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INSTITUTE OF SCIENCE AND TECHNOLOGY

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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

SAU1305 -AUTOMOTIVE CHASSIS DESIGN

UNIT 1 - INTRODUCTION TO ERGONOMICS

UNIT 1

1. Introduction:

The word "**Ergonomics**" originated from two Greek words "**Ergon**" means "**work**" and"**Nomos**" means"**natural laws**" International Ergonomics Association (IEA) defined Ergonomics (or human factors) as the scientific discipline concerned with the understanding of interactions among humans and other elements of a system, and the profession that applies theory, principles, data and methods to design in order to optimize human well-being and overall system performance. An ergonomist is an individual whose knowledge and skills concern the analysis of human-system interaction and the design of the system in order to optimize human well-being and overall system performance (IEA, 2000).

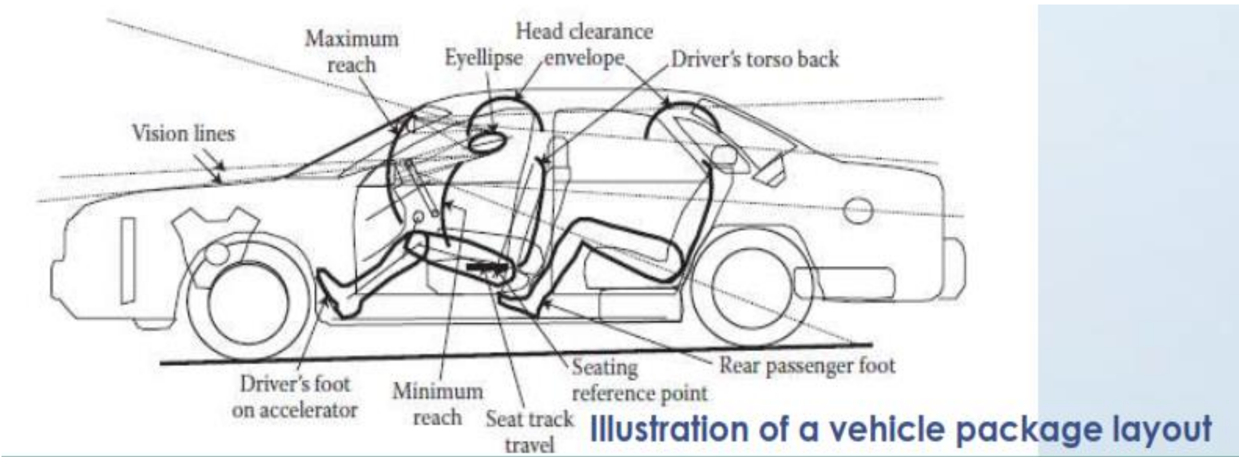


Fig: 1 vehicle package layout

Ergonomics:

Ergonomics is concerned with the health of the people and the productivity of the system. It is to get proper fit between people and their technological tools and environments. It takes account of the user's capabilities and limitations in seeking to ensure that tasks, equipment, information and the environment suit each user. Simply expressed we can say that Ergonomics is fitting the task to the person rather than fitting the person to the task.

Domains of Specialization:

According to international Ergonomics Association (IEA) ergonomics can be broadly classified into:

• Physical Ergonomics

• Cognitive Ergonomics

• Organizational Ergonomics

Physical Ergonomics:

It is concerned with human anatomical, anthropometric, physiological and biomechanical characteristics as they related to physical activity. Relevant topics may include working postures, material handling, repetitive movements, work related musculoskeletal disorders, workplace layout, health and safety.

Cognitive Ergonomics:

A proper fit of a product to a user does not end with physical interfaces. Cognitive / perceptual ergonomics is concerned with mental processes, such as perception, memory, reasoning, and motor response, as they affect interactions among humans and other elements of a system. Relevant topics include mental workload, decision-making, skilled performance, human-computer interaction, human reliability, work stress and training as these may relate to human-system and Human computer interaction design.

Organizational Ergonomics:

It is concerned with the optimization of socio technical systems, including their organizational structures, policies, and processes. Relevant topics include communication, crew resource management, and work design, design of working times, teamwork, community ergonomics, cooperative work, new work programs, virtual organizations, telework, and quality management.

Applications and Benefits:

Applications:

Ergonomics continues to be successfully applied in the fields of workplace design, occupational health, safety, product design, aerospace engineering, mechanical engineering, health care, IT sectors, transportation, training, nuclear power plant, virtual environments, industrial design and so on.

Benefits:

Application of ergonomic principles in various fields provides to better man-machine interaction, healthy and comfortable working environments, enhancement of human performance and efficiency and thus ultimately leads to overall improvement of system's (man-machine-environment) productivity with reduction of error and accidents.

Key benefits of application of ergonomics are listed below:

• Human fatigue and error can be reduced.

• increase productivity and safety

• Increase work quality

• Decrease risk of accidents

• Improve people attitude

• More user satisfaction

• Less absenteeism

• Reduced lost time, etc.

Aspects of Ergonomics:

Study of compatibility issues for proper man-machine interface is very important in ergonomics. Here, focus is generally made on user's requirement, user's characteristics and user's capabilities/limitations for user friendly design. Human compatibility with machine/instrument/work elements are discussed in terms of anthropometric, biomechanical, physiological and cognitive/ psychological aspects.

Anthropometry:

Anthropometry is the subject which deals with the measurements of the human external body dimensions in static and dynamic conditions. Anthropometric data is used for product and workplace design.

Anthropometry is of two types:

• Static Anthropometry

• Dynamic Anthropometry

Static Anthropometry:

External human body dimensional measurement taken when a man is placed in a rigid static position i.e. standing, sitting, or other adopted postures.

Dynamic Anthropometry:

The dimensional measurement of human body with various movements taken into consideration in different adopted postures which the work context demands are termed dynamic anthropometry.

Introduction to Automotive Ergonomics:

Automotive ergonomics focuses on the role of human factors in the design and use of automobiles. This includes analysis of accommodation of driver and/or passengers; their comfort; vision inside and outside vehicle; control and display design; pedal behavior information processing and cognitive load during driving etc. In the present module attempt will be made to discuss various physical aspect of occupant packaging for providing comfortable driving posture, clearance dimensions, proper view field, easy reach of the controls etc. to
The driver.

This module highlights the following:

- Spatial accommodation
    - Seating Position
    - Leg Room
    - Head Clearance
    - Lateral Clearance
- Sitting comfort /discomfort
- Reach and limitations of human
- Visual field and Visual Obstruction.

To establish the required interior space and arranging the interior and structural components, the design methods relies on the human factors data base through years of research and practical applications. The anthropometry for automotive design is consistent with the driver and passenger safety, comfort, convenience and accommodation. The study of human capabilities and limitations gives the measurements for designing automobiles. The anthropometry for automotive design is consistent with the driver and passenger safety, comfort, convenience and accommodation. The study of human capabilities and limitations gives the measurements for designing automobiles.

Anthropometric Measurements for Automotive Ergonomics:

Automobile is designed as per the anthropometry of the targeted user population. Measurement process can be broadly classified into two categories.

Conventional Static Measurements:

The measurements taken on human body with the subjects in rigid, standardized position (fig.10). They are typically length, width, height and circumferences. These measurements includes standing height, seated height, seated eye height, upper leg length, knee height, seat length, upper and lower arm length, reach (total arm length), shoulder width, hip or seat width, weight, etc. These measurements are referenced to non-deflecting horizontal or vertical surfaces supporting the subject.

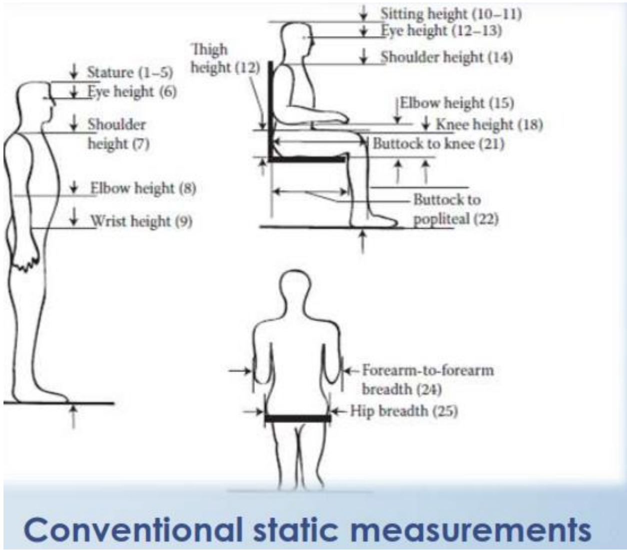


Fig: 2 conventional static measurements

Functional Task Oriented Measurements:

The measurements are taken with the human body dimensional co-ordinates x, y, z with respect to body land marks as reference points. At work or motion in the workspace (fig. 11). Typically they are represented in three dimensional co-ordinates x, y, z with respect to body land marks as reference points.

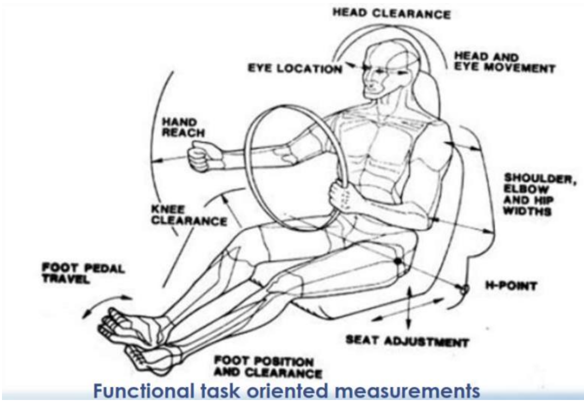


Fig:3 Seat measurements

Functional Task Oriented Measurements:

Few reference points e.g. H-point, BOF, AHP etc. are used as standard practice to define driver's position while SRP, NSRP and SgRP are generally used to define seat position in relation to driver.

• H-point (Hip pivot): Mid point of the line connecting two hip joints.

• BOF (Ball of Foot): Ball joint of Foot.

AHP (Accelerator Heel Point): position of the heel while placed on the accelerator.

• SRP (Seat Reference Point): Intersection point between midline of compressed seat back and compressed seat pan.

• NSRP (Neutral Seat Reference Point): 50th percentile person selected SRP.

• SgRP: 95th percentile person selected SRP.

These landmarks relate the occupant to components in the vehicle interior such as foot controls, seat and floor. For example, the foot is related to the ball of foot and accelerator heel point, where as hip, elbow and shoulder width are related to the h-point location. To accommodate wide range of target population, 5th and 95th percentile anthropometric data are used in general.

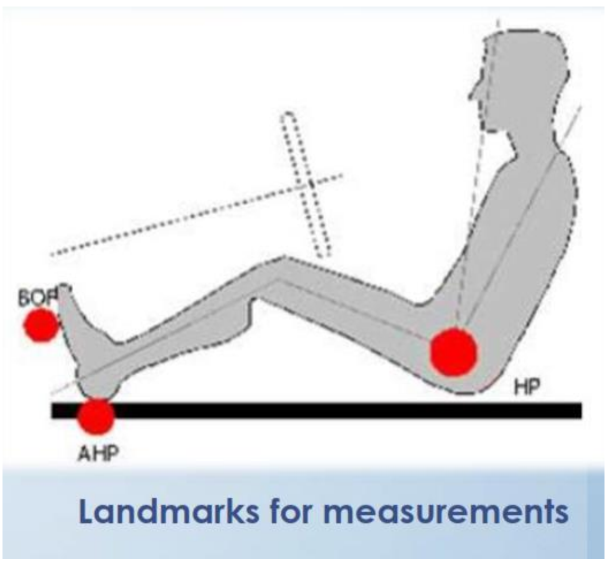


Fig:4 Landmarks for measurements

Seat Track Travel Limit:

Seat track travel limit is decided in such a way so that individuals with smaller body dimensions as well as larger body dimensions can seat comfortably on the seat and can access all the controls

Including accelerator, break and clutch. Seat track travel limits in forward-backward and upward-down ward direction are decided as per operational requirement. Figure depicts forward-backward movement of the seat as per the different percentile driver selected seat position.

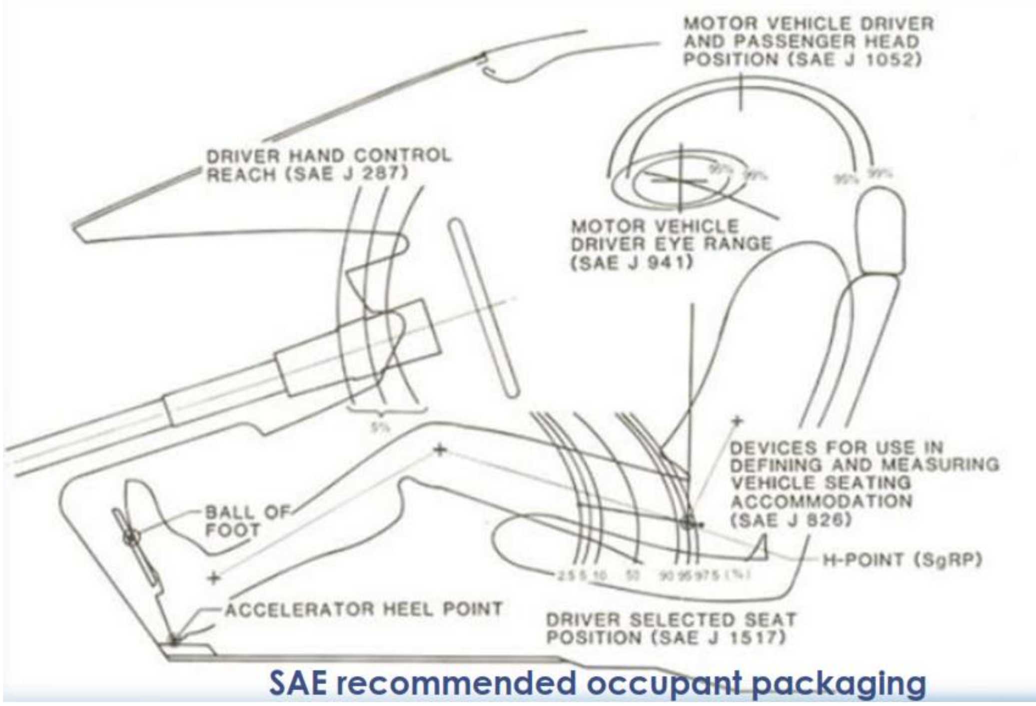


Fig:6 Seat Track Travel Limit:

Spatial Arrangement:

After defining the position of the driver on the seat, all other interior and structural components inside the vehicle are arranged accordingly with the intension to provide sufficient clearance dimensions around him/her. This process relies on human factor database. Larger anthropometric data (95th percentile value) are generally considered for this purpose. Spatial arrangement includes the positioning of driver’s seat and passenger’s seat in the allocated space in side, arrangement of various controls/components according to seating arrangements. In this module leg room, head room and lateral space are to be described in brief.

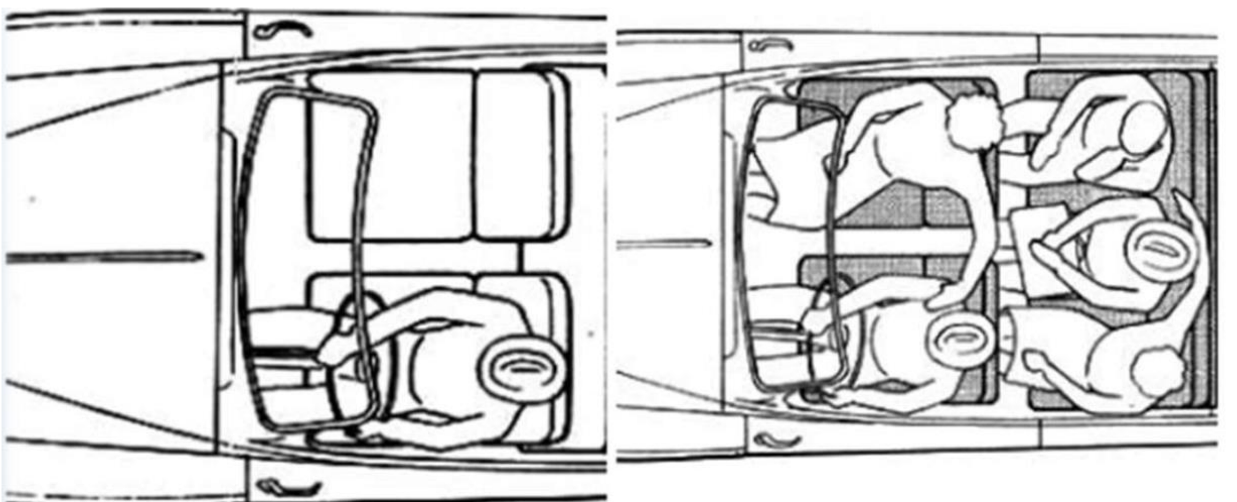
Legroom: The sufficient space for keeping legs of the driver/passenger in a comfortable position in an automobile. Proper legroom enables drivers to access structural component with ease. There should not be any obstacle to keep feet comfortably and at the same time for accessing controls like pedals (break/accelerator/clutch). Measurement of horizontal distance between H-Point and AHP is useful for this purpose. Care should be taken to ensure that any parts of lower body like thighs/knees should not touch with steering wheel or dash board or any other component.

Headroom:

The height. It is the vertical clearance space above the head of driver/passenger in an automobile. A minimum 5.0 cm head clearance for jolt in a vehicle is recommended (Galer 1987, Woodson et al. 1992). In vehicular workstation, available head clearance must be sufficient for wearing and removing the helmet in seated posture in seat.

Lateral Space:

Lateral space is the space pertaining to the side of driver/passenger. Lateral space is important for physical or psychological comfort. Conventionally, 95th percentile bi-deltoid breadth of the population with an additional allowance of 10% on each side can be considered adequate for lateral clearance during normal sitting side by side.



Lateral clearance for sitting side by side

Fig:7 Reach and Limitation of Human:

In many work situations, individuals perform their activity within a specified 3D space of fixed location which is sometimes referred to as ‘work-space envelope’ (Sanders and McCormick 1993). This envelope preferably should be circumscribed by the functional arm reach of the operator and most of the things they need to handle should be arranged within this envelope. In figure describe human capabilities and limits in terms of reach on horizontal work-surface with their measurements.

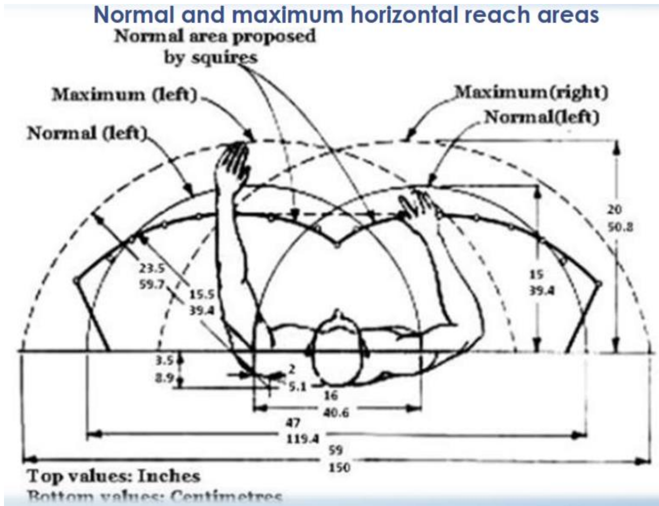


Fig:8 Maximum reach areas

Normal and maximum horizontal arm reach does not correlate with reach capabilities in actual vehicle workstation. Factors such as seat position, seat deflection, shoulder articulation, and lean allowed by lack in a shoulder harness (if one is worn) affects a drivers reach capabilities. Forward arm reach of the driver according to anthropometry and seat track travel as described in SAE J287 shown in figure.

Strength Capability

Strength for Control Operation:

Strength is one type of human performance limiting factor and concerns the application of force in the operation of controls and in other physical tasks. Often, limitation of strength imposes a one-way constraint and it is sufficient to determine the level of force that is acceptable for a weak limiting user. The capabilities of human body are considered to make the operational components in the vehicle while driving. For example, the force is required for the ease of operation of clutch, steering, opening and closing of doors etc. Actuating force limits for some important tractor controls for Indian male agricultural workers (CIAE, Bhopal, 2009) are given Below:

- Brake Pedal:
- 5th p Rt leg strength (male)=261 N.
- Maximum actuating force for break operation should be less than 260 N.
- Clutch Pedal:

- 5th p Lt leg strength (male)=247 N.
- Frequently operated compared to break pedal.
- 50% of 5th p Lt leg strength (male)= 123.5 N.
- Maximum actuating force for Clutch operation should be less than 124 N.

Accelerator Pedal:

5th p Rt foot strength (male)=163 N.

Continuously operated, 30% of 5th p Rt foot strength (male)=49 N (upper limit). Maximum actuating force for accelerator operation should be less than 49 N. Weight of leg = 9%= .09 of body wt., part of this wt. is supported by heel. Lower limit of force exertion for accelerator= 54.7kg x9.81x.09x0.5=24N.

Steering Wheel:

5th p torque strength with both hands, sitting (male) =36 Nm (force 171 N with lever arm of 0.21 m). Frequently operated, 30% of 5th p = 51 N. Maximum actuating force for steering wheel operation should be less than 51 N.

Gear Selection/ Speed Selection Lever: 5th p RT hand push strength = 49 N, limiting force for operation. Maximum actuating force for gear operation should be less than 49 N.

INTRODUCTION TO DRIVER SEAT

Driver seat is an inseparable part of any automobile. Its main function is not only to provide a seating space to driver but also support, protect and to provide comfortable seating posture to the occupants. Today driver seat design has been given very importance because poorly designed seat affect badly on human health as well as psychological condition of driver hence increases the chances of accidents. It is evolved after evolution of first car at the start of nineteenth century. Following table shows the evolution of driver seats with period and car where it is used.

1.1 Parts of Driver seat

Driver seat is very complicated, consists of large number of parts and mechanisms. Main parts of driver seat are frame, padding, seat pan, head restraints system, reclining mechanism with lever, trim (seat cover), and suspension system, air bags, seat belt, fore and aft adjustment, height adjustment etc.

Fig:11 Seat

1.2 Function of Major Components of Seat

1.2.1 Seat Frame

It is most important part of any seat over which all other adjustment systems and components are mounted. It is made from HSLA (High Strength low alloy steel) tube

1.2.2 Anchorage

It is nothing but the space at which driver seat is mounted.

1.2.3 Seat Cushion/ Padding

It is that part of seat on which driver sit. It is soft and made from a resilient material such as PU foam of varying stiffness. Base and back cushions are used for seat.

1.2.4 Seat Back

It is that part of seat which is vertical or somewhat inclined and supports the driver lumbar, shoulder and head. At the top of seat back generally a head restraint system is mounted. Angle of seat back can be adjusted with the help of back reclining mechanism.

1.2.5 Seat Adjustments

It includes height, fore and aft as well as back reclining adjustment systems used to adjust height, fore and aft distance and angle of back respectively.

1.2.6 Head Restraint

It is mounted over the seat back at top, its main function is to support head also restrict the backward displacement and protect the cervical vertebrae. There are four types of head restraints namely integrated, detachable, separate and proactive head restraints .Proactive is advanced version of head restraint.

1.2.7 Suspension

Generally at two places suspension is used namely seat base and seat back. For suspension springs are used. Main purpose of suspension system is to attenuate the vibrations from road at driver seat and his body.

1.2.8 Trim

It is nothing but outermost covering of a driver seat, made from a cloth or leather of good quality. It has pleasant colour, appearance as well as styling.

PARAMETER AFFECTING DRIVER SEAT DESIGN

2.1 Ergonomics Related

Ergonomics is branch of design engineering applied to driver seat design requires that we take into consideration how the products we design fit the people that are using them. When seat fit to the driver it gives more comfort, less stress and maintains good psychological and health condition of driver. Ergonomics can be an integral part of design, manufacturing and use. Knowing how the study of anthropometry, posture, repetitive motion, and work space design affects the user is critical to a better understanding of ergonomics as they relate to end-user needs.

2.1.1 Comfort Related Parameters

Comfort is feelings like relief, encouragement, enjoyment and stable. Comfort is dependent on pressure distribution over the seat, thermal comfort, vibration at driver body, geometric parameters.

2.1.1.1 Pressure Distribution Over Seat

For better comfort and proper maintenance of driver health uniform pressure distribution along the human body over the seat is very important parameter. Pressure distribution over a seat is dependent on the properties of cushion material such as stiffness, deflection and design of cushion. Also it depends on nature of loading, seat pan design and backrest design. A proper seat pan contour distributes a uniform pressure over seat and avoids concentration of stress in human buttocks. Properly inclined back rest avoids stresses at buttocks.

2.1.1.2 Thermal Comfort

It is most important ergonomic parameter on which comfort as well as health of driver dependent. Generally seat cushion materials (seat cover, PU foam and coconut fibers) absorb heat from driver body any acts as heat reservoir, this phenomenon is good for health in winter season but responsible for un-comfort in summer season. This problem can be overcome by using different types of seat covers depending on seasons.

2.1.1.3 Vibration

Road surface is not uniform everywhere yet it is constructed properly by advanced technology. Such surface is responsible for vibrations at driver seats. Human body can sustain 4-7 Hz frequency vibrations. Some tractor semi active hydraulic and active mechanical hydraulic actuators (i.e. suspension systems) are used to attenuate vibration transferred to the driver. Driver seat is so designed that very less vibrations transferred to driver.

2.2 Geometric Parameters

It includes the parameters like lumbar support, backrest slope angle, seat width, depth and height, seat pan angle.

2.2.1 Lumbar Support

Driver works extended periods of times therefore proper lumbar support is very important in order to maintain the position of vertebrae. Large number of anthropometric data is required for proper design of seat which provides support to lumbar. Proper lumbar support can be achieved by maintaining profile of seat cushions of PU foam material.

2.2.2 Other Adjustments

It includes other parameters apart from lumbar support. Seat adjustments totally depend on driver body shape and size i.e. anthropometry. Therefore seat designer must study the huge amount of data related to anthropometry before building a first prototype. Seat angle is also responsible for Pressure distribution over seat. Seat pan can be tilted in between 0-10 0 angle depending on driver body shapes and sizes.

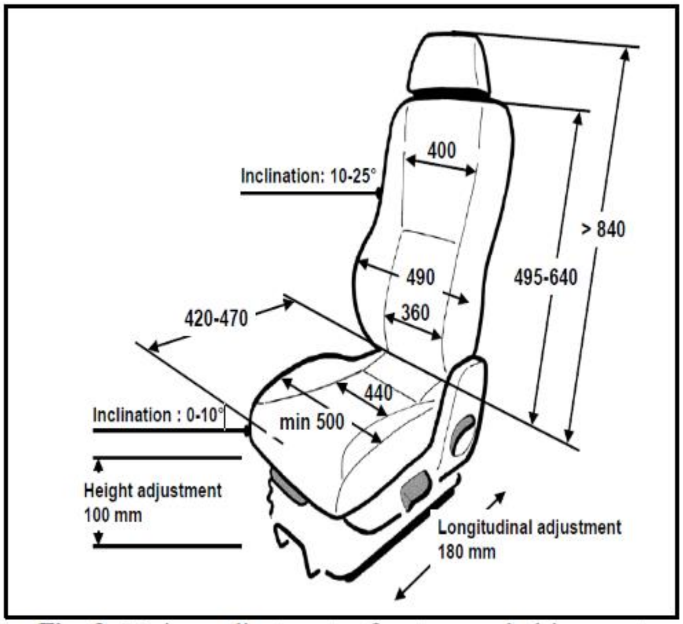


Fig -3: Various adjustments of motor coach driver seat

Dash board instruments:

- Speedometer tells you the speed of your vehicle in MPH and KPH.
- Tachometer shows how many rotations your engine is making per minute.
- Odometer shows how many miles your car has traveled in its lifetime.
- Fuel Gauge shows how much fuel remains in your car's tank.
- Gear Display shows which gear your car is currently in.
- Turn Signal Indicators flash when your turn signals are on; both will flash if you turn on your hazard lights.
- Active System Lights alert you to parts of the vehicle that are activated, such as an open trunk or door.

Automotive displays:

Sharp's ground-breaking Free Form Display has set the automotive world abuzz. Now, automakers and OEM suppliers can integrate display panels in shapes and sizes never seen before. With 25 years of experience developing and manufacturing high-end displays and other components for automotive applications, Sharp Devices Europe also supplies the European automotive industry with a full line of LCD technologies.

And there are more innovations waiting for you at Sharp, such as our 3D displays. Or our progressive super view technology (PSV), which increases sunlight readability to enhance safety and comfort. Pioneering Sharp Dual View displays allow drivers and co-pilots to view different content on the same screen simultaneously.

These pioneering technologies combined with Sharp's extensive consumer electronics experience enable OEMs to succeed at what may be their toughest challenge: delivering in-vehicle experiences that live up to the standards set by the latest consumer electronics.

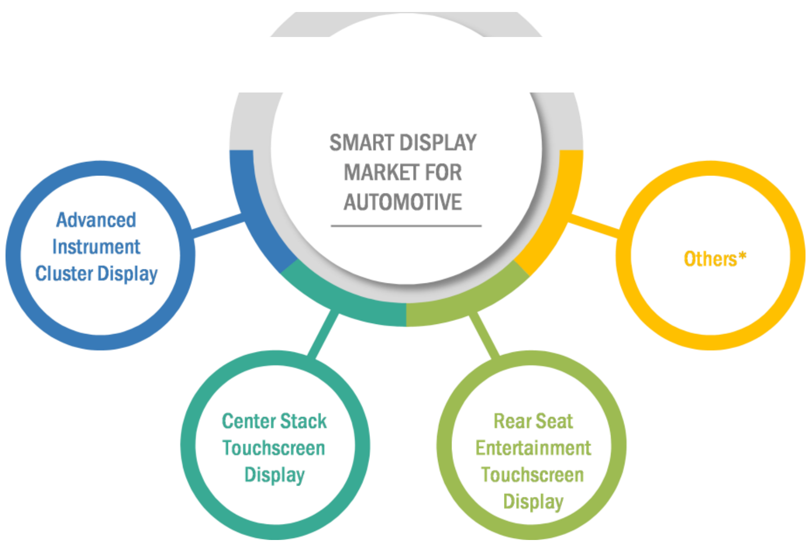


Fig:4 Automotive displays

Electronic Stability Control

Electronic Stability Control (ESC), also called an Electronic Stability Program (ESP), or Dynamic Stability Control (DSC), or Vehicle Stability Control (VSC), depending on the automaker and the market it is being offered in. To summarize, VSC uses the vehicle's brakes to help steer the vehicle during times of slipping or possible spinout. Braking is applied to wheels individually to counter over or understeer. Most VSC systems also reduce engine power automatically during these operations to further improve traction. Toyota calls its systems VSC or Vehicle Dynamics Integrated Management (VDIM).

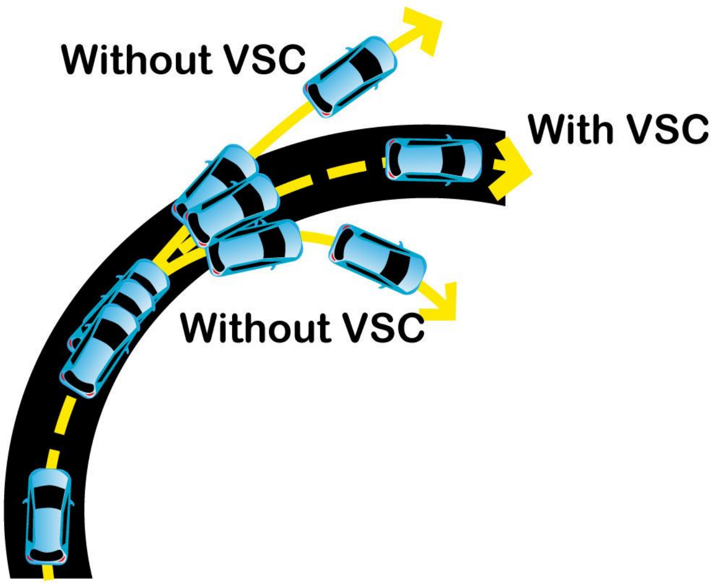


Fig:5 Electronic Stability Control

About one third of fatal traffic accidents could be avoided if a VSC system were employed, according to both the Insurance Institute for Highway safety (IIHS) and the National Highway Traffic Safety Administration (NHTSA). As of 2009, ESC is mandated in vehicles of 10,000 pounds gross vehicle weight or lower sold in the United States on a rolling scale with all vehicles of this size being required to have it after November, 2013.

ESC and VSC first appeared in production vehicles in 1995. It was introduced simultaneously that year by Mercedes-Benz, BMW, Volvo and Toyota. Suppliers included Bosch and ITT Automotive (now owned by Continental Automotive). By the end of 2009, both Ford (which gained their ESC from Volvo) and Toyota had made ESC/VSC standard in all vehicles sold in North America, with Toyota rolling it into all of their brands (including Scion) by 2011.

How VSC Works

In the background, as the vehicle is driven, VSC continually monitors the driver's intended direction (steer) with the vehicle's actual direction (lateral acceleration, yaw, and wheel speeds). When these become disjointed, with the driver's intended direction not being the same as the vehicle's actual direction, VSC intervenes appropriately.

Most commonly, VSC will engage during misjudged cornering (understeer or oversteer due to excessive speed), evasive swerves, and hydroplaning. It operates on all driving surfaces and has proven effective as a means of maintaining vehicle control and reducing accidents.

The system works through a combination of vehicle sensors, control of the anti-lock braking system (ABS), and traction control systems (TSC/ASR) for drive wheels. Unlike these individual systems, VSC considers input from the driver to add stability or correct for steering loss.

The VSC computer, usually located in the vehicle's main fuse box or as part of the ABS system computer, continuously measures yaw (rotation around vertical axis, or left-right spin), individual wheel spin rates, and traction. On most Toyotas with VSC, four sensors are incorporated in this system:

- Steering wheel angle sensor
- Yaw rate sensor
- Lateral acceleration sensor
- Wheel speed sensor

Some larger vehicles will also include a roll rate sensor for rollover prevention. Input from these sensors determines what the VSC is to do, if anything, based on the computer's comparison of data with a total vehicle "state space" (equations used to model vehicle dynamics in real-time). Commands are issued appropriately to vehicle components such as the ABS. A hydraulic modulator in each wheel measures and dynamically adjusts brake pressure individually to the wheel according to instructions from the VSC computer. The driver's steering wheel angle as well as traction sensing is used to determine the amount of correction required.The VSC "Off Switch"

Most sport models of vehicles and some offroad-capable vehicles will have switches to allow the driver to disable VSC. In many sport driving conditions, such as on the track, in rally driving, and some hard offroad applications, VSC can interfere with advanced driving techniques. Corner drifting, for example, common in both track and dirt GT driving, is countered by VSC and thus counterproductive to the sport driver's wishes. Lateral sliding is also sometimes used as a maneuvering tool in offroad driving as a means of avoiding obstacles or gaining traction.

In addition, on some vehicles, the VSC may interfere with the use of a smaller spare tire, which will often give a different wheel spin rate than the other wheels on the car. Some newer Toyotas will compensate for a small spare automatically, however, if the sensed difference is constant (as it should be under normal driving). VSC is also automatically disabled when any of the four wheel speed sensors is disabled, so many shop and tow truck service persons will unplug the wheel speed sensor on the wheel onto which the spare has been temporarily mounted.

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DEPARTMENT OF AUTOMOBILE ENGINEERING

UNIT 2 - VEHICLE FRAME AND SUSPENSION

UNIT -2

VEHICLE FRAME AND SUSPENSION

Loads acting on Vehicle Frame:

Chassis Operating Conditions

The design of an automobile chassis requires prior understanding of the kind of conditions the chassis is likely to face on the road. The chassis generally experiences four major loading situations, that include,

(i) vertical bending,

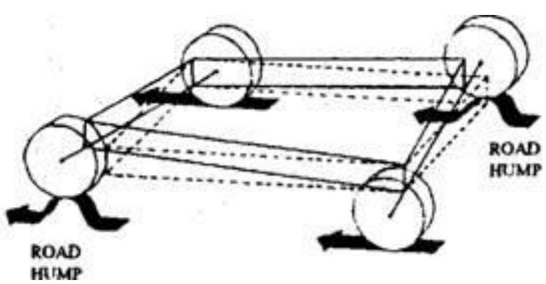
(ii) longitudinal torsion,

(Hi) lateral bending, and

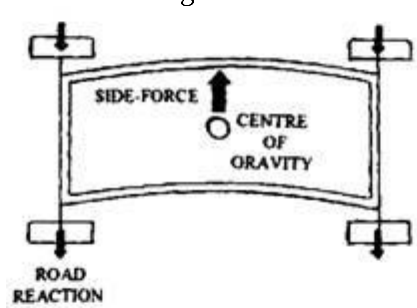
(iv) horizontal lozenging.

Vertical Bending. Considering a chassis frame is supported at its ends by the wheel axles and a weight equivalent to the vehicle's equipment, passengers and luggage is concentrated around the middle of its wheelbase, then the side-members are subjected to vertical bending causing them to sag in the central region.

Longitudinal Torsion. When diagonally opposite front and rear road-wheels roll over bumps simultaneously, the two ends of the chassis are twisted in opposite directions so that both the side and the cross-members are subjected to longitudinal torsion, which distorts the chassis.



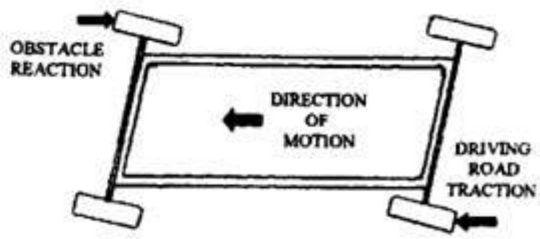
Longitudinal torsion.



Lateral bending.

Lateral Bending. The chassis is exposed to lateral (side) force that may be due to the camber of the road, side wind, centrifugal force while turning a corner, or collision with some object. The adhesion reaction of the road-wheel tyres opposes these lateral forces. As a net result a bending moment acts on the chassis side members so that the chassis frame tends to bow in the direction of the force.

Horizontal Lozenging. A chassis frame if driven forward or backwards is continuously subjected to wheel impact with road obstacles such as pot-holes, road joints, surface humps, and curbs while other wheels produce the propelling thrust. These conditions cause the rectangular chassis frame to distort to a parallelogram shape, known as ‘lozenging’.



Lozenging.

Chassis Frame Sections:

During movement of a vehicle over normal road surfaces, the chassis frame, is subjected to both bending and torsional distortion as discussed in the previous section. Under such running conditions, the various chassis-member cross-section shapes, which find application, include.

(i) Solid round or rectangular cross-sections,

(ii) Enclosed thin-wall hollow round or rectangular box-sections,
(iii) Open thin-wall rectangular channelling such as ‘C, T, or ‘top-hat’ sections.

The chassis side-members, which span the wheelbase between the front and rear axles must be able to take the maximum of the sprung weight. The sprung weight is the weight of the part of the vehicle supported by the suspension system. The binding stiffness of these members must resist their natural tendency to sag. The use of either pressed-out open-channel sections or enclosed thin-wall hollow round or rectangular box-sections can provide the maximum possible bending stiffness of chassis members relative to their weight.

A comparison of the bending stiffnesses of different cross-sections having the samecross-sectional area and wall thickness is presented in F. Considering a stiffness of 1 for the solid square section, the relative bending stiffnesses for other sections are,

Square bar	1.0
Round bar	0.95
Round hollow tube	4.3
Rectangular C-channel	6.5
Square hollow section	7.2

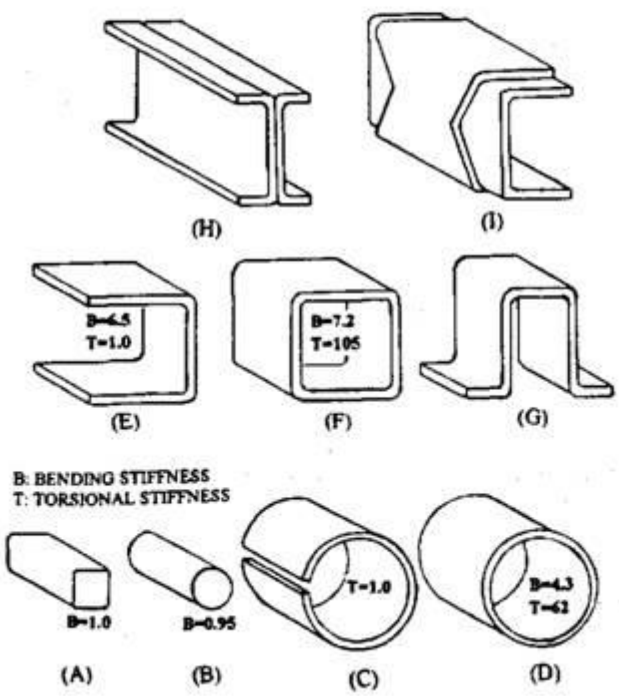
Practically, a 4 mm thick C-section channel having a ratio of channel web depth to flange width of about 3:1 are used as chassis side-members. This provides a bending resistance of 15 times greater than that for a solid square section with the same cross sectional area. For heavy-duty applications, two C-section channels may be placed back to back to form a rigid load-supporting member of I-section (To provide additional strength and support for an existing chassis over a highly loaded region (for example, part of the side-member spanning a rear tandem-axle suspension), the side-members may have a double-section channel. This second skin is known as a flitch frame or plate.

Side-and Cross-member Torsional Resistance.

The open-channel sections exhibit excellent resistance to bending, but have very little resistance to twist. Therefore, both side and cross-members of the chassis must be designed to resist torsional distortion along their length.

F illustrates the relative torsional stiffness between open-channel sections and closed thin-wall box-sections. Comparisons firstly between the open and closed circular sections and secondly between the rectangular sections are made, considering the open section has a resistance of 1 in each case.

Longitudinal split tube	= 1.0
Enclosed hollow tube	= 62.0
Open rectangular C-channel	= 1.0
Closed rectangular box-section	= 105.0

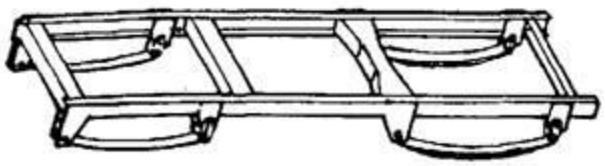


Chassis-member sections. A. Square solid bar. B. Round solid bar.
C. Circular tube with longitudinal slit. D. Circular closed tube.
E. C-section. F. Rectangular box section.
G. Top-hat-section. H. I-section.
I. Channel flitch plate.

This clearly explains the advantages of using channel sections over the hollow tube due to high torsional stiffness. The chassis frame, however, is not designed for complete rigidity, but for the combination of both strength and flexibility to some degree.

Chassis Frame Design

A frame suitable for a light truck or minibus is shown in Fig. The frame uses a non-independent suspension system and is consisted of two channel-shaped side-members, which are joined together with the help of a series of cross-members. These cross-members are placed at points of high stress and are cold-riveted to the side-members. The channel section must be chosen to minimize deflection. Most frames of light vehicles are made of low-carbon steel having the carbon content of 0.15 – 0.25 percent. Since the load varies at each point of the frame, so to reduce its weight either the depth of channel is to be decreased, or a series of holes are to be drilled along the neutral axis in the regions where the load is relatively less.

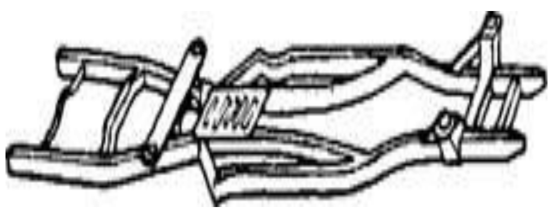


Frame for light truck.

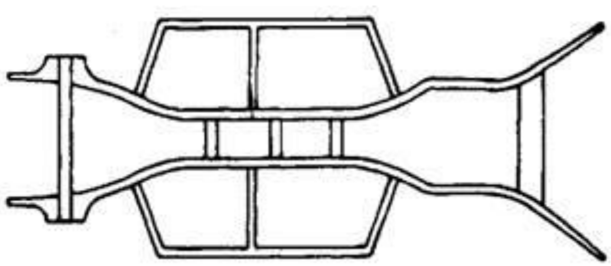
To safeguard the frame against lozenging, gusset plates are fitted to reinforce the joins between the side- and cross-members, or an *X’ type bracing is placed between two or more of the cross members. The frame shown in Fig does not have sufficient rigidity against torsion, so the body has to meet this requirement. If the body is not designed to resist these stresses, the problems like movement between doors and pillars, broken windscreens and cracking of the body panels may occur.

Since body jigs for pressing the integral bodies are generally very expensive, it is usual to use a separate chassis frame when the production of a given model is not large in number. Most of the cars have independent suspension, so the frame must be extremely rigid at the points of joining the main components with the body. To achieve this, box-section members are welded together and suitably reinforced in the regions of high stress.

A backbone frame, an alternative construction to the conventional rectangular frame. In this construction two longitudinal box section members are welded together at the centre and separated at the front and rear to accommodate the main components. A series of out-rigger frame members are welded to the spine to support the floor of the body.



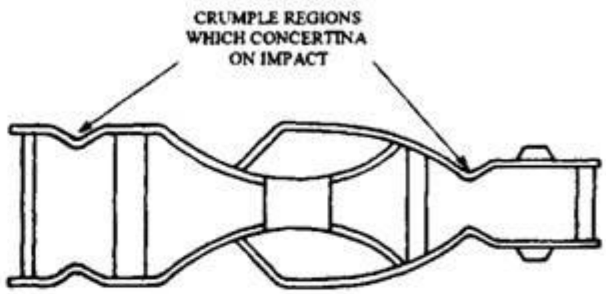
Box section frame.



Backbone-type frame.

Energy-absorbing Frame.

The chassis frames in older designs were made very stiff in order to improve safety for the occupants of a car when involved in a collision. This is not truly correct because on impact the structure provides the occupants an extremely high deceleration and the force acting on the human body as it dashes against a hard surface is likely to cause serious injury or death.



Energy-absorbing frame.

This problem has been overcome in most modern frames by constructing the front- and rear-end of the frame in a manner so that it crumples in a concertina manner during collision and absorb the main shock of the impact. Actually the body panel in the vicinity of these crumple zones are generally damaged beyond repair, but this is a small price to pay to minimize the injury to the occupants. Figure illustrates the principle of designing a frame to absorb the energy of front- and rear-end impacts.

Formulas Used:

1. Bending equation

$$\frac{\mathrm{M}}{\mathrm{I}} = \frac{f}{y}$$

Where, M→Max bending moment

f→ Bending stress

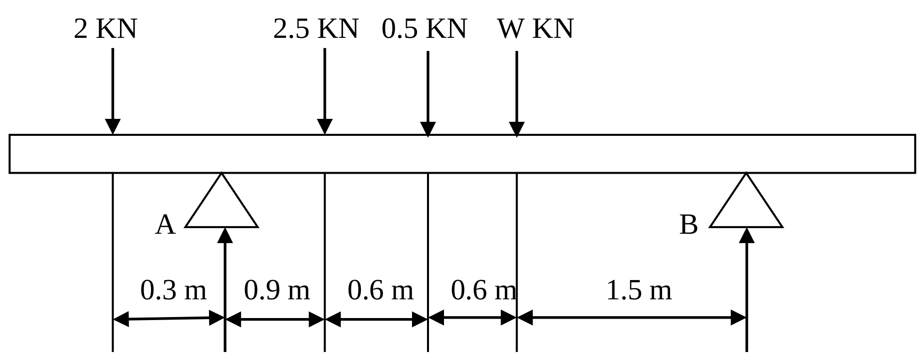
y→Max distance of the outer fibre

I→Moment of Inertia of the section.

PROBLEMS

1. A bus chassis, 5.4 m long, consists of two side member and a number of cross members. Each side member can be considered as a beam, simply supported at two points A&B, 3.6m apart, A being positioned 0.9 m from the front end of the frame and subjected to the following concentrated loads. Engine support (front) 2 KN, engine support(rear) 2.5KN gear box support 0.5KN, and body W,KN. The distance of these loads from the front end of the frame are respectively 0.m, 1.8m, 2.4m and 3m. If the frame are respectively 0.6m,1.8m,2.4m and 3m. If the reaction at A is 8.5KN,determine
   a) The magnitude of the load 'W'due to vehicle body,
   b) The magnitude of the support reaction at B.

The below figure represents diagrammatically the loading system.



Solution:

Clokwise moment about B= Anticlockwise moment about B

$8.5\times 3.6 = 2\times 3.9 + 2.5\times 2.7 + 0.5\times 2.1 + W\times 1.5$

W= 10KN

Now,

$\sum V=0$ (or) upward forces =total downward forces

R_A+R_B= Total downward forces

$8.5+R_B=2+2.5+0.5+10$

$R_B=6.5KN$

2) Calculate the max. Bending moment and maximum section modulus assuming the following particulars.

Wheel base= 180 cm

Overall length= 360cm

Equal overhang on either side.

270 kgf acting at CG of load 45 cm in front of front axle

180 kgf acting at CG of load 45 cm behind front axle

180 kgf acting at CG of load 45 cm in front of rear axle

67.5 kgf acting at CG of load 45 cm behind rear axle

In addition there is a uniformly distributed load of 1.75K per cm run over the entire length of the chassis. Assume dynamic stress=twice the static stress induced.

$\sum MR_2=0 \quad \therefore R_1 \times 180 = 67.5 \times 225 + 180 \times 135 + 180 \times 45 - 270 \times 45$

$R_1=197$ Kgf

$\sum V=0 \quad \therefore R_1+R_2 = 270+180+180+67.5$

$R_2= 500.5$ Kgf

Max B.M, At H, B.M $=270\times 45 + 1.75 \times 90 \times {}^{90}/_{2}$

$=19{,}238$ Kgf.cm

Consider f, = ${}^{600Kgf}/_{cm^2}$

z= Section modulus

Then, B.M =f z

Also max.bending moment due to dynamic force is twice then due to static forces.

$Z= \frac{2\times 19{,}238}{600}$

$Z= 64.1\ cm^3$

3) The load distribution between the front and the rear axle of motor vehicle weighing 1350Kgs is that 48% of the total load is taken by the front axle. The width of the track is 140cm and the distance between the centres of the spring pads is 66cm. Design a suitable ‘I’ section for the front axle assuming that the width of the flange and its thickness are 0.6 and 0.2 of the overall depth of the section respectively and the thickness of the web 0.25 of the width of the flange. Assume a working stress of 915 Kgf\cm^2.

Solution:

Total weight of the vehicle =1350 Kgf

The load taken by the front axle=0.48$\times$ 1350

=648 Kgf.

Assume, the front axle behaves like simply supported loaded at the spring centres.

Max. B.M = 37$\times$ 324

=11988 Kgf m

For the desining of 'I' section,

If, the overall depth = d cm

Then, flange width =0.6 dcm

Web thickness=0.25$\times$ 0.6d

=0.15 d cm

Using the relation $\frac{M}{I} = \frac{f}{y}$,$\rightarrow$ 1

Where, M$\rightarrow$Max B.M =11988Kgf.cm

f $\rightarrow$ Bending stress =915Kgf/cm

y $\rightarrow$Max distance of the fibre =d/2 cm,

I $\rightarrow$ M.O.I of I section

$$I= 1/12\ o.6d\times d^3 - 1/12\times 0.45d\times (0.6d)^3$$

$$I=1/12(0.64^4)\ [1-0.45\times(0.6)^2]$$

$$I= 0.0419\ d^4\ cm^4$$

Sub in 1

$$\frac{12136}{0.0412d^4} = \frac{915}{d/2}$$

$$d^3 = 1582$$

$$d= 5.42\ cm$$

The dimensions of I section are,

Flange = 0.6 ×5.42 =3.252 cm

Flange Thickness= 0.2× 5.42 =1.084 cm

Web tichness= 0.15×5.42 =0.813 cm.

DESIGN OF LEAF SPRING

FORMULAS

1. Deflections, $\delta = \frac{3pl^3}{Enwt^3}$

2. Bending stress, $f_b = \frac{3pl}{nwt^2}$

3. Radius of the curvature, R= $\frac{l^2}{8Z} + \frac{Z}{2} \cong \frac{l^2}{8Z}$
   Where,
   P→ Total load m spring, N
   L →Distance b/w the eyes of spring, m (length of main leaf)
   E →Modulus of Elasticity
   N→ No of leaves
   W→ Width of leaf,m
   t →Thickness of leaf,m
   Z →Perpendicular distance b/w the line joining the centres of the eyes of a Half elliptic spring and inner side of main leaf,m

4. Energy stored per unit volume = $\frac{\tau^2}{6E}$

PROBLEMS

1) A vehicle spring of semi-elliptic type has leaves of 75 width and 10mm thickness and effective length 900mm .If the stress is not to exceed 220725 KPa when the spring is loaded to 4905N, estimate the required number of leaves and the deflection under this condition. If the spring is just flat under load, what is the intial radius? Take E=196.2×10^6 KPa

   Solution

   From, $f_b = \frac{3pl}{nwt^2}$, $n = \frac{3pl}{wt^2 f_b}$

   $$n = \frac{3\times4905\times0.9}{0.075\times(0.01)^2\times220725\times10^3}$$

$n = 8$

Deflection, $\delta = \frac{3pl^3}{Ewnt^3}$

$= \frac{3\times4905\times(0.9)^3\times10^3}{192.2\times10^9\quad\times0.075\times8\times(0.01)^3}$

$\delta = 86.57$ mm

The initial radius, R$=\frac{l^2}{8Z} + \frac{Z}{2} \quad = \frac{l^2}{8\delta} + \frac{\delta}{2}$ $(\therefore z=\delta)$

R$= \frac{(900)^2}{8\times86.57} + \frac{86.57}{2}$

R= 1212.9 mm

DESIGN OF COIL SPRINGS

FORMULAS

1. Deflection, $\delta = \frac{8nPD^3}{GD^4} = \frac{\pi D^2 n\tau}{Gd}$

2. Allowable stress, $\tau = \frac{G\ d\ S}{\pi D^2} = \frac{8PD}{\pi d^3}$, Pa

3. Energy stored in spring $=\frac{p}{2}\times\delta$, Nm

4. Energy stored per unit volume $= \frac{\tau^2}{4G}$

Problems

1) A typical coil suspension spring has 10 effective coils a mean diameter 12.5 mm and made out of wires of dia 15mm. The sprig is designed to carrya max load of 3531.6 N. Calculate the shear stress and the deflection underthe above loading. If amax shear stress of 637650kPa is allowable in the material, then what is the possible clearance in the spring? Take thevalue of G= 73575×10^3 kPa

Solution

Deflection, $\delta = \frac{8n\ PD^3}{G\ d^4} = \frac{8\times10\times3531.6\times(0.125)^3\times10^3}{73575\times10^6\quad\times(0.015)^4}$

$\delta = 148$ mm

Shear stress, $\tau = \frac{Gd\ S}{\pi D^2 n} = \frac{73575\times10^3\ 0.015\times0.148}{\pi\times(0.125)^2\times10}$

$T = 332915$ kPa

$$\delta_{max} = \frac{\pi D^2 \tau_{max}}{G\ d} = \frac{\pi(125)\times^2\times 10\times 637650}{73575\times 10^3\times 15}$$

$$\delta_{max} = 284\text{mm}$$

$\therefore$Allowable clearance $= 284-148$

$=126$mm

$\therefore$Total allowable clearance is 126mm.

DESIGN OF TORISON BAR:

FORMULAS

1) Deflection under torsional moment = $\frac{TL}{GJ} = \frac{32TL}{\pi d^4 G}$

$$\therefore \theta_{max} = \frac{5760\ TL}{\pi^2 d^4 G}\text{ , degrees.}$$

2) Torsional shear stress produced, $\tau = \frac{16\ \tau}{\pi d^3}$

$$\therefore \tau_{max} = \frac{16 T_{max}}{\pi d^3}$$

3) Shear stress, $\tau = \frac{G.\theta,d}{2\ L}$, Pa

$\tau_{max} = \frac{G\theta_{max}\times d}{2L}$,Pa

4) Length of effective part of torsion bar, $L = \frac{G.\theta_{max}\times d}{2\tau_{max}}$

$L = \frac{\pi G \theta_{max} d}{360\tau_{max}}$,m (∴$\theta_{max}$ is in degrees)

Where, T→ Torque,Nm

G→ Modulus of Rigidity, Pa

5) Load rate, r $= \frac{\pi d^4 + 32Wx}{32L(l^2 - x^2)}$

6) Angular rate of deflection, $r_a = \frac{W_{max}}{\theta_{max}}$,N

7) Load, $M = \frac{T}{y} = \frac{T}{l\cos\alpha} = \frac{r_a(\alpha+\beta)}{l\cos\alpha}$, N

Where,

L→ length of lever arm

X→ linear deflection from horizontal =l sin α

Y → effective arm length = l cos α

Problems:

1. A torsion-bar suspension is to be designed to support a maximum static load of 3433.5 N at the end of the lever arm 250 mm long. The deflection of the lever above the horizontal is to be 30° with a total angle of deflection of 90°. Assuming a safe allowable stress of 784800 kPa, calculate (a) the diameter of the torsion bar (b) the effective length and (c) the load rate.

Solution:

Dynamic load= 2 x static load = 2 x 3433.5 = 6867 N

Effective length of the arm = l cos 30° = 250 x 0.866 = 216.5 mm

Moment acting on the bar T = Wy = 6867 x 0.2165 = 1488.7 Nm

Diameter of the bar, $d = \sqrt[3]{\frac{16T_{max}}{\pi f_{smax}}} = 31.2$ mm

Length of the bar is given as, $L = \frac{\pi G\ \theta_{max} d}{360\ f_{s\,max}} = 2.3$ m

Deflection above the horizontal, x= l sin 30° = 12.5 cm

Load rate $r = \frac{\pi d^4 G + 32\ L\ W_x}{32\ L\ (l^2 - x^2)} = 81766$ N/m.

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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

UNIT 3 - FRONT AXLE AND STEERING SYSTEM

UNIT 3

FRONT AXLE AND STEERING SYSTEM

Front Axle and Steering System

Front axle carries the weight of the front part of the automobile as well as facilitates steering and absorbs shocks due to road surface variations. The front axles are generally dead axles, but are live axles in small cars of compact designs and also in case of four-wheel drive. The steering system converts the rotary motion of the driver's steering wheel into the angular turning of the front wheels as well as to multiply the driver's effort with leverage or mechanical advantage for turning the wheels.

The steering system, in addition to directing the vehicle in a particular direction must be arranged geometrically in such a way so that the wheels undergo true rolling motion without slipping or scuffing. Moreover, the steering must be light and stable with a certain degree of self-adjusting ability. Steering systems may also be power assisted.

The chapter discusses the front axle construction and its alignment, and steering geometry and steering systems.

Front Axle

The front axle is designed to transmit the weight of the automobile from the springs to the front wheels, turning right or left as required. To prevent interference due to front engine location, and for providing greater stability and safety at high speeds by lowering the centre of gravity of the road vehicles, the entire centre portion of the axle is dropped.

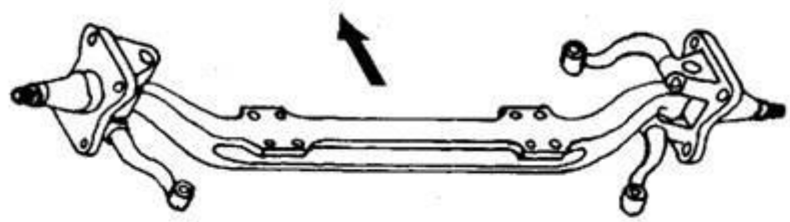
As shown in Fig, front axle includes the axle-beam, stub-axles with brake assemblies, u ack-rod and stub-axle arm.

Front axles can be live axles and dead axles. A live front axle contains the differential mechanism through which the engine power flows towards the front wheels.

For steering the front wheels, constant velocity joints are contained in the axle half shafts. Without affecting the power flow through the half shafts, these joints help in turning the stub axles around the king-pin.

The front axles are generally dead axles, which does not transmit power. The front wheel hubs rotate on antifriction bearings of tapered-roller type on the steering spindles, which are an integral part of steering knuckles. To permit the wheels to be turned by the steering gear, the steering spindle and steering knuckle assemblies are hinged on the end of axle. The pin that forms the pivot of this hinge is known as king pin or steering knuckle pin. Generally dead front axles are three types. In the Elliot type front axles the yoke for king spindle is located on the ends of I-beam.

The axle ends are forked to hold the steering knuckle extension between them. The reverse Elliot front axles have hinged spindle yoke on spindle itself instead of on the axle. The forked portion is integral with the steering knuckle. This type is commonly used as this facilitates the mounting of brake backing plate on the forged legs of the steering knuckle. In the Lemoine type front axle, instead of a yoke type hinge, an L-shaped spindle is used which is attached to the end of the axle by means of a pivot. It is normally used in tractors.



Front axle.

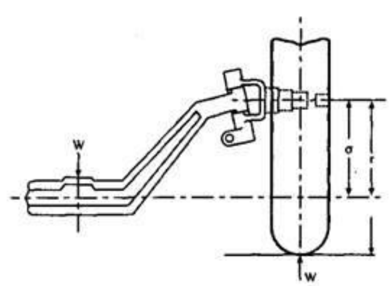
The axle beam in use is of I or H-section and is manufactured from alloy forged steel for rigidity and strength. As compared to dead front axles, a totally different type of swivelling mechanism is used on the live front axle. To connect the wheel hub axles with driving axle shafts, constant velocity joints are used for the vehicles fitted with the front live axles. Tracta, Rzeppa (or Sheppa) on Bendix constant velocity or universal joints are normally used.

Front axles are subjected to both bending and shear stresses. In the static condition, the axle may be considered as a beam supported vertically upward at the ends i.e. at the centre of the wheels and loaded vertically downward at the centres of the spring pads. The vertical bending moment thus caused is zero at the point of support and rises linearly to a maximum at the point of loading and then remains constant.

Thus the maximum bending moment = Wl, Nm
where, W = The load on one wheel, N
I = The distance between the centre of wheel and the spring pad, m

Under dynamic conditions, the vertical bending moment is increased due to road roughness. But its estimate is difficult and hence is generally accounted for through a factor of safety. The front axle also experiences a horizontal bending moment because of resistance to motion and this is of a nature similar to the vertical one but of very small magnitude and hence can be neglected except in those situations when it is comparatively large.

The resistance to motion also causes a torque in the case of drop type front axle as shown in Fig. Thus the portions projected after the spring pads are subjected to combined bending and torsion.



Loads on front axle

The magnitude of the torque = Rδ Nm.

Where, R = the resistance to motion N

δ = the drop from the spindle axis to the centre of the section m

The shear stress in the axle is due to braking torque and its magnitude (as shown in above Fig)

LOADS ACTING IN FRONT AXLE

i) Shear tress (Torsion)
ii) Bending stress

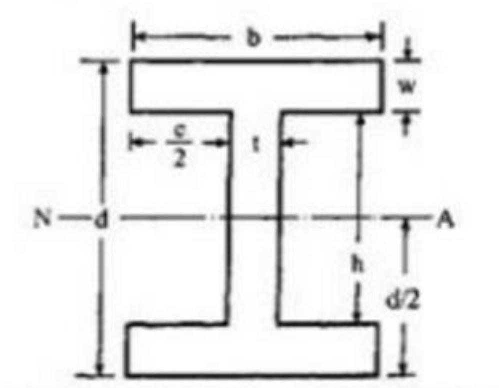
Shear stress in the axle due to braking torque = μWr

FRONT AXLE CROSS SECTION BETWEEN THE SPRING PADS

Bending moment predominates since the braking is lower b/w the spring pads. Thus I-section is used for the portion between the spring pads.

AT THE STEERING HEAD

Since the torsion predominates at the steering head the I-section at the centre gradually changes to circular or oval.



FOR I-SECTION

Max Bending moment,

$$\frac{M}{I} = \frac{f_b}{y}$$

M→ Max bending moment, Nm

F_b→ Allowable bending stress, N/m^2

Y→ Max.distance of the fibre from neutral axis =d/2

I →Moment of inertia of the section =$(bd^3 - ch^3)/12$,m^4 .

Where, d →Overall depth

b→ Flange width

t →Flange thickness

w →web thickness

c=b-t

h = d-2w

Generally d = 6t+b = 4.25t+w = 2.5t

For circular or oval section

Max Torsion is given by,

$$\frac{T}{I_p} = \frac{f_s}{y}$$

Where, T →Max Torque in the plane

F_s →Allowable shear stress

Y→ Distance from neutral axis to the outermost fibre = d/2

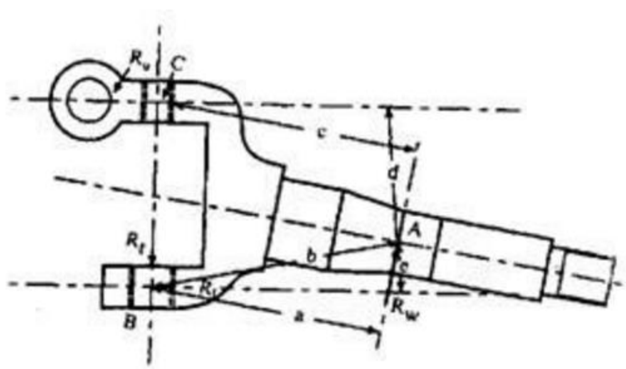
D→ Diameter of circular section= major axis for oval section

I_p →Polar moment of inertia of the section

= (π/32) d^4 for circular section

= (π/32) d^3b, for oval section with minor axis

Bearing loads on front axle



Let,

R_w = Reaction of wheel in spindle acting vertically through the centre of tyre on ground

R_t = Load on thrust bearing

R_{u} = Load on upper knuckle bearing

R_l = Load on lower knuckle bearing

From the figure, $R_l = \frac{c}{d+e}$ kw

$$R_u = \frac{a}{a+e} R_w$$

$$R_t = \frac{ce+ad}{b(d+e)} R_w$$

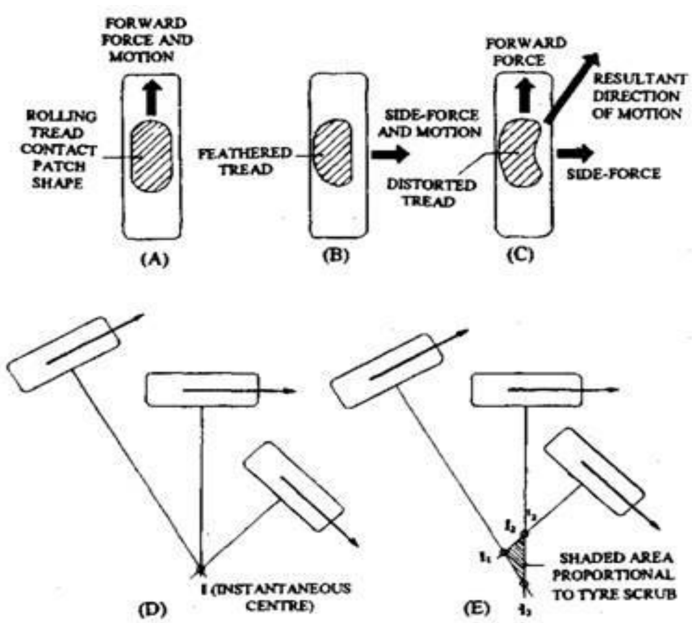
STEERING SYSTEM

Condition for True Rolling

True rolling occurs only when the direction of motion of the vehicle is perpendicular to the wheel axis, i.e. the wheel is subjected to forward force. When wheel is subjected to side force that acts parallel to the wheel axis, a true scrub action is produced. When the wheel is subjected to both forward and side forces, the movement is compounded of true rolling and lateral distortion

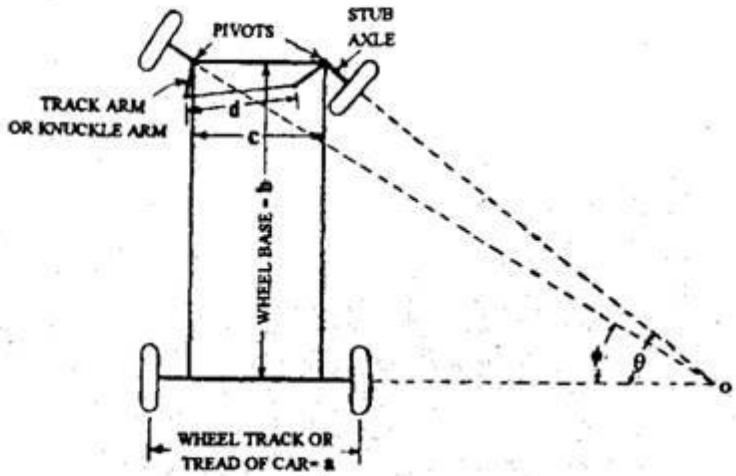
This condition occurs when the wheels are being steered, i.e. the direction of motion is neither parallel nor perpendicular to the axis of rotation.

On a circular path, true rolling condition occurs when the projected axes of several wheels all moving in different curved paths intersect at a single point called the instantaneous centre When these projected axes do not intersect at a single point, a degree of tyre scrub results



Road-wheel and tyre rolling conditions. A. True-rolling. B. True scrub. C. Tyre steer. D. Condition for true rolling. E. Condition for tyre scrub.

Whenever a vehicle takes a turn, the front wheels must turn in a definite manner both in relation to each other and to the axis of the rear wheels so that the lateral slip may be avoided



Condition for true rolling of a vehicle.

and true rolling for all the wheels is obtained. For this, as explained above, all the wheels must always rotate about the instantaneous centre. Since the rear wheels have a common and fixed axis, it is quite obvious that this common centre, 0, would lie somewhere on its extension.

Cot $\boldsymbol{\phi}$ – Cot θ $=\frac{c}{b}$

Where,

θ→ Angle of inside lock

$\boldsymbol{\phi}$ →Angle of outside lock

a →Wheel track

b →Wheel base

c →distance b/w pivot centers

d→ length of track rod

Turning circle Radius

Radius of arc described by the centre of the track made by the outside front wheel of the vehicle when making its shortest turn.

Inner front wheel, $R_{IF} = \frac{b}{Sin\ \theta} - \frac{a-c}{2}$

Outer front wheel, $R_{OF} = \frac{b}{Sin\ \varphi} + \frac{a-c}{2}$

Inner rear wheel, $R_{IR} = \frac{b}{tan\ \theta} - \frac{a-c}{2}$

Outer rear wheel, $R_{OR} = \frac{b}{\tan \varphi} + \frac{a-c}{2}$

Problems:

1) A motor car has a wheel base of 2.743m and pivot centre of 1.05m. The front and rear wheel track is 1.217m. Calculate the connect angle of outside lock and turning circle radius of outer front and inner rear wheels when the angle of inside lock is 40° .

Solution:

For correct steering angle, Cot ϕ – Cot θ $=\frac{c}{b}$

$$\text{Cot } \phi = \frac{1.065}{2.743} + \cot 40°$$

$$\phi = 32.4°.$$

Turning circle radius of outer front wheel, $R_{OF} = \frac{b}{Sin\ \varphi} + \frac{a-c}{2}$

$$= \frac{2.743}{\sin 32.4°} + \frac{1.217-1.065}{2}$$

$$R_{OF} = 5.196\text{m}$$

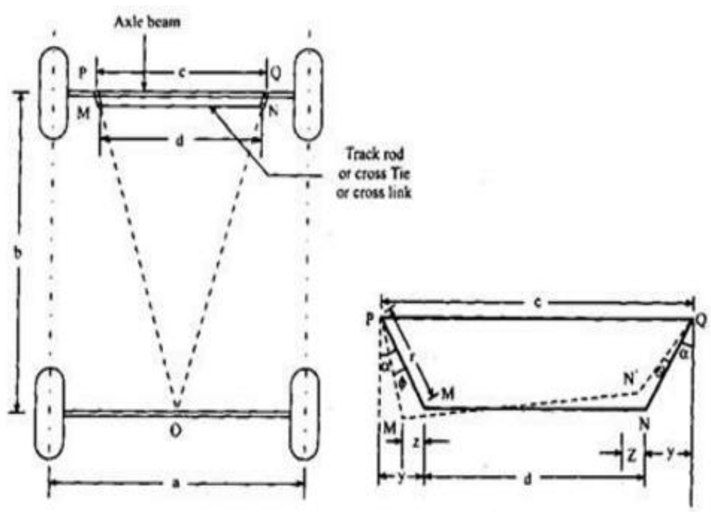
Turning circle radius of inner rear wheel $R_{IR} = \frac{b}{tan\ \theta} - \frac{a-c}{2}$

$$= \frac{2.743}{\tan 40°} - \frac{1.217-1.065}{2}$$

$$R_{IR} = 3.2\text{m}$$

Steering Mechanisms

Ackermann linkage geometry(Minimum error)



From the diagram,

$$\text{Sin }(\alpha+\theta)+\text{ Sin }(\alpha-\varphi)=\frac{2y}{r}=2\text{Sin }\alpha$$

Problems

1) A track has pivot pins 1.37m apart the length of each tack arm is 0.17m and the track rod is behind front axle and 1.17m long. Determine the wheel base which will give true rolling for all wheels base which will give the rolling for all wheels when the car is turning so that the inner wheel stub axle is 60° to the centre link of the car, A geometrical construction may used.

From the above figure, $\text{Sin }\alpha=\frac{c-d}{2r}$

$$\text{Sin }\alpha=\frac{1.37-1.17}{2\times0.17}$$

$$=0.178$$

$$\alpha=16.12\,°$$

Now, $\text{Sin }(\alpha+\theta)+\text{Sin }(\alpha-\theta)=2\text{Sin }\alpha$

$$\text{Sin }(16.12°+30°)+\text{ Sin }(16.12°-\varphi)=2\times0.178$$

$$\varphi=25.56°$$

For correct steering, $\text{Cot }\boldsymbol{\phi}-\text{Cot }\theta=\frac{c}{b}$

$$\text{Cot }25.26°-\text{Cot }30°=\frac{1.37}{b}$$

Wheel base, b= 3.92m

2) The distance between the king pins of a car is 1.3m. The track arms are 0.1525m long and the length of the track rod is 1.2m. For a track of 1.42m are a wheel base of 2.85m, find the radius of curvature of the path followed by the near-side front wheel at which correct steering is obtained when the car is turning to the right.
   Solution:

   For correct steering, $\text{Cot }\boldsymbol{\phi}-\text{Cot }\theta=\frac{c}{b}$

$$\text{Cot }\boldsymbol{\phi}-\text{Cot }\theta=\frac{1.3}{2.85}=0.4562$$

$$\text{Sin}=\frac{c-d}{2r}=\frac{1.3-1.2}{2\times0.1525}=0.328$$

$$\alpha=19.2°$$

   Now the value of 'θ' is to be calculated for correct steering,

   Using the relation,

$\mathrm{Sin}\,(\alpha+\theta) + \mathrm{Sin}\,(\alpha-\theta) = 2\mathrm{Sin}\,\alpha$

Putting $\theta=30^0$, $\mathrm{Sin}\,(19.2^0 - \boldsymbol{\phi}) = 0.656-0.757$

$= -0.101 = \mathrm{Sin}\,(-5.8^0)$

$\boldsymbol{\phi}= 19.2+ 5.8= 25^0$

Now, $\mathrm{Cot}\,\boldsymbol{\phi} - \mathrm{Cot}\,\theta = \mathrm{Cot}\,25^0 - \mathrm{Cot}\,30^0 = 0.4124$

Putting $\theta=35^0$, $\mathrm{Sin}\,(19.2^0 - \boldsymbol{\phi}) = 0.656-0.811$

$= -0.115 = \mathrm{Sin}\,(-8.9^0)$

$\boldsymbol{\phi}= 19.2+ 8.9= 28.1^0$

Now, , $\mathrm{Cot}\,\boldsymbol{\phi} - \mathrm{Cot}\,\theta= \mathrm{Cot}\,28.1^0 - \mathrm{Cot}\,35^0 =0.4446$

Putting $\theta=37^0$, $\mathrm{Sin}(19.2^0 - \boldsymbol{\phi}) = 0.656-0.831$

$= -0.175 = \mathrm{Sin}(-10.05^0)$

$\boldsymbol{\phi}= 19.2+ 10.05= 29.25^0$

Now, $\mathrm{Cot}\,\boldsymbol{\phi} - \mathrm{Cot}\,\theta = \mathrm{Cot}\,29.5^0 - \mathrm{Cot}\,37^0 = 0.4562$

The values for θ and $\boldsymbol{\phi}$ are 37^0 & 29.25^0 respectively

The radius of curvature of path followed by near side front wheel

$$R_{IF} = \frac{b}{Sin\theta} - \frac{a-c}{2}$$

$$R_{IF} = \frac{2.85}{sin37} - \frac{1.42-1.3}{2} = 4.68\text{m}$$

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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOBILE ENGINEERING

UNIT 4 - CLUTCH DESIGN CALCULATION

UNIT 4

CLUTCH DESIGN CALCULATION

CLUTCH:

It is a mechanism designed to disconnect and re-connect the engine power from and to the transmission system.

A Clutch is a mechanical device which is used to connect or disconnect the source of power from the remaining parts so the power transmission system at the will of the operator. The flow of mechanical power is controlled by the clutch.

Types of Clutches

(i) Positive Clutches (ii) Friction clutches

Positive Clutches: In this type of clutch, the engaging clutch surfaces interlock to produce rigid joint they are suitable for situations requiring simple and rapid disconnection, although they must be connected while shafts are stationery and unloaded, the engaging surfaces are usually of jaw type. The jaws may be square jaw type or spiral jaw type. They are designed empirically by considering compressive strength of the material used.

The merits of the positive clutches are

. (i) Simple (ii) No slip (iii) No heat generated compact and low cost.

Friction Clutches: Friction Clutches work on the basis of the frictional forces developed between the two or more surfaces in contact. Friction clutches are usually – over the jaw clutches due to their better performance. There is a slip in friction clutch. The merits are (i) They friction surfaces can slip during engagement which enables the driver to pickup and accelerate the load with minimum shock.

(ii) They can be used at high engagement speeds since they do not have jaw or teeth

(iii) Smooth engagement due to the gradual increase in normal force.

The major types of friction clutches are

(i) Plate clutch (Single plate) (multiple plate)

(ii) Cone clutch

(iii) Centrifugal clutch

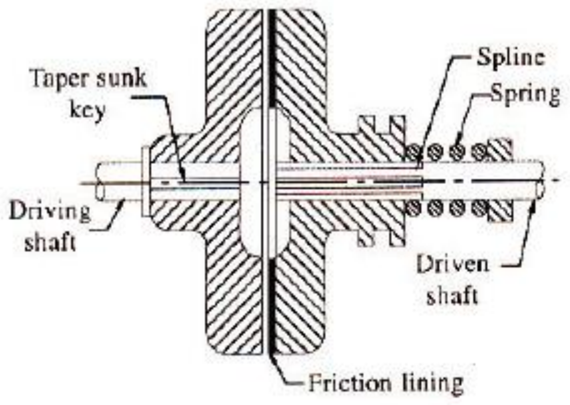
(iv) Dry

(v) Magnetic current clutches

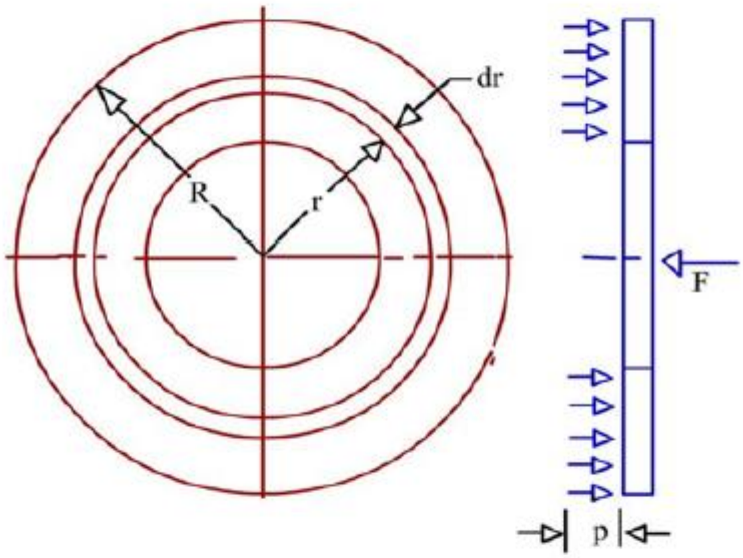
(vi) Eddy current clutches

Single plate clutch:

A single plate friction clutch consisting of two flanges shown in fig 2. One flange is rigidly keyed in to the driving shaft, while the other is free to move along the driven shaft due to spliced connection. The actuating force is provided by a spring, which forces the driven flange to move towards the driving flange. The face of the drive flange is linked with friction material such as cork, leather or ferodo.



Torque transmitted by plate or disc clutch:



A friction disk of a single plate clutch is shown in above fig

The following notations are used in the derivation

Do = Outer diameter of friction disc (mm)

Di = Inna diameter of friction disc (mm)

P = pressure of intensity N/mm2

F = Total operating force (N) (Axial force)

T = torque transmitted by friction (N-mm)

Consider an elemental ring of radius r and radial thickness dr

Area of the friction surface = 2πr.dr

Axial force on the ring, $\partial w = p \times A$ = P$p \times 2\pi r\ dr$

Total axial load $W = \int_{r_1}^{r_2} p\ 2\pi r\ dr$

Frictional torque acting on the ring T= $\mu\ \int_{r_1}^{r_2} p\ 2\pi r^2\ dr$

There are two criteria to obtain the torque capacity – uniform pressure and uniform wear

1. Uniform pressure Theory:

In case of new clutches, un playing assumed to be uniformly distributed over the entire surface area of the friction disc. With this assumption, P is regarded as constant.

$$p = \frac{W}{\pi\ (r_2^2 - r_1^2)}$$

$$T = \frac{2}{3}\ \mu\ W\ \frac{(r_2^3 - r_1^3)}{(r_2^2 - r_1^2)}$$

Uniform Wear Theory:

According to this theory, it is assumed that the wear is uniformly distributed over the entire surface --- of the friction disc. This assumption is used for workout clutches. The axial wear of the friction disc is import ional to the frictional work. The work done by the frictional force (μ P) and subbing velocity (2prN) where 'N' is speed in rpm. Assuming speed N and coefficient of friction 'μ' is constant for given configuration

Wear μ Pr

Pr = constant C

When clutch plate is new and rigid. The wear at the outer radius will be more, which will release the pressure at the outer edge due to the rigid pressure plate this will change the pressure distribution. During running condition, the pressure distribution is adjusted in such a manner that the product pressure is constant, C.

$$W = \int_{r_1}^{r_2} 2\pi C \, dr = 2\pi C \, [\, r_2 - r_1]$$

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

CLUTCH COMPONENTS:

Driving member

(i) Pressure plate assembly

a) Pressure springs

b) Pressure plate (pressure spring type/Diaphragm type)

c) Release levers

(ii)Fly wheel

Driven member

a) Clutch disc or plate

b) Clutch shaft

Operating member

a) Clutch pedal

b) Release bearing

c) Release lever

d) Linkages

SINGLE PLATE CLUTCH PROBLEMS:

1. An automobile power unit gives a maximum torque of 13.56Nm.the clutch is of a single plate dry disc type, having effective clutch lining of both sides of the plate disc. The coefficient of the friction is 0.3 and the maximum axial pressure is 8.29×10^4 pa, and external radius of the friction surface is 1.25 times the internal radius.

Calculate the dimensions of the clutch plate and the total axial pressure that must be exerted by the clutch springs.

Given data:

T=13.56Nm, μ=0.3, P_{max}=8.29 x10^4 pa, r_2=1.25r_1

Solution:

$$T = \pi\mu C(r_2^2 - r_1^2)\text{ , C=p}r_1\text{=8.29 x}10^4 r_1$$

Substituting,

$$13.56 = 2\pi \text{ x } 0.3 \text{ x } 8.29 \text{ x}10^4 r_1 \,(1.5625\, r_1^2 - r_1^2)$$

$$=2\pi \text{ x } 0.3 \text{ x } 8.29\text{x}10^4 \text{ x}0.5625 r_1^3$$

$$13.56 = 87897.835 r_1^3$$

$$r_1^3 = (13.56 \div 87897.835)$$

$$r_1 = \sqrt[3]{(1.542\times^{-04})}$$

$$r_1 = 0.053\text{m and } r_2 = 0.067\text{m}$$

Total axial pressure, W =$2\pi C(r_{2-}r_1)$

$$= 2 \text{ x } \pi \text{x } 8.29 \text{ x}10^4 \text{ x } 0.0536\ (0.067 – 0.0536)$$

W= 373.92N.

2. A motor car engine develops 5.9 KW at 2100 rpm. Find the suitable size of clutch plate having friction linings riveted on both sides to transmit the power, under the following conditions:

a) Intensity of pressure on the surface not to exceed 6.87 x 10^4 pa.
b) Slip torque and losses due to wear etc. is 35% of engine torque.
c) Coefficient of friction on contact surface is 0.3.
d) Inside diameter of the friction plate is 0.55 times the outside diameter.

Solution:

$$T = \frac{p_{\omega\times 60}}{2\pi N}$$

$$T = \frac{60000\times 5.9}{2\pi\times 2100}$$

T = 26.84Nm

Taking account of the losses the total torque, T=26.84 x 1.35

=36.23Nm.

$$T = \pi\mu C(r_2^2 - r_1^2) \text{ X } 2$$

$$36.23 = 2\pi \times 0.3 \times 6.87 \times 10^4 \times r_1 \left(\left(\frac{r_1}{0.55}\right)^2 - r_1^2\right)^2$$

$$= \pi \times 4.122 \times 10^4 \left(\frac{1}{0.303} - 1\right) r_1^3$$

$$r_1^3 = \frac{36.23 \times 0.303}{\pi \times 4.122 \times 10^4 \times 0.697} = 1.216 \times 10^{-4}$$

$$= \sqrt[3]{1.216 \times 10^{-4}}$$

$r_1 = 0.049m$ and $r_{2=}0.09m$

Hence inside diameter =0.098m, outside diameter =0.18m.

3. A plate clutch has three discs on the driving shaft and two discs on the driven shaft, providing four pairs of contact surfaces. the outside diameter of the contact surface is 240mm and inside diameter 120mm. assuming uniform pressure and μ =0.3, find the total spring load pressing the plates together to transmit 23Kw power at 1575 revolution per minute. If there are 6 springs each of stiffness 13 KN/m and each of the contact surfaces has worn away by 1.25mm, find the maximum power that can be transmitted, assuming uniform wear.

Solution:

Given:

$n_{A=3,} n_{B=2,} n = 3 + 2 - 1 = 4, \mu = 0.3, p_{\omega}$=25KW, N=1475rpm,$r_{2=}$120mm,$r_{1=}$60mm

Therefore,

$$\omega = \frac{2\pi N}{60} {rad}/{s}$$

$$\omega = \frac{2\pi \text{ x } 1575}{60} = 52.5\pi \, {rad}/{sec}$$

$p_{\omega=}$T ω

$$T = \frac{P_\omega}{\omega} = \frac{25\times10^3}{52.5\pi} = 151.6\ Nm.$$

For uniform pressure condition,

$$T = n\frac{2}{3}\mu W\left[\frac{r_2^3 - r_1^3}{r_2^2 - r_1^2}\right]$$

$$151.6 = 4 \times \frac{2}{3} \times 0.3\ W\ \left[\frac{0.12^3 - 0.06^3}{0.12^2 - 0.06^2}\right]$$

W=1355N.

Given no, of springs =6

Contact surface of the spring =8

Wear on each contact surface =1.25mm.

Total wear = 1.25 x 8 =10mm.

Stiffness of each spring = $13 \times 10^3\ {N}/{m}$.

Therefore, reduction in spring force

= total wear x no. of springs x stiffness per spring

$$= 0.01 \times 6 \times 13 \times 10^3 = 780N$$

New axial load = $1355 - 780 = 575N.$

Considering uniform wear, torque transmitted,

$$T = n\mu W\left(\frac{r_2 + r_1}{2}\right)$$

$$=4 \times 0.3 \times 575\left(\frac{0.12+0.06}{2}\right) = 62Nm.$$

Maximum power transmitted, $p_{\omega=}$T ω =$62 \times 52.5\,\pi = 10230W.$

CONE CLUTCH PROBLEMS:

In a cone clutch, the semi-angle of cone is 15^0, coefficient of friction is 0.35 and the contact surface have an effective mean diameter of 80mm.if the axial force applied is 196.2N find the torque required to produce the slipping of the clutch under uniform wear.

Calculate the time required attaining the full speed and also the energy lost in the slipping of the clutch, if the clutch is employed to contact an electric motor, running uniformly at 1200 rpm, with a flywheel which is stationary and has a moment of inertia of3.4 Nm^2.

Solution:

Given:

$\theta = 15^0$,$\mu = 0.3$, and $W = 193.2$ N.

$d_{m=}\frac{d_2 + d_1}{2}$=80 =0.08mm.

Now, $T = \frac{\mu(r_2+r_1)W}{2\sin\theta} = \frac{0.35 \times 0.08 \times 196.2}{2\sin 15^0} = 10.6\text{Nm}$

Given N =1200 rpm and I =3.4 Nm^2.

Since,$T = \frac{I}{g}\alpha$, substituting,

$$10.6=\frac{3.4}{9.81}\alpha$$

$$\alpha = \frac{9.81 \times 10.6}{3.4} = 30.58\,{}^{\text{rad}}/_{\text{sec}^2}$$

$$t = \frac{\omega}{\alpha} = \frac{2\pi N}{60\alpha} = \frac{2\pi \times 1200}{60 \times 30.58} = 4.11\text{s}$$

$$\text{Energy supplied} = T\omega t = \frac{T2\pi Nt}{60}$$

$$= \frac{10.6 \times 2\pi \times 1200 \times 4.11}{60} = 5471.9 \text{ Nm}.$$

$$\text{Energy of flywheel} = \frac{\mathrm{I}\omega^2}{2\mathrm{g}} = \frac{3.4}{2 \times 9.81}\left(\frac{2\pi \times 1200}{60}\right)^2 = 2733.8\mathrm{Nm}.$$

Energy lost during slipping $=5471.9 - 2733.1 = 2738.1Nm.$

2. A cone clutch with a cone semi-angle of 12^0 is to transmit 11.19Kw at 750rpm. The width of the face is $1/4th$ of the mean diameter and the normal pressure between the contact faces is not to exceed 8.27×10^4 . Allowing the coefficient of friction of 0.2, determine the main dimensions of the clutch and the axial force required.

Solution:

T$=\frac{60000\times \mathrm{p}_{\omega}}{2\pi \mathrm{N}} = \frac{60000 \times 11.19}{2\pi \times 750} = 142.5\mathrm{Nm}$

$$\omega = \frac{1}{4} \times \frac{d_2 + d_1}{2} = \frac{r_2 + r_1}{4} = \frac{r_2 - r_1}{sin\theta} = \frac{r_2 - r_1}{sin 12^0}$$

$4(r_2 - r_1) = sin12^0(r_2 + r_1) = 0.208\ (r_2 + r_1)$

$$3.792\, r_2 = 4.208\, r_1$$

$$r_{2=}\frac{4.208}{3.792}r_1{=}1.11r_1$$

$$T = \frac{\mu\pi C}{\sin\theta}\ (r_2^2 - r_1^2) = \frac{\mu\pi p}{\sin\theta} r_1(r_2^2 - r_1^2)$$

$$142.5 = \frac{0.2\pi \times 8.27 \times 10^4}{\sin 12^0} r_1(1.231 - 1)\ r_1^2$$

$$= \frac{0.2\pi\ \times 8.27\ \times 10^4 \times 0.231}{0.208}\ r_1^3$$

$= 57678.5\ r_1^3$

$$r_1^3 = \frac{142.5}{57678.5} = 0.0024706\ m^3$$

$$r_{1=0.135\ m}$$

Hence $r_{2=1.11\times 0.135=0.14\ m}.$

$$\omega = \frac{0.135 + 0.14}{4} = 0.7125m.$$

Now, $W = 2\pi C(r_2 - r_1)$

$=2\pi pr_1(r_2 - r_1) = 2\pi \times 8.27 \times 10^4 \times 0.135\ \times 0.014$

=1051.7N.

CENTRIFUGAL CLUTCH

1. A centrifugal clutch is to transmit 14.72 kW at 900 rpm. The shoes are four in number. The speed at which the engagement begins is 3/4th of the running speed. The inside radius of the pulley rim is 140 mm and the centre of gravity of the shoe lies at 120 mm from the centre of the spider. The shoes are lined with Ferrodo for which the coefficient of friction may be taken as 0.25. Determine

(a) Weight of the shoe and

(b) Size of the shoes if angle sustained by the shoes, at the centre of the spider is 60^0 and the pressure exerted on the shoes is 9.81 x 104 N/m^2.

Solution:

Given:

P_w=14.72kw, n=4, N=900rpm, r_c=0.12m, r=0.15,μ=0.25

$$\omega = \frac{2\pi \times 900}{60} = 94.26\,{}^{rad}/_{sec}$$

$$\omega_{e=\frac{3}{4}}\omega=70.7\,{}^{rad}/_{sec}$$

$$T = \frac{p_{\omega\times 60}}{2\pi N} = \frac{14.72\times 60000}{2\pi 900} = 156.26Nm.$$

We have $\frac{p_{\omega\times 60}}{2\pi N} = n\mu\frac{W}{g}(\omega^2 - \omega_e^2)r\,r_c$

$$W = \frac{156.26\times g}{n\mu r_c r(\omega^2-\omega_e^2)} = \frac{156.26\times\times 9.81}{4\times 0.25\times 0.12\times 0.15[(94.26)^2-(70.7)^2]} = 21.9N$$

Given $2\alpha = 60 = \pi/3\,rad$

P=$9.81\frac{N}{m^2}$

Therefore l=contact length of the shoe =2 $\alpha r = (\pi/3)\times 0.15 = 0.1571m.$

$R = F - P = \frac{W}{g}(\omega^2-\omega_e^2)r_{c=}$ blp.

Therefore, b=

$$\frac{Wr_{c(\omega^2-\omega_e^2)}}{glp} = \frac{21.9\times 0.12\times[(94.26)^2-(70.7)^2]}{9.81\times 0.1571\times 9.81\times 10^4} == 0.0674\text{m}.$$

2. A centrifugal clutch is to transmit 25.8 P_ω at 750rpm when engaged at 75 percent of the running speed. The inside diameter is 0.36 m and the radial distance of the centre of gravity of each shoe from the shaft axis is 0.15 m.

Assuming μ = 0.3, determine the necessary weight of each of shoe of the above clutch.

Solution: N=750rpm, $\omega = \frac{2\pi N}{60}\,{}^{rad}/_{s}$, $\omega = \frac{2\pi \text{ x } 750}{60} = 25\,\pi\,{}^{rad}/_{sec}$, r=0.18 m, $\boldsymbol{r_{c=}}$0.15m, μ=0.3, x=0.75, $P_\omega = 25.8KW$.

Now$\frac{60000\, P_\omega}{2\pi N} = 4\mu \frac{W}{g}\omega^2 r_c r(1-x^2)$,

$$W = \frac{60000\ \times 25.8\ \times g}{2\pi N 4\mu\omega^2 r_c r(1-x^2)}$$

$$= \frac{60000\ \times 25.8\ \times 9.81}{2\pi \times 750 \times 4 \times 0.3(25\pi)^2 \times 0.15 \times 0.18[1-(0.75)^2]}$$

$$= \frac{60000\ \times 25.8\ \times 9.81}{8\pi^3 \times 750 \times 0.3 \times 625 \times 0.15 \times 0.18 \times 0.4375} = 36.91N.$$

3. A centrifugal clutch has four shoes which slides radically in a spider keyed to the driving shaft and make contact with the internal cylindrical surface of a rim keyed to the driven shaft When the clutch is at rest, each shoe is pulled against a stop by a spring so as to leave a radial clearance of 5mm between the shoe and the rim .the pull exerted by the spring is then 500N. The mass centre of the shoe is 160mm from the axis of the clutch.

If the internal diameter of the rim is 400mm, the mass of each shoe is 8kg, the stiffness of each spring is 50N/mm and the coefficient of friction between the shoe and the rim is 0.3, find the power transmitted by the clutch at 500rpm.

Solution

n=4 , C=5mm, m=8kg, S=500N, s=500 N/mm, r_c=160mm, μ=0.3, r=200mm, N=500rpm.

Therefore,

$$\omega = \frac{2\pi N}{60} {rad}/_{s}$$

$$= \frac{2\pi \times 500}{60} = 52.37 {rad}/_{s}$$

The operating radius, r_o= $r_c + C = 160\ + 5 = 0.165m$

$$F = m\omega^2 r_0 = 8(52.37)^2 \times 0.165 = 3620\ N.$$

$$P = S + C_s = 500 + 5 \times 50 = 750\ N.$$

Frictional force acting tangentially on each shoe $= \mu R = \mu(F - P)$

$$= 0.3(3620 - 750) = 861\ N$$

Total friction torque,

$$T = \mu Rrn = 861\ \times 0.2 \times\ 4 = 688.8\ Nm$$

Power transmitted, $P_\omega = T\omega = 688.8\ \times 52.37 = 36100\ W$.

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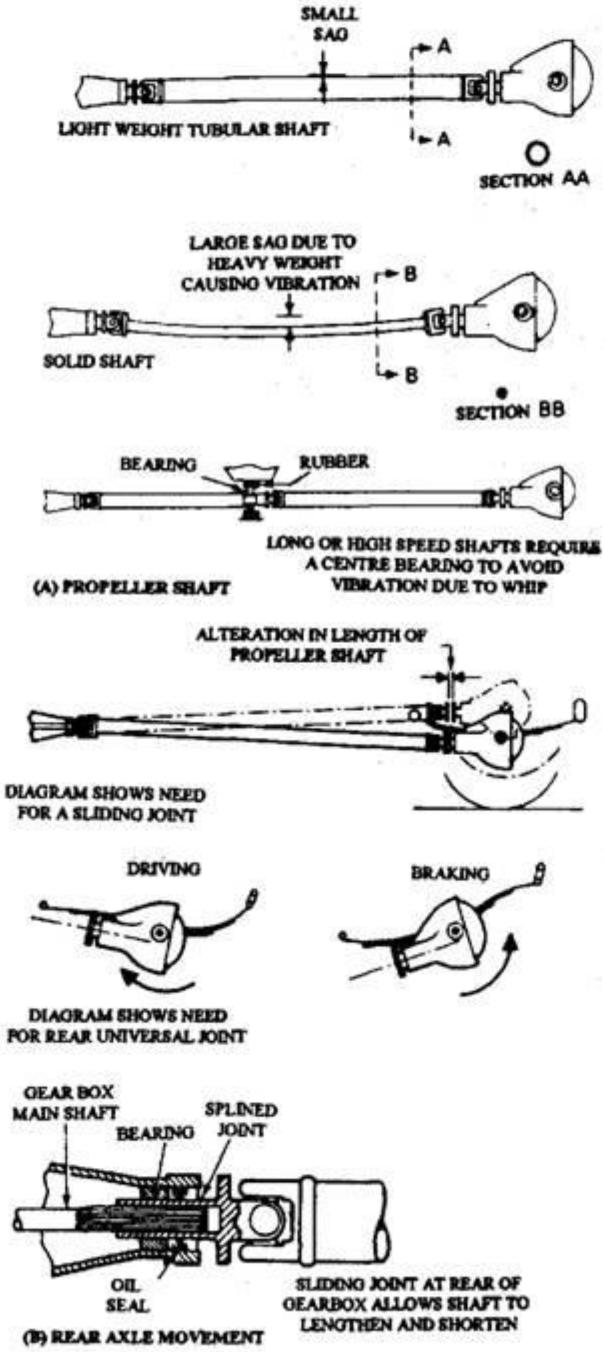
UNIT-5-GEAR BOX, FINAL DRIVE AND REAR AXLE

UNIT V

FINAL DRIVE AND REAR AXLE

Propeller Shaft and Drive Shaft

Propeller shaft, sometimes called a carden shaft, transmits power from the gearbox to the rear axle. Normally the shaft has a tubular section and is made in one- or two-piece construction. The two-piece arrangement is supported at the midpoint by a rubber mounted bearing. Short drive shafts are incorporated for the transmission of power from the final drive assembly to the road wheels in both front and rear wheel drive layouts.



Design of propeller shaft:

Formulas used

1) Critical speed, $N_c = \frac{60}{2\pi}\frac{\pi^2}{l^2}\sqrt{\frac{EI}{\rho A}}$, rpm

Where,

I→ Moment on inertia of shaft section= $\frac{\pi}{64}(d_o^4 - d_i^4)$,

A→ Cross sectional area $\frac{\pi}{4}(d_o^2 - d_i^2)$,

d_o→ Outside dia,m

d_i →Inside dia,m

l→ Length of the shaft,m

g →acceleration due to gravity,$\frac{m}{s^2}$

E→ Modulus of elasticity for steel

ρ→ Density of steel, =7860 $\frac{kg}{m^3}$

Note

A tubular section propeller shaft is normally used because it has,

i) Low weight

ii) Provides large resistance to misalignment, especially sag

iii) has good torisonal strength

iv) Provides lo resistance to changes in angular speed

2) The torque equation is given by

$$\frac{T_d}{I_p} = \frac{f_s}{y}$$

Where,

T_e → Engine torque, Nm

T_t →torque tom be transmitted by propeller shaft

G→ Over all gear rates

f_s →Safe shear stress , N/m^2

I_p →Polar moment of inertia of the shaft section $= \frac{\pi}{32}(d_o^4 - d_i^4)$, m^4 (for hollow section)

Y distance from neutral axis to the outcome fire of shaft =$d_0/2$

Then, $T_t = T_e.G$

T_d =design torque for the shaft

3) $T_d = T_t \times$ Factor of safety (F.O.S =3) generally.

Problems:

1) An automobile engine develops 28KW @1500 rpm and its bottom gear ratio in 3.06. If a propeller shaft of 40mm outside diameter is to be used, determine the inside diameter of mild steel tube to be used, assuming a safe shear stress of 55×10^3 kpa for MS.

Solution

$$P=\frac{2\pi N T_e}{60}$$

$$28=\frac{2\pi\times1500\times T_e}{60000}$$

$\therefore T_e = 178.34$ Nm

Torque to be transmitted by propeller shaft, $T_t = T_e\times G$

$$=178.34\times 3.06$$

$$T_t = 545.72 \text{ Nm}$$

$$I_p = \frac{\pi}{32}(d_o^4 - d_i^4), \text{ m}^4$$

Now, $\frac{T_d}{I_p}=\frac{f_s}{y}$

$$\frac{545.72}{\frac{\pi}{32}(d_o^4-d_i^4)}=\frac{55\times10^6}{0.02}$$

d_i =27mm (sub d_o = 40mm)

Differential

This mechanism rotates the wheels at different speeds, while maintaining a drive to both wheels.

Problems:

1) The steering set of a vehicle provide a turning circle radius of 6.6m with a wheel track width of 1.2m. The effective road wheel rolling diameter is 0.72m. Calculate the number of revolutions made by the inner and outer wheels for one turning circle.

Solution

Mean turning radius, R_m = 6.6m

Outer wheel turning Radius, $R_o = 6.6+\frac{1.2}{2}=7.2$m

Outer wheel turning Radius, $R_i = 6.6-\frac{1.2}{2} = 6$m

Rolling circumferences of road wheel = $\pi D=0.72\pi m$

Distance travelled by outer wheel for one complete turning circle $2\pi R_o = 2\pi\times7.2$

Distance travelled by inner wheel for one complete turning circle $2\pi R_i = 2\pi\times6$

Revolutions completed by outer wheels = $\frac{14.4\pi}{0.72\pi}$ = 20 revolutions.

Revolutions completed by outer wheels = $\frac{12\pi}{0.72\pi}$ = 16.6 revolutions.

The difference = 20-16.6 =3.4 revolutions for one complete turning circle wheel

Note

The difference must be absorbed by tyre scrub or accommodated by some mechanical means

2) The below figure shows a differential gear as applied to a motor vehicle. The pinion A on the propeller shaft has 12 teeth and gear with crown wheel B of 60 teeth, shafts R and S from the rear axles to wheels the road wheels are attached

a) Show that the speed of the caring is the mean of the speeds of the two rear wheels, when the vehicle is taking a turn

b) If the propeller shaft has a speed of 1000rm and the road wheel which is driven by C has a speed of 210 rpm. What is the speed of the road wheel driven by D?

c) In which direction is the vehicle taking a turn?

Solution

a) $\frac{N_{C-}N_B}{N_D - N_B} = -\frac{T_D T_B}{T_B T_C} = -\frac{T_D}{T_C} = -1$ [$T_D = T_C$]

(or) $N_c - N_B$ =-) $N_D + N_B$ => $N_B = \frac{N_C + N_D}{2}$

The carring or crown wheel has the mean speed of two wheels

b) The speed of the crown wheel, $N_B = \frac{T_A}{T_B} \times$ Speed of propeller shaft

$$= \frac{12}{60} \times 1000 = 200\text{rpm}$$

$N_D = 2N_B - N_C$ = 400-210 [N_C = 210rpm]

N_D = 190 rpm

c) The vehicle will take a turn towards left.

3)An engine develops a torque of 104Nm at 2500rpm ad drives through a gearbox having constant mesh of 15 and 30 teeth respectively. The second gear wheel of the mainshaft has 36 teeth and the meshing pinion has 18 teeth. The rear axle ratio is 5:1the effective radius of tyre is 0.42mm.If the overall transmission efficiency is 85%.Calculate

a) The speed of the vehicle in second year

b)Torque in each half shaft

solution:

IInd year ratio of G.B = $\frac{Product\ of no\ of\ teeth\ on\ driven\ gears}{Product\ of\ no\ of\ teeth\ on\ driver\ gears}$

$$\frac{30\times36}{15\times18} = 4:1$$

Overall gear ratio= G.B ratio$\times$ Rear axle ratio

$=4\times 5=20:1$

Speed of Road wheels in 2nd year = $\frac{Engine\ speed}{Overall\ gear\ ratio}$

$=\frac{2500}{20}$ = 125rpm

Linear speed of road wheel = Angular speed × Circumference

$=125\times 2\pi \times 0.42$ M/min

= 19.8 km/hr

Assuming no slip, vehicle speed = linear speed of wheel

Torque delivered to rear axle in second gear =Engine torque ×GR× η

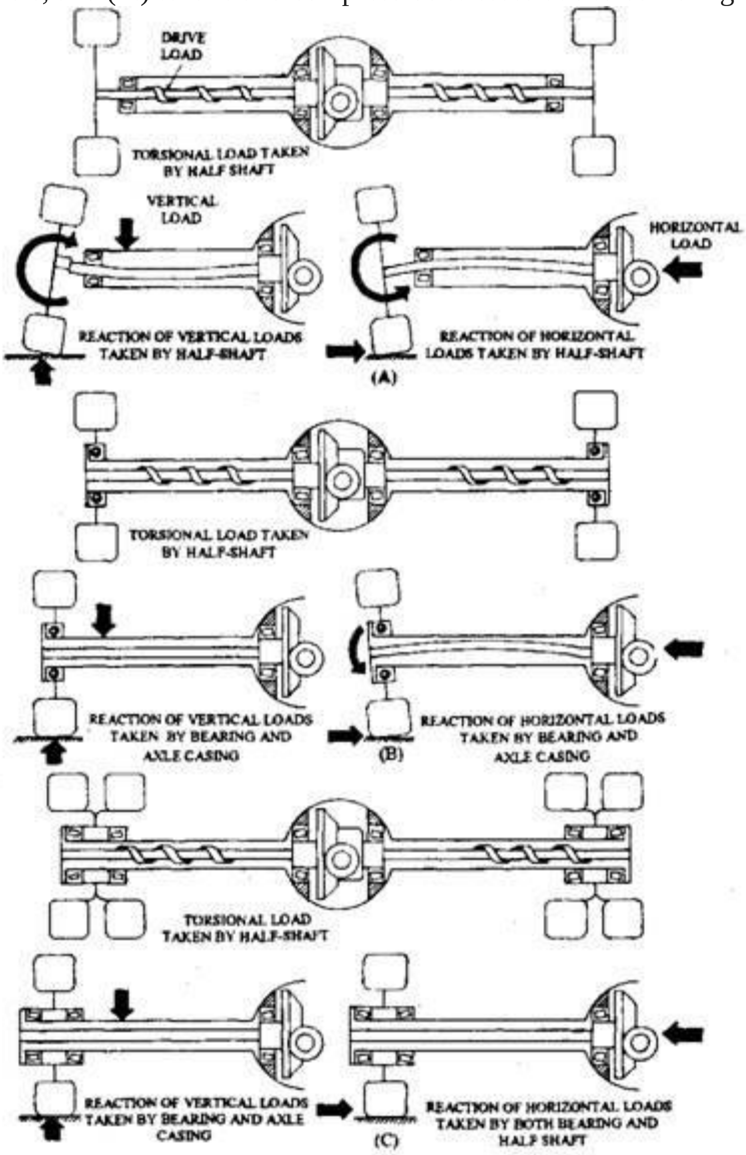
$=104\times20\times0.85$

T_r = 1768 Nm

Torque in each half shaft = $\frac{1768}{2}$ = 884 Nm

Axle Shafts and Hub Arrangements

The axle shaft transmits the drive from the differential sun wheel to the rear hub. The various types of shafts may be compared based on the stresses they resist. A simple automobile shaft has to withstand (i) torsional stress due to driving and braking torque, («) shear and bending stresses due to the weight of the vehicle, and (lit) tensile and compressive stresses due to cornering forces.



Loading of different axle-hub arrangements. A. Semi-floating axle hub. B. Three-quarter floating axle hub.
C. Fully floating axle hub.

Axle shafts are divided into semi-floating, three-quarter floating and fully floating depending on the stresses to which the shaft is subjected. Axle half-shafts are situated on each side of the final drive and convey motion to the road-wheels.

There are basically three different arrangements of supporting axle wheel hubs on the rear-axle casing. These include :

(i) Semi-floating axle hub (commonly used on cars).

(ii) Three quarter floating axle hub (rarely used today).

(m) Fully floating axle hub (commonly used on heavy vehicles).

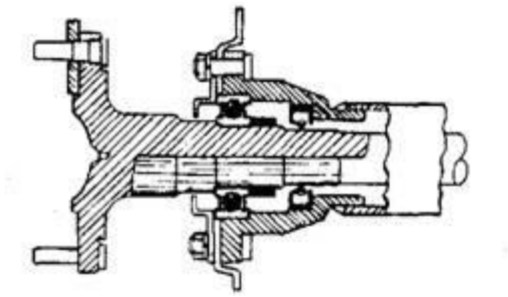
Figure demonstrates how loads are resisted with different axle-hub arrangements. A tough, hard material is used for the axle shaft to withstand the various stresses, resist spline wear and provide good resistance to fatigue. Medium carbon alloy steel containing nickel, chromium and molybdenum is generally used to manufacture axle shafts.

Semi-floating Axle Hub.

The road-wheel is attached to the axle hub, which is an extension of the axle half-shaft. A single bearing inside the tubular axle-casing supports the outer end of the shaft. The inner end of the shaft is splined and supported by the final-drive unit, which itself is mounted on bearings within the axle casing.
The semi-floating axle along with its overhanging hub is subjected to the driving torque as well as to both vertical and horizontal loads. The vertical load produces a shearing force, and the distance between the wheel and the suspension-spring seat on the axle causes a bending moment, the reaction of which is shared between the axle bearing and the final-drive-unit bearings.

The horizontal load due to tilting of the vehicle, cornering centrifugal force, or side wind gives rise to both side-thrust and a bending moment. This bending moment may add to the vertical bending moment or may oppose it, depending on the direction of application of the side-force. A semi floating axle, suitable for small and medium sized cars, is illustrated in Fig.

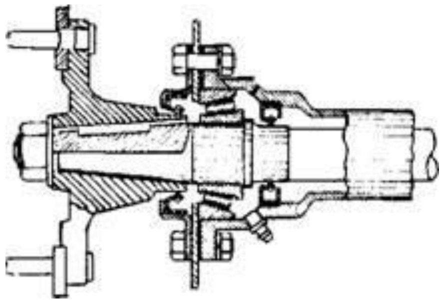
The axle half shaft and flanged hub are forged from a single piece of nickel chrome steel. The hub end of the shaft is provided with a larger diameter than the rest of its length, which resists the vertical and horizontal loads. The outer face of the flanged hub is shouldered so that it centralizes accurately the brake drum. The flange is provided with evenly spaced holes around it for wheel studs.



Semi-floating ball-race-bearing axle.

A pre-greased and sealed deep grooved ball-race bearing is pressed over and along the shaft up to its shoulder. The bearing is sandwiched on one side by the axle-casing and on the other by the brake back-plate and the retaining plate by four nuts and bolts. To prevent excess oil leakage to the end of the axle-casing, a radial-lip oil-seal is pressed into a recess in the casing. Oil level of the final-drive is considerably lower than the axle shaft. However, the large amount of splash may cause the lubricant to spread along the shaft and enter the brake drum. An oil retainer is fitted at the hub end and the lip of the seal is positioned towards the final drive in the sealing arrangement.

A semi-floating axle shown in Fig. uses a taper-roller bearing, which is suitable for larger and higher-performance cars because of its greater load-carrying capacity. A separate hub is wedged on to a keyed and tapered half-shaft and a castellated nut holds it is position. The taper-roller-bearing inner cone fits with a light force inside the mouth of the casing. The exact position of the bearing in the casing is provided by shims packed between the casing flange and the brake back-plate. Increasing the thickness of the shims on one side and decreasing it on the other shifts both half-shafts further to one side relative to the axl casing. On either road-wheel the outward thrust is absorbed by the adjacent hub bearing, while inward thrust is transmitted to the opposite bearing through the axle half-shafts and a slotted axle-shaft spacer (not shown). Therefore, each hub bearing takes thrust in one direction only.



Semi-floating taper-roller-bearing axle.

Bearing Loads.

Bearing loads due to side thrust on a wheel in semi-floating axle is shown in Fig.

Let F = lateral force at the rim of the wheel

r = radius of the wheel

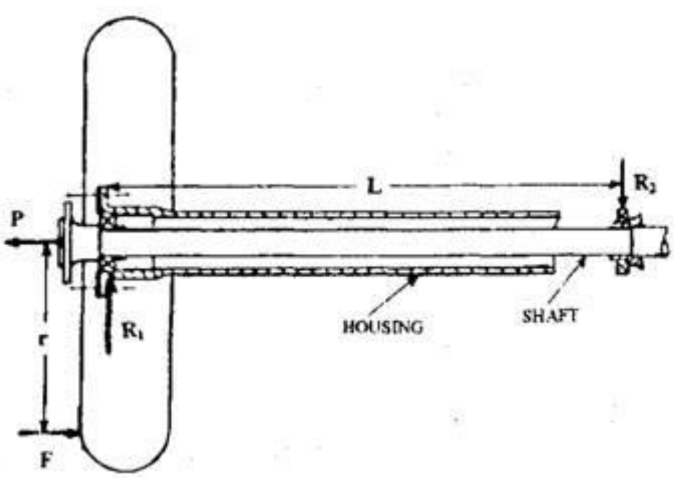
L = distance between the centres of wheel bearings R\ and

R2 = radial reactions of the wheel bearing on the wheel hub

P = the thrust reaction of the bearing In practice, the radio rIL ~ 0.6.

Considering the forces in the horizontal and vertical directions,

P = F and R\=R2



Bearing loads due to side thrust on semi-floating axle.

Taking moments $R_1 L = R_2 L = Fr$

Hence, $R_1 = R_2 = \frac{r}{L} F \approx 0.6\ \mathrm{F} \approx \frac{3}{5}\ \mathrm{F}$.

Therefore, for semi-floating type axles, P is equal to F, and Ri and R2 each approximately equal to three fifth off. Ri adds to the normal static load on the bearing, whereas R2 opposes it.

Axle Shaft.

Shafts for semi-floating type axles are subjected to both bending and torsion. Hence, the diameter, of the shaft should vary with the bending moment along the length. Accordingly, the diameter is minimum near the differential end where the shaft is subjected to nearly total torsion alfcd is maximum at the outboard bearing, where bending moment is maximum. The maximum stress in shaft occurs when the wheels slip or lock due respectively to a sudden application of power or braking on dry, hard pavement.

The frictional force under these conditions $= \mu W$

where, μ = the coefficient of friction between tyre and the road,

= 0.6 for dry, hard pavement

W = the load supported on one wheel under fully loaded condition.

The frictional force, μW produces a torque $= \mu W r$ and a bending moment at the centre of the outboard bearing $= \mu W l$

where r = the effective radius of the wheel

l = the distance between the centre planes of the road wheel and the outboard bearing of the axle.

The load, W produces a bending moment on the axle at the centre of the outboard bearing $= Wl$.

Again driving torque on each wheel $= T_w/2 = \frac{1}{2}\ G\eta_t\ T_e$ (Refer section 31.3)

But driving torque should be less than the torque due to frictional force. Thus, there are two bending moments on the shafts at the outboard bearing, a vertical moment, Wl and a horizontal moment, μWl. The resultant of these bending moments in the shaft is given by,

$$M_b = \sqrt{(Wl)^2 + (\mu\ Wl)^2} = \sqrt{1+\mu^2}\ Wl$$

and the resulting bending stress in the shaft, $f_b = \dfrac{M_b y}{I}$

where, y = the distance from the neutral axis to the outer-most fibre of the shaft

$= \dfrac{d}{2}$ for solid circular shaft of diameter, d

I = the moment of inertia of the shaft section

$= \dfrac{\pi d^4}{64}$ for solid circular shaft.

$$\text{Therefore, } f_b = \frac{M_b \frac{d}{2}}{\frac{\pi d^4}{64}} = \frac{32Wl\sqrt{1+\mu^2}}{\pi d^3},\ \text{N/m}^2 \text{ or Pa.}$$

The shear stress in the shaft is given by, $f_s = \frac{Ty}{I_p}$

where, I_p = the polar moment of inertia of the shaft section

$$= \frac{\pi d^4}{32} \quad \text{for solid circular section}$$

T = the maximum torque acting on the section of the shaft = μWr

Therefore, $$f_s = \frac{16\mu Wr}{\pi d^3}, \text{ N/m}^2 \text{ or Pa.}$$

Using the formula for combined stress, the maximum tensile (or compressive) stress in the shaft is given by

$$\sigma_n = \frac{1}{2} f_b \pm \frac{1}{2} \sqrt{f_b^2 + 4f_s^2}$$

$$= \frac{16W}{\pi d^3} [1 \sqrt{1 + \mu^2} \pm \sqrt{\mu^2 (r^2 + l^2) + l^2}], \text{ N/m}^2 \text{ or Pa}$$

(–ve sign for compressive) and the maximum shear stress in the shaft is given by,

$$\sigma_s = \frac{1}{2} \sqrt{f_b^2 + 4f_s^2}$$

$$= \frac{16W}{\pi d^3} \sqrt{\mu^2 (r^2 + l^2) + l^2}, \text{ N/m}^2 \text{ or Pa}$$

To use the above equations, W must be either known by way of assumption or by choice. The value of W should be that under which the engine would just be able to spin the wheels in lowest gear.

Since the tractive effort on each wheel, $F = T_e G\eta_t/2r$ (Refer section 31.3) and torque on each wheel, $T_w = T_e G\eta_t/2$, W (by the latter approach) = F/μ.

Using torsion formula, the minimum diameter can be calculated and at the bearing the equations for combined stresses can be used, assuming a diameter proportionately larger than the minimum and thus, both the allowable tensile and shear stresses in the shaft at the outboard bearing can be compared with the calculated values.

Axle Housing.

Semi-floating axle housing, like axle shafts, is subjected to both bending and torsion. The static load on the ourboard bearing depends on the driving force and the retarding force, and attains its maximum value when the wheel is either spun by the engine or locked by the brake. The bearing load produces a bending moment on the axle housing, which is zero at the centre of the bearing and increases uniformly to a maximum value at the centre of the spring seats and thereafter remains constant. As discussed in the case of axle shafts, the

maximum bending moment in this case may be taken as $\sqrt{1 + \mu^2}\, W_s\, l$,

where W_s = the static load on the bearing

and l = the distance between the centers of the bearing and the spring seat or pad.

The axle housing is subjected to torque under all operating conditions. The maximum torque can be taken as $\mu\, W_r$.

Now in this case also similar to axle shaft, $f_b = \frac{32\, W_s\, l \sqrt{1 + \mu^2}}{\pi d^3}$, Pa

$$f_s = \frac{16\, \mu\, W_r}{\pi d^3}, \text{ Pa}$$

$$\sigma_n = \frac{16}{\pi d^3}\left[W_s\, l \sqrt{1 + \mu^2} \pm \sqrt{(\mu\, W_r)^2 + (W_s l)^2 (1 + \mu^2)}\right], \text{ Pa}$$

$$\sigma_s = \frac{16}{\pi d^3} \sqrt{(\mu W_r)^2 + (W_s l)^2 (1 + \mu^2)}, \text{ Pa.}$$

Three-quarter-floating Axle-hub.

The road-wheel, in this case also, is bolted to the hub forming part of the axle-shaft. The outer end of the shaft and hub is supported by a bearing located over the axle-casing. The bearing in this case is positioned between the hub and the casing unlike between the axle and the casing as in the semifloating layout. The inner end of the half-shaft is splined to the final-drive assembly, same as the semi-floating half-shaft.

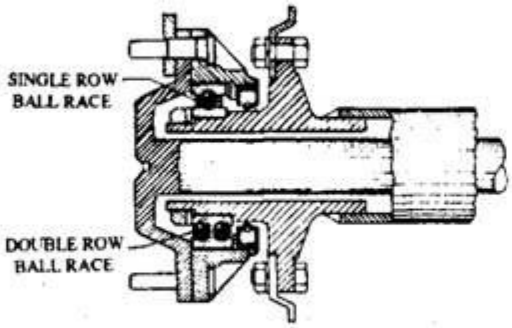
In the three-quarter-floating axle and hub arrangement, the driving torque is transmitted by the shaft, but the shear force and bending moment are absorbed by the tubular axle-casing through the hub bearing, only if the road-wheel and the hub bearing lie in the same vertical plane. Practically, a slight offset of wheel and bearing centres exist so that the hub is tilted relative to the axle-casing. This is resisted by the bearing, but incase this offset is large, the half-shaft provides the additional resistance. Horizontal loads, which create end-thrust, are opposed by the hub bearing and casing. However, the side-forces create a bending moment, which tends to twist the wheel relative to the axle-casing. This tilting tendency is resisted mostly by the hub bearing and partly by the axle-shaft.

A large tilting force therefore tends to overload the bearing if it is not adequately sized. A three-quarter-floating axle shown in was once very popular for cars and light commercial vehicles when semi-floating half-shafts frequently failed due to fracture, specifically in cold weather. However, due to availability of the compact, cheap and reliable semi-floating axle, the three-quarter-floating arrangement is rarely used today.

The half-shaft uses an upset-forged flange at the outer end, which is clamped to the bearing hub by the wheel studs. Either a large-diameter single-row or a double-row ball-race bearing is used, depending upon light- or heavy-duty applications. This bearing is located on the axle-casing and is secured in position by a large nut. The outer bearing track supports the hub. An oil-seal is placed at the back of the hub to prevent excess oil, coming from the final drive, to escape to the brakes from the hub.

Fully Floating Axle-hub.

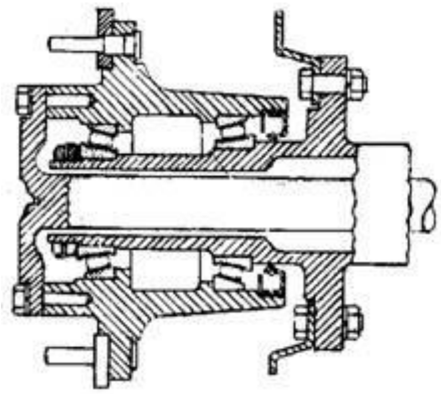
This axle-hub arrangement incorporates a flanged sleeve, which is positioned over the axle casing. The flange is provided to accommodate the road-wheel or wheels. Two bearings widely



Three-quarter-floating axle.

spaced are installed between the hub and the casing to support the hub assembly This provides am improvement on the first two types of hub support. The axle-shaft in this case takes only the turning-effort or torque. Both the vertical and horizontal load reactions are resisted by a pair of widely spaced taper-roller bearings installed on the axle-casing. The axle half-shaft, therefore, is free from all the loads except the torsional drive to the wheel.

Figure illustrates a fully floating axle-hub, based on a concept of the three-quarter-floating axle. The construction is such that the two hubs on their bearing rotate independently of the half-shaft. Studs connecting the shaft to the hub transmit the drive and when the nuts on these studs are removed, the shaft may be removed without jacking up the vehicle and without interfering with the load-supporting role of the hub. This layout, therefore, allows the vehicle to be towed with a broken half shaft. This is a larger and more expensive construction than both the other layouts. This is specifically suitable for all truck and heavy-duty vehicles employing live axles and for trailers using dead axles where torque and axle loads are greater. Depending upon the application, single or twin road-wheels are used.



Fully floating axle.

Bearing Loads.

The diagram showing the bearing loads due to side thrust on a full floating axle is presented in Fig.

In the figure,

F = lateral force at the rim of the wheel

r = radius of the wheel

L = distance between the centres of wheel bearing

Rl and i?2 = radial reactions of the wheel bearing on the wheel hub

P = the thrust reaction of the bearing

In practice, the ratio r/L = 4.

Considering the force in the horizontal and vertical directions,

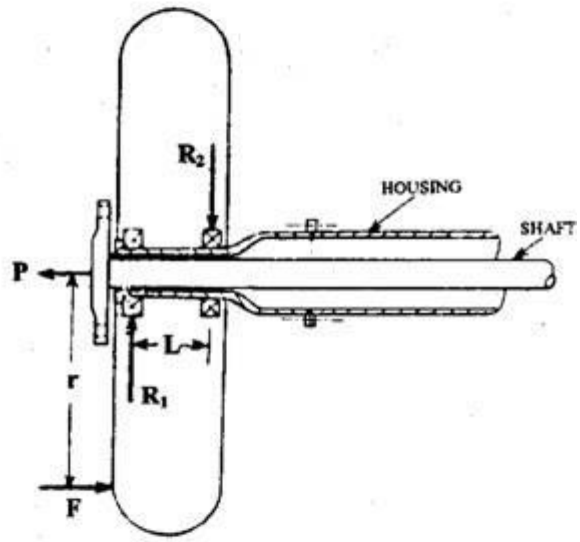
P = F andRi=R2

Taking moment, Fr = R\L and

Fr = R%L

Hence, Ri=R2=^F=4F

Rl adds to the normal static load on the bearing, whereas i?2 opposes it. The thrust load on the bearing is equal to the shock load, F on the wheel, whereas radial shock loads, Rl and R2 on the bearing are each approximately four times F.



Bearing loads due to side thrust on full-floating axle.

Axle Shaft.

In this case the axle shafts are subjected to torsional stresses only. The shaft is also of constant cross-section. Thus, the formula for the torsional strength of shafts can be applied to calculate the diameter of the shaft.

i.e. $$f_s = \frac{T}{I_p} y = \frac{16T}{\pi d^3}, \text{ N/m}^2 \text{ or } \text{P}_\text{a}$$

For better grades of alloy steel that are generally used for axle shafts, shear stress averages to 295 to 325 MPa.

Axle Housing.

The housing for full-floating axle transfers the load to the road wheels. This acts as a simply-supported beam at the ends with distributed loads. But for the sake of simplicity in calculations without appreciable error, it can be considered as a beam with concentrated loading at the centre of the spring seats.

Let W = the maximum load on one wheel

I = The distance between the centre planes of the wheel and the spring seat.

Then the bending moment produced by the reaction of the ground on the housing at the centre of the spring seat = Wl

Let CW = the maximum propelling or retarding force.

Then CW produces a bending moment = CWl and a torque = CWr

The combined bending moment $= WL\sqrt{1+C^2}$

Now the following set of equations can be obtained in a fashion similar to that discussed above under semi-floating axle.

$$f_b = \frac{32Wl\ \sqrt{1+C^2}}{\pi d^3},\ \text{Pa}$$

$$f_s = \frac{16\mu Wr}{\pi d^3},\ \text{Pa}$$

$$\sigma_n = \frac{16W}{\pi d^3}\ [l\sqrt{1+C^2} \pm \sqrt{C^2(r^2+l^2)+l^2}],\ \text{Pa}$$

$$\sigma_s = \frac{16W}{\pi d^3}\ \sqrt{C^2(r^2+l^2)+l^2},\ \text{Pa}$$

GEAR BOX

Resistance to motion:

This is the resistance a vehicle faces while attempting to move from a stall condition or while accelerating. This resistance must be overcome by the power plant of the engine in order to sustain motion. When the power produced is smaller than the resistance to motion, the vehicle will gradually slow down. We must have experienced the slowing down of bicycles if we stop pedalling. The bicycle also slows down if we go uphill or if wind blows from front. A poorly inflated tire also causes the vehicle to groan more and slow down. These are the resistances that force the vehicle to slow down under their effect.

Broadly the resistances can be categorized into the following categories:

- Aerodynamic drag
- Gradient resistance
- Rolling resistance
- Inertia

All the above produce a restraining force working against the tractive force. The tractive force must be greater than or equal to the resistive forces in order to maintain a sustainable motion. We can balance them as

$F = F_{req} = F_A + F_G + F_R + F_I = R_r + R_a + R_g$

Where

F_A= Force due to air resistance

F_G = Force due to gradient of a slope

F_G = Force due to rolling resistance

F_I = Force due to moving or static inertia

The last one F_I comes into the picture only when the vehicle accelerates or decelerates, while the first three always offer a resistance even when the vehicle is moving at a constant speed.

Air resistance/ Aerodynamic drag:

When a body travels within a dense medium, the molecules of the medium collide with the moving object and thereby absorb some of the energy. This is felt as a resistance to the moving object. If the medium is denser, then the resistance is more. Also when the object moves at a faster speed, the resistance increases proportionately. Mathematically it can be expressed as:

$F_A = -\frac{1}{2} \times C_d \times P \times V^2$

Where

C_d = Co−efficient of discharge

P = Pressure

V = Velocity of the vehicle

Gradient resistance:

When the vehicle travels uphill, a component of its weight works in a direction opposite to its motion. If some energy is not supplied to overcome this backward force, then the vehicle would slow down, stall and roll backwards. If the vehicle is trading uphill at a slope of θ, then the weight of the vehicle, W has two components: one perpendicular to the road surface (with a value $W \cdot \cos\theta$) and the other along the road surface (with a value $W \cdot \sin\theta$). The component along the road surface is the one that tries to restrict the motion.

The gradient resistance is given by: $F_G = W \cdot \sin\theta$

Rolling resistance:

When a vehicle rolls, it rolls with its tires in contact with the road surface. The relative motion of two hard surfaces produces a friction. Further, neither the road, nor the tire are perfectly rigid. Hence, both flex under the load slightly. As there is a gradual deformation at the contact between the road and the tire, greatest at the bottom most point and least at the entry and exit points, the slip of the tire w.r.t. the road produces another type of loss of energy which results in a resistance.

Rolling resistance is composed of the following components:

- Tire Rolling resistance: $F_{R,T}$
- Road rolling resistance: $F_{R,Tr}$
- Resistance due to tire slip angle: $F_{R,\alpha}$
- Resistance due to bearing friction and residual braking: $F_{R,fr}$

Hence the rolling resistance offered may be written as:

$F_R = F_{R,T} + F_{R,Tr} + F_{R,\alpha} + F_{R,fr}$

The tire rolling resistance $F_{R,T}$ is a result of the resistance due to flexure of the tire, air resistance on the tire and friction of tire with the road. These three can be summed up and written as:

$F_{R,T} = F_{R.T.flex} + F_{R.T.A} + F_{R.T.fr}$.

In a simplified manner the total rolling resistance can be related to the vertical load on the wheels and can be written as:

Co−efficient of rolling friction, $k_R = {}^{F}{}_{R}/_{FZ.w}$

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

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 = $\frac{\mathrm{T_B}}{\mathrm{T_A}}\frac{\mathrm{T_J}}{\mathrm{T_I}} = 3.3$ 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

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

= 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 $5.2m^2$. 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 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 $V = \frac{2\pi Nr}{G} m/min$

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

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+0.1685$\boldsymbol{V^2}$) 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 0.0827$\boldsymbol{V^2}$, transmission efficiency 88% in top speed, car weights 19934N when fully loaded. Calculate

a) The bkW required for a top speed of 144 km/hr.
b) The acceleration in $\boldsymbol{m/s^2}$ at 48 km/hr, assuming the torque at 48 km/hr in the top gear 25% more than at 144km/hr.

c) The bkW 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 $\boldsymbol{m/s^2}$.

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 bkw $= \frac{RV}{3600\, \eta_t} = \frac{2173.5 \times 144}{0.88 \times 3600} = 98.8\ KW.$

(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,

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

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 2.23$\boldsymbol{m^2}$ 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.

$$\textit{generally the gear ratios are in geometric progression then}$$

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

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

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

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

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

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

$$-----------$$