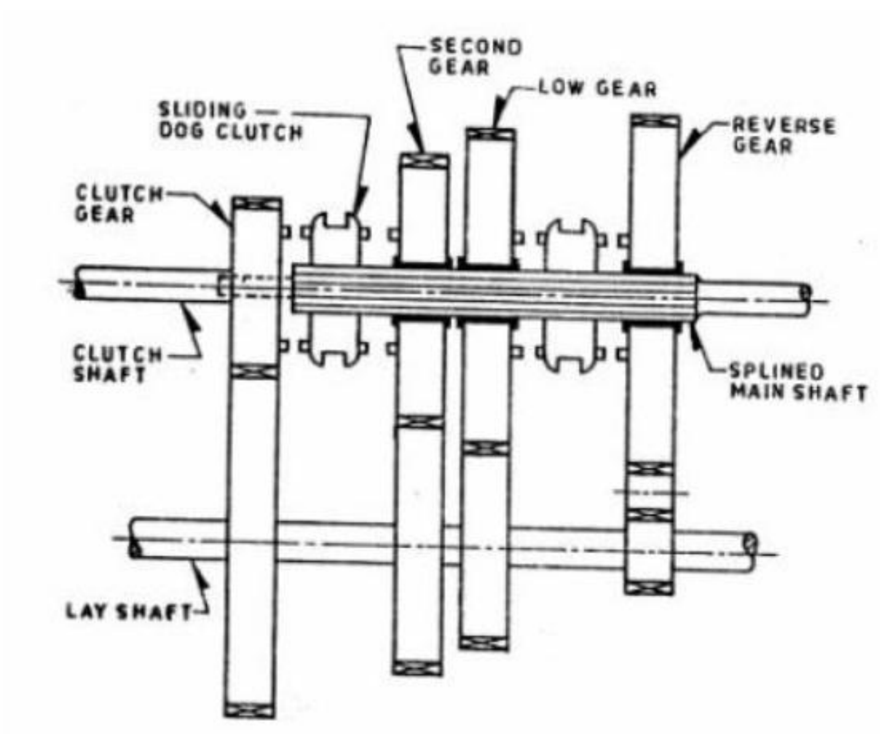
REVERSE GEAR

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

NEUTRAL GEAR

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

CONSTANT MESH GEARBOX



In this type of gearbox, all the gears of the main shaft are in constant mesh with corresponding gears of the countershaft. The gears on the main shaft which are bushed are free to rotate. The dog clutches are provided on main shaft. The gears on the lay shaft are, however, fixed. When the left Dog clutch is slid to the left by means of the selector mechanism, its teeth are engaged with those on the clutch gear and we get the direct gear. The same dog clutch, however, when slid to right makes contact with the second gear and second gear is obtained. Similarly movement of the right dog clutch to the left results in low gear and towards right in reverse gear. Usually the helical gears are used in constant mesh gearbox for smooth and noiseless operation.

SYNCHROMESH GEARBOX

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

SYNCHRONIZERS

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

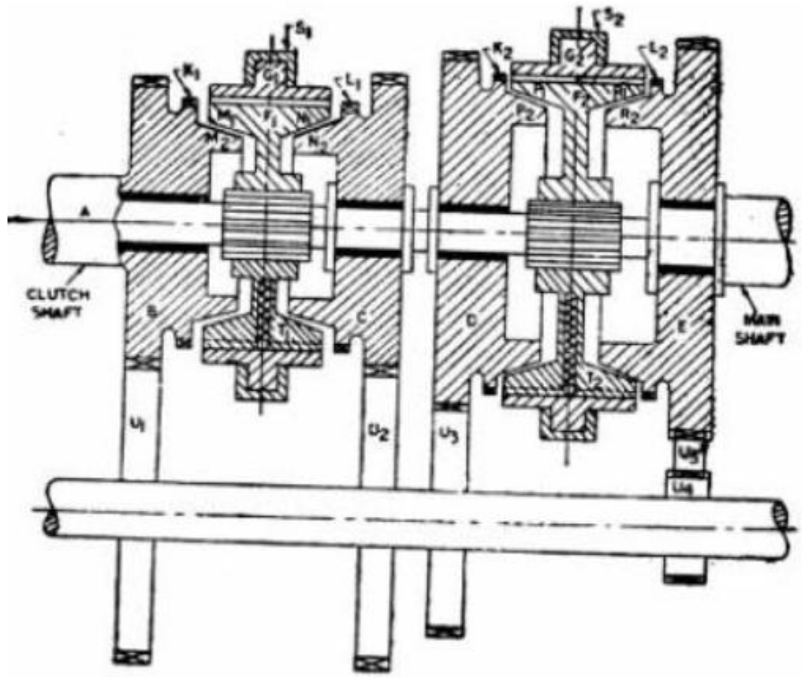


Figure shows the construction and working of a synchromesh gearbox. In most of the cars, however, the synchromesh devices are not fitted to all the gears as is shown in this figure. They are fitted only on the high gears and on the low and reverse gears ordinary dog clutches are only provided. This is done to reduce the cost. In figure A is the engine is the engine shaft, Gears B, C, D, E are free on the main shaf t and are always in mesh with corresponding gears on the lay shaft. Thus all the gears on main shaft as well as on lay shaft continue to rotate so long as shaft A is rotating. Members F1 and F2 are free to slide on splines on the main shaft. G1 and G2 are ring shaped members having internal teeth fit onto the external teeth members F1 and F2 respectively. K1 and K2 are dogteeth on B and D respectively and these also fit onto the teeth of G1 and G2. S1 and S2 are the forks. T1 and T2 are the balls supported by spring. These tend to prevent the sliding of members G1 (G2) on F1 (F2). However when the force applied on G1 (G2) slides over F1 (F2). These are usually six of these balls symmetrically placed circumferentially in one Synchromesh device. M1, M2, N1, N2, P1, P2, R1, R2 are the frictional surfaces.

To understand the working of this gearbox, consider figure which shows in steps how the gears are engaged. For direct gear, member G1 and hence member F1 (through spring- loaded balls) is slid towards left till cones M1 and M2 rub and friction makes their speed equal. Further pushing the member G1 to left causes it to overdrive the balls and get engaged with dogs K1. Now the drive to the main shaft is direct from B via F1 and the spines. However, if member G1 is pushed too quickly so that there is not sufficient time for synchronization of speeds, a clash may result. Likewise defect will arise in case springs supporting the balls T1 have become weak. Similarly for second gear the members F1 and G1 are slid to the right so that finally the internal teeth on G1 are engaged with L1. Then the drive to main shaft will be from B via U1, U2, C, F1 and splines. For first gear, G2 and F2 are moved towards left. The drive will be from B via U1, U2, D, F2 and splines to the main shaft. For reverse gear, G2 and F2 are slid towards right. In this case the drive will be from B via U1, U2, U5, E, F2 and splines to the main shaft. A synchro's purpose is to allow the collar and the gear to make frictional contact before the dog teeth make contact. This lets the collar and the gear synchronizes their speeds before the teeth need to engage, like this: The cone on the blue gear fits into the cone-shaped area in the collar, and friction between the cone and the collar synchronize the collar and the gear. The outer portion of the collar then slides so that the dog teeth can engage the gear.

DETERMINATION OF GEAR RATIOS

From one point of view, the ideal type of transmission is the so called indefinitely variable gear, in which torque ratio can be varied continuously within wide limits, because it permits of operating the engine at all times under optimum conditions with respect to both fuel consumption and wear and tear. However in an ordinary transmission only a small number of gear ratios can be provided to cut down expenses and weight. The most desirable number of gear changes depends in part on the use to which the transmission is to be put, and each field of application must be considered separately. The most desirable number of gear ratios depends on the difference between highest and lowest gear ratios. The larger the ratio between corresponding gear ratios, the more difficult it is to make the change from one gear to another, it must be done by either shifting gears into mesh laterally or securing a gear to its shaft by means of a jaw clutch. A ratio of 2:1 is about the limit and is frequently used in trucks although a ratio of 1.5:1 is considered better from a standpoint of ease shifting. Formerly a ratio of 1:8 was used in passenger cars and high speed was a direct drive with a ratio of 1:1, the intermediate speed a reduction gear with a1:8 ratio and the low speed a reduction gear with a ratio of (1.8 x 1.8): 1 Or 3.24:1. When gear ratios are arranged in such an order they form a geometrical series which offers certain advantages from the standpoint of operation. In most automotive transmission the ratios of the different gears come fairly close to forming a geometric series. A slight deviation from the series is made at high speed end. There are certain limitations on the number of gear tee th which can be provided, and it is therefore not always possible to obtain an exact geometrical series of ratios, even if that should be desired.

Traction and tractive effort:

The force available at the contact between the rear wheel tyres and road is known as tractive effort. The ability of the rear wheels to transmit this effort without slipping is known as traction. Hence usable tractive effort will never exceed traction.

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