Subject :vehicle structures design	:
Weekly Hours: Theoretical: 2	**2**: :
Tutorial:	:
Experimental :	:
Units: 2	**2** :

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VEHICLE STRUCTURE

Integral body construction

The integral or unitary body structure of a car can be considered to be made in the form of three box compartments; the middle and largest compartment stretching between the front and rear road wheel axles provides the passenger space, the extended front box built over and ahead of the front road wheels enclosing the engine and transmission units and the rear box behind the back axle providing boot space for luggage.

These box compartments are constructed in the form of a framework of ties (tensile) and struts (compressive), pieces (Figure 1) made from rolled sheet steel pressed into various shapes such as rectangular, triangular, trapezium, top-hat or a combination of these to form closed box thin gauge sections. These sections are designed to resist direct tensile and compressive or bending and torsional loads, depending upon the positioning of the members within the structure.
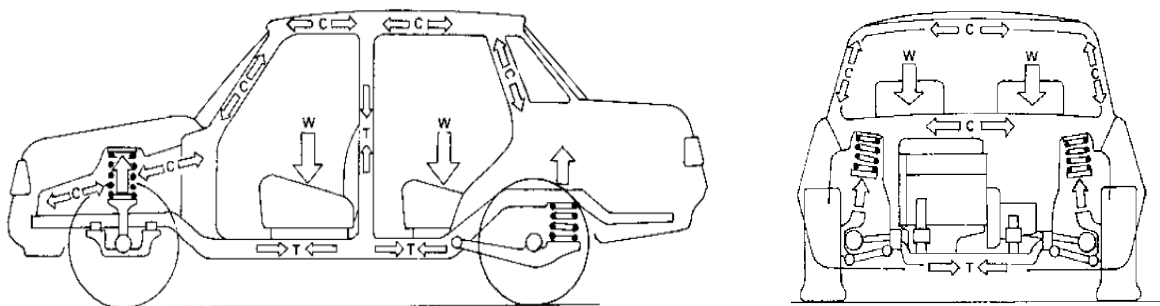

Figure 1 Structural tensile and compressive loading of car body.

Description and function of body components *(*Figure 2)

The major individual components comprising the body shell will now be described separately under the following subheadings:

1. Window and door pillars
2. Windscreen and rear window rails
3. Cantrails
4. Roof structure
5. Upper quarter panel or window
6. Floor seat and boot pans
7. Central tunnel
8. Sills
9. Bulkhead
10. Scuttle
11. Front longitudinals
12. Front valance
13. Rear valance

Window and door pillars (Figure 2 (3, 5, 6, and 8)) Window screen and door pillars are identified by a letter coding; the front windscreen to door pillars are referred to as *A post,* the centre side door pillars as *Be post* and the rear door to quarter panel as *D post.* These are illustrated in Figure 2.
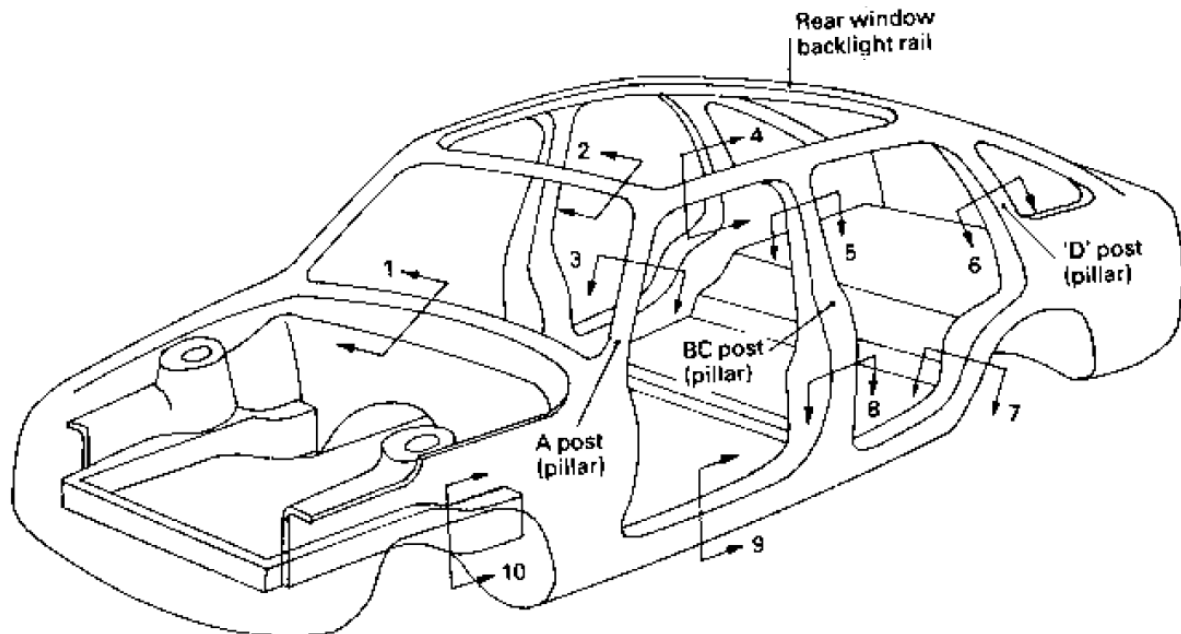

Figure 2   Load bearing body

Windscreen and rear window rails (Figure 2(2)) These box-section rails span the front window pillars and rear pillars or quarter panels depending upon design, so that they contribute to the resistance opposing transverse sag between the wheel track by acting as compressive members. The other function is to support the front and rear ends of the roof panel. The undersides of the rails also include the glazing channels.

Cantrails (Figure 2(4)) Cantrails are the horizontal members which interconnect the top ends of the vertical A and BC or BC and D door pillars (posts). These rails form the side members which make up the rectangular roof framework and as such are subjected to compressive loads. Therefore, they are formed in various box-sections which offer the greatest compressive resistance with the minimum of weight and blend in with the roofing. A drip rail (Figure 2(4)) is positioned in between the overlapping roof panel and the cantrails, the joins being secured by spot welds.

Roof structure (Figure 2) The roof is constructed basically from four channel sections which form the outer rim of the slightly dished roof panel. The rectangular outer roof frame acts as the compressive load bearing members. Torsional rigidity to resist twist is maximized by welding the four corners of the channel-sections together. The slight curvature of the roof panel stiffens it, thus preventing wrinkling and the collapse of the unsupported centre region of the roof panel. With large cars, additional cross-rail members may be used to provide more roof support and to prevent the roof crushing in should the car roll over.

Upper quarter panel or window (Figure 2(6)) This is the vertical side panel or window which occupies the space between the rear side door and the rear window. Originally the quarter panel formed an important part of the roof support, but improved pillar design and the desire to maximize visibility has either replaced them with quarter windows or reduced their width, and in some car models they have been completely eliminated.

Floor seat and boot pans (Figure 1.3) These constitute the pressed rolled steel sheeting shape to enclose the bottom of both the passenger and luggage compartments. The horizontal spread-out pressing between the bulkhead and the heel board is called the *floor pan,* whilst the raised platform over the rear suspension and wheel arches is known as the *seat* or *arch pan.* Thus joins onto a lower steel pressing which supports luggage and is referred to as the *boot pan.*
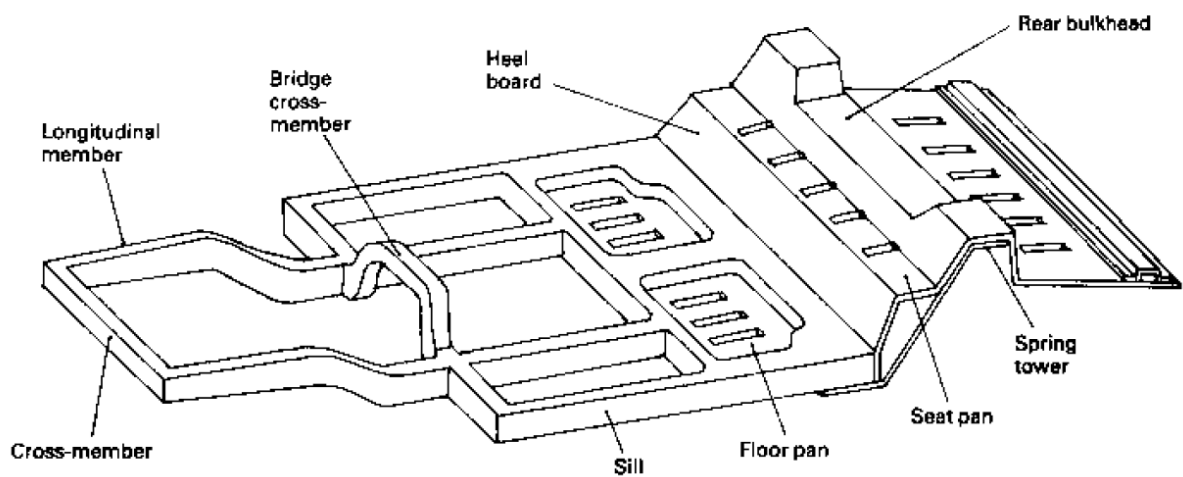
Central tunnel (Figure 3(a and b)) This is the curved or rectangular hump positioned longitudinally along the middle of the floor pan. Originally it was a necessary evil to provide transmission space for the gearbox and propeller shaft for rear wheel drive, front-mounted engine cars, but since the chassis has been replaced by the integral box section shell, it has been retained with front wheel drive, front-mounted engines as it contributes considerably to the bending rigidity of the floor structure. Its secondary function is now to house the exhaust pipe system and the hand brake cable assembly.
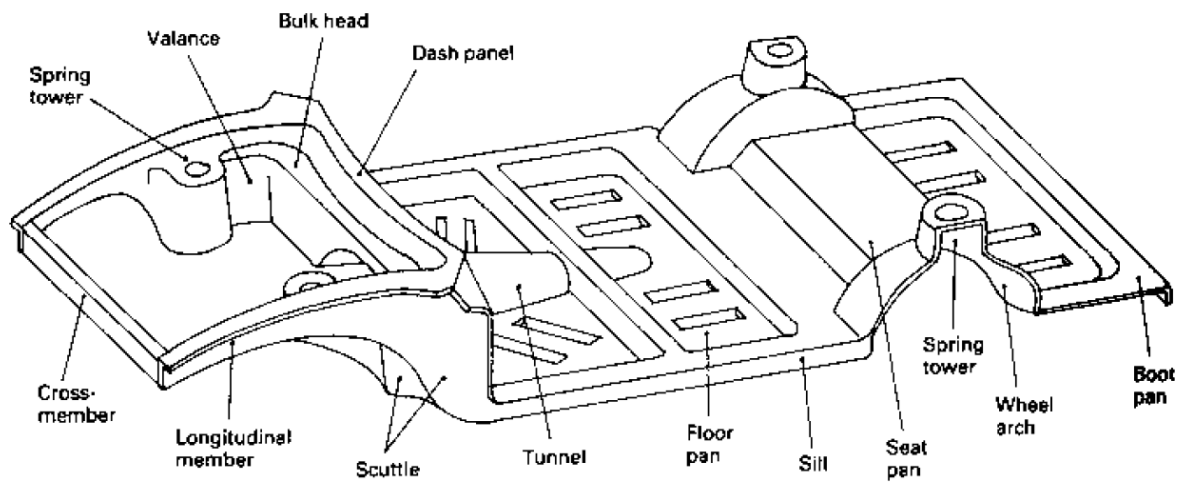
(a) Small central tunnel, sills, front valance, rear wheel arches and all round spring towers
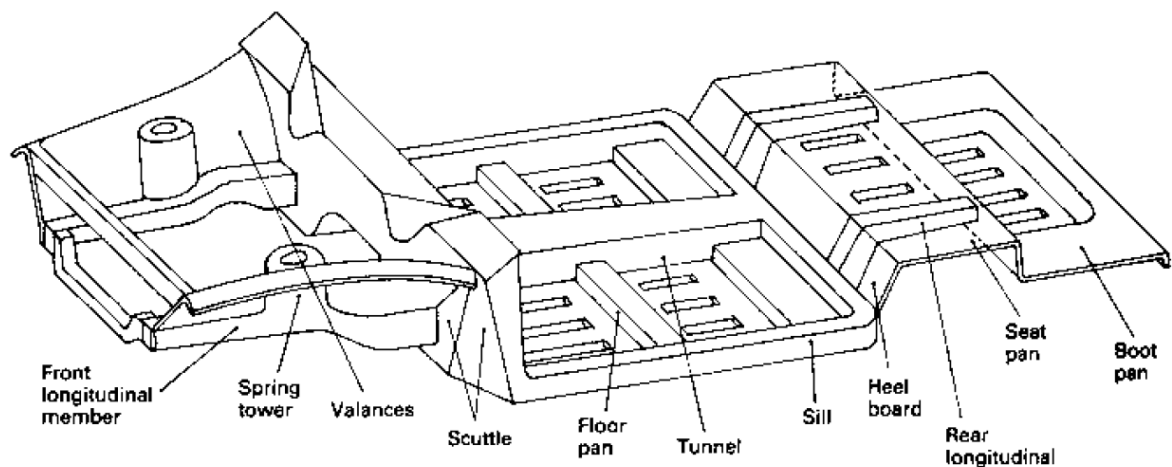
(b) Large central tunnel, sills, front valance and spring tower with rear box-section reinforced seat pan

(c) Longitudinal front members, sills, rear seat pan with bulkhead and spring towers

Figure 3 Platform chassis

Central tunnel (Figure 3(a and b)) This is the curved or rectangular hump positioned longitudinally along the middle of the floor pan. Originally it was a necessary evil to provide transmission space for the gearbox and propeller shaft for rear wheel drive, front-mounted engine cars, but since the chassis has been replaced by the integral boxsection shell, it has been retained with front wheel drive, front-mounted engines as it contributes considerably to the bending rigidity of the floor structure. Its secondary function is now to house the exhaust pipe system and the hand brake cable assembly.

Sills (Figures 2(9) and 3(a, b and c)) These members form the lower horizontal sides of the car body which spans between the front and rear road-wheel wings or arches. To prevent body sag between the wheelbase of the car and lateral bending of the structure, the outer edges of the floor pan are given support by the side sills. These sills are made in the form of either single or double box-sections (Fig. 1.2(9)). To resist the heavier vertical bending loads they are of relatively deep section. Open-top cars, such as convertibles, which do not receive structural support from the roof members, usually have extra deep sills to compensate for the increased burden imposed on the underframe.

Bulkhead (Figures 2(1) and 3(a and b)) This is the upright partition separating the passenger and engine compartments. Its upper half may form part of the dash panel which was originally used to display the driver's instruments. Some body manufacturers refer to the whole partition between engine and passenger compartments as the dash panel.

Scuttle (Figure 3(a and b)) This can be considered as the panel formed under the front wings which spans between the rear end of the valance, where it meets the bulkhead, and the door pillar and wing. The lower edge of the scuttle will merge with the floor pan so that in some cases it may form part of the toe board on the passenger compartment side.

Front longitudinals (Figures 2(10) and 3(a and b)) These members are usually upswept box-section members, extending parallel and forward from the

bulkhead at floor level. Their purpose is to withstand the engine mount reaction and to support the front suspension or subframe. A common feature of these members is their ability to support vertical loads in conjunction with the valances. However, in the event of a head-on collision, they are designed to collapse and crumble within the engine compartment so that the passenger shell is safeguarded and is not pushed rearwards by any great extent.

Front valance (Figures 2 and 3(a and b)) These panels project upwards from the front longitudinal members and at the rear join onto the wall of the bulkhead. The purpose of these panels is to transfer the upward reaction of the longitudinal members which support the front suspension to the bulkhead. Simultaneously, the longitudinals are prevented from bending sideways because the valance panels are shaped to slope up and outwards towards the top.

Rear valance (Figure 2(7)) This is generally considered as part of the box-section, forming the front half of the rear wheel arch frame and the panel immediately behind which merges with the heel board and seatpan panels. These side inner-side panels position the edges of the seat pan to its designed side profile and thus stiffen the underfloor structure above the rear axle and suspension.

Transmission System Design

MANUAL GEARBOXES

Power from a petrol or diesel reciprocating engine transfers its power in the form of torque and angular speed to the deriving wheels of the vehicle to produce motion. The object of the gearbox is to enable the engine's turning effect and its rotational speed output to be adjusted by choosing a range of under- and overdrive gear ratios so that the vehicle responds to the driver's requirements within the limits of the various road conditions. An insight of the forces opposing vehicle motion and engine performance characteristics which provide the background to the need for a *wide* range of gearbox designs used for different vehicle applications will now be considered.

Resistance to vehicle motion

To keep a vehicle moving, the engine has to develop sufficient power to overcome the opposing road resistance power, and to pull away from a standstill or to accelerate a reserve of power in addition to that absorbed by the road resistance must be available when required.

Road resistance is expressed as *tractive resistance* (kN). The propelling thrust at the tyre to road interface needed to overcome this resistance is known as *tractive effect* (kN) (Fig. 3.1). For matching engine power output capacity to the opposing road resistance it is sometimes more convenient to express the opposing resistance to motion in terms of *road resistance power.*

The road resistance opposing the motion of the vehicle is made up of three components as follows:

1. Rolling resistance
2. Air resistance
3. Gradient resistance

Rolling resistance (Figure 1) Power has to be expended to overcome the restraining forces caused by the deformation of tyres and road surfaces. Secondary causes of rolling resistance are wheel bearing, oil seal friction and the churning of the oil in the transmission system. It has been found that the flattening distortion of the tyre casing at the road surface interface consumes more energy as the wheel speed increases and therefore the rolling resistance will also rise slightly as shown in Figure 1. Factors which influence the magnitude of the rolling resistance are the laden weight of the vehicle, type of road surface, and the design, construction and materials used in the manufacture of the tyre.
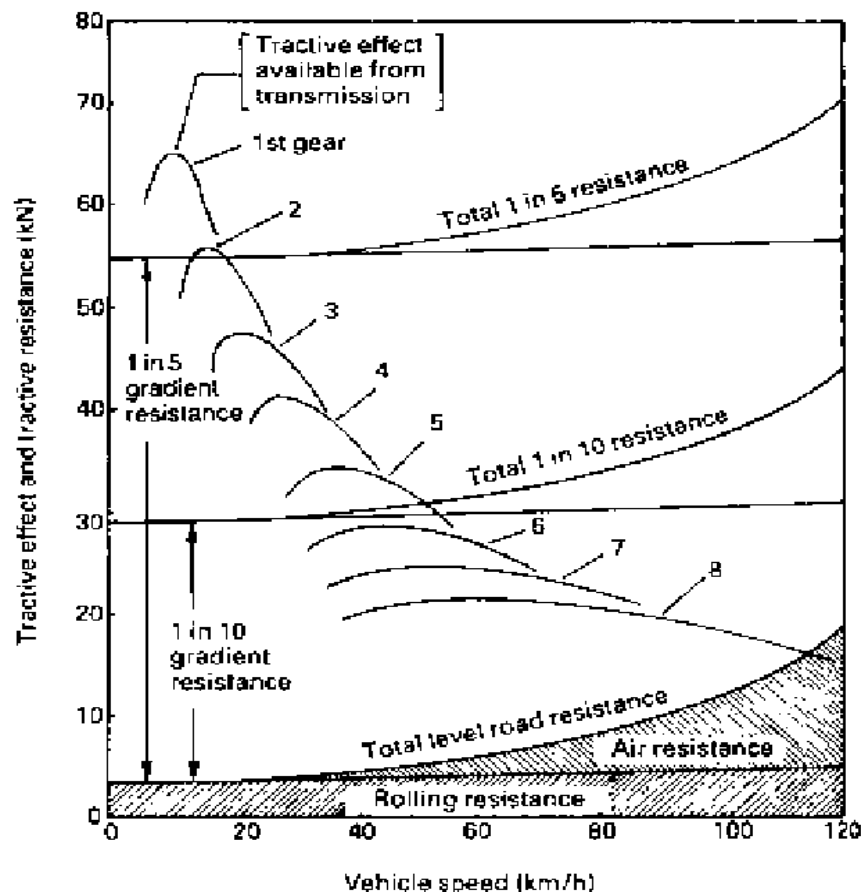

Figure 1 Vehicle tractive resistance and effort performance chart.

Air resistance (Figure 1) Power is needed to counteract the tractive resistance created by the vehicle moving through the air. This is caused by air being

pushed aside and the formation of turbulence over the contour of the vehicle's body. It has been found that the air resistance opposing force and air resistance power increase with the square and cube of the vehicle's speed respectively. Thus at very low vehicle speeds air resistance is insignificant, but it becomes predominant in the upper speed range. Influencing factors which determine the amount of air resistance are frontal area of vehicle, vehicle speed, shape and streamlining of body and the wind speed and direction.

Gradient resistance (Figure 1) Power is required to derive a vehicle and its load not only along a level road but also up any gradient likely to be encountered. Therefore, a reserve of power must be available when climbing to overcome the potential energy produced by the weight of the vehicle as it is progressively lifted. The gradient resistance opposing motion, and therefore the tractive effect or power needed to drive the vehicle forward, is directly proportional to the laden weight of the vehicle and the magnitude of gradient. Thus driving up a slope of 1 in 5 would require twice the reserve of power to that needed to propel the same vehicle up a gradient of 1 in 10 at the same speed, as shown in Figure.

Power to weight ratio

When choosing the lowest and highest gearbox gear ratios, the most important factor to consider is not just the available engine power but also the weight of the vehicle and any load it is expected to derive. Consequently, the power developed per unit weight of laden vehicle has to be known. This is usually expressed as the *power to weight ratio.*

$$Power\ to\ weight\ ratio = \frac{Brake\ power\ developed}{Laden\ weight\ of\ vehicle}$$

There is a vast difference between the power to weight ratio for cars and commercial vehicles which is shown in the following examples.

Example: Determine the power to weight ratio for the following modes of transport:

1. A car fully laden with passengers and luggage weighs 1.2 tonne and the maximum power produced by the engine amounts to 120 kW.
2. A fully laden articulated truck weighs 38 tonne and a 290 kW engine is used to propel this load.

Solution

a) Power to weight ratio $= \frac{120}{1.2} = 100 \, kW/tonne$

b) Power to weight ratio $= \frac{290}{38} = 7.6 \, kW/tonne$

Ratio span

Another major consideration when selecting gear ratios is deciding upon the steepest gradient the vehicle is expected to climb (this may normally be taken as 20%, that is 1 in 5) and the maximum level road speed the vehicle is expected to reach in top gear with a small surplus of about 0.2% gradeability.

The two extreme operating conditions just described set the highest and lowest gear ratios. To fix these conditions, the ratio of road speed in highest gear to road speed in lowest gear at a given engine speed should be known. This quantity is referred to as the *ratio span.*

$$Ratio \; span = \frac{\text{Road speed in highest gear}}{\text{Road speed in lowest gear}}$$

(both road speeds being achieved at similar engine speed).

Engine torque rise and speed operating range (Figure 2)

Commercial vehicle engines used to pull large loads are normally designed to have a positive torque rise curve, that is from maximum speed to peak torque with reducing engine speed the available torque increases (Figure 2). The amount of engine torque rise is normally expressed as a percentage of the peak torque from maximum speed (rated power) back to peak torque.

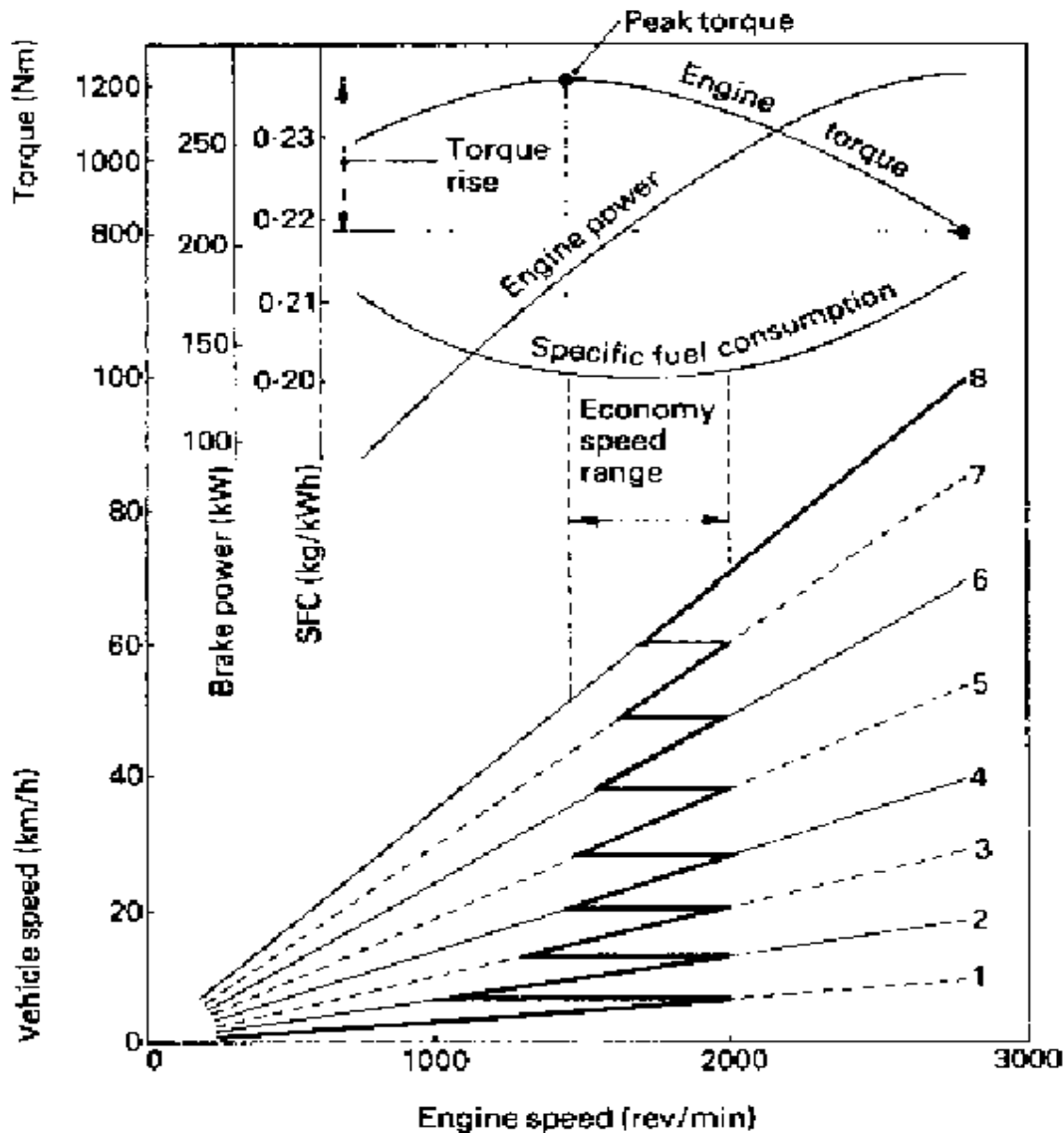$$\%torque\ rise = \frac{Maximum\ speed\ torque}{Peak\ torque} \times 100$$

Fig.3.2 Engine performance and gear split chart for an eight speed gearbox

The torque rise can be shaped depending upon engine design and taking into account such features as naturally aspirated, resonant induction tuned, turbocharged, turbocharged with intercooling and so forth. Torque rises can vary from as little as 5 to as high as 50%, but the most common values for torque rise range from 15 to 30%.

A large torque rise characteristic raises the engine's operating ability to overcome increased loads if the engine's speed is pulled down caused by changes in the road conditions, such as climbing steeper gradients, and so tends to restore the original running conditions. If the torque rise is small it cannot help as a buffer to supplement the high torque demands and the engine speed will rapidly fade. Frequent gear changes therefore become necessary compared to engines operating with high torque rise characteristics. Once the engine speed falls below peak torque, the torque rise becomes negative and the pulling ability of the engine drops off very quickly.

Vehicle driving technique should be such that engines are continuously driven between the speed range of peak torque and governed speed. The driver can either choose to operate the engine's speed in a range varying just below the maximum rated power to achieve maximum performance and journey speed or, to improve fuel economy, wear and noise, within a speed range of between 200 to 400 rev/min on the positive torque rise side of the engine torque curve that is in a narrow speed band just beyond peak torque. Figure 2 shows that the economy speed range operates with the specific fuel consumption at its minimum and that the engine speed band is in the most effective pulling zone.

Five speed and reverse doable stage synchromesh gearbox (Figure 3)

With this arrangement of a five speed double stage gearbox, the power input to the first motion shaft passes to the layshaft and gear cluster via the first stage pair of meshing gears. Rotary motion is therefore conveyed to all the second stage layshaft and mainshaft gears (Figure 3). Because each pair of second stage gears has a different size combination, a whole range of gear ratios are provided. Each mainshaft gear (whilst in neutral) revolves on the mainshaft but at some relative speed to it. Therefore, to obtain output powerflow, the selected mainshaft gear has to be locked to the mainshaft. This then completes the flow

path from the first motion shaft, first stage gears, second stage gears and finally to the mainshaft.
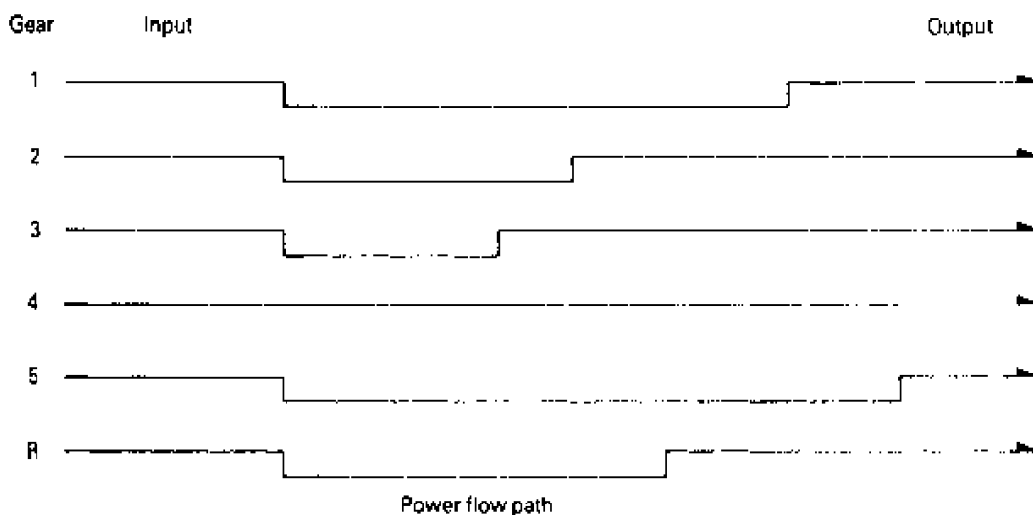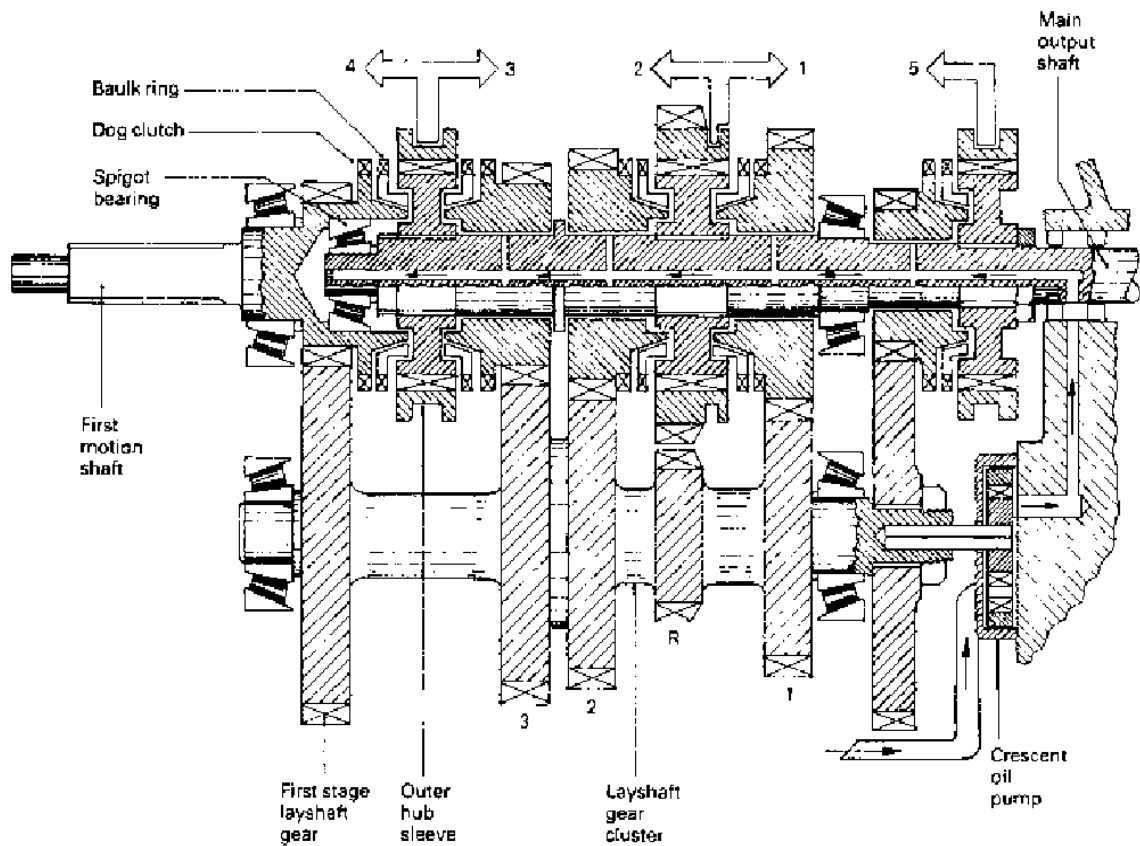

Figure 3 Five speed and reverse double stage synchromesh gearbox

In this example the fifth gear is an overdrive gear so that to speed up the mainshaft output relative to the input to the first motion shaft, a large layshaft fifth gear wheel is chosen to mesh with a much smaller mainshaft gear.

For heavy duty operations, a forced feed lubrication system is provided by an internal gear crescent type oil pump driven from the rear end of the layshaft (Figure 3). This pump draws oil from the base of the gearbox casing, pressurizes it and then forces it through a passage to the mainshaft. The oil is then transferred to the axial hole along the centre of the mainshaft by way of an annular passage formed between two nylon oil seals. Lubrication to the mainshaft gears is obtained by radial branch holes which feed the rubbing surfaces of both mainshaft and gears.

OVERDRIVE CONSIDERATIONS

Power is essential to propel a vehicle because it is a measure of the rate of doing work, that is, the amount of work being developed by the engine in unit time. With increased vehicle speed, more work has to be done by the engine in a shorter time. The characteristic power curve over a speed range for a petrol engine initially increases linearly and fairly rapidly. Towards mid-speed the steepness of the power rise decreases until the curve reaches a peak. It then bends over and declines with further speed increase due to the difficulties experienced in breathing at very high engine speeds (Figure 4).

A petrol engined car is usually geared so that in its normal direct top gear on a level road the engine speed exceeds the peak power speed by about 10 to 20% of this speed. Consequently, the falling power curve will intersect the road resistance power curve. The point where both the engine and road resistance power curves coincide fixes the road speed at which all the surplus power has been absorbed. Therefore it sets the maximum possible vehicle speed.

By selecting a 20% overdrive top gear, say, the transmission gear ratios can be so chosen that the engine and road resistance power curves coincide at peak engine power (Fig. 3.26). The undergearing has thus permitted the whole of the engine power curve to be shifted nearer the opposing road resistance power curve so that slightly more engine power is being utilized when the two curves intersect. As a result, a marginally higher maximum vehicle speed is achieved. In other words, the engine will be worked at a lower speed but at a higher load factor whilst in this overdrive top gear.
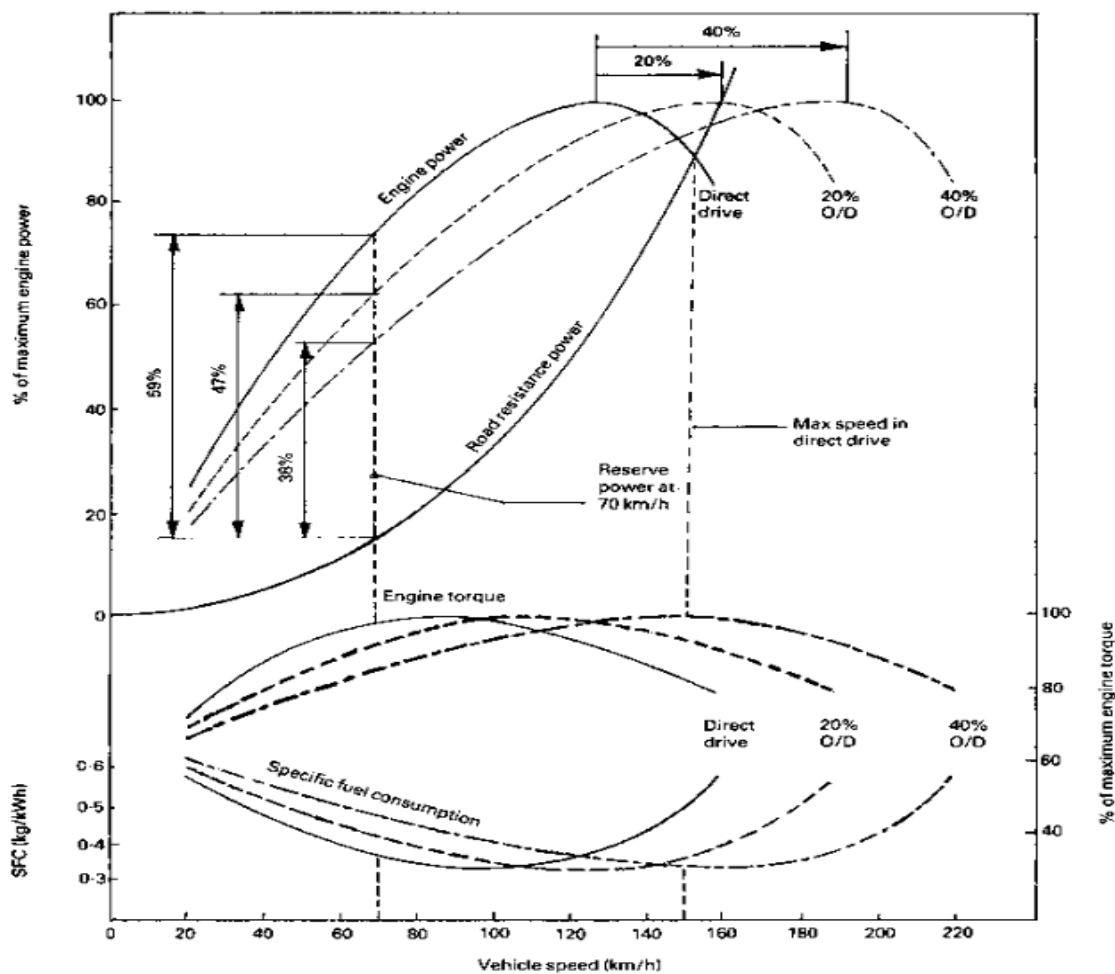

Figure 4 Effect of over and undergearing on vehicle performance

If the amount of overdrive for top gear is increased to 40%, the engine power curve will be shifted so far over that it intersects the road resistance power curve before peak engine power has been obtained (Figure 4) and therefore the maximum possible vehicle speed cannot be reached.

Contrasting the direct drive 20% and 40% overdrive with direct drive top gear power curves with respect to the road resistance power curve at 70 km/h, as an example, it can be seen (Figure 4) that the reserve of power is 59%, 47% and 38% respectively. This surplus of engine power over the power absorbed by road resistance is a measure of the relative acceleration ability for a particular transmission overall gear ratio setting.

A comparison of the three engine power curves shows that with direct drive top gear the area in the loop made between the developed and opposing power curves is the largest and therefore the engine would respond to the changing driving conditions with the greatest flexibility.

If top gear is overdriven by 20%, as shown in Figure 4, the maximum engine power would be developed at maximum vehicle speed. This then provides the highest possible theoretical speed, but the amount of reserve power over the road resistance power is less, so that acceleration response will not be as rapid as if a direct drive top gear is used. Operating under these conditions, the engine speed would never exceed the peak power speed and so the engine could not 'over-rev', and as a result engine wear and noise would be reduced. Benefits are also gained in fuel consumption as shown in Figure 4. The lowest specific fuel consumption is shifted to a higher cruising speed which is desirable on motorway journeys.

Indulging in an excessive 40% overdrive top gear prevents the engine ever reaching peak power so that not only would maximum vehicle speed be reduced compared to the 20% overdrive gearing, but the much smaller difference in power developed to power dissipated shown on the power curves would severely reduce the flexibility of driving in this gear. It therefore becomes essential for more frequent down changes with the slightest fall-off in road speed. A further disadvantage with excessive overdrive is that the minimum specific fuel consumption would be shifted theoretically to the engine's upper speed range which in practice could not be reached.

An analysis of matching an engine's performance to suit the driving requirements of a vehicle shows that with a good choice of undergearing in top gear for motorway cruising conditions, benefits of prolonged engine life, reduced noise, better fuel economy and less driver fatigue will be achieved. Another major consideration is the unladen and laden operation of the vehicle, particularly if it is to haul heavy loads. Therefore most top gear overdrive ratios are arrived at as a compromise.

SETTING GEAR RATIOS

Matching the engine's performance characteristics to suit a vehicle's operating requirements is provided by choosing a final drive gear reduction and then selecting a range of gear ratios for maximum performance in terms of the ability to climb gradients, achievement of good acceleration through the gears and ability to reach some predetermined maximum speed on a level road.

Setting top gear

To determine the maximum vehicle speed, the engine brake power curve is superimposed onto the power requirement curve which can be plotted from the sum of both the rolling (R_r) and air (R_a) resistance covering the entire vehicle's speed range.

The total resistance opposing motion at any speed is given by:

$$Total\ Resistance = R_r + R_a = C_r\,W + C_d\,A\,v^2$$

where C_r = coefficient of rolling resistance

C_d = coefficient of aerodynamic resistance (drag)

W = gross vehicle weight (N)

A = projected frontal area of vehicle (m^2)

v = speed of vehicle (km/h)

The top gear ratio is chosen so that the maximum road speed corresponds to the engine speed at which maximum brake power is obtained (or just beyond) (Figure 5).
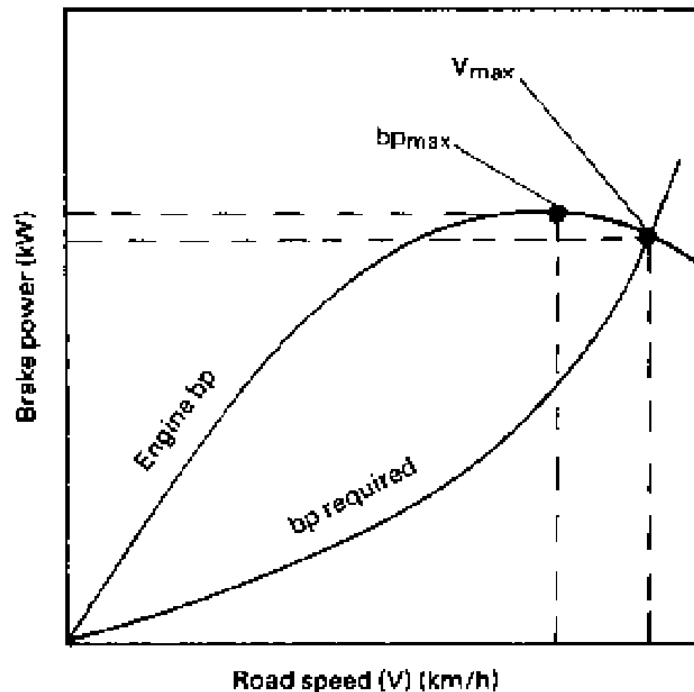

Figure 5 Relationship of power developed and road power required over the vehicle's speed range

Gearing is necessary to ensure that the vehicle speed is at a maximum when the engine is developing approximately peak power.

Thus   Linear wheel speed = Linear road speed

$$\frac{1}{G_F}\left(N\frac{2\pi}{60}\right)\left(\frac{d}{2}\right) = v\frac{1000}{3600}$$

$\therefore$ Final drive gear ratio $G_F = 0.06\dfrac{\pi d\, N}{v}$

where G_F = final drive gear ratio, N = engine speed (rev/min), d = effective wheel diameter (m) and v = road speed at which peak power is developed (km/h).

Example A vehicle is to have a maximum road speed of 150 km/h. If the engine develops its peak power at 6000 rev/min and the effective road wheel diameter is 0.54m, determine the final drive gear ratio.

Solution

$$G_F = 0.06 \frac{\pi d\ N}{v}$$

$$= 0.06 \frac{\pi \times 0.54 \times 6000}{150}$$

$$= 4.07 : 1$$

Setting bottom gear

The maximum payload and gradient the vehicle is expected to haul and climb determines the necessary tractive effort, and hence the required overall gear ratio. The greatest gradient that is likely to be encountered is decided by the terrain the vehicle is to operate over. This normally means a maximum gradient of 5 to 1 and in the extreme 4 to 1. The minimum tractive effort necessary to propel a vehicle up the steepest slope may be assumed to be approximately equivalent to the sum of both the rolling and gradient resistances opposing motion.

The rolling resistance opposing motion may be determined by the formula:

$$R_r = C_r\ W$$

Average values for the coefficient of rolling resistance for different types of vehicles travelling at very slow speed over various surfaces have been determined and are shown in the following table.

Table 1 Average values of coefficient of rolling resistance

Vehicle type	Coefficient of rolling resistance (C_r)		
	Concrete	Medium hard soil	Sand
Passenger Car	0.015	0.08	0.30
Trucks	0.012	0.06	0.25
Tractors	0.02	0.04	0.20

Likewise, the gradient resistance R_g (Figure 6) opposing motion may be determined by the formula:
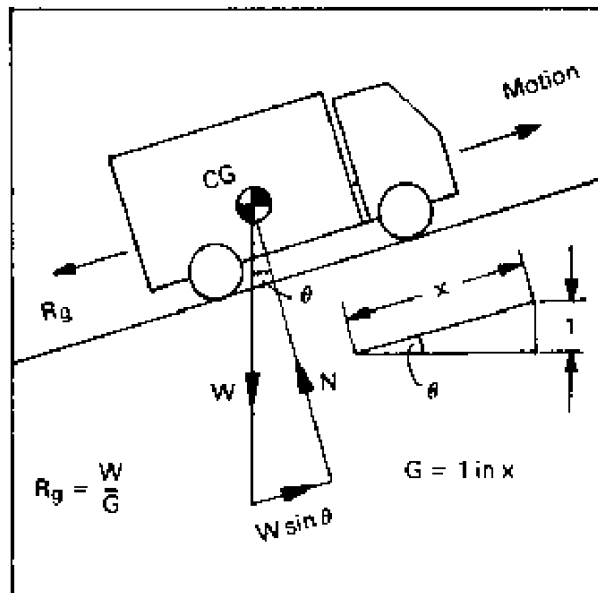
$$R_g = W \sin \theta$$

Figure 6 Gradient resistances to motion

Tractive effort = Resisting forces opposing motion

$$E = R_r + R_g$$

Where E is the tractive effort (N)

Once the minimum tractive effort has been calculated, the bottom gear ratio can be derived in the following way:

Driving torque = Available torque

$$E\,r = T\,G_B\,G_F\,\eta_m$$

$$\therefore Bottom\;gear\;ratio\;G_B = \frac{E\,r}{T\,G_F\,\eta_m}$$

where G_F = final drive gear ratio

G_B = bottom gear ratio

η_m = mechanical efficiency

E = tractive effort (N)

T = maximum engine torque (Nm)

r = effective road wheel radius (m)

Example A vehicle weighing 1500kg has a coefficient of rolling resistance of 0.015. The transmission has a final drive ratio 4.07:1 and an overall mechanical efficiency of 85%.

If the engine develops a maximum torque of 100Nm, as shown in figure 7, and the effective road wheel radius is 0.27m, determine the gearbox bottom gear ratio.

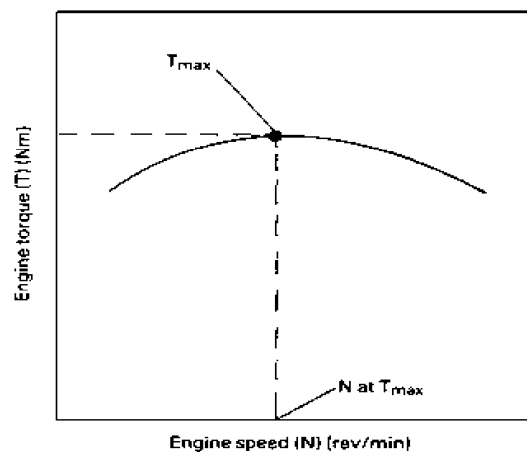Assume the steepest gradient to be encountered is a one in four.

Figure 7 Engine torque to speed characteristics

Solution

$$R_r = C_r\,W = 0.015 \times (1500 \times 10) = 225\,N$$

$$R_g = \frac{1500 \times 10}{4} = 3750\,N$$

$$E = R_r +\ R_g = 3750 + 225\ = 3975N$$

$$G_B = \frac{E\,r}{T\,G_F\,\eta_m} = \frac{3975 \times 0.27}{100 \times 4.07 \times 0.85}$$

$$= 3.1\!:\!1$$

Setting intermediate gear ratios

Ratios between top and bottom gears should be spaced in such a way that they will provide the tractive effort-speed characteristics as close to the ideal as possible. Intermediate ratios can be best selected as a first approximation by using a geometric progression. This method of obtaining the gear ratios requires the engine to operate within the same speed range in each gear, which is normally selected to provide the best fuel economy.

Consider the engine to vehicle speed characteristics for each gear ratio as shown (Figure 8). When changing gear the engine speed will drop from the highest N_H to the lowest N_L without any change in road speed, i.e. v_1, v_2, v_3 etc.

Let $G_1 = 1^{st}$ overall gear ratio

$G_2 = 2^{nd}$ overall gear ratio

$G_3 = 3^{rd}$ overall gear ratio

$G_4 = 4^{th}$ overall gear ratio

$G_5 = 5^{th}$ overall gear ratio

Where Overall gear ratio $= \dfrac{Engine\ speed\ (rev/min)}{Road\ wheel\ speed\ (rev\ /min)}$
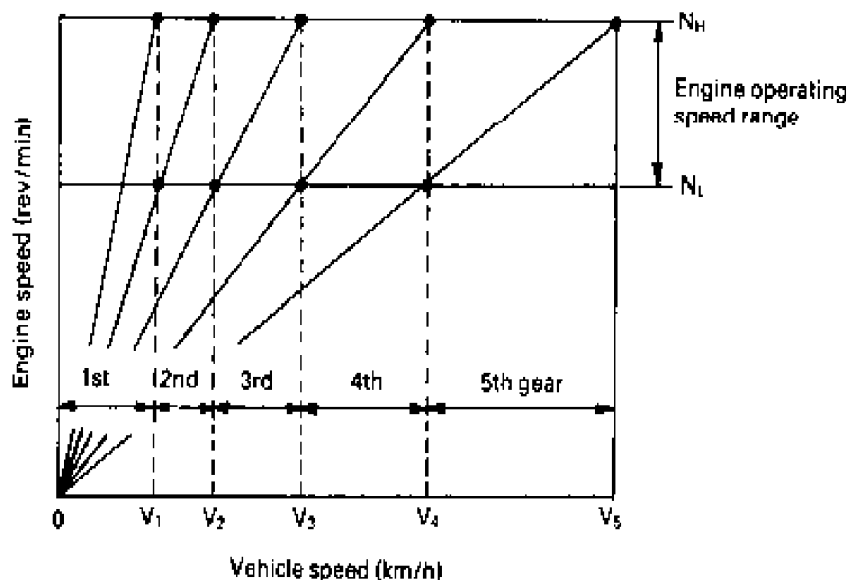

Figure 8 Gear ratios selected on geometric progression

Body design: Aerodynamics

INTRODUCTION

Throughout the history of the motor car there have been individual vehicles that have demonstrated strong aerodynamic influence upon their design. Until recently their flowing lines were primarily a statement of style and fashion with little regard for the economic benefits. It was only rising fuel prices, triggered by the fuel crisis of the early 1970s, that provided a serious drive towards fuel-efficient aerodynamic design. The three primary influences upon fuel efficiency are the mass of the vehicle, the efficiency of the engine and the aerodynamic drag. Only the aerodynamic design will be considered in this section but it is important to recognize the interactions between all three since it is their combined actions and interactions that influence the dynamic stability and hence the safety of the vehicle.

AERODYNAMIC FORCES

Aerodynamic research initially focused upon drag reduction, but it soon became apparent that the lift and side forces were also of great significance in terms of vehicle stability. An unfortunate side effect of some of the low drag shapes developed during the early 1980s was reduced stability especially when driven in cross-wind conditions. Cross-wind effects are now routinely considered by designers but our understanding of the highly complex and often unsteady flows that are associated with the airflow over passenger cars remains sketchy. Experimental techniques and computational flow prediction methods still require substantial development if a sufficient understanding of the flow physics is to be achieved.

The aerodynamic forces and moments that act upon a vehicle are shown in coefficient form in Figure 1. The force and moment coefficients are defined respectively as:

$$C_f = \frac{F}{\frac{1}{2}\rho v^2 A} \qquad C_m = \frac{M}{\frac{1}{2}\rho v^2 Al}$$

where F is force (lift, drag or side), M is a moment, ρ is air density, v is velocity, A is reference area and l is a reference length. Since the aerodynamic forces acting on a vehicle at any given speed are proportional to both the appropriate coefficient and to the reference area (usually frontal area) the product $C_f A$ is commonly used as the measure of aerodynamic performance, particularly for drag.
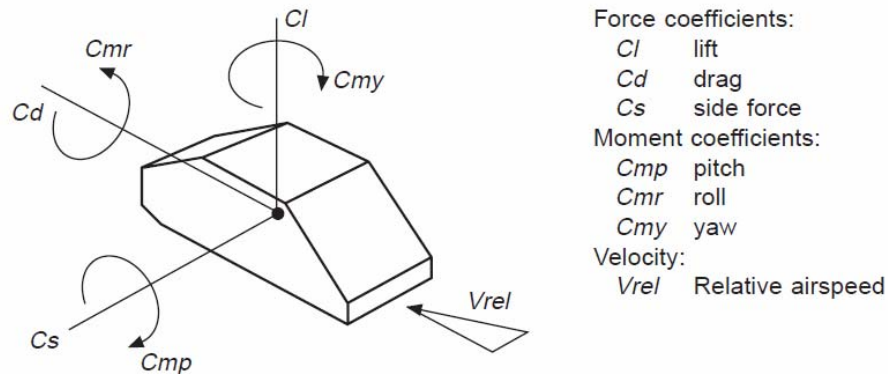

Figure 1 Lift, drag, side force and moment axes

The forces may be considered to act along three, mutually perpendicular axes. Those forces are the drag, which is a measure of the aerodynamic force that resists the forward motion of the car, the lift which may act upwards or downwards; and the side force which only occurs in the event of a cross-wind or when the vehicle is in close proximity to another. The lift, drag and pitching moments are a measure of the tendency of those three forces to cause the car to rotate about some datum, usually the centre of gravity. The moment effect is most easily observed in cross-wind conditions when the effective aerodynamic side force acts forward of the centre of gravity, resulting in the vehicle tending to steer away from the wind. In extreme, gusting conditions the steering correction made by the driver can lead to a loss of control.

DRAG

The drag force is most easily understood if it is broken down into five constituent elements. The most significant of the five in relation to road vehicles is the *form drag* or *pressure drag* which is the component that is most closely identified with the external shape of the vehicle. As a vehicle moves forward the motion of the air around it gives rise to pressures that vary over the entire body surface as shown in Figure 2a. If a small element of the surface area is considered then the force component acting along the axis of the car, the drag force, depends upon the magnitude of the pressure, the area of the element upon which it acts and the inclination of that surface element Figure 2b. Thus it is possible for two different designs, each having a similar frontal area, to have very different values of form drag.
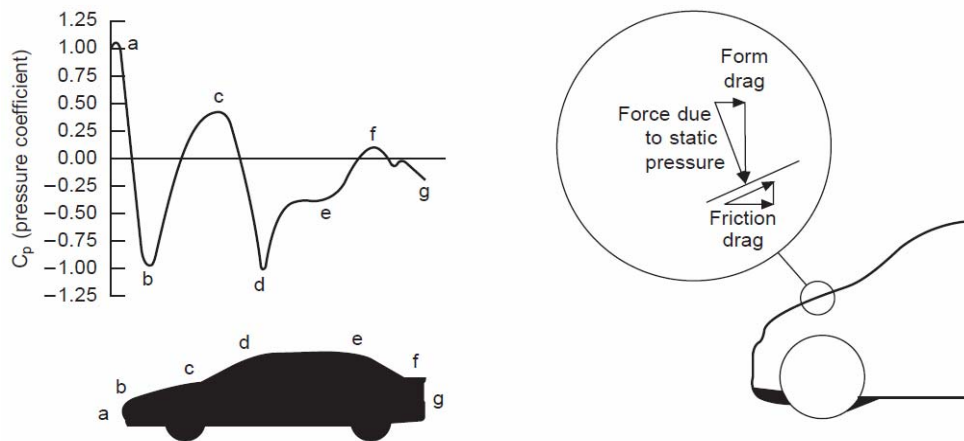

Figure 2 (a) Typical static pressure coefficient distribution; **(b)** The force acting on a surface element

As air flows across the surface of the car frictional forces are generated giving rise to the second drag component which is usually referred to as *surface drag* or *skin friction drag*. If the viscosity of air is considered to be almost constant the frictional forces at any point on the body surface depend upon the shear stresses generated in the boundary layer. The boundary layer is that layer of fluid close to the surface in which the air velocity changes from zero at the

surface (relative to the vehicle) to its local maximum some distance from the surface. That maximum itself changes over the vehicle surface and it is directly related to the local pressure. Both the local velocity and the thickness and character of the boundary layer depend largely upon the size, shape and velocity of the vehicle.

A consequence of the constraints imposed by realistic passenger space and mechanical design requirements is the creation of a profile which in most situations is found to generate a force with a vertical component. That lift, whether positive (upwards) or negative induces changes in the character of the flow which themselves create an *induced drag* force.

Practical requirements are also largely responsible for the creation of another drag source which is commonly referred to as *excrescence drag.* This is a consequence of all those components that disturb the otherwise smooth surface of the vehicle and which generate energy absorbing eddies and turbulence. Obvious contributors include the wheels and wheel arches, wing mirrors, door handles, rain gutters and windscreen wiper blades but hidden features such as the exhaust system are also major drag sources.

Although some of these features individually create only small drag forces their summative effect can be to increase the overall drag by as much as 50%. Interactions between the main flow and the flows about external devices such as door mirrors can further add to the drag. This source is usually called *interference drag.*

The last of the major influences upon vehicle drag is that arising from the cooling of the engine, the cooling of other mechanical components such as the brakes and from cabin ventilation flows. Together these *internal drag* sources may typically contribute in excess of 10% of the overall drag.

DRAG REDUCTION

Under the heading of drag reduction the designer is concerned not only with the magnitude of the force itself but also with a number of important and directly related topics. Firstly there are the effects of wind noise. Aerodynamic noise is closely associated with drag creation mechanisms which often exhibit discrete frequencies and which tend to arise where the air flow separates from the vehicle surface. Flow separation is most likely to occur around sharp corners such as those at the rear face of each wing mirror and around the 'A' pillar of a typical passenger car. Because of the close relationship between drag and noise generation it is not surprising that drag reduction programmes have a direct and generally beneficial effect upon wind noise. Such mutual benefits are not true of the second related concern, that of dynamic stability. The rounded shapes that have come to characterize modern, low drag designs are particularly sensitive to cross-winds both in terms of the side forces that are generated and the yawing moments. Stability concerns also relate to the lift forces and the changes in those forces that may arise under typical atmospheric wind conditions.

The broad requirements for low drag design have been long understood. Recent trends in vehicle design reflect the gradual and detailed refinements that have become possible both as a result of increased technical understanding and of the improved manufacturing methods that have enabled more complex shapes to be produced at an acceptable cost. The centre-line pressure distribution arising from the airflow over a typical three-box (saloon) vehicle has been shown in Figure 2a. A major drag source occurs at the very front of the car where the maximum pressure is recorded (Figure 2a, point 'a') and this provides the largest single contribution to the form drag. This high pressure, low velocity flow rapidly accelerates around the front, upper corner (b) before slowing again with equal rapidity. The slowing air may not have sufficient momentum to carry it along the body surface against the combined resistance of

the pressure gradient and the viscous frictional forces resulting in separation from the body surface and the creation of a zone of re-circulating flow which is itself associated with energy loss and hence drag. The lowering and rounding of the sharp, front corner together with the reduction or elimination of the flat, forward facing surface at the very front of the car addresses both of these drag sources. A second separation zone is observed at the base of the windscreen and here a practical solution to the problem is more difficult to achieve. The crucial influence upon this drag source is the screen rake. Research has clearly demonstrated the benefits of shallow screens but the raked angles desired for aerodynamic efficiency lead to problems not only of reduced cabin space and driver headroom but also to problems of internal, optical reflections from the screen and poor light transmission. Figure 3 demonstrates the benefits that may be achieved by changing the bonnet slope and the screen rake.
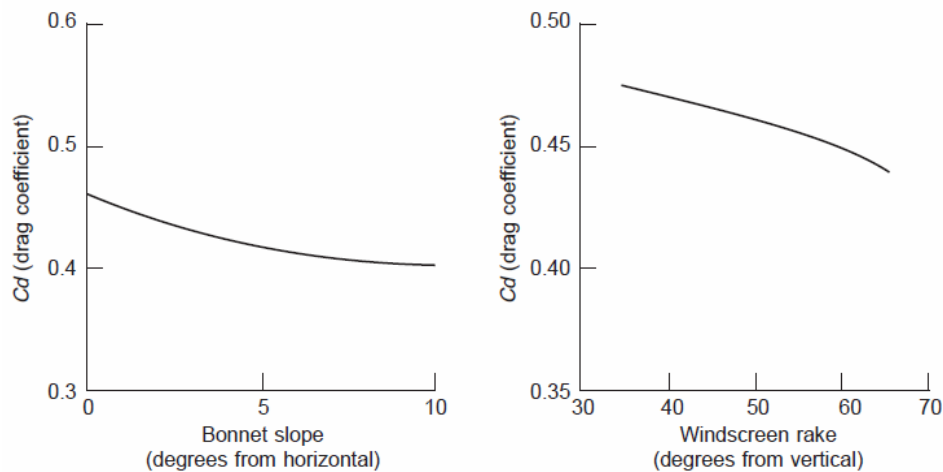

Figure 3 Drag reductions by changes to front body shape

There is further potential for flow separation at the screen/roof junction which similarly benefits from screen rake and increased corner radius to reduce the magnitude of the suction peak and the pressure gradients.

The airflow over the rear surfaces of the vehicle is more complex and the solutions required to minimize drag for practical shapes are less intuitive. In

particular the essentially two-dimensional considerations that have been used to describe the air flow characteristics over the front of the vehicle are inadequate to describe the rear flows. Figure 4 demonstrates two alternative flow structures that may occur at the rear of the vehicle. The first (Figure 4a) occurs for 'squareback' shapes and is characterized by a large, low pressure wake. Here the airflow is unable to follow the body surface around the sharp, rear corners. The drag that is associated with such flows depends upon the cross-sectional area at the tail, the pressure acting upon the body surface and, to a lesser extent, upon energy that is absorbed by the creation of eddies. Both the magnitude of the pressure and the energy and frequency associated with the eddy creation are governed largely by the speed of the vehicle and the height and width of the tail. A very different flow structure arises if the rear surface slopes more gently as is the case for hatchback, fastback and most notchback shapes (Figure 4b). The centerline pressure distribution shown in Figure 2a shows that the surface air pressure over the rear of the car is significantly lower than that of the surroundings. Along the sides of the car the body curvature is much less and the pressures recorded here differ little from the ambient conditions. The low pressure over the upper surface draws the relatively higher pressure air along the sides of the car upwards and leads to the creation of intense, conical vortices at the 'C' pillars. These vortices increase the likelihood of the upper surface flow remaining attached to the surface even at backlight angles of over 30 degrees. Air is thus drawn down over the rear of the car resulting in a reacting force that has components in both the lift and the drag directions. The backlight angle has been shown to be absolutely critical for vehicles of this type. Figure 5 demonstrates the change in the drag coefficient of a typical vehicle with changing backlight angle. As the angle increases from zero (typical squareback) towards 15 degrees there is initially a slight drag reduction as the effective base area is reduced. Further increase in backlight angle reverses this trend as the drag inducing influence of the upper surface pressures and trailing vortex

creation increase. As 30° is approached the drag is observed to increase particularly rapidly as these effects become stronger until at approximately 30° the drag dramatically drops to a much lower value. This sudden drop corresponds to the backlight angle at which the upper surface flow is no longer able to remain attached around the increasingly sharp top, rear corner and the flow reverts to a structure more akin to that of the initial squareback. In the light of the reasonably good aerodynamic performance of the squareback shape it is not surprising that many recent, small hatchback designs have adopted the square profiles that maximize interior space with little aerodynamic penalty.
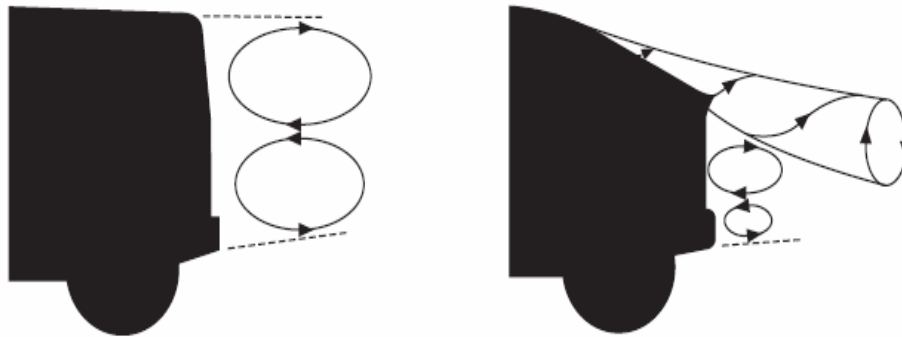
Figure 4(a) 'Squareback' large scale flow separation. **(b)** 'Hatchback/Fastback' vortex generation

The more traditional notchback or saloon form, not surprisingly, is influenced by all of the flow phenomena that have been discussed for the forms discussed above. As the overcar flow passes down the rear screen the conditions are similar to those of the hatchback and trailing; conical vortices may be created at or near the 'C' pillar. The inclination of the screen may be sufficient to cause the flow to separate from the rear window although in many cases the separation is followed by flow re-attachment along the boot lid. Research has shown that in this situation the critical angle is not that of the screen alone but the angle made between the rear corner of the roof and the tip of the boot. This suggests that the effect of the separation is to re-profile the rear surface to

something approximating to a hatchback shape and consequently the variation in drag with this effective angle mimics that of a continuous, solid surfaced 'hatch'. It follows that to achieve the minimum drag condition that has been identified to correspond to a backlight angle of 15° (Figure 5) it is necessary to raise the boot lid, and this has been a very clear trend in the design of medium and large saloon cars (Figure 6). This has further benefits in terms of luggage space although rearward visibility is generally reduced. Rear end, boot-lid spoilers have a similar effect without the associated practical benefits.
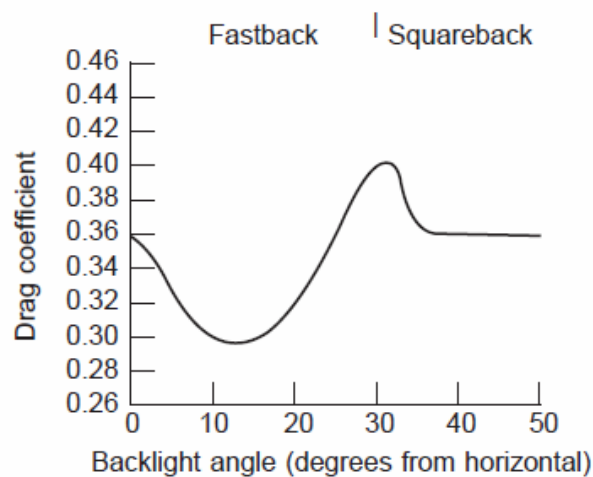

Figure 5 The influence of backlight angle on drag coefficient
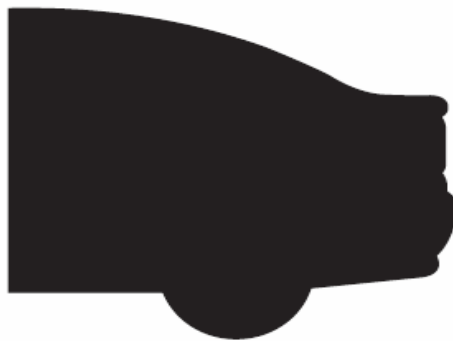

Figure 6 High tail, low drag design

Attention must also be paid to the sides of the car. One of the most effective drag reduction techniques is the adoption of boat-tailing which reduces the effective cross-sectional area at the rear of the car and hence reduces the volume

enclosed within the wake (Figure 7). In its most extreme configuration this results in the tail extending to a fine point, thus eliminating any wake flow, although the surface friction drag increases and the pressures over the extended surfaces may also contribute to the overall drag. Practical considerations prevent the adoption of such designs but it has long been known that the truncation of these tail forms results in little loss of aerodynamic efficiency.
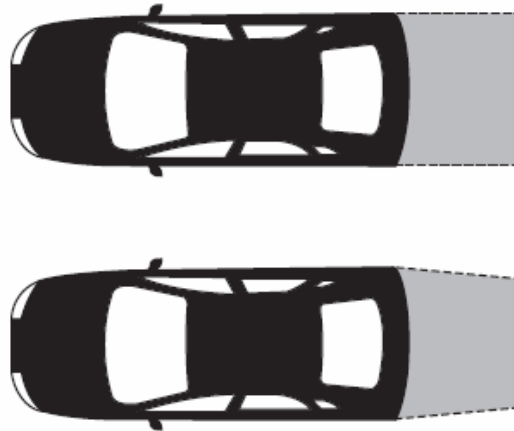

Figure 7 Boat tailing: reduced wake

Despite the efforts that have been made to smooth visible surfaces it is only recently that serious attempts have been made to smooth the underbody. The problems associated with underbody smoothing are considerable and numerous factors such as access for maintenance, clearance for suspension and wheel movement and the provision of air supplies for the cooling of the engine, brakes and exhaust must be given considerable weight in the design process. Just as the airflow at the extreme front and rear of the car were seen to be critical in relation to the overcar flow, so it is necessary to give comparable consideration to the air flow as it passes under the nose of the vehicle and as it leaves at the rear. It comes as a surprise to many to learn that the sometimes large air dams that are fitted to most production vehicles can actually reduce the overall drag forces acting on the car despite the apparent bluntness that they create. The air dam performs two useful functions. The first is to reduce the lift force acting on

the front axle by reducing the pressure beneath the front of the car. This is achieved by restricting the flow beneath the nose which accelerates with a corresponding drop in pressure. For passenger cars a neutral or very slight negative lift is desirable to maintain stability without an excessive increase in the steering forces required at high speed. For high performance road cars it may be preferred to create significant aerodynamic down-force to increase the adhesion of the tyres. The side effects of aerodynamic down-force generation such as increased drag and extreme steering sensitivity are generally undesirable in a family car. Lowering the stagnation point by the use of air dams has also been shown in many cases to reduce the overall drag despite the generation of an additional pressure drag component.

The shaping of the floorpan at the rear of the car also offers the potential for reduced drag (Figure 8). As the flow diffuses (slows) along the length of the angled rear underbody the pressure rises, resulting in reduced form drag and also a reduced base area, although interactions between the overcar and undercar air flows can result in unexpected and sometimes detrimental effects. Such effects are hard to generalize and detailed experimental studies are currently required to determine the optimum geometry for individual vehicle designs, but typically it has been found that diffuser angles of approximately 15° seem to provide the greatest benefits.
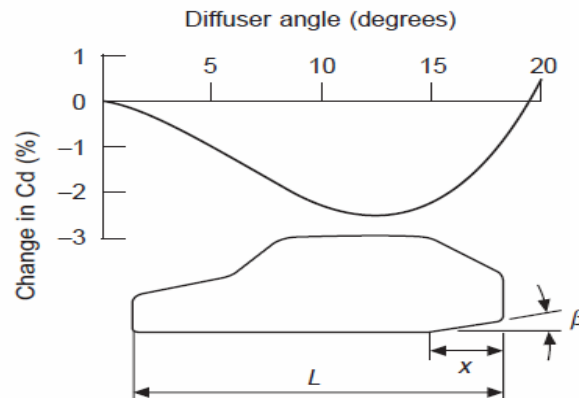

Figure 8 Rear, underbody diffusion

STABILITY AND CROSS-WINDS

The aerodynamic stability of passenger cars has been broadly addressed as two independent concerns. The first relates to the 'feel' of a car as it travels in a straight line at high speed and in calm conditions and to lane change maneuverability. The second concerns the effects of steady cross-winds and transient gusts that are associated with atmospheric conditions and which may be exaggerated by local topographical influences such as embankments and bridges.

The sources of straight line instability in calm conditions has proved to be one of the most difficult aerodynamic influences to identify. This is largely because of the complex interactions between the chassis dynamics and relatively small changes in the magnitude of lift forces and centre of pressure. Qualitative observations such as driver 'feel' and confidence have proved hard to quantify. New evidence suggests that stability and particularly lane change stability degrade with increases in the overall lift and with differences in lift between the front and rear axles.

The influence of cross-winds is more easily quantifiable. Steady state cross-winds rarely present a safety hazard but their effect upon vehicle drag and wind noise is considerable. Most new vehicles will have been model-tested under yawed conditions in the wind tunnel at an early stage of their development but optimization for drag and wind noise is almost always based upon zero cross-wind assumptions. Some estimates suggest that the mean yaw angle experienced in the U.K. is approximately 5º and if that is correct then there is a strong case for optimizing the aerodynamic design for that condition.

The influence of transient cross-wind gusts such as those often experienced when passing bridge abutments, or when overtaking heavy vehicles in the presence of cross-winds is a phenomenon known to all drivers. To reduce the problems that are encountered by the driver under these conditions it is

desirable to design the vehicle to minimize the side forces, yawing moments and yaw rates that occur as the vehicle is progressively and rapidly exposed to the cross wind. The low drag, rounded body shapes that have evolved in recent years can be particularly susceptible to cross-winds. Such designs are often associated with increased yaw sensitivity and commonly related changes of lift distribution under the influence of cross-winds can be particularly influential in terms of reduced vehicle stability. The influence of aerodynamics is likely to be further exaggerated by anticipated trends towards weight reduction in the search for improved fuel efficiency. Although methods for testing models under transient cross-wind conditions are under development, reliable data can, as yet, only be obtained by full scale testing of production and pre-production vehicles. At this late stage in the vehicle development programme the primary vehicle shape and tooling will have been defined so any remedial aerodynamic changes can only be achieved at very high cost or by the addition of secondary devices such as spoilers and mouldings; also an undesirable and costly solution. To evaluate the transient behaviour of a vehicle at a much earlier stage of its design it is necessary not only to develop model wind tunnel techniques to provide accurate and reliable data but most importantly to fully understand the flow mechanisms that give rise to the transient aerodynamic forces and moments. Initial results from recent developments in wind tunnel testing suggest that the side forces and yawing moments experienced in the true transient case exceed those that have been measured in steady state yaw tests.

NOISE

Although some aerodynamic noise is created by ventilation flows through the cabin the most obtrusive noise is generally that created by the external flow around the vehicle. Considerable reductions have been made to cabin noise levels which may be attributed in part to improved air flows with reduced noise creation and also to improved sealing which has the effect both of reducing

noise creation and insulating the occupants from the sound sources. Figure 9 provides an approximate comparison between the different noise sources (engine, tyres and aerodynamics) that have been recorded in a small car moving at 150 km/h. The creation of aerodynamic noise is mostly associated with turbulence at or near the body surface and moves to reduce drag have inevitably provided the additional benefit of noise reduction. Although there is a noise associated with the essentially random turbulence that occurs within a turbulent boundary layer it is the sound associated with eddy creation at surface discontinuities that has both the greatest magnitude and also the most clearly defined (and annoying) frequencies. Improvements in rain gutter design and the positioning of windscreen wipers reflect some of the moves that have been made to reduce noise creation and improved manufacturing techniques and quality control have also resulted in major noise reduction as a consequence of improved panel fit. Protrusions such as wing mirrors and small surface radii such as at the 'A' pillar remain areas of particular concern because of their proximity to the driver and because of the relatively poor sound insulation provided by windows. It has been demonstrated that it is the noise associated with vortex (eddy) creation that is the dominant aerodynamic noise source over almost the entire audible frequency range. One of the largest, single noise generators is the sun roof. Its large size results in low frequencies and large magnitudes and poorly designed units may even lead to discernible low frequency pressure pulsing in the cabin. Despite customer demand for low cabin noise there has been a parallel increase in the number of sun roofs that have been fitted to new cars. Open windows can create similar problems. Increased use of air conditioning is the best practical solution to this particular problem.
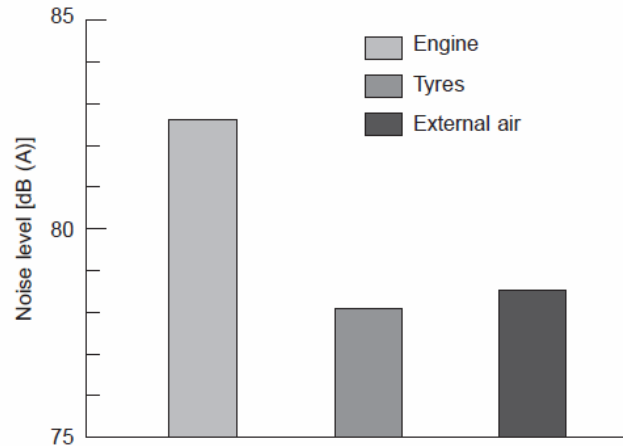
Figure 9 Noise sources

UNDERHOOD VENTILATION

The evidence from numerous researchers suggests that the engine cooling system is responsible for between 10% and 15% of the overall vehicle drag, so it is not surprising to note that considerable effort has been focused upon the optimization of these flows. Traditionally the cooling drag has been determined from wind tunnel drag measurements with and without the cooling intakes blanked-off. The results from those wind tunnel tests must be treated with caution since the closure of the intakes may alter the entire flow-field around a car. Underhood flow restrictions arising from the ever-increasing volume of ancillary equipment under the bonnet has further focused attention on cooling air flows, and this is now one of the primary applications for the developing use of computational flow simulation codes. Many of the sources of cooling drag are readily apparent such as the resistance created by the relatively dense radiator matrix and the drag associated with the tortuous flow through the engine bay. In general any smoothing of the flow path will reduce the drag, as will velocity reductions by diffusion upstream from the cooling system, although the implications of the latter upon the heat transfer must be considered. Less obvious but also significant is the interaction between the undercar flow and the cooling flow at its exit where high turbulence levels and flow

separations may to occur. Careful design to control the cooling exit flow in terms of its speed and direction can reduce the drag associated with the merging flows but in general the aerodynamics are compromised to achieve the required cooling.

The potential for underhood drag reduction is greatest if the air flow can be controlled by the use of ducting to guide the air into and out from the radiator core. Approximate relationships between the slowing of the cooling airflow and the pressure loss coefficient, are widely described in the published texts (e.g. Barnard, 1996). The high blockage caused by the radiator core has the effect of dramatically reducing the air velocity through the radiator and thus much of the air that approaches the radiator spills around it. The relatively small mass flow that passes through the core can exhibit substantial non-uniformity which reduces the effectiveness of the cooling system. These problems can be much reduced if the flow is ducted into the radiator in such a way as to slow the flow in a controlled and efficient manner, and careful design of the degree of diffusion can greatly improve the efficiency of the cooling flow. Increasing the diffusion slows the air flowing through the radiator which reduces both the drag force and the heat transfer. Although the reduced heat transfer rate results in a requirement for a larger radiator core surface area, the drag reduction is proportionately greater than is the corresponding reduction in heat transfer. A low speed, large core area therefore creates less drag for a given heat transfer rate. Inevitably, compromises are necessary. The larger core adds weight and cost and the generally close proximity of the radiator to the intake leaves little scope for the use of long, idealized ducting. Too much diffusion will lead to flow separation within the intake which may result in severe flow non-uniformities across the face of the radiator. Gains are also available if the air is ducted away from the radiator in a similarly efficient manner, but in most cases

the practical complexity of such a system and the requirement for a source of cooling air to the ancillaries has prevented such measures.

CABIN VENTILATION

Sealing between the body panels and particularly around the doors has achieved benefits in terms of noise reduction and aerodynamic drag, but the almost complete elimination of leakage flows has also led to changes in the design of passenger compartment ventilation. To achieve the required ventilation flow rates greater attention must be paid not only to the intake and exit locations but also to the velocity and path of the fresh air through the passenger compartment. The intake should be located in a zone of relatively high pressure and it should not be too close to the road surface where particulate and pollutant levels tend to be highest. The region immediately ahead of the windscreen adequately meets all of these requirements and is also conveniently located for air entry to the passenger compartment or air conditioning system. This location has been almost universally adopted. For the effective extraction of the ventilation air a zone of lower pressure should be sought. A location at the rear of the vehicle is usually selected and in many cases the air is directed through the parcel shelf and boot to exit through a controlled bleed in the boot seal. Increasing the pressure difference between the intake and exit provides the potential for high ventilation air flow rates but only at the expense of a flow rate that is sensitive to the velocity of the vehicle. This is particularly noticeable when the ventilation flow is heated and the temperature of the air changes with speed. A recent trend has been to use relatively low pressure differences coupled with a greater degree of fan assistance to provide a more controllable and consistent internal flow whether for simple ventilation systems or for increasingly popular air conditioning systems.

WIND TUNNEL TESTING

Very few new cars are now developed without a significant programme of wind tunnel testing. There are almost as many different wind tunnel configurations as there are wind tunnels and comparative tests have consistently shown that the forces and moments obtained from different facilities can differ quite considerably. However, most manufacturers use only one or two different wind tunnels and the most important requirement is for repeatability and correct comparative measurements when aerodynamic changes are made. During the early stages in the design and development process most testing is performed using small scale models where 1/4 scale is the most popular. The use of small models allows numerous design features to be tested in a cost effective manner with adequate accuracy.

For truly accurate simulation of the full scale flow it is necessary to achieve geometric and dynamic similarity. The latter requires the relative magnitudes of the inertia and viscous forces associated with the moving fluid to be modelled correctly and the ratio of those forces is given by a dimensionless parameter known as Reynolds number (Re):

$$\mathrm{Re} = \frac{\rho u d}{\mu}$$

where ρ is the fluid (air) density, u is the relative wind speed, d is a characteristic dimension and μ is the dynamic viscosity of the fluid. For testing in air this expression tells us that the required wind speed is inversely proportional to the scale of the model but in practice the velocities required to achieve accuracy (using the correct Reynolds number) for small scale models are not practical, and Reynolds number similarity is rarely achieved. Fortunately, the Reynolds numbers achieved even for these small models are sufficiently high to create representative, largely turbulent vehicle surface boundary layers, and the failure to achieve Reynolds number matching rarely results in major errors in the character of the flow. The highest wind speeds at

which models can be tested in any particular wind tunnel are more likely to be limited by the ground speed than by the air speed. The forward motion of a vehicle results not only in relative motion between the vehicle and the surrounding air but also between the vehicle and the ground. In the wind tunnel it is therefore necessary to move the ground plane at the same speed as the bulk air flow, and this is usually achieved by the use of a moving belt beneath the model. At high speeds problems such as belt tracking and heating may limit the maximum running speed, although moving ground plane technology has improved rapidly in recent years with the developments driven largely by the motor racing industry for whom 'ground effect' is particularly important. A considerable volume of literature is available relating to the influence of fixed and moving ground planes upon the accuracy of automotive wind tunnel measurements.

The use of larger models has benefits in terms of Reynolds number modelling and also facilitates the modelling of detailed features with greater accuracy, but their use also requires larger wind tunnels with correspondingly higher operating and model construction costs.

The forces acting upon a wind tunnel model are usually measured directly using a force balance which may be a mechanical device or one of the increasingly common strain gauge types. The latter has clear benefits in terms of electronic data collection and their accuracy is now comparable to mechanical devices. Electronic systems are also essential if unsteady forces are to be investigated. Lift, drag and pitching moment measurements are routinely measured and most modern force balances also measure side force, yawing moment and rolling moment. These latter three components relate to the forces that are experienced in cross-wind conditions.