

RACING
AND
SPORTS CAR
CHASSIS
DESIGN

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PREFACE

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The aim of this book is to provide information on the more advanced types of chassis and suspension in a form which will be understood by the large majority of motoring enthusiasts. Technical terms and formulae have been kept to a minimum, but those which are considered essential for the more serious student of design are contained in appendices at the end of the text. And for those to whom "stress" and "strain" would appear to mean much the same, there is a list of such expressions, with brief explanations, in the Glossary.

It is hoped that this work will do something to promote better understanding of the finer points of design—a subject on which very little has been published in the past—and perhaps help would-be chassis builders to avoid some of the more obvious pitfalls which await them. It cannot be too strongly emphasised that the basic requirements for success in this sphere, as in many others, are awareness of all the difficulties involved and ability to integrate innumerable conflicting elements into a balanced design. If expense is no object it should be possible to achieve almost any goal, but most cars—even racing cars—are designed with cost very much in mind and, strange as it may seem, these are often the most successful ones.

It would be impossible to write a book of this kind without outside assistance, and in this case our thanks are due particularly to James Allington, who supplied the drawings, Gilbert McIntosh, who was responsible for most of the material in Appendix I, and Keith Duckworth, who helped with the compilation of Appendix III. Valuable assistance was also afforded by personnel of several of the firms mentioned in the text.

Spring, 1961

M.C
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CHAPTER I

THE PURPOSE OF THE CHASSIS

Ideally, the purpose of a motor car chassis is to connect all four wheels with a structure which is rigid in bending and torsion—that is one which will neither sag nor twist. It *must* be capable of supporting all components and occupants and *should* absorb all loads fed into it without deflecting unduly. Whatever the car and whatever its purpose, however, the chassis is only a means to an end. As an alternative to the definition given in the first sentence, the ideal chassis could be summed up as a method of locating and linking together, by means of a complete structure, all the various mounting brackets on a car. Here, as in the opening sentence, due emphasis should be given to the word “structure”.

As will be shown later, in a well-designed car the chassis is the last major item to go on the drawing board. As a means of connecting mounting brackets, it should be designed to locate every one in the position in which it serves to the greatest advantage, and from which the loads involved can be taken out in the simplest possible manner.

Considering the chassis in this way, its chief purpose is to provide suitable mountings for all components of the car. In descending order of magnitude, the major loads involved are: 1. rear suspension and final drive; 2. front suspension and steering; 3. engine and/or gearbox; 4. fuel tank; 5. seats (and occupants), steering column, pedals and other controls, including hydraulic cylinders; 6. radiator; 7. battery; 8. spare wheel.

It is the layout of these components which largely decides the design of the chassis, and in the early stages of planning the greatest attention should be given to front and rear suspension pick-ups, which must be catered for as point loads put into the structure. In connection with the front suspension, the number of point loads, their magnitude and direction, depends on the type of suspension employed. The loads per wheel can be as low as two for suspension and one for steering with twin trailing link front suspension, or as high as six with certain types of wishbone suspension (four wishbone pick-ups, steering arm pick-up and spring mounting). These loads vary in type and degree from a very large, complex loading, as in the trailing link layout, to the comparatively small, simple loadings of the latest types of wishbone suspension.

Rear suspension loads are basically similar but generally of a higher order, owing to the greater weight normally concentrated on the rear wheels and the fact that the rear suspension normally has to accommodate driving torque. In each case the chassis structure has to deal with all these

loads. Types of rear suspension range from the simplest form of live axle to the most complicated independent layout. The simplest of all forms is the live axle mounted on quarter elliptic springs. The de Dion type of suspension—widely used on sports and racing cars until very recently—is one of the most complicated, in that transmission loads are absorbed into the chassis by an entirely separate means from those employed to absorb suspension loads. Normally radius arms deal with the longitudinal loads involved in acceleration, while torsional loads are taken out through the chassis-mounted final drive unit. Torsional loads due to braking are taken out into the chassis members through the differential mountings if the brakes are mounted inboard, or via the radius arms in the case of outboard brakes.

The mounting of the final drive unit is extremely important in a front engined/rear drive car with independent rear suspension, especially as it is often situated in part of the chassis from which it is difficult to take out high point loads. This is particularly true in the case of a multi-tubular or space frame structure. The design of mountings for the final drive unit depends on the proximity of points at which mounting loads can be absorbed. In many cases the main chassis structure into which loads are to be fed may not be conveniently near mounting points designed into the final drive unit. If this is so, a sub-frame must be designed to bridge the gap; great care is necessary in the design of such structures.

Almost invariably, final drive loads have to be taken out through a minimum of three basic mountings. The disposition of loads is normally symmetrical around the centre line of the unit, and this resolves the mountings into one bolt at either top or bottom and two in the other plane. In detail these bolts can be in single shear, double shear or tension and compression, and in most cases it is desirable to introduce some form of rubber bushing in order to reduce shock loadings to a minimum—and in the case of a road car to provide sound insulation. The type and disposition of mountings must be borne in mind throughout all stages of design, to make the best use of the various rubber bushings available for this purpose.

In the case of a rear-engined car a different set of conditions applies. Static loads may be greater, but the distances over which torque reactions can be resolved, and the fact that power unit mountings take transmission loads, reduce the necessity to consider final drive as a separate detail. On rear-engined cars the two sets of mountings—engine and transmission—are complementary. In this case the minimum number of mountings is three—one above or below the gearbox and two—one on either side—towards the front of the engine.

Whether a car is front- or rear-engined, the design of engine mountings should be complementary to the chassis design; the wider apart the mountings the lower the over-all loads. Additional chassis members, apart from those necessary for the primary structure, may be required to disperse point loads into the chassis, and as a basic design feature—since the power unit mountings carry a large proportion of the sprung weight—it is

desirable to have good load paths direct from engine mountings to suspension pick-ups to dissipate energy. On a front-engined car, with engine and gearbox mounted in unit, the location of the rear mounting often presents difficulty—particularly when it is desirable to use a proprietary gearbox mounting. In this case extra chassis members or some form of sheet metal structure may become essential.

Mountings for smaller items are almost invariably disregarded in the initial stages of design, but it is a great advantage if *all* components can be borne in mind all the time. For instance, it would help to bear in mind two or three alternative positions for the battery, a dead weight of around 40 lb which can materially affect weight distribution and can be regarded as a useful balance weight.

In a car with two or more seats placed side by side it is structurally preferable to use a front-mounted engine, as the chassis normally provides better supports for this layout. In the case of a single-seater, however, the use of a front-mounted engine driving the rear wheels leads to numerous problems, most of them based on the transmission line. In itself, however, a single-seater chassis presents far fewer problems than a two-seater. The simplest and most efficient layout is to place the engine in unit with the transmission and this, as a result of experiments with front-wheel-drive racing cars, leads to a rear-engined layout. (Front-wheel-drive can function extremely well, and has many practical advantages for passenger cars, but thus far it has not been shown to provide cornering power in line with that of the more advanced rear-wheel-driven sports and racing cars.)

In the case of a front-engined, rear-wheel-driven single-seater, the transmission line must pass either beneath or alongside the driver's seat. With the current trend for the centre of gravity to be at the lowest possible point this makes an offset transmission line virtually essential. Numerous complications result, however, such as asymmetrical engine mountings, angles in the transmission line (and further complications to obviate power losses) and complex offsets in the final drive unit. The basic alternatives here are either an offset differential unit with unequal length drive shafts or extremely complicated gearing. It is also possible to offset the driver (in which case, taken to extremes, the driver may have less feeling of control than when placed symmetrically) but unless the transmission is also offset this leads to the production of a car having two-seater proportions.

The position of the gearbox also has an important influence in this respect. If in unit with the engine it is liable to increase the amount of offset required. Thus it is best in this case to have both clutch and gearbox at the rear, with the propeller shaft running at engine speed. This also leads to a reduction of the rotating mass between engine and gearbox and may help weight distribution, allowing the engine to be mounted further back than would otherwise be possible. These problems have been largely responsible for the swing to rear-engined racing cars in recent years. For

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anyone still contemplating building a front-engined, rear-drive single-seater, however, the simplest method is to avoid complication by using a central propeller shaft passing under the seat and driving the rear wheels through vertical transfer gears. It might also be possible to use an offset engine layout with horizontal transfer gears.

To sum up, whatever type of chassis is used, it should, ideally, be a perfect structure designed to link up the mounting points for all the components that go to make up the car.

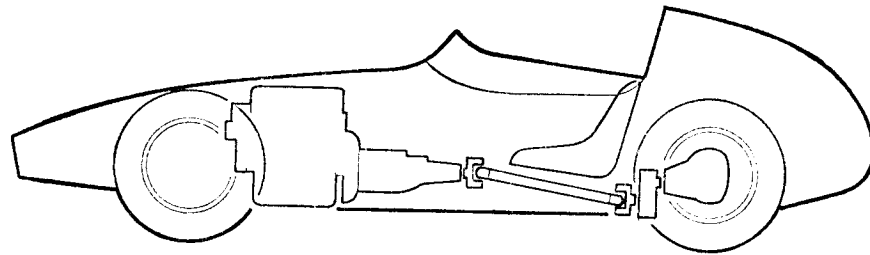
Chassis History

The development of chassis having adequate torsional stiffness is a fairly recent one. The early sports car chassis was constructed on massive lines, and its design owed more to bridge-building than to light engineering. Even today the chassis of many production sports cars are stiff only in bending.

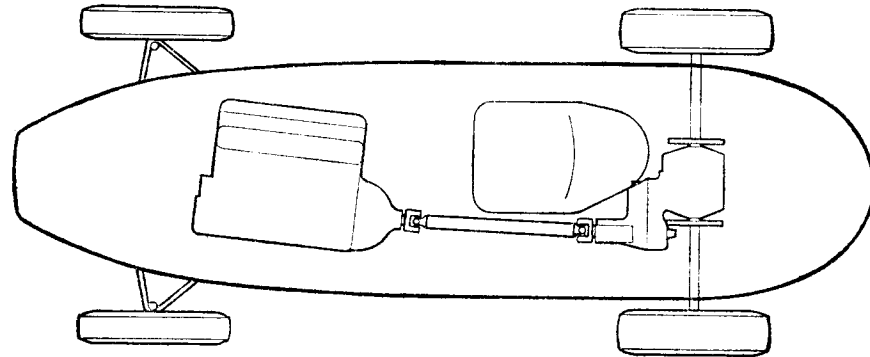
The history of the development of the more advanced types of chassis is an interesting one. Prior to the Second World War almost all sports car chassis were of the girder type, generally with live axles at both front and rear. In 1934 Auto Union initiated a change to twin tube chassis on racing cars, with a layout of this type composed of round-section tube. In the same year Mercedes-Benz used box-section members in a similar layout but, in 1937, they too went over to tubes—in this case of oval section. At the same time they also “progressed” from independent rear suspension to a de Dion layout, of the type first used in the late nineteenth century.

The twin tube chassis, with de Dion rear suspension, remained in vogue for racing cars in the years following the war, and indeed the de Dion axle was not superseded until the end of the 1950s. The need for increased chassis stiffness was recognised by the addition of tubular superstructure to the basic twin tube layout, but this contributed more to ease of body mounting than to torsional capacity. Early attempts at chassis of the space frame type also appeared at this time, but lacked the triangulation necessary to form complete structures.

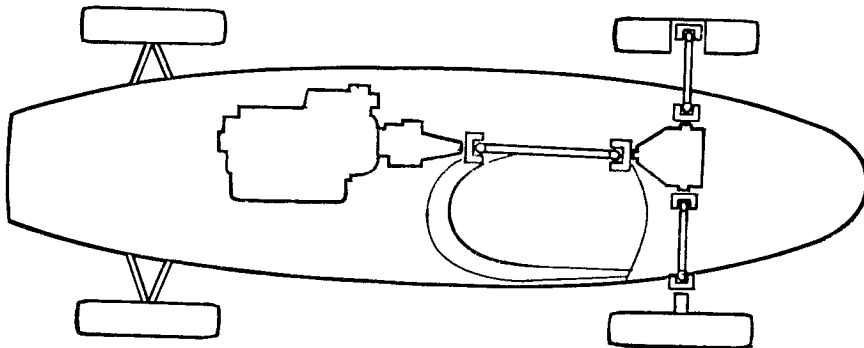
Two chassis designed on general space frame principles appeared in 1952—the Lotus Mark Six and the Mercedes-Benz 300SL. One came from a small concern with very limited resources and a largely part-time staff, the other from a major manufacturer with an impressive record in both racing and passenger car construction. The success of both led to this type of chassis becoming very widely used on sports/racing and racing cars, and also to its being adopted for a number of specialist road cars. However, for this purpose the space frame has several disadvantages—notably the high door sill which is virtually unavoidable in such a layout—making unitary body/chassis construction, as used in many mass-produced saloon cars, more practical for large-scale manufacture. Unitary construction has also become fashionable for single-seaters since the introduction, in 1962, of the Lotus 25. The term “monocoque” is widely used to describe



1 *Single-seater offsets. One of the simplest ways of lowering the driving position—and centre of gravity—on a front-engined car is to lower the transmission line, bringing the drive back up to hub height by means of transfer gears just ahead of the differential unit*



2 *Complex offsets. On the 1958 Formula One Lotus the centre of gravity was lowered by means of an angled engine and offset transmission and seat. Owing largely to the complications involved this car was not particularly successful*



3 *The 1960 Formula Junior Lola used offset engine, seat and final drive unit in the interests of a low centre of gravity and minimum frontal area. The transmission layout necessitated the use of unequal length drive shafts and—as these also constituted suspension linkages—asymmetrical radius arms*

this type of structure, and means, simply, that chassis and body are one and the same thing.

In its centre section, the modern rear-engined racing car is admirably suited to monocoque construction. The general requirements of the area ahead of the driver's feet can also be met by this type of structure, but it is not well suited for use in the engine bay, the chief problems being lack of torsional stiffness, interference with the exhaust layout and accessibility—the latter being far more important than some designers seem to think.

As regards torsional stiffness, a great deal can be gained in this sphere by making the engine a structural member. However, this practice is not to be recommended unless the crankcase has been designed for this purpose—in addition to its major function of providing positive location for the crankshaft and cylinder head(s). Furthermore, a complex series of mountings at both front and rear is required to deal with both torsional and bending loads, whereas only two front and one rear mountings are normally necessary when the engine is not used structurally.

The practice of using the power unit as a structural member was taken to its logical extremes on the first monocoque Ferrari, the engine (aided by a fabricated bulkhead bolted to it in the vicinity of the rear suspension) being the only structural component behind the seat back bulkhead, where the chassis proper stopped short. However, the 1964 Ferrari has vestigial "wheelbarrow arms" beneath the engine, which presumably take some of the bending loads and help to locate the rear bulkhead.

With a wide engine, the only satisfactory alternative to the use of rigid mountings may well be the use of a tubular structure for the engine bay. There are difficulties in taking out the point loads involved into a sheet metal centre section, but it is possible to do this with careful design and it should also be possible to lay out a space frame for the engine bay which will provide adequate torsional stiffness without adverse effect on exhaust layout or accessibility—both of which are liable to suffer when "wheelbarrow arms" are used. It may be necessary to make some members detachable, to facilitate installation and removal of the engine, but it is possible to do this without any loss of torsional capacity. A tubular frame may also provide better load paths from the rear bulkhead to the centre section than either the engine or the combination of engine and wheelbarrow arms.

A useful alternative to the monocoque for a two-seater is the backbone chassis, as used on the Lotus Elan (front-engined) and the Lotus 30 (rear-engined). Although this type of chassis requires separate bodywork it has a great deal in common—from a structural point of view—with the Lotus 25-style monocoque, the central part being compressed into a single "tube" to permit the fitting of two seats, doors and all the other appurtenances of a sports car.

The space frame chassis is, of course, a direct alternative to unitary construction, its small diameter tubes transmitting loads in much the same

way as the underframe and body panels of such a structure. Both types of construction owe a great deal to aircraft practice; this is hardly surprising, as a great deal of time and money has been spent on research in this sphere—far more than in the car world.

Outwardly it may appear strange that the most advanced sports car chassis of today are the work of specialist firms rather than major manufacturers. It is all a question of economics. The tubular space frame, although involving little in the way of tooling costs, is relatively expensive to manufacture, requiring a great deal of skilled welding. The only comparable alternative, unitary construction, involves very considerable tooling costs and is less satisfactory for open cars than saloons owing to lack of bracing in the cockpit area. But it has the great advantage over all other designs that separate bodywork is not required. By comparison, the conventional chassis is much simpler to make and involves far fewer snags in the fitting and servicing of mechanical components. For ultimate chassis performance, however, there is no substitute for torsional stiffness, and this can only be provided by the more advanced type of layout.

Suspension History

As has been mentioned above, the development of chassis having adequate torsional stiffness is a fairly recent one. Sports cars of an earlier era made up for this deficiency by using extremely stiff suspension; many, in fact, gave the impression that there were no suspension springs in their make-up. Today the ride is more acceptable, but the cornering speed of the mass-produced sports car has not improved as much as might have been expected—particularly in view of the tremendous advances in tyre design. On many production cars one of the only ways of improving roadholding is the fitting of stiffer springs or anti-roll bars—both of which have much the same effect. This applies particularly at the front, where the types of independent suspension most commonly used are subject to considerable camber change on roll. Increased roll stiffness—whether via the springs or the anti-roll bar—increases the adhesion of the front tyres and thus improves the car's roadholding by bringing the cornering power of the front wheels almost up to that of the back ones, which are kept almost upright on their live axle. Such perfection is only achieved on very smooth surfaces, however. With stiffer springs at the front the tyres frequently lose contact with the road on rough surfaces, while each rear wheel reacts to every movement of the other one. Stiffened or flattened rear springs may also show some advantages in terms of circuit speeds, but a car fully modified in this way usually suffers the penalty of a very harsh ride.

On the whole, the production sports car of today offers a reasonable compromise between roadholding and ride. With a more up-to-date approach, however, both of these conflicting features can be vastly improved at one and the same time. The basic requirements are a stiff chassis, advanced independent suspension and soft springs.

CHAPTER II

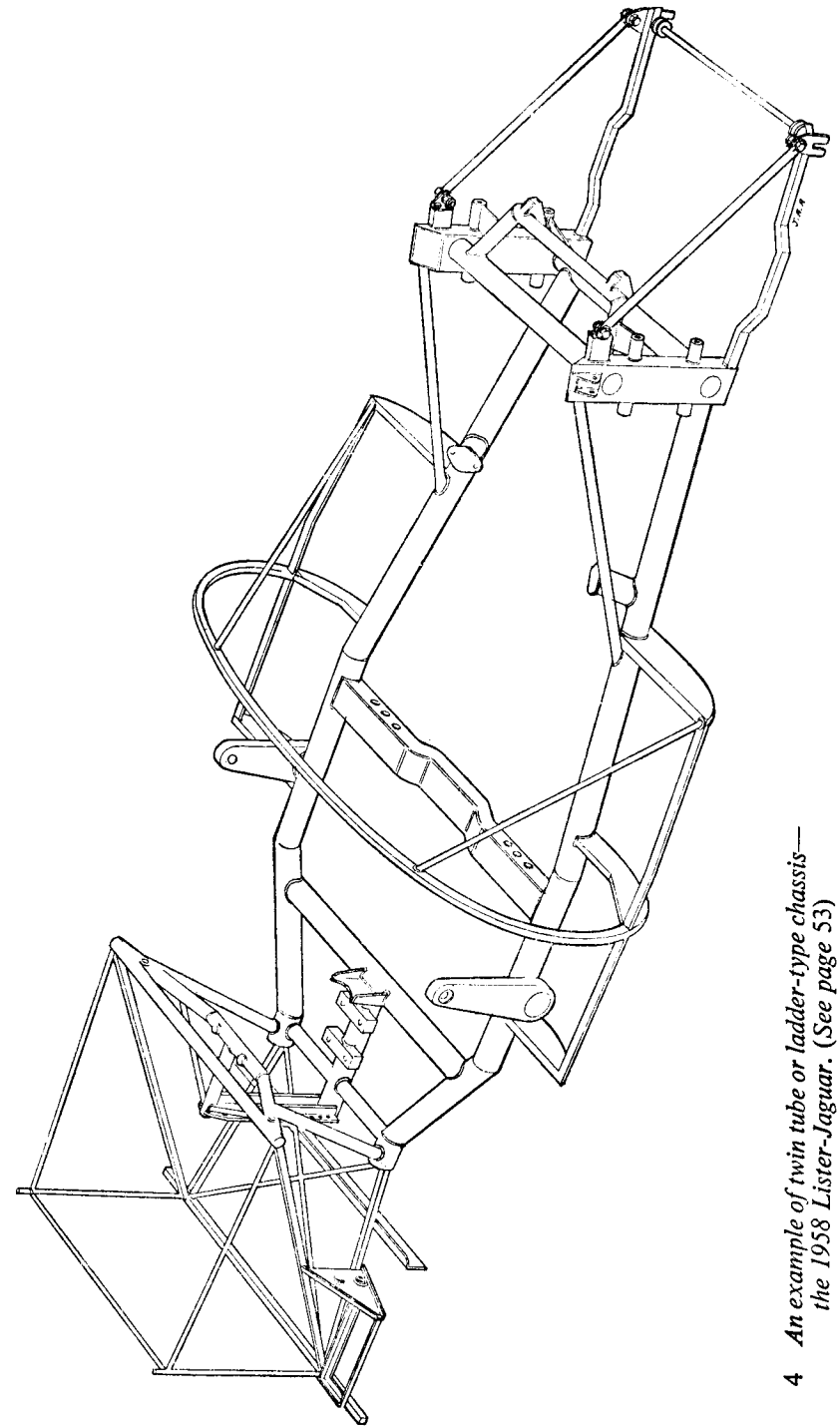
THE MAIN TYPES OF CHASSIS

Twin Tube or Ladder Frame Chassis (See fig. 4)

For many years now the girder type of chassis has been used on the majority of mass production sports cars, but specialist builders have time and again shown the advantages of the various types of tubular chassis. Thus, disregarding the torsionally inefficient girder type, the simplest form of chassis is the twin tube or ladder frame, with two large diameter side members and either lateral or diagonal bracing, or a mixture of both, in either similar or smaller diameter tubes; the latter provide both increased torsional rigidity and mountings for main and subsidiary components. The most common material for this type of frame is 3 or 3½ inch 16 gauge mild steel tube, and the normal method of construction is by electric or gas welding. Care is necessary when joining tubes of differing diameters to avoid local failure, and this is normally overcome by the use of gusset plates (as illustrated in fig. 10) to spread the load to the full diameter of the tube.

The twin tube type of chassis, although relatively heavy and lacking torsional stiffness, is easy to make and very durable, due to the use of heavy gauge material. Furthermore, it is not prone to accidental damage and provides good accessibility of mechanical components. By the use of sub-frames it is fairly easy to arrange mountings for all components. In designing a chassis of this type it is necessary first of all to locate suspension, engine, gearbox, differential and seat mountings. Having decided these, it is fairly easy to find the optimum position for the chassis tubes. With a twin tube car this is not very critical, as torsional stiffness does not alter greatly with varying tube positions. The load capacity of such a chassis in bending is poor, as the frame has to cater for quite high loads and round tube is not suitable for heavy loading in this manner. The torsional capacity also is very low, because the torsional stiffness of the chassis depends solely on the section of the tube used. Cruciform bracing helps by taking out torsional loads in bending. Bearing in mind the loads involved, oval section tube would possibly prove better than round for secondary installations where only bending loads are encountered. Oval tube is not satisfactory for main members, however, because of its lack of torsional rigidity.

For any given weight of tubing, the torsional stiffness of a cross section of twin tube frame could be improved by using a single tube of slightly greater diameter. However, although this illustrates the relativity of



4 An example of twin tube or ladder-type chassis—the 1958 Lister-Jaguar. (See page 53)

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torsional stiffness, it is not practicable for a sports car chassis, owing largely to mounting complications.

A twin tube chassis must always be relatively heavy, owing to the extra weight made necessary by the low efficiency of the frame. Complication in construction is kept to a minimum because of the small number of individual members involved, their greater wall thickness and the use of arc welding—all of which cut down the cost of construction. In addition, in this type of chassis the minimum of distortion due to welding is encountered during fabrication.

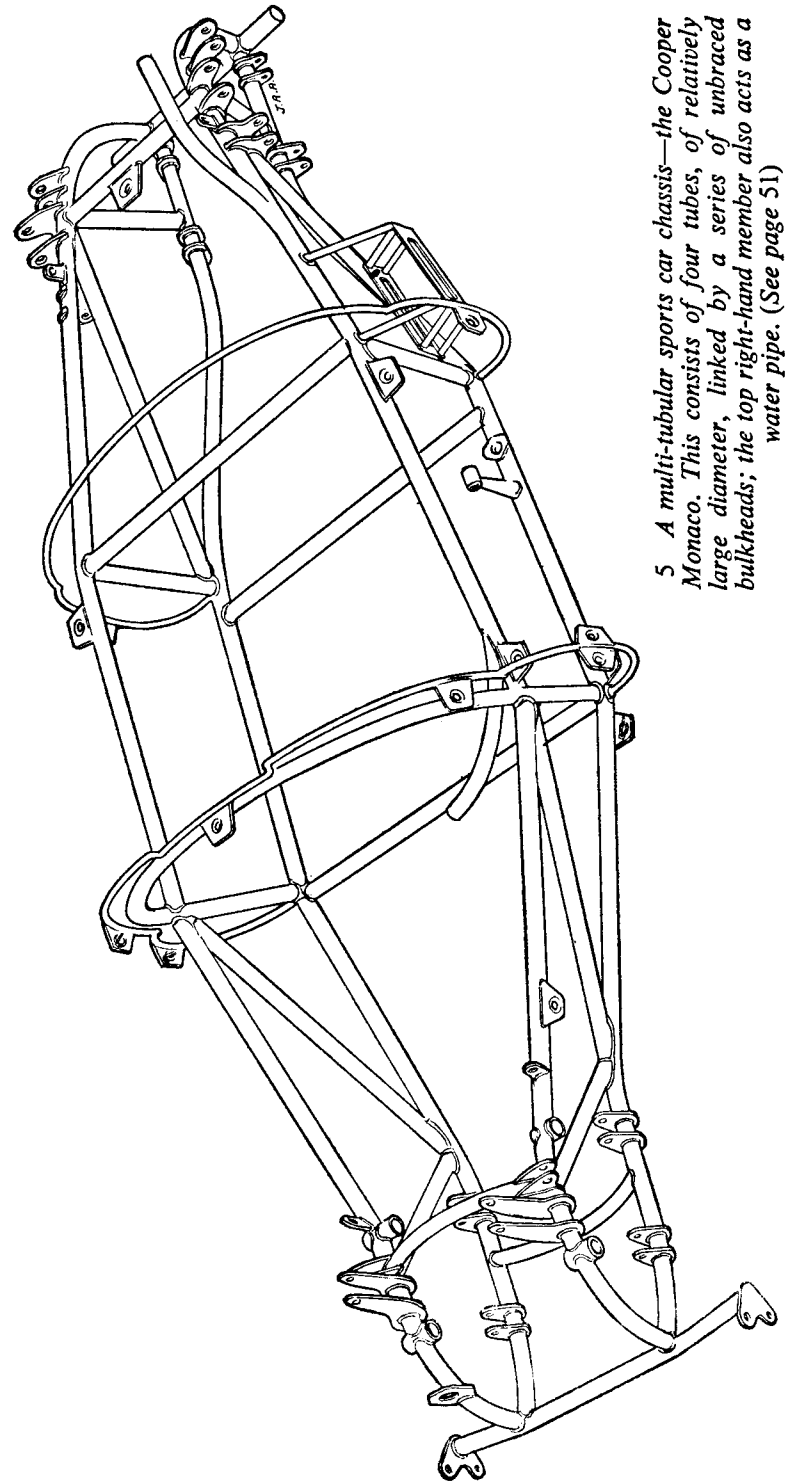
With a twin tube chassis it is important to build up a front suspension frame which gives adequate support to the suspension units and at the same time transmits suspension loads into the main chassis structure. In this instance square or rectangular section tube may well be better than round tube from the design point of view. Much the same can be said for rear suspension if coil springs are used, but with independent rear suspension a load-carrying frame is required in the centre of the chassis, and this must be capable of feeding transmission loads into the main structure. One heavy cross-frame, usually located at the scuttle on a front-engined car, should provide support for such relatively heavy items as the battery, steering column, body, door hinges, flame-proof bulkhead and brake, clutch and accelerator pedals.

For simplicity, cheapness and general ease of building, a twin tube type structure is quite satisfactory for a low- or medium-powered road sports car, and in this particular application could possibly be more satisfactory than a high performance chassis structure. However, because of the current rate of chassis development, the twin tube frame is not advised for any serious competitive motoring of National or International class.

Multi-tubular Chassis (See fig. 5)

In theory the term “multi-tubular” could be used to describe all chassis other than the twin tube type referred to in the previous section. In practice, however, the term can perhaps be best applied to those chassis which utilise four main side rails but cannot be classified in the true space frame category.

Multi-tubular frames are basically of very low efficiency, but have proved to be a successful compromise between the twin tube chassis and the space frame in terms of both stiffness and production cost. With a multi-tubular layout, relatively large section tubes are necessary to attain stiffness from welded joints and from the torsional rigidity of individual members. The load capacity of a multi-tube frame in bending is generally quite adequate, provided there is sufficient diagonal bracing throughout the length of the chassis to prevent lozenging. Torsional capacity depends largely on the number of members and the diameter and section of tubing employed, but is very much inferior to that of a space frame. An effective



5 A multi-tubular sports car chassis—the Cooper Monaco. This consists of four tubes, of relatively large diameter, linked by a series of unbraced bulkheads; the top right-hand member also acts as a water pipe. (See page 51)

THE MAIN TYPES OF CHASSIS

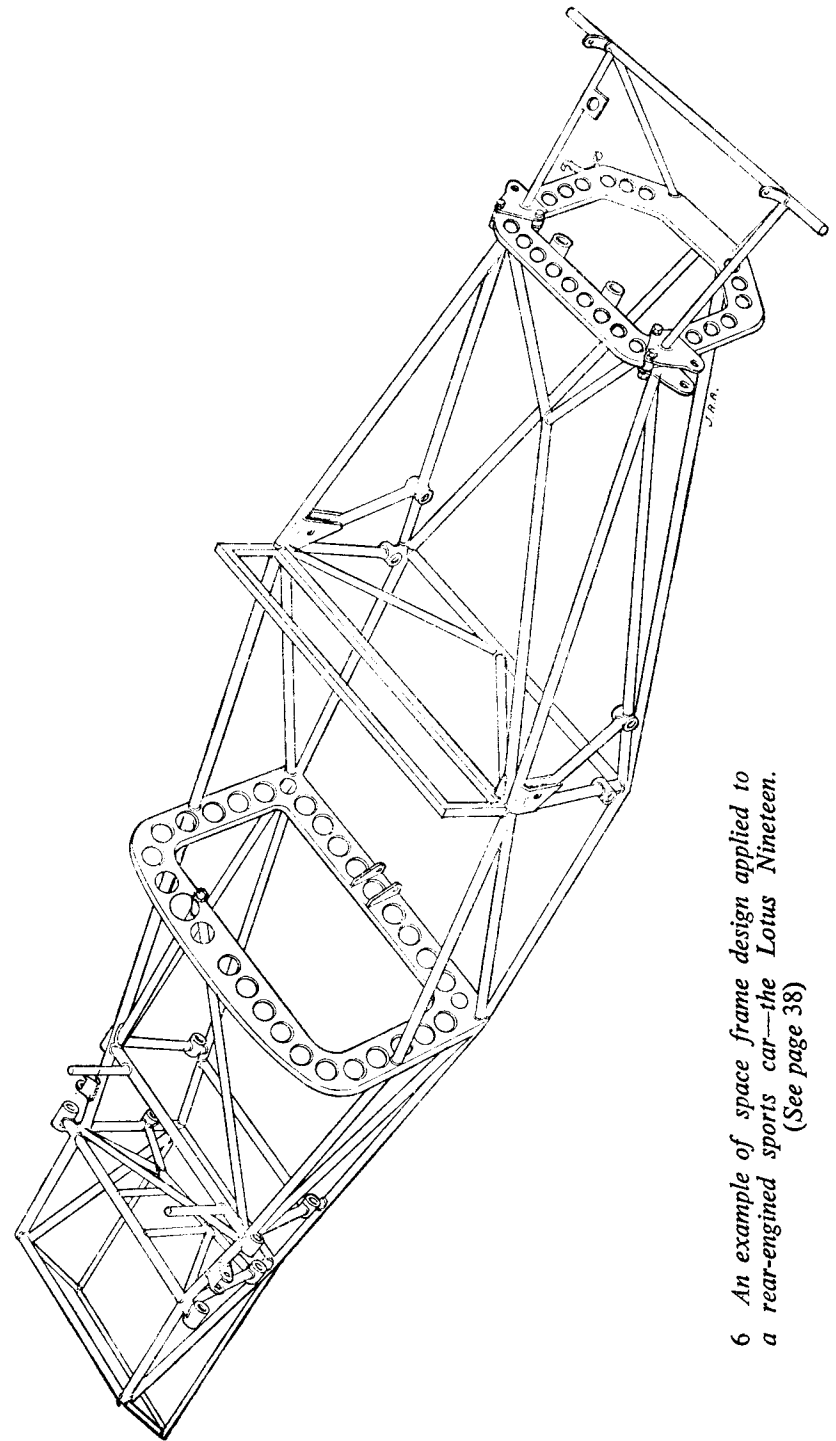
multi-tubular chassis must also of necessity be considerably heavier than a space frame. Accessibility is hardly likely to be as good as on a twin tube car, but this depends largely on the design. Durability depends mainly on weight, but even a heavily-constructed chassis of this type is more liable to structural failure than a well-designed lightweight space frame because of the bending loads taken by the welded joints. Chassis of this type can also be very difficult to repair after even quite slight impacts, owing to the fact that loads are transmitted throughout the frame rather than restricted to a small area, as is often the case with a space frame. Construction costs are similar to those for a space frame, but unless the chassis is made to wide manufacturing tolerances and the brackets are fitted afterwards, irrespective of their position, it is sometimes extremely difficult to make the chassis and the components fit together. Experience generally shows that it is very difficult to keep a chassis of this type dimensionally accurate during manufacture.

In the multi-tube category can be included many chassis which are normally described as being of the space frame type but which are, in fact, merely four-rail chassis with diagonal bracing where convenient. Conversely, as very few true space frames have ever been made, it is necessary to consider in this category many chassis which come near to the ideal but with some compromise in important areas. Of these, the cockpit of a sports car is usually the most critical area and the torsional stiffness of almost all space frames could be improved by as much as 100 per cent by running a diagonal across the top of this bay. In general, uniform stiffness is essential for a proper structure, and if one part is too stiff the concentration of loads and deflection at one point may lead to fatigue failure.

Although in some respects the multi-tubular frame is an advance on the twin tube chassis, it is not to be encouraged, as it is neither simple nor efficient. The Cooper racing car may be cited as the exception which proves the rule.

Space frames (See fig. 6)

The space frame is the most efficient type of chassis which it is possible to build in limited production. Unitary construction may be superior in some instances but there are many factors against it, as will be shown later. As regards the space frame, it is difficult to imagine a chassis of this type having adequate torsional rigidity without automatically having ample rigidity in bending. However, the criterion of chassis design—and in fact the primary function of a high-performance chassis—is torsional rigidity. In a sports car chassis it is almost impossible to arrive at a true, complete structure because of the necessity to compromise. The best example of a space frame chassis from the point of view of torsional rigidity would be a square-section rectangular box, with ends, sides, top and bottom triangulated by diagonals running from one corner through the centre to the opposite corner. This would be the lightest, stiffest, simplest and cheapest



6 An example of space frame design applied to a rear-engined sports car—the Lotus Nineteen. (See page 38)

THE MAIN TYPES OF CHASSIS

type of chassis, but at the same time it would be entirely impractical for automotive applications, particularly from the point of view of accessibility. Thus a chassis has to be split into bays, preferably two, but normally compromise creeps in, making three or four bays simpler from the point of view of localising the effects of the compromise. However, a major advantage of the space frame is that, as the very best use is being made of the material, the minimum of material is required.

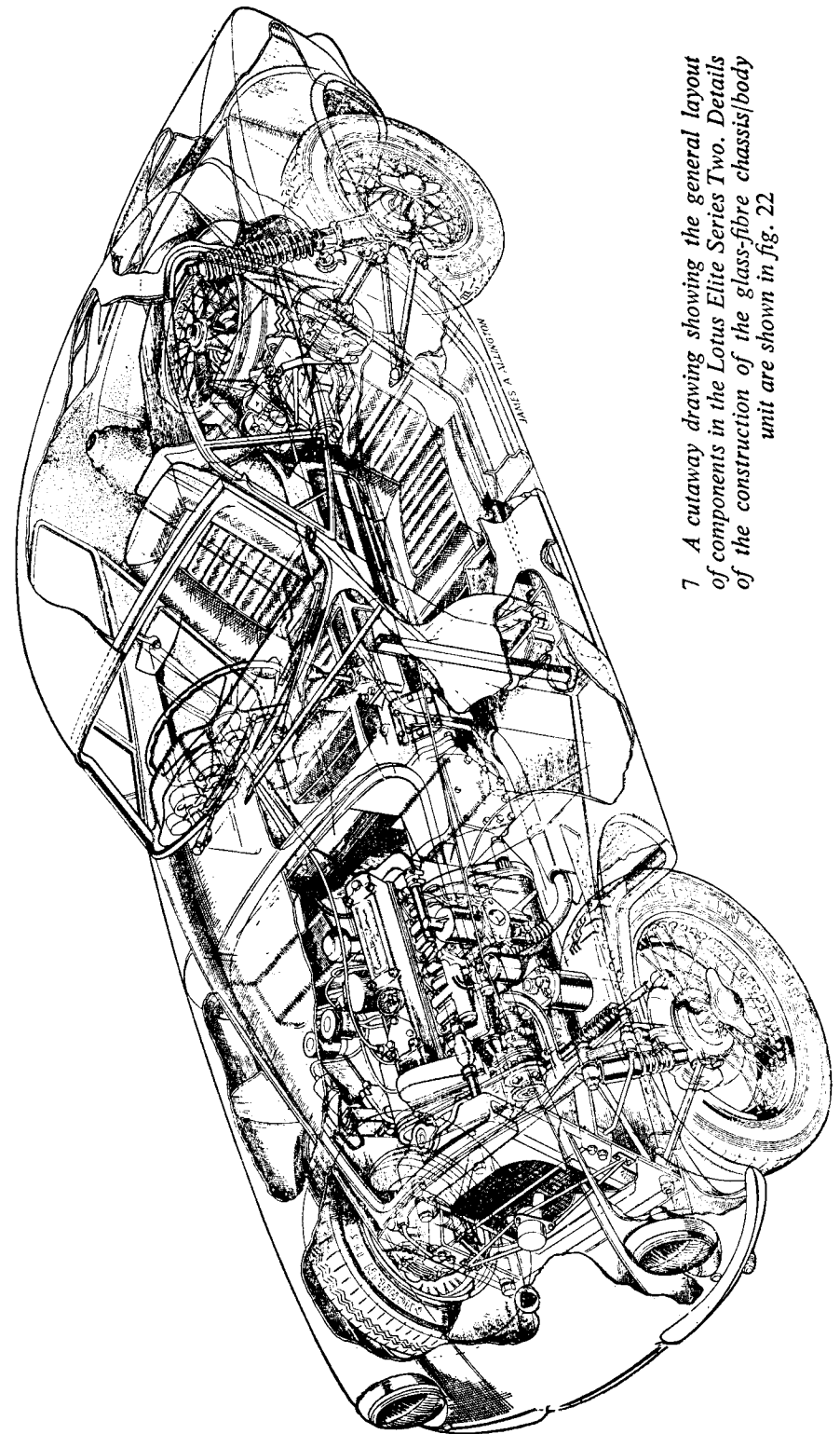
In over-all car design, accessibility is almost as important a factor as chassis rigidity, because it is essential to be able to service the car and to get all components into or out of it with the minimum of delay. For instance, on a sports or sports/racing car it is extremely important to be able to get the engine in and out. If a chassis member—usually a diagonal—interferes with this, it can be made removable. In some cases the engine can be used as a chassis member but this is to be discouraged because of difficulty with engine vibration and the need for rigid mountings. Among other problems in this department are complication of the exhaust system, owing to the need to avoid chassis members, and the difficulty of accommodating carburettors. In addition, the fitting of the rear axle is made more difficult by a multiplicity of tubes, especially if independent or de Dion-type suspension is used, when the drive unit is mounted centrally and maintenance work must be carried out at arm's length from above, behind or at one side.

A well-made space frame chassis should be very durable. The only danger of failure due to long service is likely to come from within the tubes because of internal rusting and corrosion. This can be avoided by suitably treating and sealing the insides of tubes, although pop-rivet holes sometimes confuse the issue in this case. Impact resistance should be good in the case of minor bumps, as damage should be limited to the bay receiving the blow. Major impacts are absorbed progressively, each bay taking part of the strain until it can no longer accommodate the rapidly rising load. Thus in the case of a high-speed collision, although the car may be extensively damaged, the fact that it slows down progressively often minimises injuries to the driver.

From the production point of view the space frame is probably the most expensive tubular chassis to make, because of the number of tubes used and the amount of welding involved. But it is undoubtedly the most efficient.

Unitary Construction (See figs. 7, 22, 23 and 24)

Under this heading can be considered those cars which have no separate chassis but combine chassis and bodywork in one unit, even though it may be made up of numerous panels formed into a structure by load-carrying joints. This category includes what are basically sheet metal platform chassis supplemented by superstructure which also provides stiffness. In large quantity production this is probably the best method of construction. For limited production, however—less than a total of 2500 cars—the very



7 A cutaway drawing showing the general layout of components in the Lotus Elite Series Two. Details of the construction of the glass-fibre chassis/body unit are shown in fig. 22

high tooling costs preclude the use of this type of chassis if pressed metal construction is envisaged. Smaller scale production should be economically possible using resin-bonded glass-fibre, but this type of construction is in a very early stage of development.

In the course of design, each panel or bay of a unitary construction chassis must be stabilised to carry out the function of transferring loads. The simplest way to achieve this would be to use a large, round section tube with the ends blanked off. However, the best design is the one which departs least from the ideal, and as a round section chassis/body cannot be considered practical the best compromise would seem to be a square section layout with large radius corners. Such complications as holes for the doors, boot and bonnet cause this type of chassis to depart even further from the ultimate, while practical considerations generally make it necessary for the bodywork to be tapered off at each end.

Any operation on a tube, such as tapering it or cutting holes in it, must have adverse effects on it structurally. With flexing, holes crossing load paths change their shape. Since there must be holes, despite the reduction in efficiency which they cause, the design must incorporate additional frames and bulkheads to stabilise the surrounding areas.

The use of this type of chassis would seem to offer many advantages, to judge from the developments carried out in recent years in aircraft manufacture. Such a structure can be made very stiff and extremely light—an essential feature in aircraft design—and in a car of this type 30 gauge material would probably be quite adequate. It is perhaps not commonly known that the engines of the De Havilland Comet are hung in stabilised 26 gauge stainless steel.

The load capacity of a unitary construction chassis in bending should be extremely good, because bending loads are resolved into pure tension and compression in the undertray and roof, to which type of loads these areas are ideally suited. Such a chassis should also be very good from the torsional point of view, but in practice everything depends on general and detail design around the apertures. This is particularly true in the case of an open car, which lacks the diagonal bracing provided in a closed car by the roof. Even here, however, with careful design adequate torsional stiffness should be available. Weight should be lower than for any equivalent chassis because, with good design, a far greater proportion of the weight of the material can be made to carry loads than is the case with a separate chassis and body.

In general, a unitary construction design should be stiffer than an equivalent tubular space frame and body for the same weight, or lighter for similar stiffness. Accessibility depends on the design, and this in turn depends on the degree of compromise reached between chassis stiffness and practicability. Durability depends on operating conditions but should be good, all other things being equal, while accident damage should be localised—given good design—making the cost of repairs fairly low.

CHAPTER III MATERIALS

In the choice of materials, the most important factors to bear in mind are fitness for purpose and cost. As has been mentioned in Chapter II, the type of production envisaged has a considerable influence on the type of chassis which can be used for any given car—and therefore on the materials which can be utilised. Unitary construction in pressed metal would be ideal for many limited production cars, but the cost of tooling renders it impractical. Unitary construction in glass-fibre or wood is more feasible in such cases, but both materials are relatively so new in this application that the use of separate chassis and bodies remains the simplest solution in the majority of instances. However, the cost of a hand-built “monocoque” may well be justified for single-seater racing at International level, and a suitable material for it is a workable medium performance aluminium alloy—e.g. B.S. 1470:1955 NS4.

Despite the recent trend towards monocoques, there still seems to be a considerable future for tubular steel chassis. For such purposes a wide variety of alloy steel tubing is available, ranging from high quality chrome molybdenum and nickel chrome steel to the ordinary mild steel used by the majority of specialist sports and racing car constructors. Each of the different types has its own special properties and uses. Twenty-five ton mild steel is quite suitable for welded tubular structures of the space frame type, but some manufacturers use a material which approximates to aircraft T45 specification, a manganese steel alloy which is particularly suitable for welding. The majority of current production sports/racing and racing chassis are made of electric resistance welded tubing which—although once somewhat variable in quality—is now normally first class. Chrome molybdenum steel is generally used for twin tube chassis, but in this case manganese steel would possibly be even better, because it retains its physical properties during gas welding much better than do other steels. If the higher quality (60–65 ton) steels—chrome molybdenum and nickel chrome—are used, electric welding becomes almost essential.

At this point it might be appropriate to digress a little to consider the respective merits of welding and brazing. Both are widely used, and recent developments in nickel bronze welding and manganese bronze welding techniques—together with improvements in the composition of the basic material—have resulted in joints becoming superior in performance to the original welded joint. Both types are applied by similar techniques, by far

MATERIALS

the best being that which uses a gaseous flux, as opposed to the older type of powder or paste flux. In this context it is important that bronze welding is not confused with brazing.

Welding

Welding is a method of joining two steel tubes together by heating them to melting point locally with an oxyacetylene flame, forming a weld fillet or puddle, and keeping this constant by adding filler rod as necessary. In general a similar method is followed for nickel bronze or manganese bronze welding, but these alloys have a much narrower heat range and greater care must be taken with them, as insufficient heat will cut down penetration and therefore the strength of the joint, whereas overheating is equally bad due to intergranular penetration of the filler rod into the surface of the steel. This not only makes a weak joint, but at the same time considerably weakens the affected steel component. It is difficult to detect a weakness caused in this way and thus the strength of the chassis depends largely on the skill of the operator.

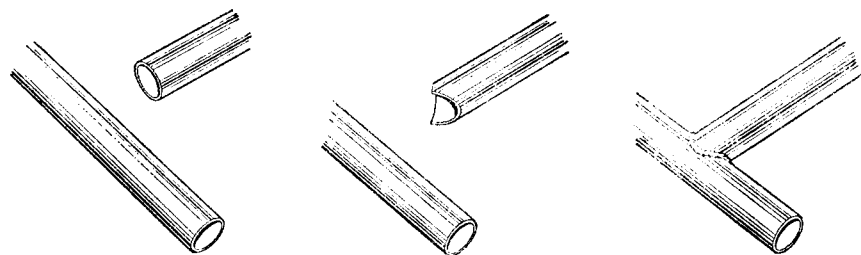
Brazing

Brazing consists of building up general heat, of a lower order than that required for welding, and applying filler rod in a stroking manner. As soon as the latter runs into the joints the area can be considered 100 per cent treated. Very similar to brazing is silver soldering, which does much the same job at lower temperatures. In this case extreme cleanliness and adequate coverage of flux is absolutely essential.

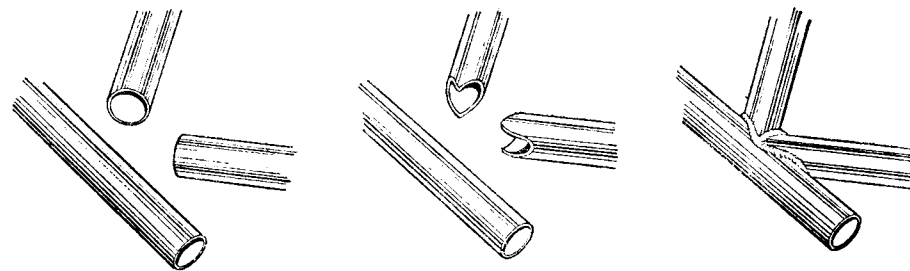
Whichever method of making joints is employed, the strength and life of the chassis are largely dependent on the cleanliness of the parts to be joined and the general preparation which is carried out before each stage of the work is commenced.

In conjunction with a space frame type chassis it is often desirable to use the undertray and other body members as load carrying panels. For such purposes clad materials—generally high-performance copper-based alloys, with high purity cladding either side for protection—are usually the most suitable. These are normally secured to the basic chassis structure by pop rivets. For this purpose, and also for general body shape panels, it is also possible to use various aluminium alloys. These fall into two different classes: copper alloys, which may be heat-treated but not welded, and manganese alloys, which may be welded but are not normally heat-treated. It is also possible to use sheet Elektron (magnesium alloy) in such cases. This material has the great advantage that it weighs approximately one-third less than comparable aluminium alloys. Elektron is also widely used on specialist high-performance cars in the form of castings.

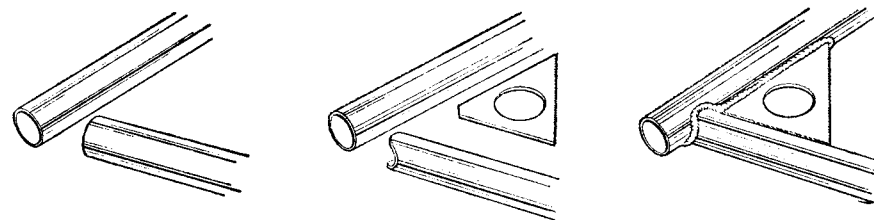
With any type of tubular chassis it is normally necessary to use an entirely separate body, but in unitary construction cars the two purposes can conveniently be combined. For limited production this virtually limits the



8 *A simple T-joint*



9 *A V-joint on to a straight tube*



10 *An oblique joint with a strengthening gusset*

field to non-metal structures, the cost and complexity of a complete stressed metal skin car being virtually prohibitive. As mentioned in the previous chapter, tooling costs make pressed metal structures uneconomical unless at least 2500 can be sold, and this leaves only glass-fibre and wood, of materials currently in use. As is shown by the Lotus Elite, glass-fibre can be used successfully with very little metal reinforcement, but the problems involved almost outweigh any advantages which are gained and it is not inconceivable that such a car could be made more economically with a separate chassis without any adverse effects on its over-all performance. Even glass-fibre involves tooling costs in the form of moulds (injection moulding would seem to be the solution here, but it in fact raises costs into the pressed steel structure range) and for even smaller scale production a possible answer is wood, as instanced by the Marcos. Here plywood is combined with glass-fibre for some of the curved panels and the result is an extremely stiff and durable structure. Both glass-fibre and wood have the advantages that they do not suffer from rust or corrosion, and in addition they are not subject to fatigue. Modern wood treatments render fears about beetles completely irrelevant in this connection, and should also obviate the risk of dry rot! Three-dimensional curves are extremely costly to make in wood, but this can normally be overcome by using glass-fibre for relatively small areas.

Glass-fibre—resin-bonded glass-fibre to give it its full name—is still a comparatively new material, and very little is generally known about its properties or method of construction. The raw materials involved are the glass-fibre “mat” itself, in the form of strands or thin sheets, and the resin which is employed to bond the fibres together. Successive layers of glass mat and resin—when set hard—form a very tough and resilient material. The ultimate in both strength and lightness is provided by two thin layers of glass-fibre separated by a layer of plastic foam.

The process of setting, or curing as it is generally known, is the most critical. Most resins—particularly polyester, the one most commonly used—either shrink or distort on curing. It is normally necessary to use heat to reduce curing time, and this is liable to increase the amount of distortion. Distortion is also more liable to occur in thicker sections—the more important ones in a stressed chassis/body unit—than in thin shaped panels. Shape is provided by the use of a mould made on a wood, plaster or metal pattern—which in many cases is a hand-made prototype car. In this mould the required amount of glass-fibre is impregnated with resin and cured.

The chief disadvantage of this method is that the reverse side always has a rough unfinished appearance which can only be avoided by match moulding—that is by lowering a male mould on to the female mould before curing in order to compress the material and provide a smooth finish on both sides. An alternative is to use two laminates bonded back to back—as on the Lotus Elite—but both these methods are rather too elaborate and much too expensive for the small-quantity producer. Further-

more, the quality of the exterior finish depends largely on the skill of the operator and the way in which the layers of glass mat are overlapped. Unless great care is taken this can lead to small ripples on the surface—which are further exaggerated by shrinkage during the curing process—giving the car a characteristic “special-bodied” appearance.

The ideal method, of course, is injection moulding, whereby sheets of glass-fibre are laid between male and female moulds into which resin is injected, the panel then being rapidly cured by electric elements incorporated in the moulds. Unfortunately, in order to withstand the pressures and temperatures involved, such moulds must be made out of metal—which makes them almost as expensive as the dies for pressed steel bodies. This method also takes much longer than the normal steel stamping, but has been successfully used by General Motors on the body of the Chevrolet Corvette. In the future, however, it may be possible to make moulds of this type from synthetic materials and to inject both the resin and the glass-fibre filaments at the same time. Such a development would obviously bring glass-fibre into the true mass production range. The costs involved would probably remain beyond the means of the small-scale producer, but for such people—employing the material for the purpose for which it is best suited and using great care at all stages of production—glass-fibre has much to recommend it, whether for stressed chassis/body units or merely for separate body panels. It has already been used structurally, with metal reinforcement, in open cars and it may not be long before we see open glass-fibre chassis/body units with no metal in them whatsoever.

Whatever type of chassis or body is used, it is normal for suspension linkages to be made of some form of steel, and in this case the choice is limited largely to the materials mentioned under this heading in connection with tubular chassis. In the majority of cases ordinary mild steel is completely adequate for wishbones, radius arms, de Dion tubes (if used) and most other forms of suspension linkage. Suspension uprights can either be steel fabrications or castings, and in the latter case the most suitable material is Elektron, on account of its excellent strength-to-weight ratio. For extremely small quantity production, however, fabrication from sheet steel is probably the best solution, and in fact the use of tubular or sheet steel throughout the car is strongly to be recommended if high quality and low cost are to be combined in the most satisfactory manner. In such a case aluminium or glass-fibre bodywork is to be preferred, and in this instance it is interesting to note that several manufacturers currently make a single aluminium body for each new model they produce, using this as a pattern from which to make moulds for subsequent glass-fibre bodies. As long as production runs into double figures, it should be possible in this way to undercut the cost of making the bodies separately from aluminium.

Specifications and properties of the most commonly used materials for chassis, bodies, suspension linkages and other components are included in Appendix II.

CHAPTER IV

THE PRINCIPLES OF A TRUE SPACE FRAME

By definition, a true space frame is a complete structure in which all joints could be flexible without the chassis losing any of its stiffness thereby. This means that stiffness must not in any circumstances be imparted by putting bending loads into members at joints. In fact joints should only be loaded in tension or compression.

In addition to the complete chassis being a structure, every section should, in itself, be a complete structure. As has already been mentioned, practical considerations require the chassis to be divided up into a number of bays. Should any one of these be structurally inferior to the others the whole chassis will suffer as a result.

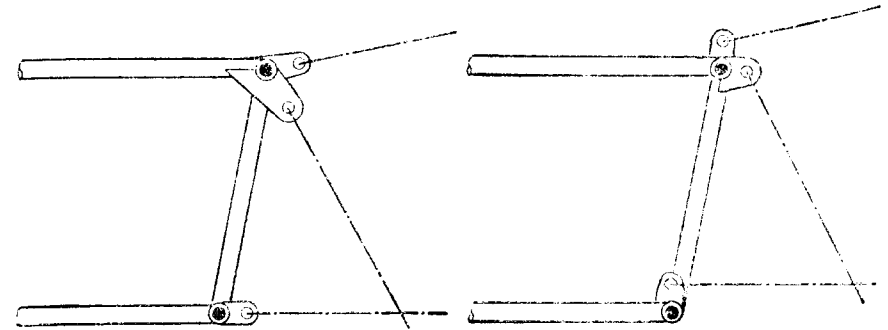
The first requisite in a chassis of this type is a balanced structure, as apart from being a space frame it is also a means of joining and reacting loads in all planes. Therefore it is extremely important that the over-all structure be borne in mind from the first stages of design. Furthermore, even after completing a satisfactory chassis frame for any given car, the designer must be not afraid to alter the whole conception because of some minor change required in suspension or other components. As has been emphasised earlier, the chassis must always be regarded as a means to an end and never as an end in itself. Having established this, it is extremely difficult to discuss space frame principles without at the same time taking into account the purpose for which the car in question is being designed.

In general, after dividing space frames into two types—single-seater and two-seater—and bearing in mind the obvious differences between them, it is still possible to consider the two chassis as being basically similar from a structural point of view. The simpler case is obviously the single-seater. For a car of this type, the starting point of design is the location of the pick-up points for all the various items of equipment which must be connected to the chassis. The most important of these are front and rear suspension, steering gear, engine and transmission, seat, gear lever and pedals—and other heavy items such as fuel and oil tanks, radiator and batteries (if carried). Having decided upon the wheelbase and front and rear track of the car, the approximate frontal area and over-all shape, the next step is to plan out as nearly as possible the exact positions of all the main chassis tubes, placing them so as to obtain the largest possible section through the structure while providing location for all the major brackets. It is also necessary to design all brackets to take their respective

loads into the chassis in the most advantageous manner, as is shown in the accompanying drawings of suspension brackets (fig. 11).

However, although these drawings show the more desirable positions for brackets feeding suspension loads into the chassis, there may well be some overriding factor considered more important by the designer, such as unjustifiable aerodynamic disturbance or undesirable effects on suspension geometry, which would make it better to have an inferior structure for the sake of a superior mechanism. This is where compromise begins.

In all cases chassis design must be completed before the choice of materials is made or work is begun. Design actually starts as a series of forces, which are later made into chassis tubes. As a series of lines on paper,



11 (Left) *The ideal method of feeding suspension loads into a chassis, through brackets which are on the same axes as the linkages involved.* (Right) *By comparison with the diagram on the left this might be termed "the wrong way". In practical terms, offsets of this nature represent the limits of acceptable compromise when other considerations prevent the use of the ideal layout*

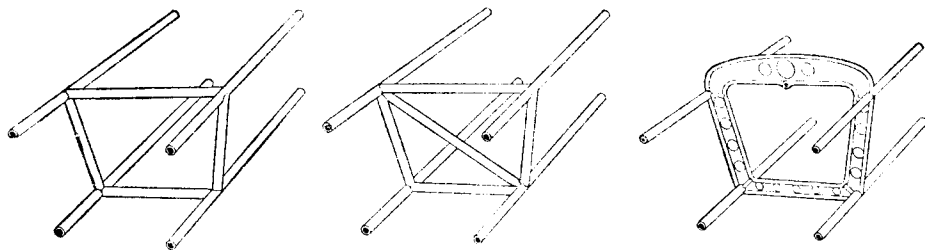
a chassis drawing shows the experienced designer the location of all chassis loads and gives him an idea of their magnitude. Using his experience he can decide which members should be made in relatively large section, thick-wall tube and which require only light tubing. The beginner must either work out a complicated series of calculations—as shown in Appendix I—or follow the example of established constructors. For the current high-performance sports or racing car the majority of needs are met by mild steel tube in the following dimensions: 1 inch diameter 16 gauge round, 1 inch 18 gauge round and 1 inch 18 gauge square. For comparatively lightly stressed members $\frac{3}{4}$ inch 18 gauge tube might prove to be quite adequate. All these dimensions, however, are dependent on the chassis being designed on space frame principles.

Ideally, to design a complete space frame starting from scratch, it is necessary to stress the whole chassis three-dimensionally. Whatever is

done in this field the designer must think in terms of deflection rather than stress and strain. In practice, experienced designers are unlikely to attempt full stress calculations on any new chassis but rather to rely on experience and to put purity of structure second to convenience of construction. When tackling design this way it is essential to know what compromise can be made without detracting from the over-all stiffness of the structure. If practical considerations demand it, poor engineering can be accepted—but only if the designer himself recognises it. An experienced man can design a satisfactory space frame—and one which in practical terms may be superior to an ideal structure—without recourse to calculations. But the beginner can hardly expect to do the same thing, even if he attempts to copy the work of a major constructor. This was clearly shown in Formula Junior racing during 1960. Even the most experienced designer will probably use a scale model to check his ideas on vital points before committing himself to the actual construction of the car. Studies of this nature must be purely qualitative, however, as it is not possible to work out loads on a scale model.

Much has been said and written in recent years about cars, particularly racing and sports cars, becoming too light. Experience has shown, however, that light weight is a major criterion of performance, particularly in terms of acceleration and roadholding. In view of the weight restrictions now imposed in almost every sphere of international racing, it must be emphasised that chassis and suspension components should always be designed to be as light as possible, any weight which must be added being in the form of useful parts. It should also be pointed out that parts of a chassis can be made too heavy, to the detriment of the structure as a whole as a result of distortion caused by built-in stresses. For ultimate performance it is still a good maxim to design light and add weight where experience dictates.

Whatever the chassis, the tubes of which it is composed must be suited to the loads which they are to carry. Ideally no tubes should be loaded in



12 *The bulkhead shown on the left has no stiffness against lozenging. That in the centre has been made into a structure by the introduction of a diagonal. On the right is shown a method of bracing a frame in which it is not possible to fit a normal diagonal. This bulkhead consists of two tubular hoops joined by perforated sheet steel*

bending, but in practice this is very difficult to achieve. In the case where an offset is required for mounting a vital component, the trend is to increase the diameter of the tube to give a greater section and avoid difficulty from compound loads. It is also important to avoid using long unsupported members, that is to say members which have an excessively high length/section ratio. And practical considerations—such as increasing the gauge of a tube which is to be used as a footrest, in order to allow for wear—must always be borne in mind, however complex the calculations which are carried out.

In all chassis, and particularly single-seaters, the provision of space for the driver prevents full triangulation of the top frame of the driving bay. This consideration alone makes the design of an ideal chassis almost impossible. However, various methods of external triangulation are possible, as shown by the Mercedes-Benz 300SL and the space frame Lister-Jaguar. The principle in this case is to build up diagonally braced beams or “torsion boxes” on either side of the car and to link them with fully triangulated bulkheads. It would also be possible to brace the top frame of the cockpit—at least on a single-seater—by means of a “perforated hoop” type structure as used for the scuttle bulkhead on some Lotus models.

CHAPTER V

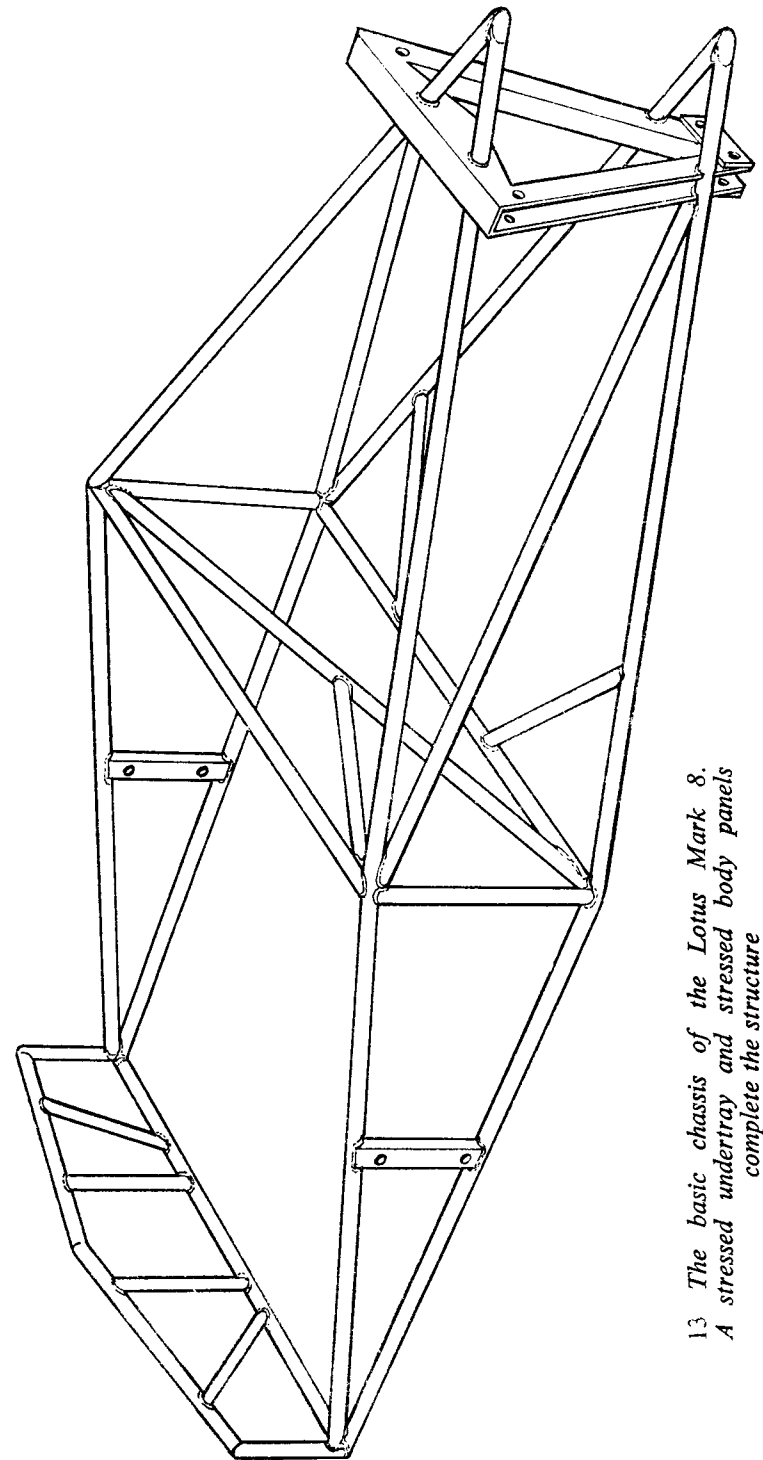
ANALYSIS —
SPACE FRAMES AND UNITARY
CONSTRUCTION CHASSIS

The Lotus Mark 8 (See fig. 13)

Structurally, the most nearly perfect sports car chassis yet made is that of the Lotus Mark 8. Very simple, extremely light and very stiff, this chassis—the only one of its type ever made—is still giving good service after six years of use, and this despite the fact that it is made up of 20 and 18 gauge tube. As can be seen from the accompanying drawing, the primary structure is fully triangulated and therefore extremely stiff. It consists of two sections on either side of a central bulkhead, the forward section being triangular in plan view, the rear triangular in side elevation. Only nineteen members are used in its construction, and the total weight is 21 lb. All members are straight and there are no structural offsets.

From a practical viewpoint, however, this chassis is open to a great deal of criticism. It was designed as a pure structure, with little thought for the loads which were to be fed into it, and thus it was necessary to add to the ideal basic structure a number of less satisfactory secondary structures through which these loads could be engineered. It has since been shown that it is far better to merge primary and secondary structures in the interests of a superior over-all chassis. In addition, practical experience soon showed that even the ideal basic layout had serious limitations from the point of view of maintenance. As an instance of this, it was necessary to dismantle the engine in order to get it into or out of the chassis. Cylinder head, manifolds, oil pump, water pump, distributor, front mounting, starter and dynamo, all had to be removed before the engine would pass through the narrow opening in the top frame. Because of the disturbance involved this almost inevitably meant that the engine produced less power in the car than when it was built up on the test bench. In addition, the use of stressed bodywork interfered seriously with the servicing of many smaller components.

By contrast with the simplicity of the primary structure, the secondary structure necessary to feed suspension and other loads into the chassis is extremely complicated. To deal with the front suspension loads involved by the use of a swing axle layout, it was decided to build a triangular frame of top-hat section sheet steel, with the base of the triangle passing across the two top chassis members some 6 inches to the rear of the front upright



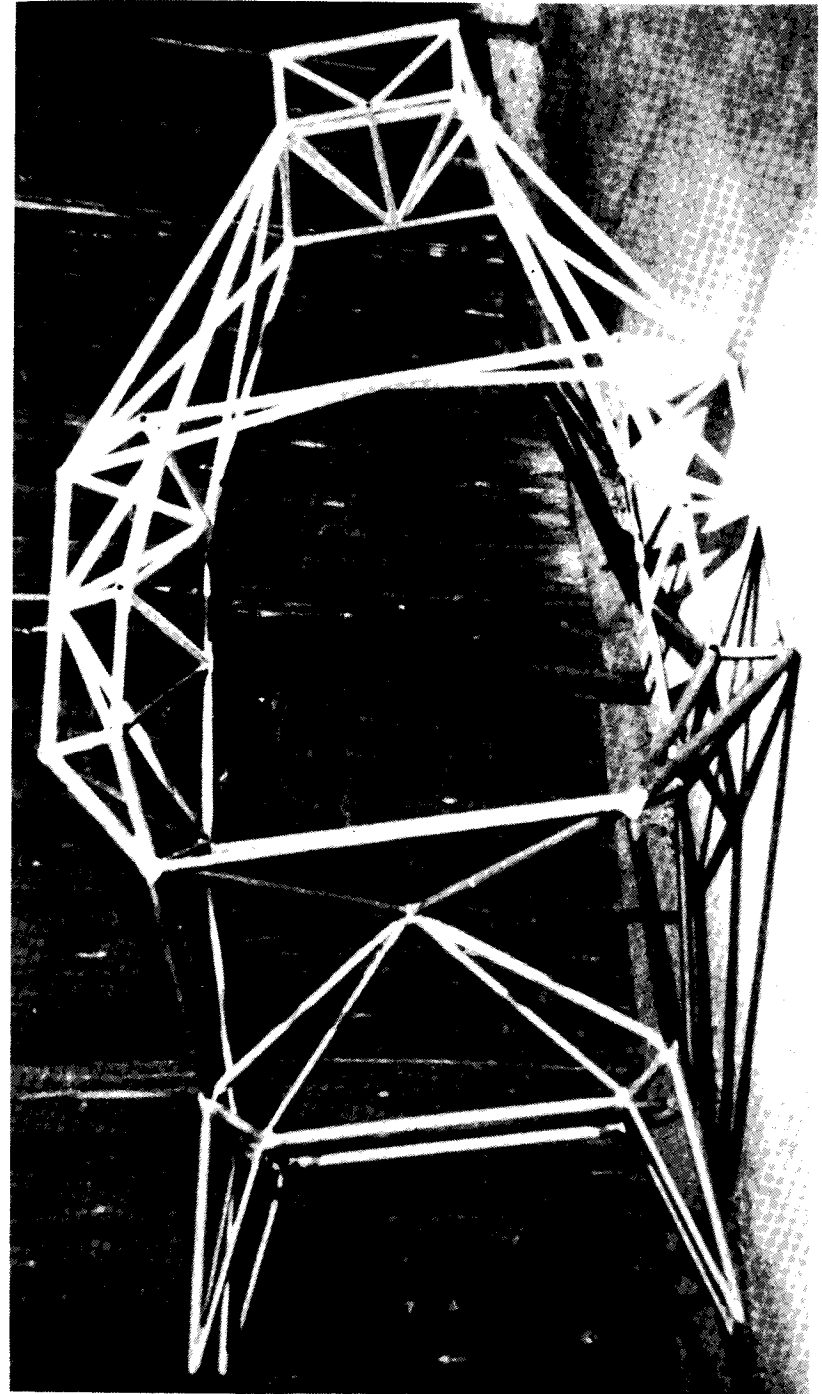
13 The basic chassis of the Lotus Mark 8.
A stressed undertray and stressed body panels
complete the structure

member. The outer ends of this transverse member were designed to accommodate the top eyes of the suspension unit. Just inboard of these pick-ups are welded the two other members of the triangle, which converge to form pick-up points for the swing axle eyes at their apex near the undertray line.

This triangular frame deals with the main front suspension loads. Suspension drag loads and brake torque reactions are taken out by radius arms located at a point approximately three-quarters of the way along the bottom chassis tubes; any consequent bending loads are taken out into the undertray by further tubes which run diagonally back towards the centre of the car, meeting the transverse bulkhead at the front of the undertray section.

One of the aims of this design was to mount the engine in such a way that loads could be reacted directly through the front suspension—in fact to hang the engine on the front suspension. This led to the construction of an extremely complicated front engine mounting, which consists of a tubular pyramid of four $\frac{3}{8}$ inch thick wall steel tubes picking up at various points on the front of the engine and converging to meet at a steel bush which acts as a housing for a Silentbloc-type bush. This is supported with the axis of its mounting bolt in a fore-and-aft plane and passes through two vertical flanges of the top-hat section member on the centre line of the chassis. From the chassis viewpoint this is quite satisfactory, but special triangular brackets are necessary to avoid twisting of the steel sandwich plate at the various pick-ups around the front timing cover of the M.G. engine. The rear engine mounting is taken from the normal M.G. position on the gearbox and loads are taken out into the propeller shaft tunnel, which is of stressed skin construction and designed to take bending loads from the gearbox as well as driving and brake loads (in the form of torque loads resolved into bending) from the differential unit.

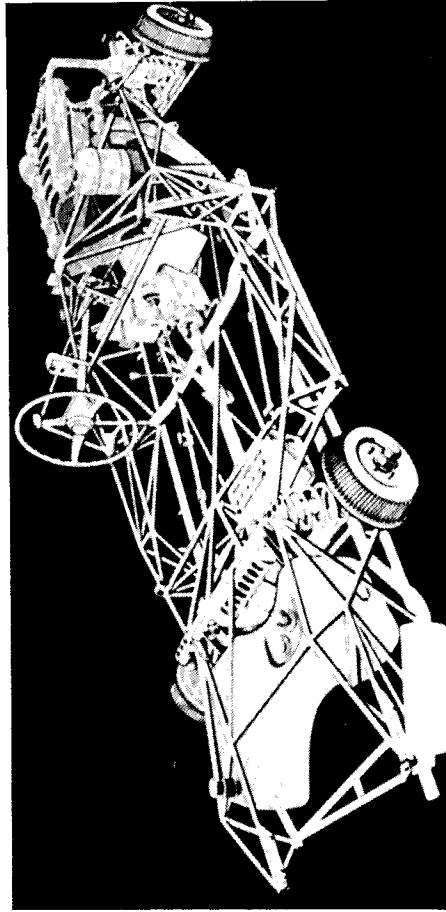
As at the front, a secondary structure is necessary to take out rear suspension loads. Vertical loads are taken out at the apexes of the triangles which form the seat-back bulkhead and lateral loads are fed into the undertray. The differential unit, which also provides lateral location for the de Dion assembly, is located by four mountings, one above and one below on the centre line and one on each side. Loads from the latter are taken out into the side walls of the propeller shaft tunnel by means of steel brackets. In practice this layout has given considerable trouble, as the passage of loads from extremely stiff brackets into 20 gauge aluminium leads to straining and eventual failure. Trouble has also been experienced with the complicated mountings for the transverse coil springs, which incorporate piston type shock absorbers as suspension linkages. Fore-and-aft suspension loads are taken out by parallel, horizontal arms into a vertical side frame member, located at the junction of the main side frame member and a secondary tube which comes forward to meet it from the seat-back bulkhead. This member also helps to overcome a possible weakness in the shallow section of the chassis side frame immediately in front of the seats.



14 The basic chassis of the space frame Lister-Jaguar. A full-length undertray is needed to complete the structure and further sub-frames are necessary for the mounting of engine and other components



15 Mercedes-Benz 300SLR sports racing car, with space-frame chassis low-pivot swing axle rear suspension, and brakes mounted inboard at both front and rear



16 The space frame chassis of the Mercedes-Benz 300SL. Torsional stiffness of this chassis (and that of the 300SLR) is derived largely from its rather complex-looking side frames, which function effectively as beams. Here outboard brakes are used

SPACE FRAMES AND UNITARY CONSTRUCTION CHASSIS

The Lotus Mark 8 was the first sports car to feature fully aerodynamic bodywork, designed by Frank Costin and supported on the chassis by light alloy sheet bulkheads. Only the front section of the bodywork is removable, the remainder being riveted to the sheet alloy which supports it. The front body mounting consists of a tubular and sheet steel structure coming forwards from the front vertical member of the chassis and also incorporating the radiator mountings.

The Space Frame Lister-Jaguar (See fig. 14)

In many ways, although outwardly it appears much more complex, the space frame chassis designed for Brian Lister by Frank Costin can be compared directly to the chassis of the Lotus Mark 8. Like the Lotus it can be summed up as a virtually ideal primary structure, with somewhat less satisfactory secondary structures dealing with engine, suspension and transmission loads. In some ways it can also be compared with the Mercedes-Benz 300SL chassis, particularly in the way it achieves torsional stiffness of the cockpit section by means of external beams on either side of this bay. In over-all design, however, it is far superior to the Mercedes, particularly as the structure is much more of a unit and great pains have been taken to ensure continuous load paths. It is also noteworthy that there is no weakening of the structure from a torsional point of view behind the rear axle line. Unfortunately, however, the purity of the structure illustrated in fig. 14 is somewhat impaired by the addition of the members whereby suspension and other loads are engineered into the chassis. It should also be remembered that the structure illustrated is incomplete due to the absence of the undertray and other sheet metal stiffening members.

Basically the chassis consists of a fully triangulated front box—on which the front suspension is mounted—and two longitudinal beams, linked by a Portal frame in the centre and a fully triangulated rear bulkhead; based on the latter is a further complete structure which deals with rear suspension loads via the differential housing, the coil spring/damper units and the leading radius arms. The full-length undertray is an essential part of the chassis, providing diagonal bracing of the entire bottom frame.

Adequate bending stiffness is imparted to the two side "beams" by the central Portal frame, which restrains any outward movement of the side members. It also withstands torsional loads, in conjunction with the front and rear bulkheads, and all three combine with the very rigid side frames to produce an extremely stiff structure. One point on which the primary structure illustrated might be open to criticism is the use of long unbraced tubes in the side of the front bay. In practice, however, small diaphragms are fitted to provide stability at these points.

Mention of load paths has already been made in connection with this chassis, and in this context it is worthy of note that front suspension loads are taken out directly by the rear suspension, and vice versa.

In any criticism of the secondary structures required to mount rear suspension elements and some other components, it must be remembered that the car was, of necessity, designed to use an existing body. Thus the rear radius arms could not pick up directly at the apexes of the rearmost triangles, and it was necessary to engineer rather complex sub-frames to bring the loads in to these points. However, neither these nor the other secondary structures in this chassis have given any trouble in use, and it is significant that subsidiary engine mountings, fitted at the request of the car's owner, have not been brought into use.

In conjunction with a de Dion axle it is desirable that the linkages controlling the longitudinal location of the wheels should be as long as possible to reduce fore-and-aft movement on bump and roll to a minimum. To achieve this, and to induce understeer with roll, leading radius arms are used. Significantly enough, a similar layout is used to control the live rear axle of the Marcos coupé.

Mercedes-Benz 300SL (See fig. 16)

The Mercedes-Benz 300SL was one of the first road-going cars to be fitted with a high-performance chassis. The aim behind its design was to produce an extremely fast touring car with luxurious appointments, and for structural reasons it was decided to use a space frame chassis. On its introduction this car attracted a great deal of attention by reason of its novel, gull-wing-type doors. This was no gimmick, but a direct result of the depth of the sides of the chassis, which precluded the use of the standard type of door opening with coupe bodywork.

Apart from the high door sill line which is involved, the chassis of the 300SL is extremely well designed from both the structural and practical engineering viewpoints. It is built up of square and round section tubes, with few members loaded other than in tension or compression, but the complexity of the design leads to the possibility of distortion during manufacture due to there being a large number of multi-tubular joints in which a considerable amount of heat is required to carry out the necessary welding operations. These lead to distortion and built-in stresses as the structure cools.

As on several other cars of this type, much of the torsional capacity of the frame is derived from an extremely stiff section in the region of the engine and scuttle bulkheads. Front suspension loads are fed directly into this area, and as a result of diagonal bracing on five of its six sides this section is easily capable of dealing with them. The difficulty of obtaining adequate torsional stiffness for the centre section of the car is overcome by taking loads round this bay by means of complex side frames, which have considerable horizontal depth and function effectively as beams. These beams run from the engine bulkhead to the seat-back bulkhead and have their maximum section at the scuttle bulkhead, which is also reinforced in

depth. This structure, involving some nine tubes on either side of the car, effectively replaces the single top diagonal which would be the ideal means of preventing lozenging in this top bay. Such a solution, however, is scarcely possible in a luxury car designed to carry at least two occupants.

There are two bulkheads behind this central section, of which the rearward one deals with the loads from the rear suspension and the chassis-mounted final drive unit; the forward one serves principally to complete the structure of the central bay. The primary structure continues to the rear and includes supports for fuel tanks and spare wheel in a rather complex series of tubes which somewhat limit the luggage carrying capacity of the car. The structure in this area is by no means as pure as in the remainder of the chassis, but as the maximum torsional stiffness is required between the axles this is not of paramount importance.

One of the chief grounds on which the 300SL is open to criticism is the use of a simple swing axle type of rear suspension in conjunction with what is otherwise a relatively advanced (and expensive) design. On the 300SLR sports/racing car the same basic chassis was retained, but a low-pivot type of swing axle rear suspension was used (see fig. 46) and this provided considerably improved rear wheel adhesion. The open 300SLR also caused a sensation on its first appearance at Le Mans by the use of an air brake in support of its massive drum brakes—mounted inboard all round on this car. The air brake, a hinged metal panel mounted behind the driving compartment, had a considerable decelerative effect at high speed and undoubtedly played an important part in the car's performance on this very fast circuit.

Lola Sports (See fig. 20)

Although interest in 1100 c.c. sports car racing decreased during 1959 and 1960—due partly to the introduction of Formula Junior—there is no doubt that the most successful car in this period was the Lola. With its fully independent suspension—of an advanced type—in conjunction with a stiff, well-designed chassis, it could well go down to posterity as one of the classic cars of the post-war era.

Despite his comparative inexperience, Eric Broadley obviously had a remarkable appreciation of the factors influencing sports cars design when he made his first Coventry Climax-engined car in 1958. His suspension layout was designed to take out loads over a wide base and to absorb them progressively into the chassis structure. He found room for a straight steering column—no mean feat in a small, front-engined car. He followed current practice in the use of cast magnesium wheels and inboard rear brakes, in the interests of unsprung weight, but he retained drum brakes while most of the opposition were changing to discs and lost nothing as a result; in fact, to judge from the evidence of the race track, he probably gained by it.

The chassis appears rather complex but is extremely interesting—and very well made. The whole design is based on providing stiffness at the mounting points for major items, and thus the frame is swept upwards towards the front to provide good mountings for the front suspension units. The front wishbones are all mounted on upright members, but in such a manner that the disadvantages of such a layout are largely overcome.

Firstly, although the suspension loads are taken out into vertical members, these are supported by a series of triangles which absorb the loads and pass them into the main chassis structure. In addition, the vertical members are of square section tubing and have bushes welded through them to provide mountings for the wishbones. Square tubing has somewhat limited torsional rigidity, but in this case it seems likely that the bending loads involved are of more importance than induced torsional loading because of the offset of the wishbone pick-ups. Square tubing is, of course, superior to round section tube in respect of bending loads, and also provides a much better base for mounting the wishbones. And although the rear arm of the front top wishbone is mounted on a short tube between the top rail and the side-frame diagonal, it is unlikely that the lateral loads it has to deal with, at this distance from the hub centre line, are of any great magnitude.

The front bulkhead, which incorporates the front suspension pick-ups at each top corner, is braced by two diagonals, with the top member—which is in any case redundant—left out; the chief reason for this is to facilitate engine installation.

The rear suspension fore-and-aft loads are taken out on each side by a single radius arm, slightly curved to clear the tyres. This radius arm and the fixed-length drive shaft form a top wishbone. Transverse loads are reacted by the drive shaft and a lower transverse link. The suspension units are mounted at the top corners of the rear bulkhead—again composed of square section tube, with the top member slightly curved to follow the profile of the bodywork. Although this is not in keeping with the rest of the chassis, the amount of curvature is not excessive for a compression member and the resultant construction is lighter than if a straight member and an additional body support tube were used.

Members running forward from the suspension unit pick-ups deal with the fore-and-aft component of the suspension unit loads, and the structure is completed by diagonals from the bottom corners of the bulkhead which meet at the centre of the top member. A vertical member which is also located at this point takes out the vertical component of the final drive unit loads, while fore-and-aft loads are dealt with by a member running forward to the top of the propeller shaft tunnel—where they are diffused in tension and compression. The odd diagonal in the rear bulkhead locates the top pick-up on the final drive unit in a lateral plane, and could be at either side. The final drive unit is also supported by two mountings at its base.

Much of the strength of the Lola chassis is derived from an extremely stiff central section. This is composed of the engine and scuttle bulkheads, braced to form a box which imparts stiffness to the bays on either side of it. From the top of the engine bulkhead—at the point where the top diagonal in the front bay meets a central upright member—diagonals run to a transverse member which is in fact slightly ahead of the scuttle bulkhead but is effectively part of it. Diagonals in the side frames and a stressed undertray make this an extremely stiff section. Much the same may be said of the front bay, which is triangulated on five sides—by the stressed undertray in the case of the floor section.

Because of the stiffness of the central bay it was possible for the cockpit sides to be fairly low without any detriment to the over-all structure, provided they had adequate bending stiffness. Design in this area was thus centred largely on providing good mountings for the rear radius arm.

The efficiency of the Lola chassis is visible not only on the race track but also in the workshop, where both routine maintenance and major overhauls can be accomplished without too much difficulty despite the use of a space-frame layout. Skilled welding is essential for a structure of this type, with as many as eight tubes having a common apex, and from this aspect—as well as from the point of view of design—the Lola should be an inspiration to small-scale constructors.

1960 Formula One Lotus

Although the success of the 1959 Formula One Cooper originally inspired the current trend towards rear-, or rather mid-engined racing cars, the chief influence in this sphere subsequently became the 1960 Formula One (and Formula Junior) Lotus. The basic requirements which prompted the creation of this car were few and simple. They were to minimise power losses, to reduce frontal area to an absolute minimum and to make the centre of gravity as low as possible. The only answer was to put the engine just ahead of the rear wheels and to make the frontal area roughly that of the seated driver; the rest followed.

The chassis is a space frame made up of 1 inch and $\frac{3}{4}$ inch, mainly straight, round mild steel tubes, with a fabricated scuttle-hoop and a small fabricated frame at the rear. No extreme weight-saving measures are adopted, so the main tubes are of 18 gauge—and some 16 gauge—steel, whereas in the past Lotus practice was to use a considerable amount of 20 gauge material.

Basically the chassis consists of three sections. Within the forward section the front suspension bay is fully triangulated and forms a complete structure in itself. Ahead of it is a separate, lighter structure which carries the radiator, oil tank and body supports. Welded on to the forward end of the front suspension bay are the rack and pinion mountings, the brake and clutch master cylinder mountings and the forward front suspension pick-

ups. Attached to the rear of this bay are the rearward front suspension pick-ups and supports for the brake and clutch pedals.

The front chassis bay extends rearwards to the scuttle hoop, which is a tubular and sheet structure so designed as to eliminate the necessity for diagonal bracing, and at the same time to provide mountings for steering column, instruments, switches, gear-lever, handbrake (and cable end stop) and seat. Forward of the scuttle the frame is a fully braced structure, but the next bay—from the scuttle to the engine bulkhead—is only braced on five sides, and only becomes torsionally stiff as part of the complete chassis, drawing its rigidity from the structures on either side of it.

In addition to the engine bulkhead, which is fully braced, the rear chassis bay consists of a bottom frame of two straight tubes, which converge at the lower rear suspension pick-ups, a fabricated triangular rear frame and a top frame which is braced by a removable Y-shaped member. As elsewhere in the car, the side frames are braced by diagonals and the bottom frame by the stressed undertray. Diagonals from the bottom of the engine bulkhead to the top of the rear frame are braced by tubes running down from the top of the engine bulkhead and meeting at the engine mounting brackets; the engine mountings are thus at the junction of three tubes.

Although the fabricated rear frame is an effective triangle it is not in fact straight-sided, its shape being influenced by the need to accommodate the inboard rear brake discs, which would foul the straight members of a normal triangular frame. The top cross-member is constructed in one unit with the Y-shaped diagonal, and can be detached, to facilitate removal of the engine, by releasing four fitted shear bolts and one tension bolt.

The lower rear suspension pick-ups are situated immediately above the apex of the two bottom frame tubes. As the upper suspension links are the drive shafts, the pivot points in this case are the universal joints at their inner ends. The coil spring/damper units are located by brackets mounted off the top corner of the triangular rear frame. The top member of this frame acts as a beam, with suspension loads fed into it at either end and engine loads reacted in the centre, the gearbox casing being located at this point via two rubber cone-bushed mountings. Fore-and-aft suspension loads are taken out into the chassis through radius arms anchored on both sides at the top and bottom of the engine bulkhead.

Front suspension (see fig. 41) is of the double wishbone type, the tubular top member having a threaded, ball-jointed outer end to allow adjustment of camber angle. The ends of the front anti-roll bar are picked up through spherical bearings beneath the ball joints and all vertical loads are passed directly into the wheel assembly. Machined steel hubs, proprietary Standard-Triumph uprights (with a nylon bush bearing for the bottom knuckle joint) and bolt-on cast magnesium wheels are used. Steering is by means of lightweight rack and pinion gear mounted just ahead of the wheel centre line. A straight steering column is used, bolted to a 14 inch alloy wheel with a leather-covered rim.

“Double transverse link” is Colin Chapman’s title for the rear suspension, in which the drive shaft forms part of the upper link (see fig. 31). Owing to the configuration of the layout, very little tensile or compressive load is taken through the drive shaft, so that although it is structurally a suspension member it is not heavily loaded in this connection. Each hub casting houses two deep-row ball races of sealed type and is extended downwards to provide a pick-up for the lower radius arm (the upper radius arm is located just above the drive shaft) and the outer ends of the lateral links. These are threaded at the inner end to allow alteration of camber angle and at the outer for adjustment of toe-in. The cast rear wheels have $6\frac{1}{2}$ inch rims.

The front brakes are $10\frac{1}{2}$ inch Girling discs, mounted outboard. The $9\frac{1}{2}$ inch rear brakes are normally mounted inboard, but can also be mounted outboard. Provision for this is made in the rear hub castings, and it is possible for the calipers to be assembled ahead of, or behind, the hubs.

Body shape is determined largely by the need to keep frontal area to a minimum. With an exposed wheel layout, tyre sizes being extremely large for such a small car, it is doubtful whether body shape makes an appreciable difference to maximum speed.

A major problem on such a small car was to find space for some 30 gallons of fuel. This has been achieved by locating a massive 22 gallon tank over the driver’s legs, with a further $9\frac{1}{2}$ gallon tank tucked in behind, and to the right of, the seat. The plan is to run on the front tank until ideal handling is obtained, then switch over to the centre tank—maintaining a virtually constant weight distribution—for as long as possible. Space also had to be found for a 6 gallon oil tank ($4\frac{1}{2}$ gallons of oil and $1\frac{1}{2}$ of air) and this was eventually made triangular in shape and mounted immediately behind the radiator.

In a rear-engined car, particular attention must be paid to the mounting of pipes and cables, to obtain the neatest possible layout commensurate with the number of items involved. No less than fifteen pipes and cables pass through the Lotus cockpit; three oil pipes (one pressure, two scavenge), two water pipes, oil pressure, water and oil temperature lines, gearbox oil temperature line, fuel pipe and fuel pressure gauge pipe, brake and clutch pipes, ignition cable and accelerator cable. It is also important that hot and cold pipes are kept apart if possible. Such details are often forgotten in consideration of racing car design, but each one plays a vital part, or it would not be there.

Vital statistics of the 1960 Formula One Lotus are 7 ft 6 in wheelbase, 49 inch front track and $47\frac{1}{2}$ inch rear track. Dry weight is approximately 770 lb. Starting line weight, with 270 lb of fuel, 180 lb of driver, 50 lb of oil and 20 lb of water, is about $11\frac{3}{4}$ cwt. Weight distribution at normal ride level is 44 per cent front, 56 per cent rear.

1961 Formula Junior Lotus (See figs. 17 and 18)

The 1960 Formula Junior Lotus is basically very similar to the Formula One car, although few parts are interchangeable. One notable difference is the use of a simple, triangular rear frame in place of the rear "hoop" used on the Formula One car. With the lighter engine—and reduced power—of the Formula Junior version, this means of construction is entirely adequate.

On its introduction this car was generally regarded as the ultimate in minimum dimensions. Yet beside the 1961 version it looks positively enormous. In his first Formula Junior car designed as such (the 1960 model was virtually a Ford-engined Formula One car) Colin Chapman contrived to fit all the necessary components into a body which had some 30 per cent less frontal area than the 1960 version and much better aerodynamic qualities.

As in 1960, the chassis basically consists of four bulkheads—for front suspension, scuttle, seat back and rear suspension—linked by longitudinal members and diagonals to form a very rigid structure. It is made up of $1\frac{1}{4}$ inch, 1 inch and $\frac{3}{4}$ inch, round and square section, 18 gauge mild steel tubes. In fact there are only two $1\frac{1}{4}$ inch tubes, the top left and bottom right side rails, which are used as water pipes between engine and radiator.

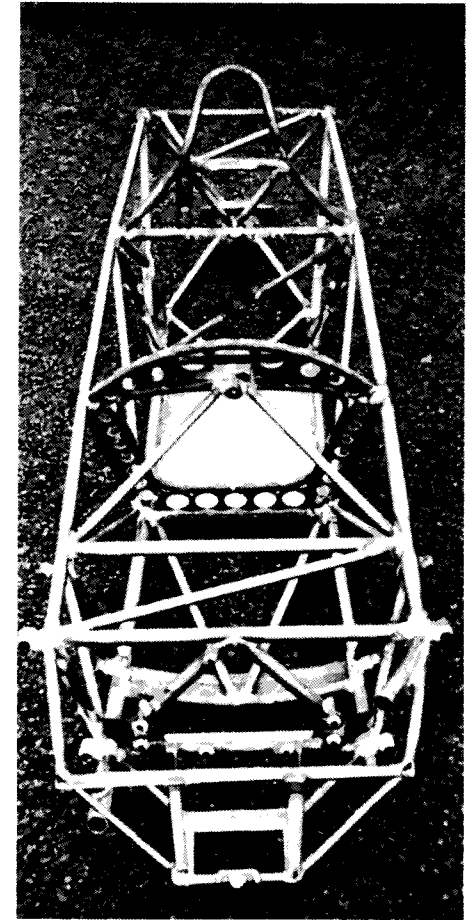
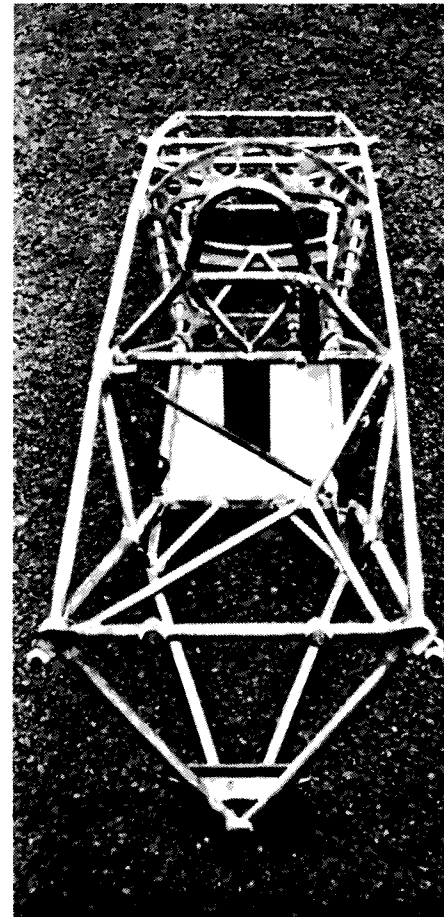
In cross-section the chassis is somewhat smaller than in 1960, but makes up for any consequent slight loss in torsional capacity by improved over-all design. As on the 1960 Formula One car, the top diagonal in the rear bay can be taken out, complete with the top section of the rear hoop, to facilitate removal of the engine and transmission as a single unit.

Practical considerations led to the structure becoming rather less pure than is customary on Chapman-designed cars, but this nevertheless proved an advantage in terms of over-all performance. For instance, the bottom tube in the secondary front bulkhead is curved to provide support for the driver's heels. And the design of the front sub-frame, supporting radiator and battery, is complicated by the fact that the bottom right longitudinal member is also a water pipe; a curved tube braces the radiator mounting on the left-hand side. In addition, the rearward front top wishbone mounting is offset from the ideal structural position in the interests of suspension geometry, and as a result the over-all chassis layout in this area appears somewhat untidy.

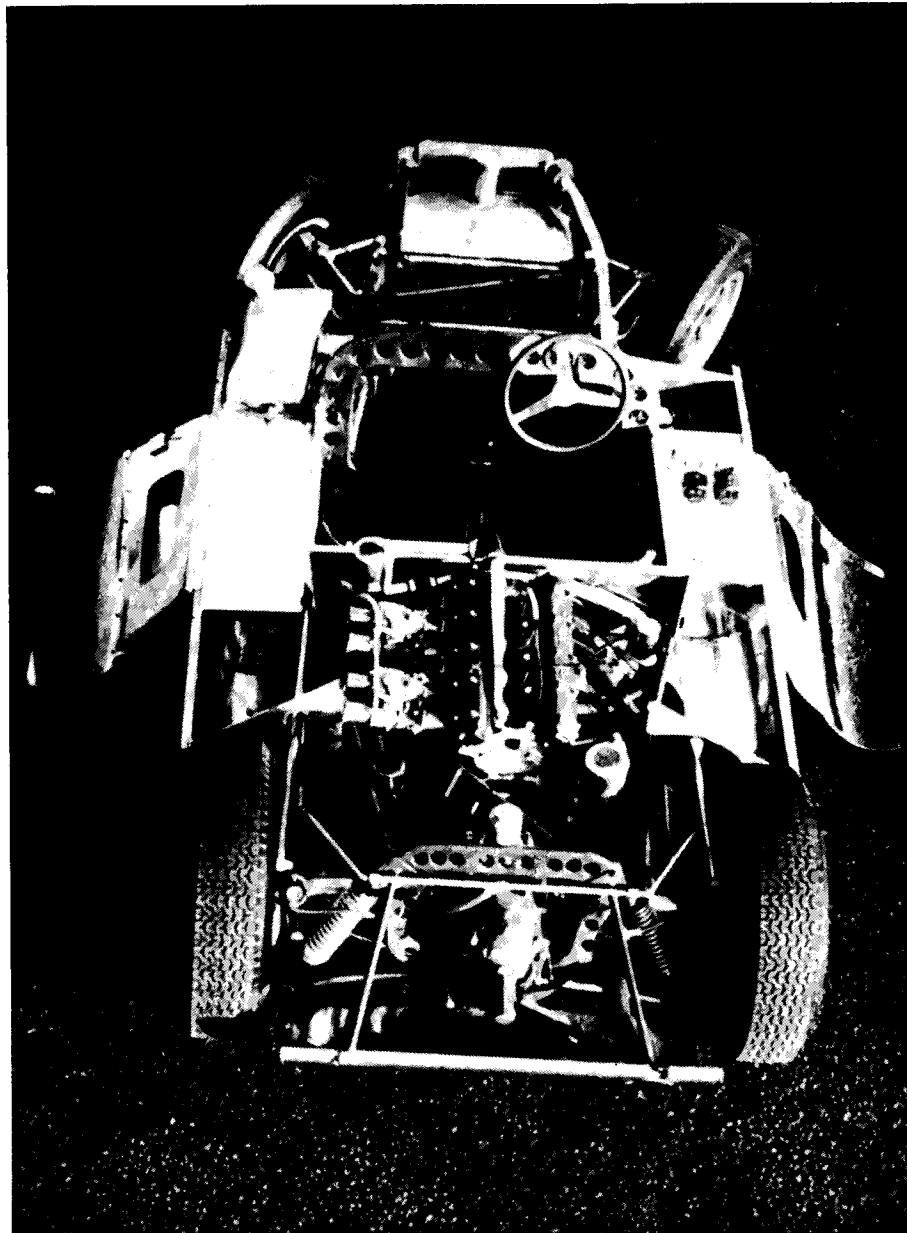
As the undertray is removable, the lower frame of the front bay is braced by diagonals which meet at the centre of the secondary front bulkhead. The top frame of this bay is also divided into three triangles by diagonals which meet at the steering column mounting. The lower frame of the rear bay is triangular, as in the 1960 car, but in this instance is composed of square section tubes to facilitate body mounting.

The chassis extends sufficiently far behind the rear wheel centre line to permit the use of a triangular rear frame in conjunction with inboard

17 *Front view: showing basic chassis lay-out, with top right and bottom left longitudinal members used as water pipes between radiator and rear-mounted engine. The front bay, extending back to the "perforated hoop" bulkhead, is triangulated on all six sides*



18 *Rear view: showing how the engine bay is fully triangulated—by a removable Y-shaped diagonal in the case of the top frame. This can be unbolted, complete with the rear cross member, to allow the engine and gearbox to be removed as a single unit*



19 Rear-engined sports/racing car: the Lotus Nineteen. Note the two "perforated-hoop" bulkheads and the triangulation of the top frame of the engine bay by a Y-shaped diagonal; note also the very good lock of the front wheels

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brakes, but this means that the mountings for the rear suspension units must be offset even more than is necessary to clear the drive shafts. The main diagonal of the right-hand side frame in this bay is curved slightly near its junction with the lower longitudinal member—to clear the oil filter bowl—but is composed of 12 gauge material to compensate for this.

In over-all terms, however, this chassis is adequately stiff for its purpose. Every frame except the top of the cockpit is fully triangulated and the structure as a whole can be resolved into a series of triangles.

The all-independent suspension system has been geometrically improved, and all suspension loads are taken out over a very wide base. The front suspension (see fig. 41) has been considerably modified, but the use of a lower front roll-centre, in the interests of minimum lateral deflection of the wheels, requires the use of a stiffer anti-roll bar to keep body roll within the limits required for optimum roadholding. At the rear (see fig. 42) there is one inch more ground clearance beneath the suspension uprights than on the 1960 car, while the inboard wishbone pick-ups are placed rather farther apart. Wheelbase and front track remain as in 1960 at 7 ft 6 in and 4 ft 1 in respectively, but rear track has been increased by one inch to 4 ft. Unsprung weight is reduced by the use of inboard brakes at the rear and 13 inch wheels, with 4.50 tyres, at the front; 15 inch wheels with 5.00 tyres are retained at the rear. To lower the centre of gravity, increase the proportion of total weight on the rear wheels and improve aerodynamic efficiency, the 8 gallon fuel tank is mounted behind and alongside the driver's seat. Starting line weight distribution with driver aboard is 60 per cent on the rear wheels, and this results in the car having a less pronounced tendency to understeer than the 1960 version.

Externally the most noticeable improvement is in stream-lining, for the 1960 car—although having very small frontal area—showed far less evidence of airflow studies than most Lotus models. The 30 per cent reduction in frontal area has been achieved mainly by the use of a smaller radiator and a more reclining seat, and in addition the symmetrical, elliptical section bodywork completely encloses all mechanical components—including carburettors and exhaust pipe—and body mountings. Even the Lotus badge is recessed into the bodywork. Air for the carburettors and cooling air for the inboard rear brakes is taken in through ducts at either side of the double curvature windscreen—yet another aerodynamic improvement. The top of the windscreen is 6 inches lower than on the 1960 car, and over-all height is 2 inches less. All the bodywork—including the undertray—is of glass-fibre, and can be removed for maintenance work in a matter of seconds.

In conjunction with the much smaller air intake—which greatly improves the car's penetration—the radiator (with its integral oil cooler) is surrounded by an elliptical shield which seals on the bodywork and forces all incoming air to pass through it. Furthermore, air exhausted from the

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radiator is deflected out through the body sides by a metal shield which surrounds the pedal compartment—thus preventing a cause of discomfort to the driver in hot weather. Praiseworthy design can also be noted in the rear section of the body, which contrives to enclose the engine and all ancillaries without any suggestion of bulbousness.

Lotus Nineteen (See figs. 6 and 19)

At the time of its introduction, the Lotus 19 was probably the most advanced sports/racing car on the circuits, and it is still very competitive given sufficient power. Based on the rear-engined 1960 Formula One Lotus, it has many features in common with the single-seater, but numerous alterations were necessary to turn it into a sports car. First and foremost, the space-frame chassis has been widened out to provide the seating space required by Appendix C of the International Sporting Code. It remains a three-section unit, as on the Formula One car, and in fact the front bulkhead (just behind the radiator) and the rear bulkhead (through which the gearbox protrudes) are Formula One components. Immediately behind the front bulkhead, whose location is determined by the forward front wishbone mountings, the chassis side rails begin to diverge. A secondary front bulkhead coincides with the rearward front suspension mountings. Diagonal bracing on five sides of this front suspension bay makes it exceptionally stiff and fully capable of absorbing all the loads passed into it by the front suspension and steering (the steering

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From the scuttle hoop rearwards the chassis side rails run, virtually parallel, back to the engine bulkhead. As on the Formula One car, this centre section is braced on only five sides, and with the greater width of the sports car becomes relatively less stiff than on the single-seater, but the rigidity of the front and rear sections compensates for this and gives the whole chassis adequate torsional stiffness.

From the engine bulkhead the main frame rails converge towards the rear bulkhead, constructed on the same lines as the scuttle hoop. Although not straight sided, the rear bulkhead is an effective triangle, its shape being determined by the need to accommodate the inboard brake discs. The upper tubes meet this bulkhead just above the pick-ups for the rear suspension units, while the lower ones converge more rapidly to meet at the bottom of the hoop, at the lower wishbone pick-ups. Fore-and-aft suspension loads are taken out into the chassis through parallel radius arms anchored at the top and bottom of the engine bulkhead side members. As this bulkhead is considerably wider on the Nineteen than on the single-seater this results in a slight alteration of geometry, but the suspension is otherwise identical to that of the Formula One car.

Also, as on the Formula One car, the upper part of the rear bulkhead is removable, together with a Y-shaped top diagonal, on the release of four fitted shear bolts and one tension bolt. In addition to absorbing suspension loads at either end, this member also deals with engine loads, the gearbox being located at its centre via two rubber cone-bushed mountings. Each front engine mounting is located at the apex of two triangles, consisting of the side frame rails and a diagonal member.

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Girling disc brakes are used, with outboard mounted 10½ inch front discs and inboard mounted 9½ inch rear discs. Provision is also made in the hub castings for mounting the rear brakes outboard; in warm climates it may well be worth the sacrifice of increased sprung weight to move the brakes away from the heat of the engine and gearbox.

The 12 gallon fuel tank is mounted on the left of the car—on the far side from the driver. The space above the driver's legs, used for the main fuel tank on the Formula One car, contains the spare wheel on the sports car. The battery is alongside the brake, clutch and throttle pedals. The cross-flow radiator and 6 gallon oil tank are as on the Formula One car, with the oil tank breathing through the chassis to a number of small holes at the rear. Aluminium water pipes link engine and radiator, and oil is circulated between engine, oil radiator and oil tank—and gearbox and gearbox oil radiator—by pipes which run outside the body panels over much of their length.

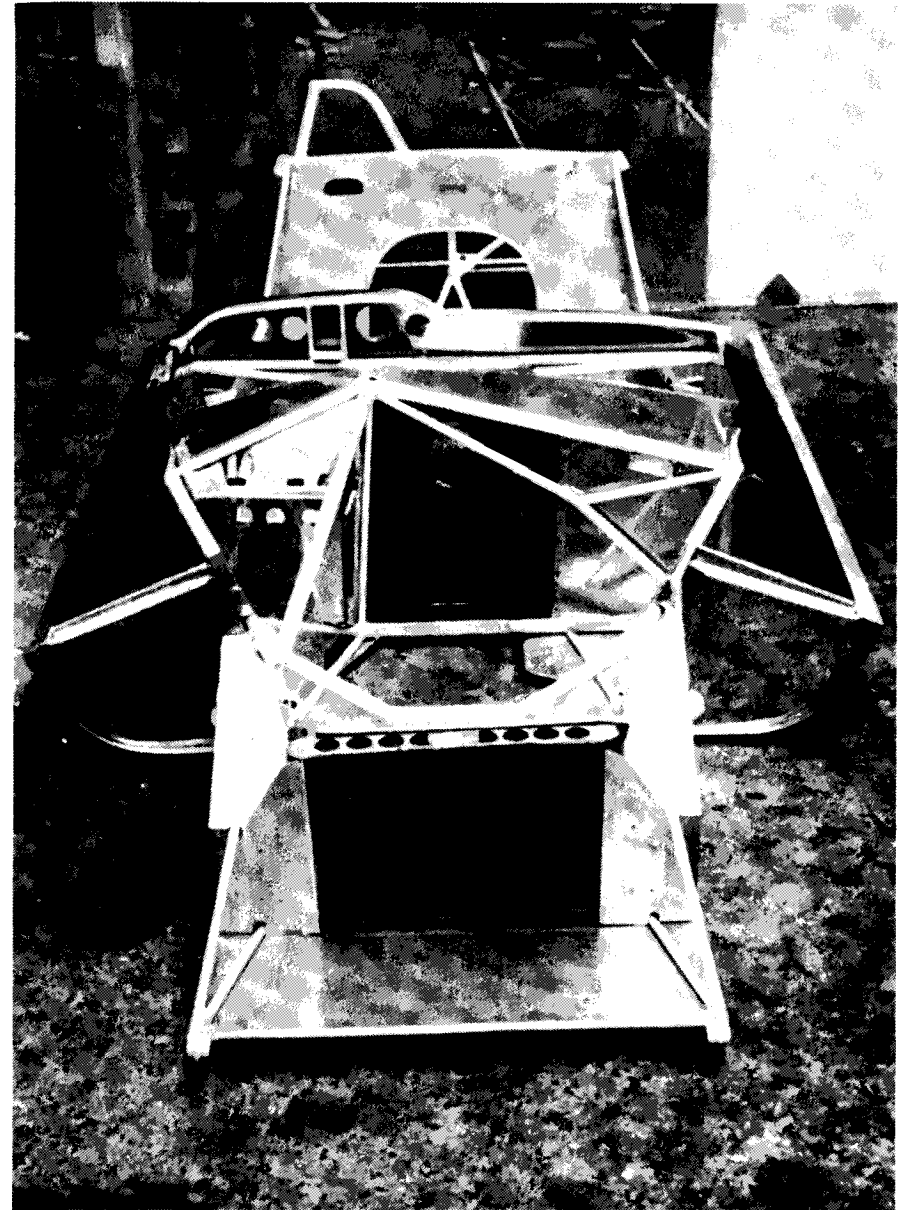
The bulk of the bodywork is in glass-fibre, consisting of two removable panels hinged at front and rear. The horizontally hinged doors, the lower body sides and the undertray are in aluminium. Lighting equipment consists of two 7 inch Lucas Le Mans headlamps, mounted behind plastic fairings, side, tail and brake lamps. To facilitate removal of the body panels, the electric wiring passes at both front and rear through waterproof plugs and chassis-mounted sockets.

The 7 ft 6 in wheelbase, 49 inch front track and 47½ inch rear track are as on the Formula One Lotus. Over-all length is 11 ft 9 in, width 5 ft 5 in, and weight, less driver, is approximately 11 cwt with 8 gallons of fuel. Weight distribution at normal ride level, with driver aboard, is 46 per cent front, 54 per cent rear.

1960 Formula Junior Lola (See fig. 21)

Although not particularly successful, the 1960 Formula Junior Lola came somewhere near the ultimate as far as the design of front-engined single-seaters is concerned. To obtain a low centre of gravity and low frontal area this car had offset engine, offset seat, offset suspension and unequal length drive shafts in connection with asymmetrical rear suspension. The choice of this type of layout presented the designer, Eric Broadley, with innumerable complications, but the resultant chassis has much to recommend it with the exception of a lack of diagonal bracing in certain important frames.

The basis of the design is four transverse bulkheads—the forward one with a secondary bulkhead to provide wishbone pick-ups—linked by a series of fully triangulated side frames. The two forward top frames are triangulated by a series of tubes with a common apex above the centre of the engine bulkhead; this asymmetric layout is necessary to allow easy removal of the engine. Owing to the use of asymmetric transmission only



20 The space frame chassis—with panelling—of the 1960 sports Lola. The asymmetrical lay-out of the front bay facilitates engine installation without in any way weakening the structure

the left-hand-side rear radius arm can be located on the driver's bulkhead, a further member being incorporated in the rear side frame to pick up the right side radius arm. All the front wishbone pick-ups are located in bending, but provision for this is made by the use of square section tubing.

The fully triangulated front bulkhead provides rack and pinion mountings, and engine mountings are taken from the secondary front bulkhead. The rear of the engine is supported at the gearbox in a special sub-structure which also forms a propeller shaft tunnel, and the pedals are mounted on a sub-frame on the engine bulkhead. Another sub-frame at the rear provides mountings for the lower transverse links, fuel tank and bodywork. The battery is located alongside the gearbox in the interests of weight distribution. But for the introduction of the rear-engined Lotus, the Lola might have been the most successful Formula Junior car of 1960. Subsequently, however, even Eric Broadley went over to rear-engined design.

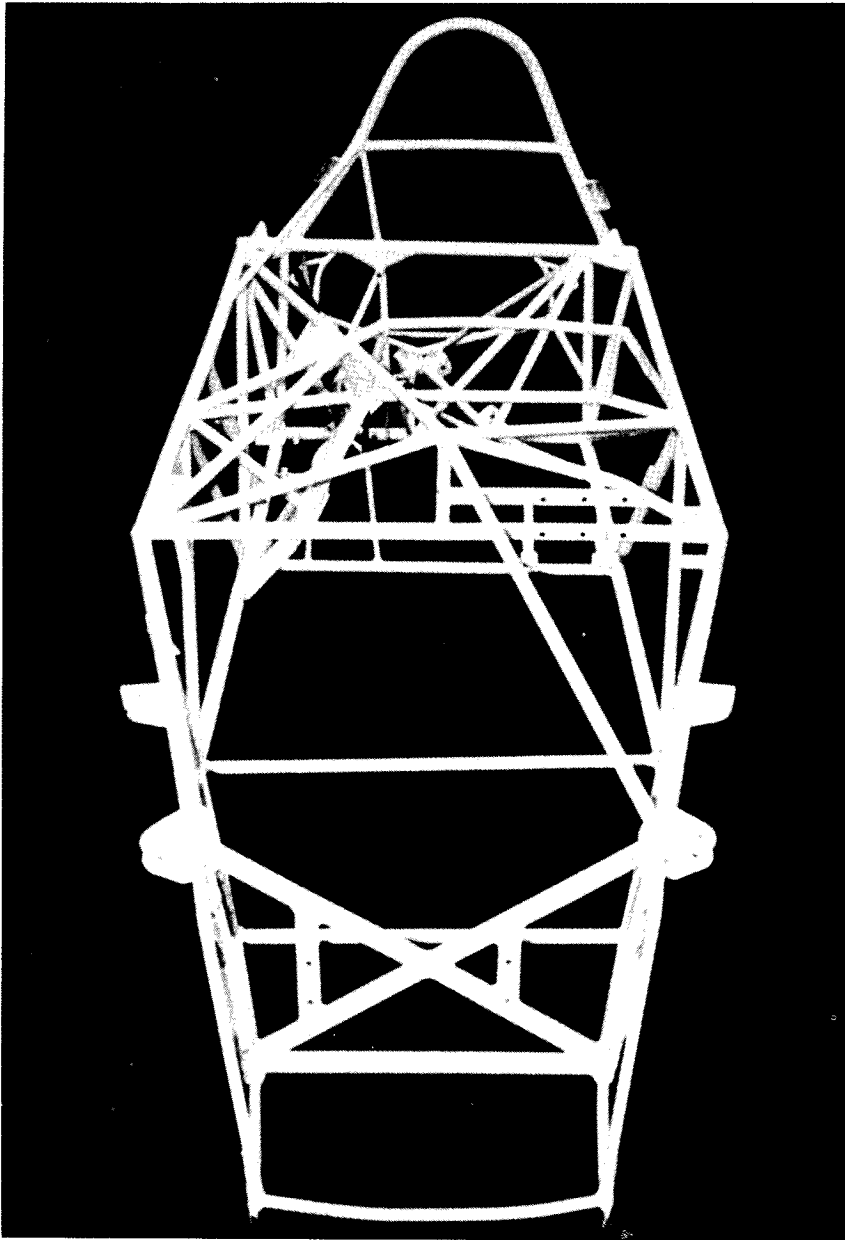
The C-type Jaguar

Chassis-wise this car is chiefly interesting in that it represents an interim stage between the girder-type layout long favoured for production sports cars and the multi-tubular type which is now extensively used on sports/racing and racing cars. The basis of the chassis, however, is tubular, and as such would seem to provide quite good torsional stiffness for an early design. As in the conventional type of chassis, the major bending loads are still taken out into the bottom frame, and the structure as a whole is relatively heavy and not particularly efficient. Nevertheless it represents a tremendous advance on the girder-type layout used in the XK120 model.

Much of the stiffness of this chassis is imparted by a central box consisting of the engine and scuttle bulkheads. The forward of these two bulkheads is braced by sheet steel, and four of the other five sides of the bay are triangulated to form a fairly stiff structure. From the engine bulkhead forwards extend two triangular frames (composed of tubes at the top, a tube and a girder at the bottom) which are linked by the front suspension bulkhead. This is a relatively massive frame of square and rectangular section tubing which provides mountings for fore-and-aft wishbone links, transverse torsion bars and telescopic shock absorbers. It also locates the radiator and—by means of outriggers—the front bodywork.

The outer side frames of the engine bay are triangulated and so are the side frames of the seat bay, which incorporate further tubes to support the rear suspension bulkhead; the latter is of the diaphragm type. This frame locates the rear torsion bars, the rear dampers and the torque reaction link, which is mounted approximately half way up the main right-hand member. From this bulkhead a subsidiary frame extends rearwards to support the fuel tank and spare wheel; this frame has merely to deal with bending loads.

If the retention of I-section members—however copiously drilled—in a



21 *Front-engined single-seater: the all-offset chassis of the 1960 Formula Junior Lola. The twin diagonals of the front suspension bulk-head eliminate the need for a top cross-member*

chassis of this type should seem at all strange, it should be pointed out that their chief purpose is to deal with bending loads and to act as attachment points for the floor and seats; they also play a part in preventing lozenging in the bottom frame. These jobs could, however, be done by small diameter, square section tubes—which would also be much lighter.

The D-Type Jaguar

By comparison with the C-type, the D-type Jaguar constitutes a tremendous advance, particularly from the point of view of torsional stiffness. Even today it is an extremely interesting design exercise. Like its predecessor, the D-type is handicapped by an extremely heavy engine/gearbox combination and by the use of a live rear axle, but it has nevertheless been extremely successful in long-distance sports car races. And just to prove that “racing improves the breed”, the basic chassis layout of the D-type Jaguar is repeated in the road-going E-type, with the addition of independent rear suspension.

The D-type chassis is a composite structure made up of square and round section tubes and stressed skin, light alloy panels. The basic box-section framework (fig. 25) deals with front suspension and engine loads, which are taken out into a central monocoque section of magnesium alloy; the loads are transmitted by members which run outwards from the front suspension bulkhead and the front engine mountings and are bolted to the central bulkhead.

Composed of 18 gauge and 20 gauge tubing, the front framework weighs 56 lb. The central monocoque structure is made up of 18 gauge magnesium alloy. At the rear it incorporates two massive box-section members which serve to locate the rear suspension. It is noteworthy that the stiffness of this central monocoque section is increased by the use of triangular “torsion boxes” along the lower body sides.

Front suspension is by relatively narrow-based double wishbones—the lower links acting on longitudinal torsion bars—and telescopic dampers. All the suspension elements and the steering rack are mounted on the basic box-section framework. The live rear axle is located by twin trailing links, composed of flat sheet steel, and an A-bracket. All these, plus the single transverse torsion bar and the telescopic dampers, pick up on the rear bulkhead of the central monocoque. As on the C-type, the rear sub-frame supports only the fuel tanks, the spare wheel and the rear bodywork.

The Lotus Elite (See figs. 7 and 22)

As shown in the accompanying drawing (fig. 22), the Lotus Elite chassis/body unit consists basically of three major standard mouldings, assisted by a bulkhead moulding. With the exception of the front suspension frame—which is bonded between the undertray and the inner moulding—and the tubular windscreen hoop, it is entirely composed of resin-bonded glass-fibre.

Of the two metal reinforcements, the front suspension frame is in sheet steel while the windscreen hoop consists of 1 inch, 10 gauge round section tube welded into 1½ inch 10 gauge square tube. This member has several functions; at its ends it provides the car’s jacking points; the square section tubes, bonded between two skins of glass-fibre, form supports for the door hinges; the “hoop” itself strengthens the windscreen pillars and roof and also forms a built-in crash bar. A further tubular cross member below the level of the windscreen stabilises the hoop laterally and supports the steering column, the instrument panel and the hand brake.

As in all unitary construction cars, some parts of the bodywork are more highly stressed than others. The radiator air intake and headlamp mountings, for instance, have no direct structural function, but the radiator inlet duct plays an important part in stabilising the front suspension frame and is built in the form of a diaphragm. Engine, as well as suspension, loads are taken out into the front suspension frame, and the latter is diagonally braced by being bonded to the inner body moulding. Bending loads in this area are taken out into the vertical sides of the engine bay—and thence into the main side members of the body.

The scuttle bulkhead moulding stabilises the body diagonally and also takes loads from the steering column, the brake and clutch pedals (and master cylinders) and the accelerator pedal. Rearwards the bulkhead contracts into the deep, box section propeller shaft tunnel, which links it to the near-vertical rear diaphragm. The tunnel is extremely rigid and provides bending as well as torsional stiffness, reducing bending loads on the undertray by supporting the seat mountings. The rear diaphragm completes the structure in this area, coupling the body sides and the tunnel by means of a beam. The rear suspension unit pick-ups are located at the top corners of this beam and the final drive is mounted in the centre of it; in this highly loaded area the glass-fibre is ¾ inch thick, tapering to ⅓ inch and finally ¼ inch. At the four mounting points the final drive unit is bolted directly to the chassis via rubber bushes. On some early examples of the Elite, mounting failures were experienced due to heat generated by the exhaust system or the inboard brakes. There have also been failures due to errors on the part of the body manufacturers, but all these problems have now been overcome, showing that direct metal-to-glass-fibre mounting can be quite practical, even where very high point loads are involved.

On the Series Two Elite the pick-ups for the rear radius arms are formed in the undertray moulding and shaped to accept the rubber bushes for the ball-type mountings. The Series Two radius arm layout (see fig. 44) is also superior to the original design both structurally and from the point of view of suspension geometry. Most noticeably, the use of two widely separated outboard pick-ups for the two-piece radius arms makes the suspension much stiffer against toe-in loads. The new geometry also obviates toe-in with vertical movement of the wheel, and this considerably improves the car’s straight-running qualities at high speed.

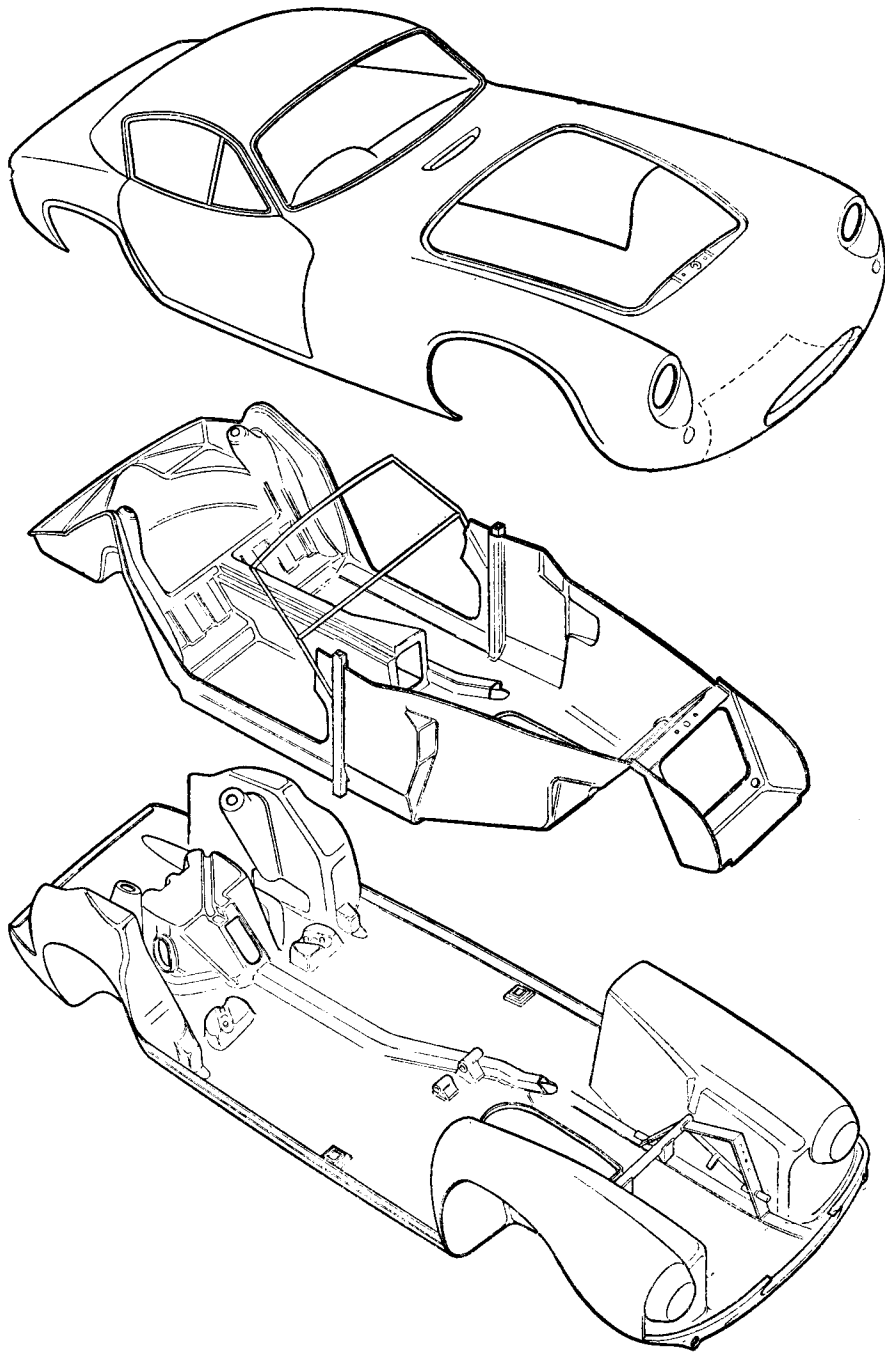
Behind the rear diaphragm the structure consists of the boot floor and two side members, all of which are designed to take the bending loads of the battery and luggage. The roof provides diagonal bracing over the top of the central bay—the one which cannot normally be triangulated on an open car—and contributes towards the Elite's very considerable torsional stiffness. As an instance of this, when a chassis/body unit was subjected to torsional loading greater than would ever be experienced in normal use, the maximum deflection across the windscreen aperture was 0.050 inch. The influence on the structure of holes for doors, engine cover and boot lid is overcome by taking out diagonal loads into the outer moulding round the edges of these apertures; further compensation is provided by the stiffness of the structure at the base of the doors and round the windscreen. The doors themselves are made by a special process which combines torsional rigidity and lightness; they are effectively of box section and the material is locally thickened to take hinge and lock mountings.

The Lotus 25 (See figs. 23 and 24)

As has already been pointed out, the Lotus 25 is in effect a twin tube chassis, a layout for which we have shown little enthusiasm. However, in this instance the tubes are of approximately oval section, measuring about 12 inches by 6 inches, and as the torsional capacity of a tube increases as the fourth power of its diameter they are obviously much stiffer than 3 inch tubes, even though they are of lighter gauge. And lest anyone should doubt that this is essentially a twin tube chassis, a section taken at mid point would show merely the two fuel tanks (of 18 swg aluminium alloy) and the undertray (20 swg aluminium alloy), with shallow inverted Vs at either side of the latter for pipes and cables. At the front is a major bulkhead, fabricated from sheet steel (mainly 16 swg), whose function is to provide mountings for front suspension, steering, pedals, master cylinders, oil tank and radiator (via long brackets) and to transfer all the loads involved into the chassis proper. (A subsidiary transverse bulkhead, which is in effect a square tube, takes out loads from the rear member of the lower wishbone.)

The scuttle bulkhead adds little to the torsional stiffness of the chassis, its main function being to locate steering wheel, instruments and switches. The main transverse member in the middle of the car is the seat-back bulkhead, which is triangular in side elevation; it consists of the actual seat-back member (which slopes at 40 degrees from the horizontal) and an almost vertical rear member, the space within being occupied, like the side "tubes", by a rubber bag fuel tank.

The Coventry Climax V8 engine is rigidly mounted, and thereby contributes considerably to the overall stiffness of the structure. Due to the shape of the engine, the chassis side members are of much smaller diameter in this bay—to such an extent that they are generally referred to as



22 The chassis/body unit of the Lotus Elite consists of three major structural mouldings. With the exception of the front suspension frame and the tubular windscreen hoop, it is entirely composed of resin-bonded glass-fibre

SPACE FRAMES AND UNITARY CONSTRUCTION CHASSIS

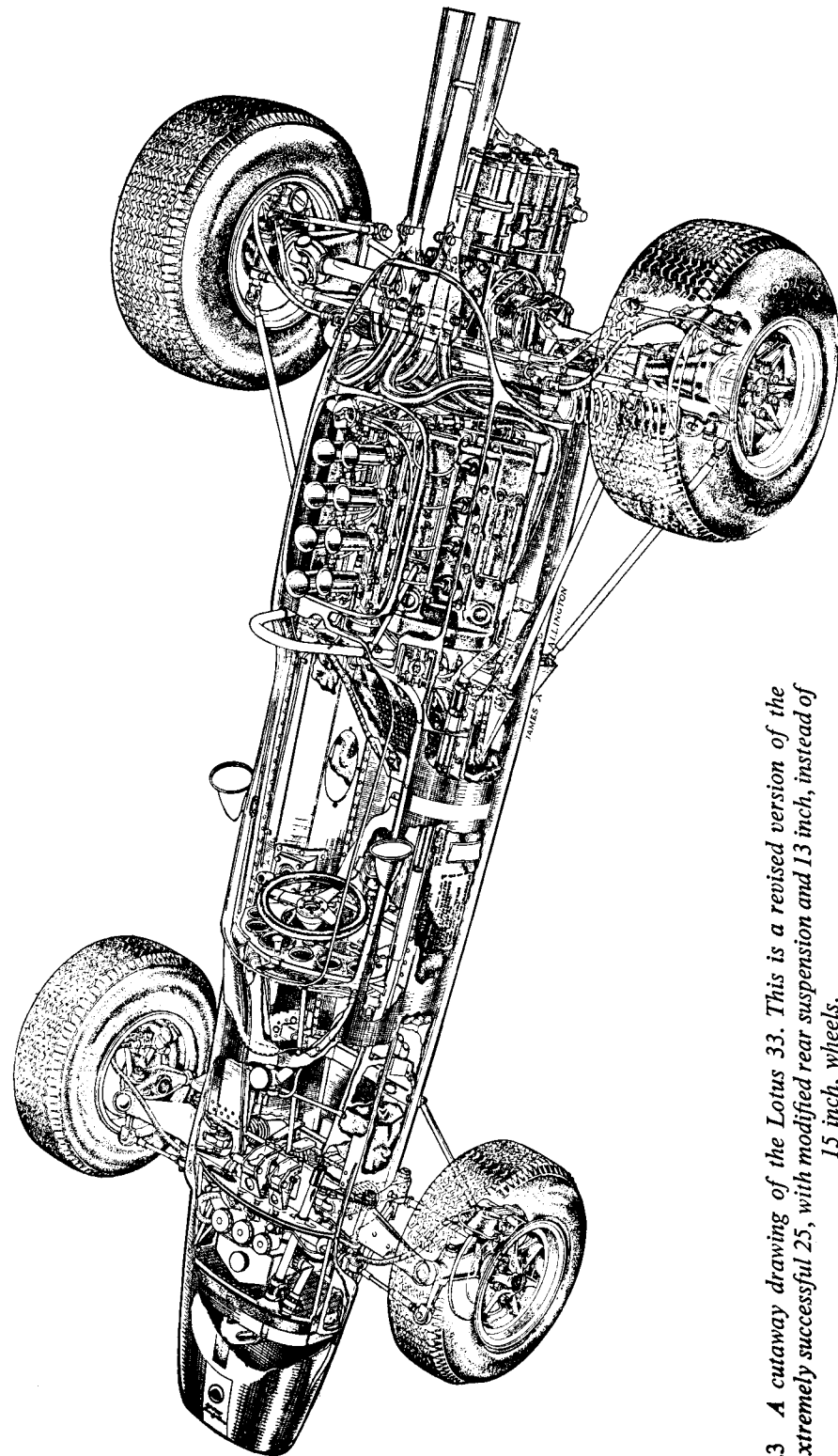
“wheelbarrow arms”, with the outer skin of 16 swg aluminium alloy and the inner skin of 20 swg mild steel. In all there are eight mounting points for the engine, four at the front and four at the rear; at the front there is one at the top and one at the bottom of each wheelbarrow arm, while at the rear there is one at the bottom of each wheelbarrow arm plus two (for the gearbox) at the top of the bulkhead; the latter also has to deal with loads from the coil spring/damper units and the transverse links of the rear suspension; longitudinal loads are taken out, via the radius arms, in the vicinity of the seat back bulkhead.

As has already been mentioned, use of the engine as a chassis member is not to be recommended unless the crankcase has been designed for this purpose (in addition to its major function of providing positive location for the crankshaft and the cylinder head). However the Coventry Climax unit seems to perform its structural purpose without detriment to its primary function, though it is not certain that the cramping of the exhaust system dictated by the proximity of the “wheelbarrow arms” has been accomplished without any power loss. And because there are no auxiliaries at the front or the bottom of the Coventry Climax V8 engine, accessibility is nothing like as bad as it might be with this type of layout.

One of the chief considerations behind the design of the Lotus 25 was to make the car as small as possible, the main limiting factor being the width of the engine. Use of an ultra-reclining driving position also allowed the height to be brought right down, and a further contribution in this sphere was made by the introduction in 1963 of a wind deflector in place of the conventional windscreen. This consists of a cut-down windscreen with a wide aperture at the front and an upcurved lip on the bodywork about 9 inches behind it; the lip deflects the air over the driver's head and the lower windscreen gives him a better view.

Another aid to improved air penetration is the use of inboard front suspension units, actuated by rocking lever top wishbones, the track rods being in line with the latter to provide the cleanest possible front end. This in turn necessitates high mountings for the rack and pinion in an area where space is already at a premium—particularly as Lotus use pendant pedals. It also increases the loads to be reacted at the suspension unit mountings and the top arm pivots, increases the time required to change springs or anti-roll bars, and necessitates the fitting of special ducts to cool the dampers. However, results tend to suggest that all this extra complication is justified—even though, in itself, it may do very little for the car's penetration—and the overall layout is far more elegant than most conventional designs.

Originally the rear suspension incorporated twin transverse links (the lower one a “reversed wishbone”, wide-based at the outer end) and parallel radius arms, but in 1964 experiments were made with the aim of reducing roll steer effect, the inboard mountings of all the suspension arms being moved to positions parallel to the centre line of the car. Experiments have

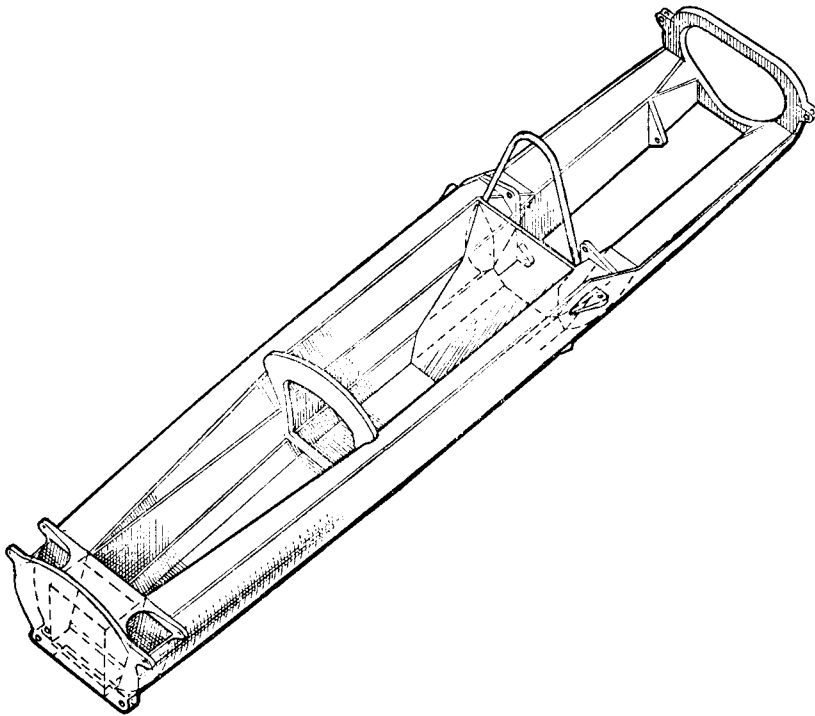


23 A cutaway drawing of the Lotus 33. This is a revised version of the extremely successful 25, with modified rear suspension and 13 inch, instead of 15 inch, wheels.

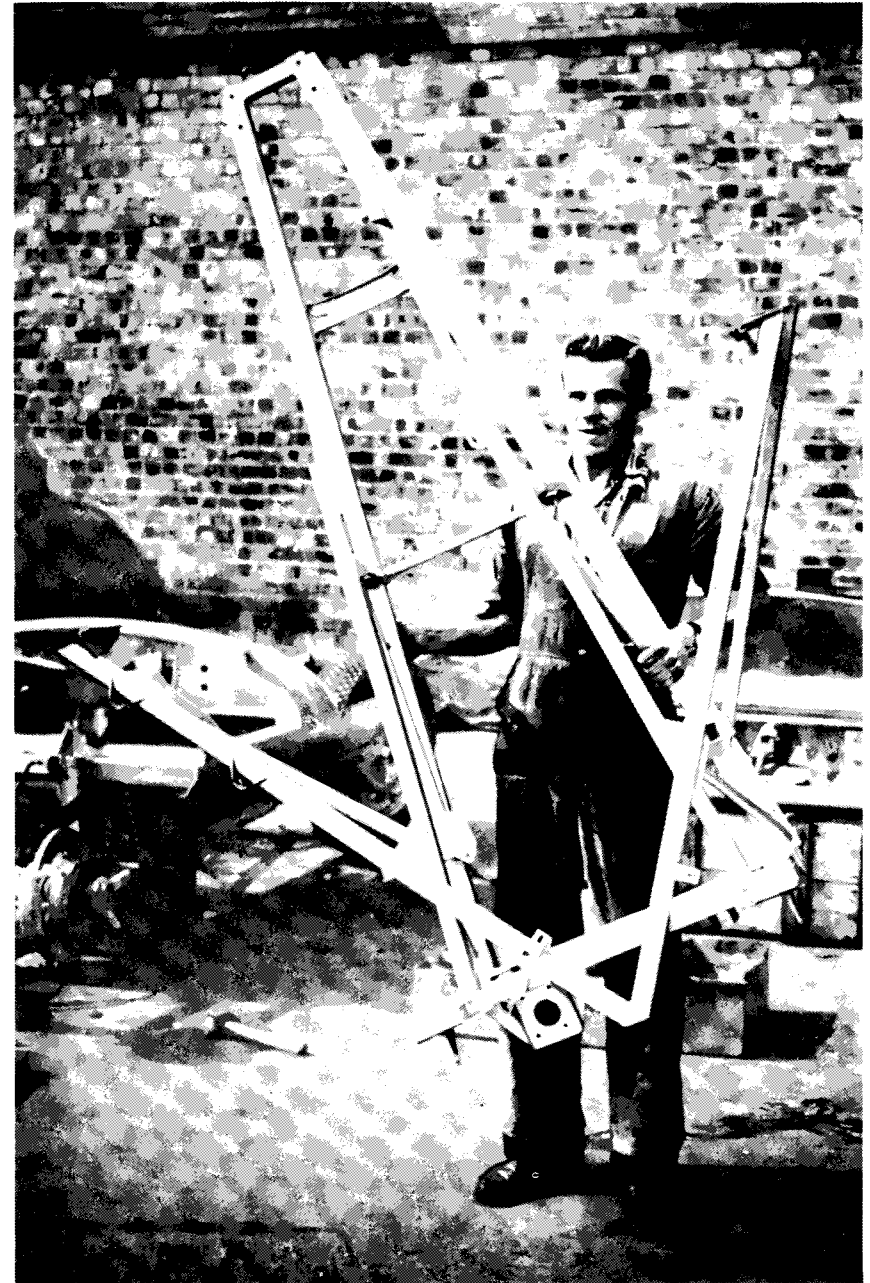
SPACE FRAMES AND UNITARY CONSTRUCTION CHASSIS

also been made with shorter virtual swing arm lengths, but these do not seem to have brought about any improvement.

When it first appeared, the Lotus 25 had a clear lead—in performance if not in reliability—over all the opposition. In 1963, with added reliability, it was virtually unbeatable, but in 1964 the opposition caught up somewhat. Nevertheless, it will undoubtedly go down in history as one of the great racing cars of all time.

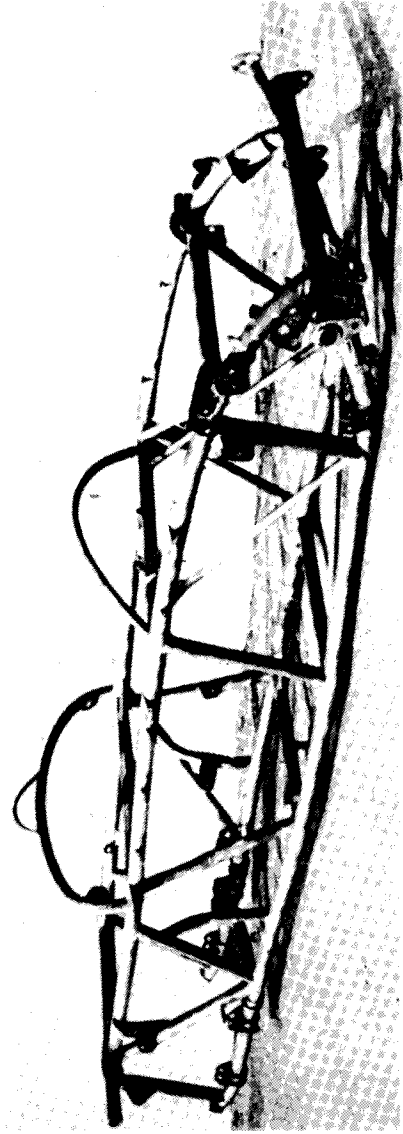


24 *The chassis/body unit of the Lotus 25/33. The stressing of such a structure is far more complicated than is the case with a tubular chassis; in fact there is no simple and straightforward method of calculating skin stresses, and anyone contemplating using this form of construction for the first time would be well advised to follow established practice regarding material gauges and to concentrate on the detail design of mounting brackets (together with associated stiffening and reinforcing features) and the layout of bolt and rivet groups*

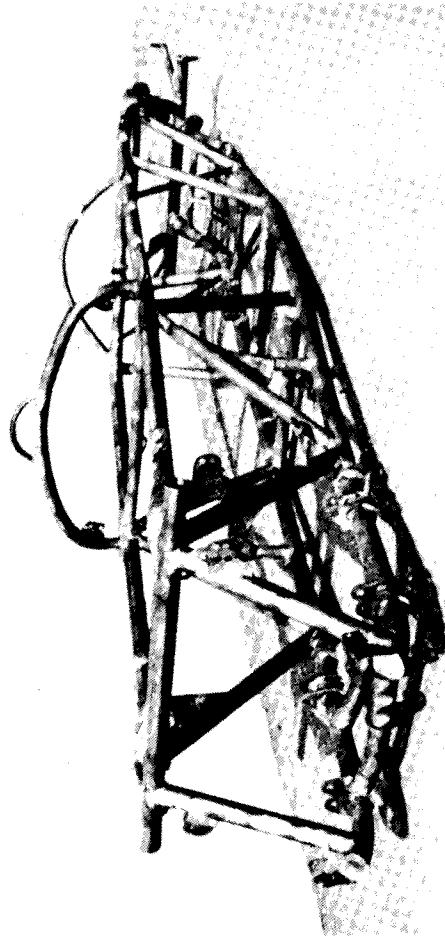


25 *The tubular front sub-frame of the sports/racing Jaguar D-type, used in conjunction with a stressed skin body. A similar type of construction is found in the road-going E-type Jaguar*

26 Front view: the use of multi-tubular joints is avoided wherever possible, while suspension mounting brackets are normally located part way along chassis members



27 Rear view: as borne out above, the Cooper chassis can be summed up as a series of brackets joined together by tubes. Alternative mounting holes are provided for top rear wishbones to allow alteration of the rear roll centre



CHAPTER VI ANALYSIS — MULTI-TUBULAR AND OTHER CHASSIS

The 1960 Formula One Cooper (See figs. 26 and 27)

It has been said that function is the final criterion of any design and on these grounds one hesitates to criticise any of the very successful racing and sports cars produced by the Cooper Car Company in recent years. From a structural viewpoint, however, there are many aspects of Cooper design which leave something to be desired. First and foremost of these is the use of curved tubes for major structural members, a practice which decreased somewhat in 1960 due to the use of coil spring rear suspension in place of the previous transverse leaf spring. In earlier models the top frame rails curved down at the rear merely to leave space for the transverse spring. Even in the 1960 design a certain amount of curved tubing is still apparent, particularly at the front of the car, where the top chassis rails curve gently downwards to meet the bottom rails—swept steeply upwards just ahead of the front bulkhead—at a common cross member.

In many ways the Formula One Cooper is the enigma of racing car design. It defies many structural laws, yet in 1959 and 1960 it won the majority of important races. Cooper engineering avoids joining three or more tubes together at any one point because it has been found that, with the comparatively large section, heavy gauge tube employed, this practice has led to failure of the weld. Similarly, suspension wishbones are normally located part-way along a chassis member rather than at the apex of two or more tubes. This means that suspension loadings are taken out directly into the tube concerned—and again Coopers use large section, relatively heavy gauge tube to allow for this. The engine mounting brackets are welded on to curved tubes in order that the length of the brackets themselves may be reduced; both ends of these curved tubes are welded to main frame members half-way along their length. In addition, the tube on the right is slightly longer than that on the left owing to the fact that the diagonal on this side meets the top chassis rail some distance behind the engine bulkhead, in order to leave space for the magneto to protrude beyond the limits of the frame. Coopers themselves do all they can to perpetuate the old fable about chalk on the wall and piles of tube on the floor, but there is a very good reason for every design feature of their cars and this is just one instance.

In terms of torsional stiffness the 1960 Formula One Cooper is considerably superior to the 1959 car, owing partly to the use of straighter tubing

and partly to a greater amount of cross bracing. Most of this is concentrated in the front bay, however, and the majority of frames in the centre and rear bays remain untriangulated. The centre bay, in fact, has no triangulation whatsoever, although some concession to added stiffness is made by short pieces of tubing welded across the apexes of the joints. Even here there is an anomaly in that these short pieces are only found on three sides of the top frame, the fourth being left open to make room for a water pipe. The long rear bays are untriangulated at top and bottom but have two diagonals in each side frame with a common apex half-way along the bottom rail. However, as these members also locate tubes carrying wishbones and engine mountings, their position is obviously determined by practical rather than structural considerations.

Towards the back of the bottom rear frame are two curved cross members, whose position is dictated largely by the location of the lower wishbone mountings; they are curved merely in order to clear the engine. Two straight cross members in the top frame locate respectively the suspension unit mountings and the sub-frame which supports the gearbox; the latter is also extended sideways as the housing for the rear anti-roll bar.

In general terms the Cooper chassis satisfies the definition "a series of brackets joined together with tubes" perhaps better than any other. It is intensely practical, accommodates all mechanical elements, fuel tanks, oil tanks, accessories and driver in the smallest possible space, and is extremely durable. Its suspension mountings allow for experimental alteration of wheel angles to suit any combination of circuit and driver, and choice of anti-roll bars allows handling characteristics to be varied to a nicety. The purist might suggest that the provision of a variety of wishbone mounting points, and easily adjusted camber and toe-in, argue a basic lack of confidence in the original suspension design. However, when the 1960 Cooper was tested for the first time, Jack Brabham lapped Silverstone over 5 seconds faster than the current lap record within a hour of the car being unloaded from its transporter. The only work done on it in this time was the substitution, after a few exploratory laps, of a slightly stiffer rear anti-roll bar.

The chief reasons for the success of the 1959/1960 Cooper are high speed on the straight—due largely to very good body shape and low frontal area—and high cornering power. The illustration of Brabham's Cooper on a fast corner at Oporto (fig. 29) shows that on the all-important question of keeping the outside wheels upright under cornering loads the Cooper reaches a very high standard. Here again the purist might argue that this is despite the wishbone loads being taken into the chassis over a very narrow base (see fig. 40) and being dissipated largely in the members involved. The front wishbone links are arranged to take both lateral and fore-and-aft loads, but despite their narrow base do the latter job extremely well, to judge from the car's excellent performance under braking. The

rear wishbones are so arranged that the rear members deal with the majority of lateral loads while the front members accommodate fore-and-aft loads, although at a very considerable angle. Here again, however, it can hardly be argued that the suspension does not perform its function more than adequately.

There are numerous ingenious detail features on this car, which was designed with the simple intention of winning races. A triangular oil tank mounted directly behind the radiator directs hot air away through holes in the body sides. The handbrake is mounted horizontally to allow the maximum space for fuel tanks alongside the driver. Transmission has been a weak point on racing cars in recent years (particularly Coopers) so the firm has built an entirely new gearbox of very straightforward design which has proved to be remarkably successful. Doubts about the adequate cooling of inboard rear brakes have led to outboard brakes being retained despite the increase in unsprung weight, fully effective braking being considered of more importance than any possible slight gain in cornering power with the more ideal layout. Simplicity, strength and reliability are far more important to the Cooper family than pioneering new ideas for the world's motor industry.

Cooper Monaco (See fig. 5)

Much the same may be said of the Cooper Monaco sports car. The 1960 version, based on the 1959 Formula One Cooper, has double wishbone and coil spring front suspension, with double wishbones and a transverse spring at the rear. As can be seen from fig. 5, the chassis is basically a broadened-out version of that used in the single-seater, with four major longitudinal members and four bulkheads. With diagonal bracing only in the side frames of the front and rear bays, this chassis has very little torsional stiffness other than that imparted by the relatively large section, heavy gauge tubes; this factor is considerably accentuated by the much increased section of the chassis, particularly in the central bay. It would also appear that even those frames which are diagonally braced lose considerable stiffness from the fact that the lower side rails curve sharply inwards on either side of the central bay and then return to a fore-and-aft plane in the region of the lower wishbone mountings.

Starting from the front, that is from the right of fig. 5, front suspension loads are taken out through a series of offset brackets, being absorbed, in the main, by the tube concerned. The suspension unit mountings are well located at the junction of the top side rails and the top front cross member, and loads from the rear arm of the top wishbone are taken out into both the top rail and the side frame diagonal, but the forward top wishbone mounting does not appear to be ideally situated, being at a point from which the top rail curves sharply downwards to meet the bottom rail. Here again, however, relatively large section tubing has its advantages.

The scuttle bulkhead follows common Cooper practice in that the lower transverse member is placed a few inches behind the uprights. This bulkhead provides adequate support for the steering column and other components mounted on it, but contributes little to the torsional stiffness of the chassis; much the same may be said of the bulkhead immediately behind the seats, which consists of a straight lower transverse member and a hoop which forms both uprights and the top transverse member. The rear bulkhead consists of curved transverse members—their shape dictated by the clutch bell housing—with short diagonal members performing the function of uprights. There are no less than four suspension mountings on each of the curved top rails—for the transverse spring, the telescopic dampers (located at the junction with the top transverse member) and the wishbone links. Neither the latter nor the offset lower wishbone mountings are provided with any form of stabilisation. To the rear of these mountings the top and bottom chassis rails converge into a common transverse member.

Subsidiary hoops of small section tube support the bodywork and the battery is carried in a cradle mounted off the front side frame. It is also worthy of note that over part of its length the top side rail acts as a water pipe between engine and radiator.

As a logical development of the basic Cooper design, the Monaco has been a very successful car—and still is, at the time of writing. At the current rate of chassis development, however, it would seem that this type of unbraced, four tube chassis will very shortly be outmoded, even at Surbiton!

The rear-engined Formula One Ferrari

Of all the major competition car designers, Ferrari seems to have found the change over from the traditional approach the most difficult to make. Ferrari, of course, makes complete cars, including engines, and power with a capital P has always been one of the leading considerations in the Ferrari scheme of things. Recent sports and racing Ferraris show the very considerable influence of British chassis designers, although it would appear sometimes that lessons learned in this way are not fully understood.

Although front-engined designs, the 1960 Formula One and sports Ferraris showed clearly the influence of Cooper successes in 1959—particularly in their suspension layout. Much the same may be said of the rear-engined Ferrari single-seater which first appeared at Monaco—and later in revised form at Solitude—and even uses the Cooper method of location for its front anti-roll bar. Later the car was fitted with Lotus-type rear suspension—that is with long radius arms to accommodate fore-and-aft loads. Although theoretically this layout should be much superior to the extremely narrow-based wishbone system employed earlier, the results were apparently disappointing, for a reversion was made to the earlier

type. With this layout complicated offset brackets are used and the suspension unit is also mounted at a considerable offset at the extreme rear of the upright. Some sections of the chassis frame are correctly triangulated, others—such as the floor of the central bay—contain redundant members in the form of two diagonals (though these are used to mount the seat and therefore serve a practical if not a structural purpose) while some large bays are left completely untriangulated. Front suspension loads are fed into the chassis on a narrow base on either side of a single bulkhead and the suspension unit mountings are offset above and to the side of the top frame rails, where they are supported by tubes welded to the centre of the top transverse member.

The rear engine mounting is hung by offset brackets from the rear transverse member, which also accommodates suspension loads via the suspension unit pick-ups. The engine bulkhead is a somewhat unsatisfactory structure in that the lower transverse member runs some distance ahead of the side members while the top member is curved—gently in the middle and more sharply at its outer end. Furthermore, this bulkhead lacks any form of triangulation. The top side rails run backwards from a point considerably higher than the side rails on the central bay, presumably in order to clear the engine, but as a result they do reach the required height for the rear suspension unit mounting. The general shape of the car is decided by the very considerable bulk of the engine, but in many other respects—such as the shape and location of the oil tank—better use might be made of the space involved. Ferrari continued to make tubular chassis in 1962 and 1963, without achieving much success in Formula One racing, but at Monza in September 1963 an entirely new car appeared, its conception obviously owing quite a lot to the Lotus 25 although there were many detail differences. Basically the chassis consists of two large diameter “tubes”, a front bulkhead, a seat-back bulkhead and the engine—to which the rear suspension is bolted via a fabricated bulkhead. Lotus 25-type suspension is used at both front and rear. The 1964 car is generally similar but has small diameter tubes beneath the engine, as already mentioned.

The 1958 Lister-Jaguar (See fig. 4)

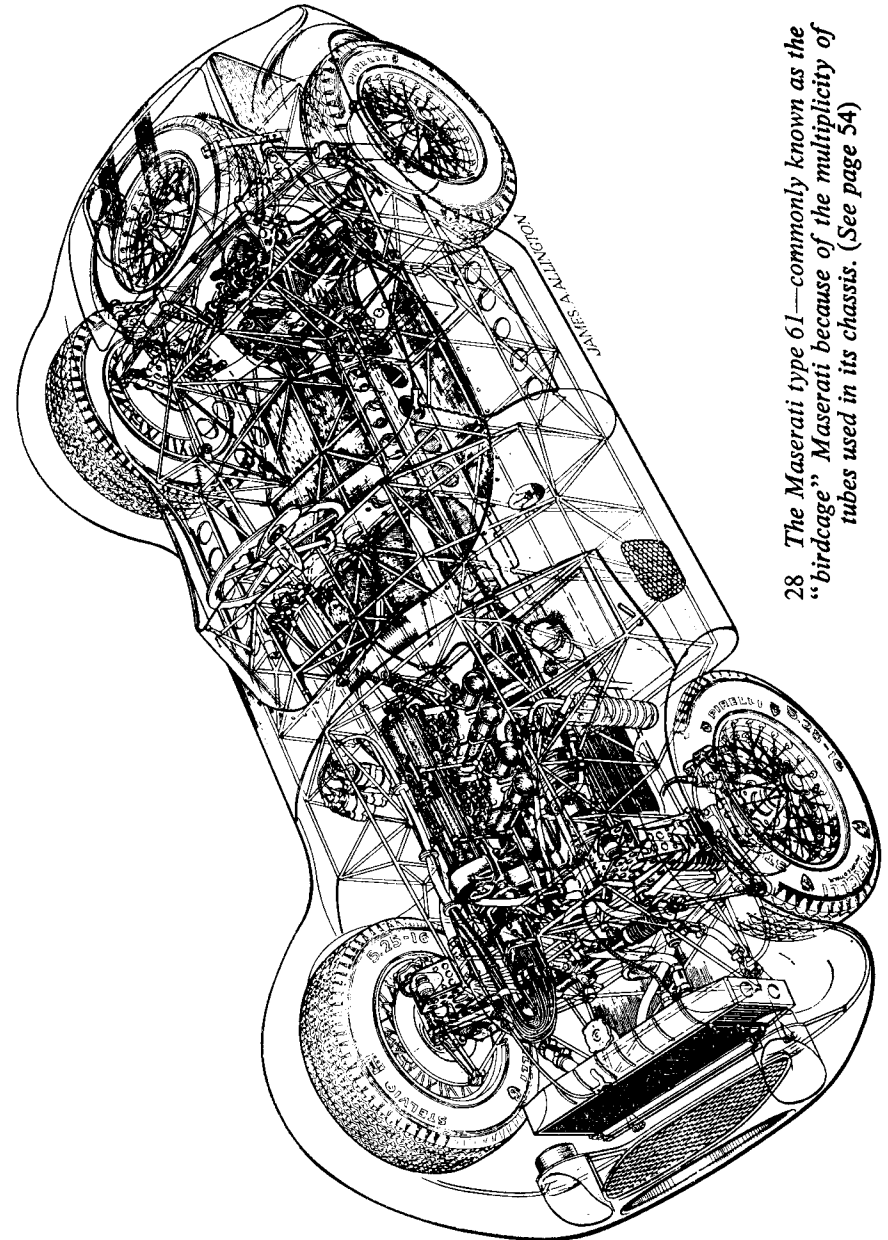
One of the most successful big sports cars of recent years is the Lister-Jaguar, a front-engined car with a twin tube chassis, double wishbone front and de Dion rear suspension. In its later stages of development nearly half the weight of this car was made up of the Jaguar engine and gearbox. Shortly before Brian Lister retired from racing he made an experimental car with a space-frame chassis (fig. 14) which might have given the Jaguar engine a new lease of life in sports car racing. This car, in fact, set up a new unlimited sports car lap record at Brands Hatch as late as October 1960.

MULTI-TUBULAR AND OTHER CHASSIS

Within the limitations of twin tube design, the Lister chassis (fig. 4) has much to recommend it. The main side rails and transverse members are made of 3 inch, 14 gauge, seamless drawn steel tube and the general shape of the chassis is largely governed by practical considerations. A massive front bulkhead provides mountings for the front suspension units, wishbones and steering gear, and a rear frame made of smaller gauge tubes provides anchorages for the rear suspension units and final drive assembly. At each side the radius arms locating the de Dion rear suspension are picked up on a bracket welded to the main longitudinal member, and lateral location is provided by a vertical slide on the back of the differential housing. A curved tubular hoop locates the steering wheel and instrument panel, and a sub-frame at the rear supports fuel and oil tanks, battery and spare wheel. A shaped cross member provides a rear engine mounting beneath the gearbox, and such items as pedals, brake and clutch master cylinders and electric fuel pumps are mounted on brackets welded directly to the main chassis tubes. These cars built up a very good reputation for reliability and were, in fact, only ousted from the awards lists by the advent of the smaller, much lighter 2- and 2½-litre Coventry Climax-engined cars.

The Maserati Type 60-61 (See fig. 28)

One of the most interesting cars produced in recent years is the type 60-61 Maserati, which very quickly earned itself the nickname "birdcage" because of its extremely complicated chassis construction. Race results have shown that the use in this way of very large numbers of small diameter tubes is no mere gimmick, but a serious attempt to obtain an extremely rigid lightweight structure. The chief disadvantage of this type of chassis is its complexity and the amount of work involved in making it. Distortion during welding would also seem to be unavoidable. And as the accompanying drawing shows, accessibility is hardly the Maserati strongpoint. Tubes have to be cut and bent to make way for vital components, oil pipes writhe in and out of chassis members and most of the engine accessories have to be dismantled before the engine can be removed from the frame. With such thin tubing, bays accommodating large loads need considerable reinforcement, and as carried out on this type 61 Maserati many of these leave much to be desired from the structural viewpoint. The rear radius arms, for instance, are located amid a maze of tubes whence the load paths out into the chassis are extremely tortuous. Traditional Italian brackets, copiously drilled, support the front suspension units and an extremely complex front bulkhead deals with suspension and steering loads. Without being churlish one might well ask whether the same job might not be done by a straightforward space frame having the minimum necessary number of members.



28 The Maserati type 61—commonly known as the "birdcage" Maserati because of the multiplicity of tubes used in its chassis. (See page 54)

CHAPTER VII

SUSPENSION PRINCIPLES

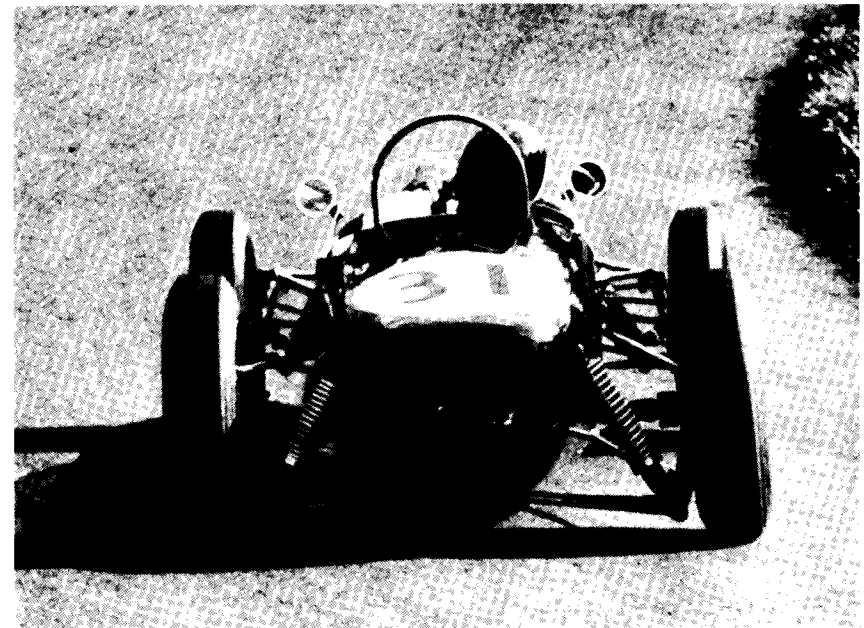
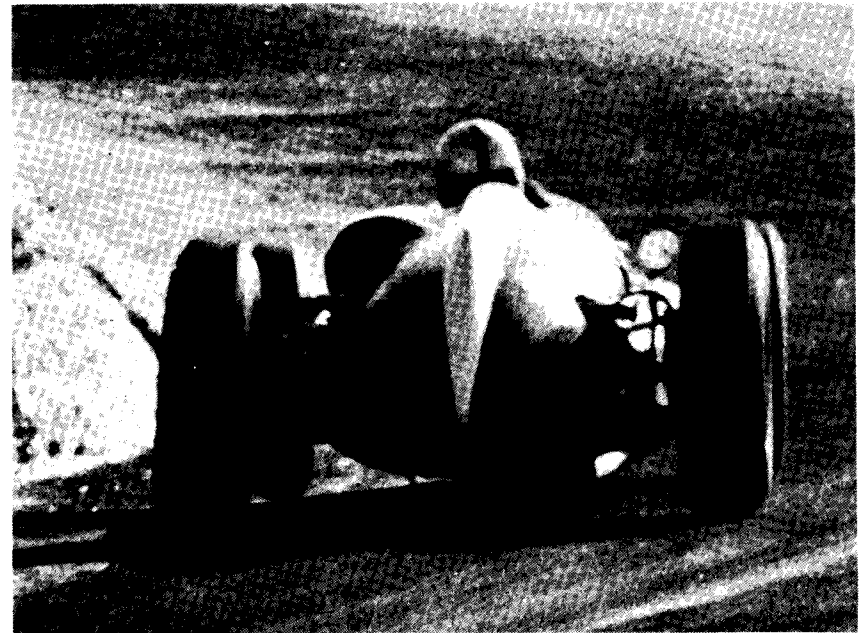
As with most other aspects of car design, suspension is very much a compromise. The main requirement is to keep all four wheels, and particularly the two driving wheels, in contact with the road at all times. It is also important for the suspension to impart suitable handling characteristics, to insulate the car from road shocks and to provide roadholding throughout the full range of power transmission. It is in the compromise between roadholding and ride that most difficulties arise.

In a sports or racing car one of the designer's chief interests is—or should be—to maintain the maximum amount of tyre contact with the road. In brief, this means keeping the wheels, and particularly the outside wheels, upright during cornering. Under cornering loads a very considerable proportion of a car's weight is transferred on to its outside wheels, and thus keeping these wheels upright is far more important than keeping the inside front wheel on the ground. Some racing drivers, in fact, would argue that it is an advantage to be able to lift the inside front wheel over low kerbs or grass verges and thus effectively increase the width of the road.

Until quite recently, stiff suspension was considered an essential feature of sports and racing car design—to restrain body roll and pitching—but the use of ultra-stiff springs generally results in the wheels being in only intermittent contact with the road, particularly on bumpy surfaces. The current trend is towards soft springs in conjunction with very effective dampers, which allow considerable wheel movement on bumpy surfaces but keep the tyre firmly in contact with the road.

On any car, particularly a high-performance or racing machine, it is essential to maintain stability under all conditions. This is influenced by several factors, including weight and weight distribution; the ratio of sprung to unsprung weight; the height of the centre of gravity; roll centre heights and their relation front to rear; wheelbase and track; suspension layout; steering geometry; loading conditions; the section, stiffness, pressure and tread depth of the tyres, and wheel and tyre balance.

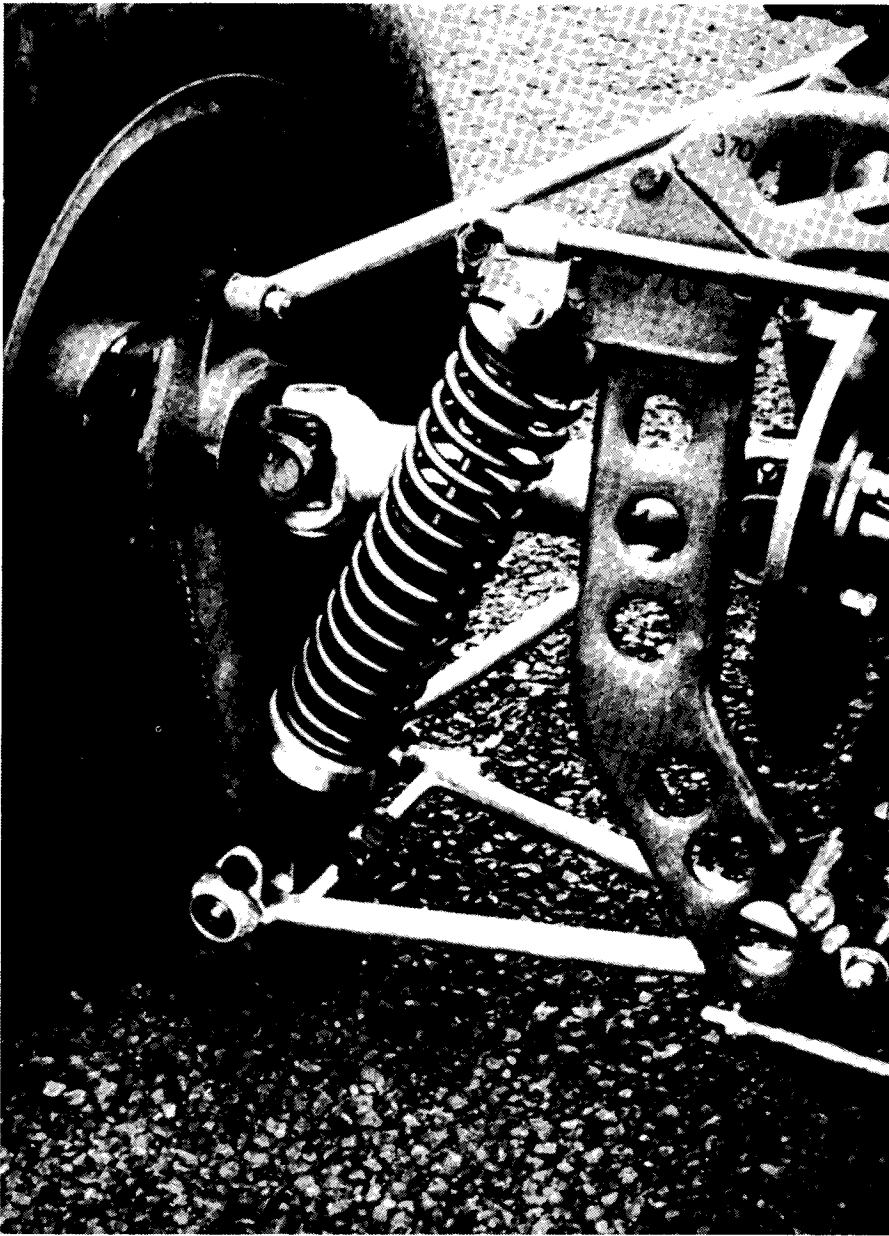
Each affects roadholding in a particular way. For instance, reducing front tyre sections on a car with neutral handling characteristics would immediately influence the car towards understeer; the smaller section tyre is less stiff, and therefore liable to greater deflections and increased slip angles, while the decreased width of tread would lead to the tyre running at larger slip angles for any given cornering force. There must be a balance of tyre stiffness front to rear in order to maintain consistent handling characteristics on both high and low speed corners.



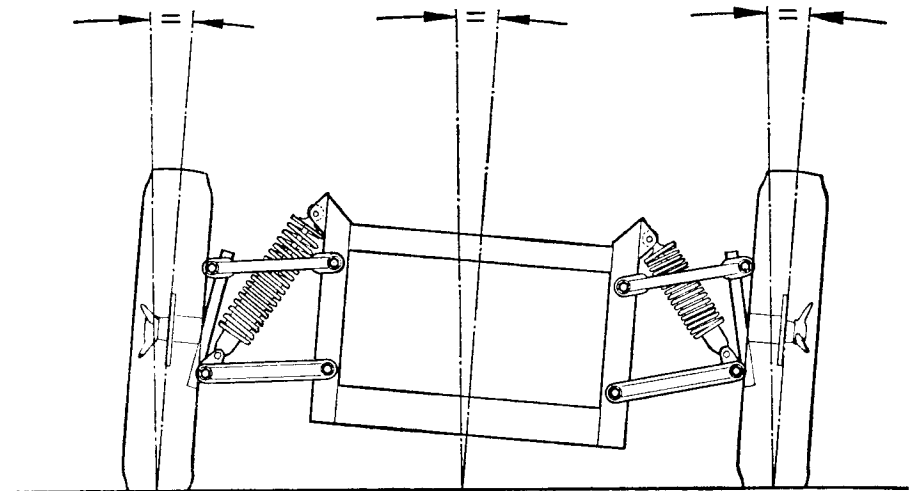
29, 30 Brabham's Formula 1 Cooper at Oporto (above) and Clark's Formula Junior Lotus at Oulton Park, show how the more advanced types of suspension keep the outside wheels almost upright during cornering. The camber of the lightly loaded inside wheels is far less important—as is the fact that Brabham's inside front wheel is clear of the road

What is Roadholding?

The maximum roadholding capacity of any car can be defined as the highest degree to which cornering force can be developed without the driver losing full control. As mentioned above, for optimum roadholding the wheels must be kept upright. In addition, the tyre tread must be kept as flat as possible to maintain the maximum contact area. In practice it is impossible to keep the tread completely flat because weight is transferred to the outside edge. A tyre needs depth to absorb bumps and thus flexes



31 An advanced type of rear suspension (1960 Formula 1 Lotus), in which loads are taken out into the chassis over a very wide base. The suspension geometry is so designed that, with weight transfer due to roll, the outside wheel remains upright—to maintain the maximum tyre contact area



32 With equal length, parallel wishbones, wheel camber is directly related to body roll. The same is true of trailing link suspension. With the more advanced types of asymmetrical wishbones, camber change due to roll is much reduced

slightly when lateral force is applied, however stiff its casing may be. Tread pattern, and the characteristics and hardness of the rubber used by the tyre manufacturer, also have an important effect. For the ultimate in roadholding the answer would be to bank the wheels in the opposite direction to the inclination of the car when rolling, in order to counteract tyre distortion. At the time of writing this ideal has not been achieved. It is possible, however, to retain negative camber in spite of body roll by starting with a very considerable degree of negative camber, but this brings the static wheel position to an undesirable angle, which considerably affects straight line performance. The best compromise is a suspension system which gives non-linear camber change, that is one in which camber is *not* directly dependent on roll.

Weight

Of all the basic factors which influence roadholding, one of the most important is weight. Since 1937 the general trend of design for racing has been towards lightening the cars to improve the power to weight ratio. From the performance point of view this trend has resulted in improved acceleration (and in some cases higher maximum speeds) but the greatest over-all increase in circuit speed has been due to improvements in controllability. These are largely the result of the decrease in weight. In the first instance this question of weight-saving involved a reduction in the region of a hundredweight a year, but nowadays the designer can rarely reduce the weight of a given type of car by more than thirty or forty pounds each year. Apart from its influence on performance, weight has a tremendous influence on suspension, from the stiffness it requires of the springs and the robustness of the linkages which are to support it. And in addition to all the mechanical advantages, reduced weight lowers the physical effort required of the driver, and this in turn plays an important part in the controllability of the car. Unfortunately, minimum weight limits are imposed in several classes of racing and these effectively inhibit further developments in this sphere, whether by the adoption of ultra-light materials or by better utilisation of normal materials. In some cars, in fact, it is necessary to carry ballast to comply with the regulations; the driver's heel-rest on the Lotus 32 weighs 80 lbs. Some manufacturers build the weight penalty into the structure—though without necessarily making it any stronger—and whichever course is adopted the aim should be to use the unwanted weight to the best advantage from the handling point of view.

Weight Distribution

Weight distribution is another extremely important factor. Historically there has been very little change in this aspect of design during the past decade in either front- or rear-engined cars, and on single-seater racing cars the weight distribution has remained almost identical for both types at approximately 45 per cent front and 55 per cent rear, with driver and fuel aboard. As the majority of successful current cars have weight distributions closely approximating to the above figures, it would seem that this is somewhere near the present optimum for maximum performance.

Centre of Gravity

A further factor which influences improvements in roadholding is the centre of gravity. Over the years the trend has always been towards lowering the centre of gravity, and the ingenuity of designers has been tested in finding ways of doing this. When the importance of this principle was first appreciated the centre of gravity was lowered by leaps and bounds, but it is now difficult to drop it more than a quarter or half an inch at the most, each year.

Reference to the centre of gravity usually brings to mind thoughts of a

solid object in space, whereas the application to car design should be in terms of height above the ground and position fore and aft. It is also important to bear in mind that the centre of gravity can quite easily be off-centre, even in a single-seater, where small deviations can be caused by offset mounting of carburettors, starter, dynamo, exhaust pipes and other mechanical components, or even more basic offset mountings of engine, seat or transmission. On a sports car the centre of gravity is always offset when only the driver is carried, but the situation should be restored to normal when the passenger seat is occupied.

Assuming a perfect suspension system, the height of the centre of gravity above the ground is the limiting factor in roadholding. If the centre of gravity was at ground level there would be no weight transfer and each pair of wheels would develop more cornering force than is possible when the inner wheel is lightly loaded and the outer one is much more heavily loaded, and therefore running at a larger slip angle. The centre of gravity is itself fixed but, due to centripetal acceleration in cornering, weight is transferred sideways. By Newton's first law of motion, the car will always tend to travel in a straight line until it is deflected from this course by a lateral load produced by steering the front wheels. When the wheels are turned the driver is attempting to accelerate the car towards the centre of a circle of the same radius as that of the corner which he is turning. The mass of the car, acting through its centre of gravity, attempts to go straight on, and is only prevented from doing so by the increasing slip angles of the tyres.

Imagine a car being driven round in a circle at the end of a rope attached to a post, with this rope going through the centre of gravity of the car so that the tyres are not called upon to develop any cornering force and can be maintained in a straight ahead position. If the rope were to break the car would carry straight on at a tangent to the circle which it was describing. In normal use, the forces in the rope act through the tyre contact points. The difference between the two cases is that, in the first instance, the wheels are not doing any work and are therefore travelling in the direction in which they are pointing, whereas when the tyres are doing the work of the rope to develop lateral force, they must run at an angle to the path of the car; this is the slip angle. In this hypothesis it must be assumed that the car in question is stable in a straight line—i.e. for any given side thrust applied through the centre of gravity the over-all slip angle effect is greater at the front than at the rear.

Weight Transfer

Weight transfer is another factor which has an important influence on roadholding and can actually be utilised to improve it. It is caused by centrifugal force, acting through the centre of gravity, being resisted by the adhesion of the tyres, and this couple results in a lifting force on the inside wheel and an increase in weight on the outside wheel. Another factor to be

SUSPENSION PRINCIPLES

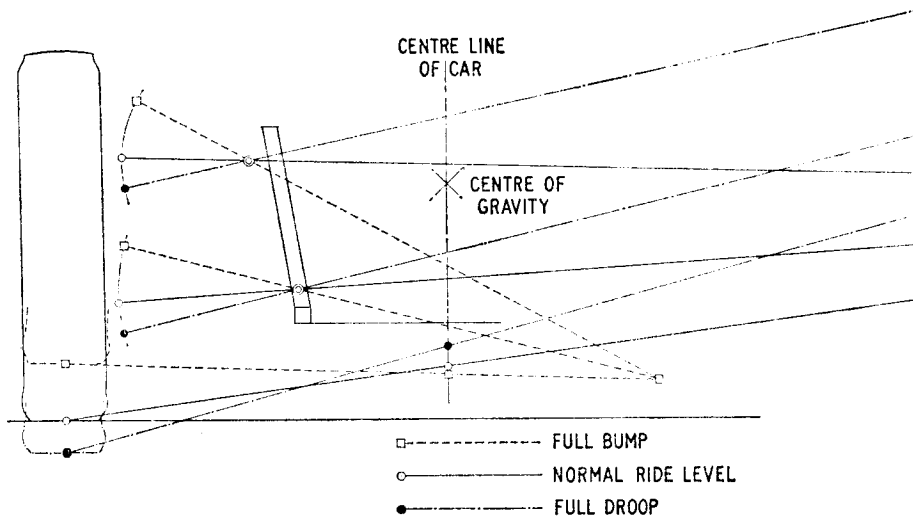
taken into consideration is the weight transfer of the axles alone. This factor is relatively unimportant in independently sprung cars, but can have a considerable influence when live axles are used.

Sprung and Unsprung Weight

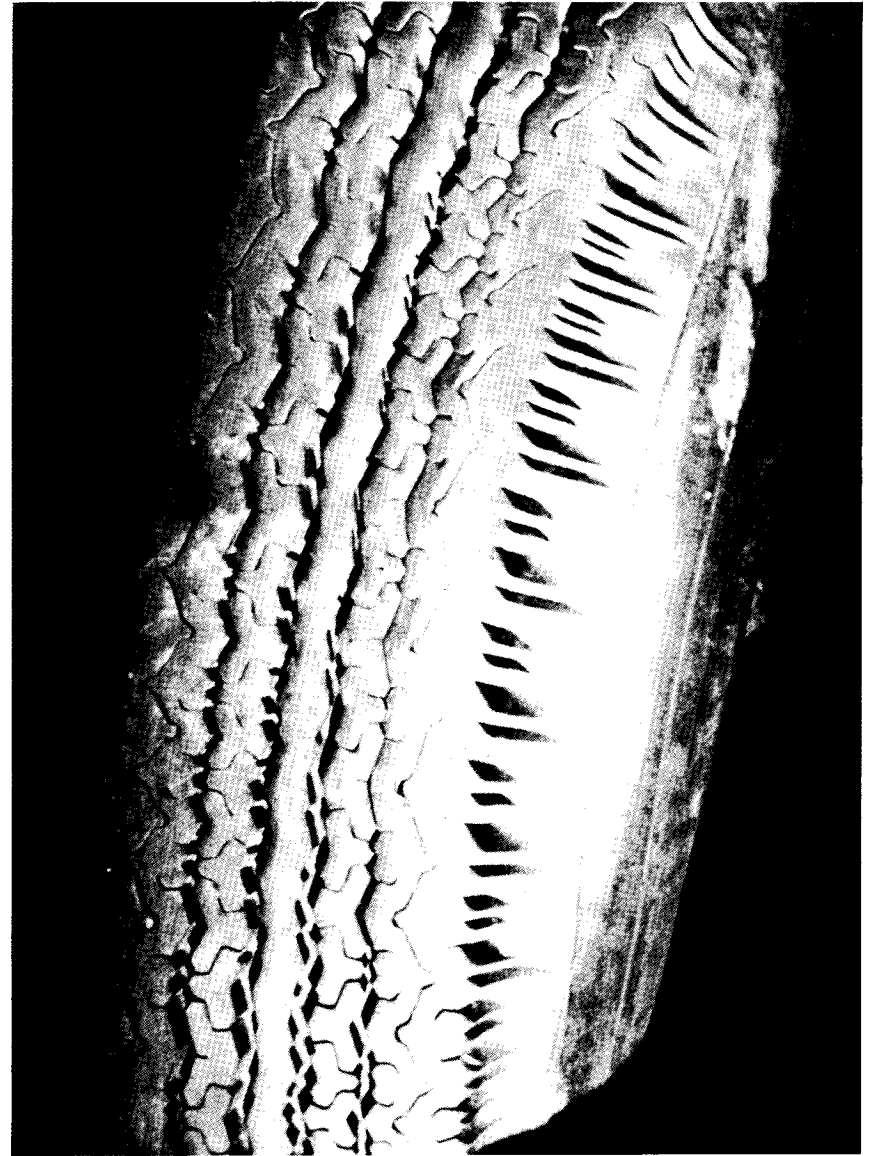
The final aspect of weight in relation to roadholding concerns the ratio of sprung to unsprung weight. This is an extremely important factor inasmuch as it is far easier to control a small moving mass than a large one. Unsprung weight, in addition to wheels, tyres, uprights and hubs, also includes parts of the suspension linkages, drive shafts and brakes—if outboard brakes are fitted. Brakes mounted on the wheels generally run cooler than those mounted inboard but cause a considerable increase in unsprung weight. Brakes mounted on the final drive housing overcome this disadvantage but are more liable to overheating—and thus poor performance. The ideal is to keep unsprung weight to a minimum and thus minimise the momentum to be controlled by suspension springs during movements caused by surface irregularities. The lower the unsprung weight the lighter the load on the chassis due to suspension movement.

Roll Centres

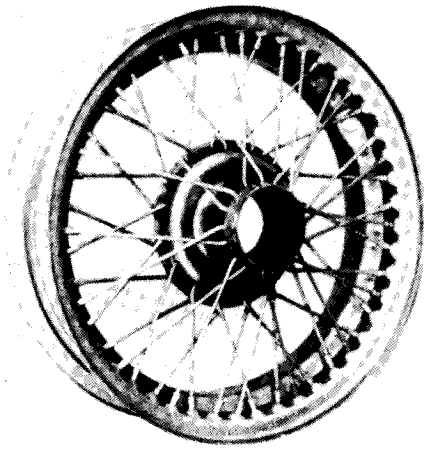
Setting aside the question of weight, the chief factor influencing roadholding through the suspension geometry is the roll centre, which must be considered not merely as a single point—like the centre of gravity—but as two separate roll centres, front and rear. Over the past few years racing practice has seen a trend towards lowering roll centres at both front and



33 Diagram showing the roll centre obtained with a typical modern double wishbone layout and the effect of suspension movement on its position

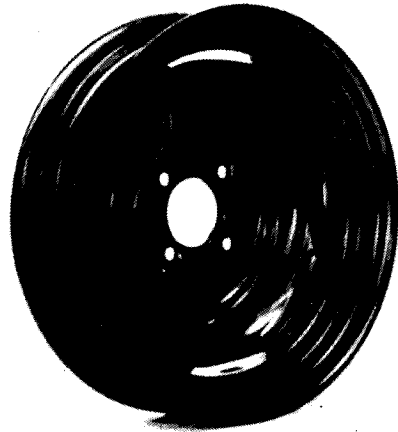


34 Some idea of the stresses to which tyres are subjected is given by this view of an ordinary road cover on the Dunlop cornering force machine. Note how the right side-wall of the tyre is distorted

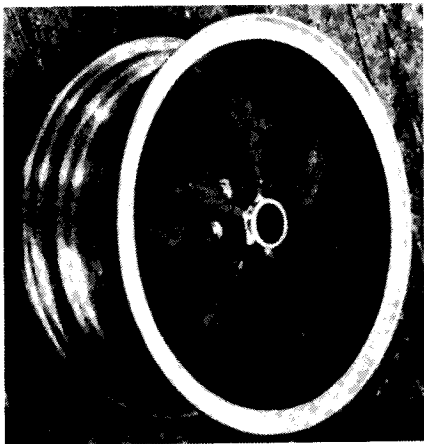


35 *The wire-spoked wheel—the traditional sports car wheel—is relatively inefficient, being the heaviest and the least stiff of the three types shown here*

36 *A pressed steel wheel, as used on the majority of mass-produced cars. Such wheels are normally somewhat stiffer than the wire-spoked type, and may also be a little lighter*



37 *A cast magnesium alloy wheel, of the type used on most current British sports and racing cars. This is by far the lightest of the three types*



rear, but there is still some divergence of opinion on the subject and there is certainly no specific rule to follow when deciding roll centres. The chief question in this case is to decide between the benefits of low roll centres—which reduce lateral deflection of the wheels to a minimum at the expense of very considerable roll angles—and high roll centres, which cut down roll angles and their associated camber effects on the wheels but which lead to considerable lateral wheel deflection on bump. High roll centres may also be impracticable due to the shortness and angularity of the linkages involved. Whichever layout is chosen, it is desirable to restrict roll centre movement to a minimum to maintain relatively constant weight transfer.

As suspension linkages are the product of roll centres, this is obviously one of the first basic design features which must be decided before a car can be made. In the extreme, very low roll centres lead to large roll angles and suspension travel being used up by roll. This leads to difficulties when suspension movement is needed to cope with uneven road surfaces. High roll centres, with their reduced roll angles, bring the attendant disadvantage of considerable lateral disturbances of the tyre tread contact patches, which leads to marked instability in cornering. Both extremes obviously have major disadvantages, and it is thus necessary for the designer to compromise in order to obtain the best of both features. As can be seen in current advanced designs, the best way of achieving the optimum is to use fairly low roll centres in conjunction with anti-roll bars. The latter, however, require considerable study, or prolonged practical experiment, before the optimum anti-roll stiffness can be achieved.

To judge from current designs, it would seem that the roll centre of a car should be lower at the front than at the rear in order to give proportionately more weight transfer to the front suspension, thereby building in the initial understeering characteristic which is generally accepted present-day practice. The ideal is a front roll centre only slightly below the rear roll centre; this is partly due to the complication arising from the use of centres at widely different heights. A very low front roll centre transfers a considerable amount of weight, a high rear roll centre very little, and in the extreme this would lead to marked understeer and lifting of the inside front wheel.

In principle, the mass of the car concentrated round the centre of gravity rolls round the roll axis. A certain amount of roll is generally considered necessary to give the driver an indication of the amount of cornering force which is being developed. Excessive roll, as well as being physically uncomfortable, leads to the suspension units running near their bump stops and the car being liable to disturbance due to the suspension bottoming; it also has adverse effects on wheel angles, which necessitate correction by large negative cambers. The maximum roll angle should not exceed 3–4 degrees at 0.75 g cornering force. In this context it should be remembered that the cornering force available decreases at higher speeds.

Suspension Frequency

Closely connected with weight, roll centres (and their associated problems) and suspension layout, is the question of suspension frequency. This is the rate of the suspension, generally referred to in cycles per minute. Here the present trend is towards lower frequencies, and current racing cars generally have frequencies of around 70/80 cycles per minute. This figure refers to the frequency of the sprung mass in free oscillation. In order to avoid the possibility of the front and rear suspension oscillating in harmony it is necessary to have a slight difference in frequency between front and rear. The normal practice is to have the rear slightly higher than the front, to help the rear suspension accommodate any given upward deflection of the front suspension and to dampen any subsequent pitching.

Care must be taken to ensure the correct relationship between suspension frequency and suspension travel. All the suspension travel must not be used up under normal conditions of operation, as bottoming of suspension seriously upsets the normal balance of the suspension, by raising the spring rate to infinity. This in its turn has drastic effects on weight transfer, and immediately influences the stability of the car. Other factors influencing roadholding are suspension rigidity, which is largely self-explanatory, wheel stiffness (together with weight as explained under the heading "Unsprung Weight") and tyres.

Tyres

As most readers will appreciate from a change of tyres on everyday road-going vehicles, there have been considerable developments in tyre adhesion during recent years. In sports and racing tyres the rate of development has been fantastic. The reasons for improvement are many but are bound up with the type of construction, the change from cotton (via rayon) to nylon casings and the development of rubber mixtures. For all this, racing tyres have also become more susceptible to changes of road surface and weather conditions. Nevertheless, the improvement in lap times for given circuits over the past few years is attributable more to tyre improvements than to any other single factor.

Wheel Stiffness

Another factor which has contributed considerably to the general development of roadholding over the last few years is increased wheel stiffness. It is noteworthy that the racing cars generally acknowledged to have the best roadholding currently use cast magnesium wheels, which are generally stiffer than the pressed steel disc wheel or the wire-spoked sports-type wheel. Another important point is that cast magnesium wheels are also considerably lighter than the other types. This both promotes

roadholding and reduces tyre wear, and it is thus possible on many circuits to use bolt-on cast magnesium wheels for quite long races. Hitherto the use of such wheels, or disc wheels, has been vetoed on the grounds that they took too long to detach when quick changes were essential. Nowadays, of course, the steel disc wheel can be used in conjunction with the knock-on type of hub, and so in fact could the cast magnesium type, at a considerable penalty in weight—unsprung weight. In view of such developments there is little to be said in favour of the wire spoke wheel, which is in many cases both the heaviest and the least stiff of the three types (as well as the most difficult to clean). On present-day racing cars it is interesting to bear in mind that, of the total unsprung weight, particularly with cast magnesium wheels, the greater proportion is still represented by the tyres. This is one of the reasons for the current trend towards smaller diameter wheels and tyres (decreased aerodynamic disturbance is another) but it would seem that the move towards wider rims and treads, in the interests of roadholding, conflicts with both these aims.

Rim Width

In conjunction with wheels and tyres, wheel rim width also plays an important part in roadholding. During the last four years there has been a considerable increase in the rim widths, tyre sections and tyre tread widths used on sports and racing cars, and this is attributable in the first place to American trends. Originally, increased tread width was required to assist power absorption with high power-to-weight ratios. This was accompanied by a flattening of the tread surface. Parallel to this came the widening of rims and a general tendency by tyre companies towards the use of wider treads and larger sections, with decreased weight due to the use of thinner, lighter materials. It would be generally agreed by the majority of technicians that, of a given five seconds per lap improvement in lap times over the last four years, two or even three seconds are attributable to tyre and wheel development.

Anti-roll Bars

The basic purpose of an anti-roll bar is to allow the use of fairly low roll centres—with the attendant advantage of minimum jacking effect due to lateral loading and minimum lateral wheel deflection on bump—without the disadvantage of large roll angles and consequent camber change. If the necessary anti-roll stiffness were provided by the suspension springs the result would probably be an extremely harsh ride, with the tyres frequently losing contact with the road on rough surfaces. On many cars a single anti-roll bar is fitted, usually in the front. The chief advantage of this is that it reduces the large changes of wheel camber which are associated with excessive body roll. However, in this connection it must always be

remembered that for one wheel bump condition the anti-roll bar effectively adds its rate to that of the suspension spring. Another important effect of fitting an anti-roll bar is that it avoids the using up of suspension movement due to excessive roll. The criterion in this case is to avoid the development of positive camber due to excessive roll.

The fitting of a front anti-roll bar also leads to reduced weight transfer on the rear wheels due to the decreased roll angle at the front. It does, however, promote weight transfer from the inside rear wheel to the outside front wheel, thereby increasing the slip angle of the outside front tyre—and increasing the degree of understeer. Great care is required with the fitting of a rear anti-roll bar, as this may lead to wheelspin on one side when the other wheel is on bump due to surface irregularities.

Going further into the application of anti-roll bars, should it be desirable to limit the roll angle but at the same time not increase suspension frequency, then it is possible to use anti-roll bars at both front and rear, in order to reduce the changes of camber angle involved by cutting down the roll angle itself. Great care must be taken to ensure that the correct ratio between front and rear roll bar effects is maintained. In brief, when an anti-roll bar is fitted at the rear as well as at the front then the stiffness of the front bar must be increased. In this instance it should be remembered that stiffness increases as the fourth power of the diameter. In any case the fitting of anti-roll bars, either at both front and rear or at the front alone, must always be considered as part of the original design, and not merely as an afterthought to make up for suspension inadequacies. It is also important to remember that two wheels connected by an anti-roll bar cease to be suspended completely independently; a really stiff bar can cause considerable interference at one wheel when the other is affected by irregularities of the road surface.

Springs

Springs have a marked effect on roadholding, largely as a result of their stiffness. Spring rates should, in general, be as soft as possible provided this does not increase the amount of roll and cause the suspension to reach its bump stops. The basic aim of the springs is to keep the wheels on the ground and allow each wheel to rise on bump without affecting the car. Stiffer springs reduce roll but bring in other undesirable effects which are particularly noticeable on rough surfaces. In principle, suspension can only be 100 per cent efficient on a completely smooth surface, and bumpy road surfaces radically decrease the efficiency of any suspension system.

Like the suspension springs, spring dampers have a very important influence on roadholding through their effectiveness in keeping the wheels on the ground all the time. In this connection it is important to remember that increased stiffness of the dampers is liable to affect bump as well as rebound. Front dampers have a slight damping effect on bump (about one

third of that on rebound), whereas rear dampers have damping only on rebound. Adjustable dampers, as fitted on current racing and sports/racing cars, enable a fine balance to be obtained between suspension bounce (when the wheel can be observed, and felt, to be working too freely) and a harsh ride, with too much shock being transmitted to the chassis and with the suspension working out of phase.

Wheelbase and Track

A generalisation on the subject of wheelbase and track is that long wheelbases tend to promote straight line stability while wide tracks with relatively short wheelbases give optimum cornering power.

SUMMARY OF WHEELBASES AND TRACKS ON RECENT FORMULA ONE AND SPORTS CARS

	<i>Wheelbase</i>	<i>Track front</i>	<i>Track rear</i>
1964 Lotus F1	7 ft 7 in	4 ft 8 in	4 ft 8 in
1964 BRM F1	7 ft 6½ in	4 ft 6 in	4 ft 5 in
1960 Cooper F1	7 ft 7 in	3 ft 10½ in	4 ft 0 in
1960 Lotus F1	7 ft 6 in	4 ft 1¾ in	3 ft 11 in
1960 B.R.M. F1	7 ft 5¾ in	4 ft 0½ in	4 ft 3¼ in
1960 Ferrari F1	7 ft 3½ in	4 ft 0 in	3 ft 11 in
1958 Vanwall F1	7 ft 6¼ in	4 ft 5¾ in	4 ft 3½ in
1957 Maserati F1	7 ft 6 in	4 ft 4 in	4 ft 2 in
Cooper Monaco	7 ft 7 in	3 ft 10½ in	4 ft 0 in
Lotus 19	7 ft 6 in	4 ft 1¾ in	3 ft 11 in
Lotus 15	7 ft 4 in	3 ft 11 in	4 ft 0 in
A.C. Ace	7 ft 6 in	4 ft 2 in	4 ft 2 in
Lola Sports	7 ft 1 in	4 ft 0 in	3 ft 11½ in
Maserati 61	7 ft 2½ in	4 ft 2 in	4 ft 1 in
1958 Lister-Jaguar	7 ft 6¾ in	4 ft 4 in	4 ft 5½ in

From the table above it can be seen that the successful racing and sports cars of recent years all fall within very narrow limits of dimensions as far as wheelbase and track are concerned. In recent years the tendency has been towards wider tracks, particularly at the rear, to increase cornering power. This has been achieved without any noticeable loss of directional stability, despite the fact that lighter cars are generally more prone to wind effects. From this point of view it is vital that the centre of pressure on the body area in side elevation, calculated on the area of body presented to a side wind, should be behind the centre of gravity in order that natural corrections can be applied by the driver whenever necessary. It is also important to avoid toe-in on bump because of the stiffness of current tyres.

Handling

“Understeer” and “oversteer” are two motoring terms in common usage, yet not everybody knows exactly what they mean. Basically,

understeer is caused by the front wheels of a car having less cornering power, in relation to the weight they support, than the rear ones. If the front wheels have more cornering power than the rear ones, then the tendency is towards oversteer. The causes of understeer and oversteer vary from car to car, but they are normally connected with the camber of the respective wheels—particularly the outside wheels—and the slip angles at which the tyres run when cornering. If the front wheels are less upright than the rear ones the tendency is towards understeer. This situation is commonly found nowadays, both on cars with independent front suspension and live rear axles (which do keep the wheels virtually upright, despite their other disadvantages), and on the most advanced rear-engined sports and racing cars. Because most early rear-engined cars had simple swing axle rear suspension—notable for rapid camber change—there grew up a legend that *all* rear-engined cars oversteer. With advanced suspension geometry, however, it is possible to make most rear- or mid-engined cars understeer strongly, if required.

In addition, many racing and sports cars have wider wheel rims and wider tyres at the rear, partly in the interests of forward traction but mainly to increase cornering power. Any relative decrease of cornering power at the front is due partly to the smaller area of rubber in contact with the road and partly to increased slip angles; this is controlled partly by camber change due to suspension geometry and partly by weight transfer.

Although a car's basic handling characteristics are decided in the drawing office, the degree of understeer or oversteer can be varied, within reasonable limits, by alteration of tyre pressures. It is not yet universally realised that—whatever the conditions—increased tyre pressure results in increased cornering power. This is not to imply that pressures should not be reduced under wet or other unfavourable conditions. It merely means that blowing up either front or rear tyres can materially affect a car's handling characteristics. More air in the rear tyres increases their cornering power, and thus promotes a tendency towards understeer. Alternatively, on an oversteering car it reduces the amount of oversteer. More air in the front tyres will either reduce understeer or increase oversteer. In a car showing a strong tendency towards one or other of these conditions it is not normally possible to bring about a complete change of characteristics in this way, but it should at least be fairly easy to make some improvement.

There is far more to this than the indiscriminate increase of tyre pressures beyond normal recommendations, however. In a road car high pressures lead to a harsh and uncomfortable ride. In sports or racing machinery they lead, on bumpy surfaces, to the tyres being only in intermittent contact with the road, and thus losing even more cornering power than they gain under ideal conditions. Once they have been determined, the correct tyre pressures for any given racing car can be altered very little either way without adversely affecting circuit speeds. Individual drivers will require a variation of two to four pounds per square inch, to obtain

the handling characteristics most nearly approaching their ideal, but rarely more than this. It may also prove advantageous to adjust tyre pressures to promote understeer on a high speed circuit or to counteract it on a twisty one. In most modern racing cars, however, provision is made for altering handling characteristics more basically by means of adjustable suspension linkages and alternative anti-roll bars.

Adjustable Suspension

The use of adjustable suspension is a fairly recent development, and one which has been described in some quarters as a “hit-or-miss” method of design. It has many advantages, however, and can be considered a very practical alternative to the ideal of building a different car for every circuit. Wheelbase and track cannot normally be altered in this way, but virtually every other factor in suspension can. Roll centres can be changed by providing alternative inboard pick-ups for the top wishbones, and camber can be altered by means of telescopic, threaded linkages. Toe-in and toe-out can be controlled in a similar way, and springs and dampers can be changed in a matter of minutes to obtain the most suitable rates for a given circuit. Where anti-roll bars are used, their thickness can be varied to give more or less roll stiffness and thus more or less weight transfer to the front wheels—with consequent effects on camber change and slip angles. The chief purpose of a front anti-roll bar is to provide the advantages of a low roll centre—minimum lateral deflection on bump—without the disadvantage of large roll angles and camber change. Increasing the thickness of the front anti-roll bar promotes understeer—and vice versa—but the variations which can usefully be made by this means are fairly small, and are normally measurable in 32nds of an inch. On a car not originally designed to use an anti-roll bar it may be possible to bring about a considerable improvement in cornering power by fitting one, but this may lead to the degree of understeer on fast corners reaching undesirable proportions.

Braking

Stability under braking is an extremely important factor in high-performance cars, and depends partly on the layout of suspension linkages, which transmit braking loads to the chassis, and partly on the stiffness of the chassis and the ease with which it dissipates these loads. Ideally, loads reacted directly on the plane in which they are developed, and taken out progressively through the chassis, will have the least effect on the car's stability. Furthermore, loads taken out over a wide base through parallel linkages will be of far lower magnitude than those fed through a single linkage. Lotus experienced serious braking problems with their 1959 Formula One car, which used a trailing anti-roll bar as one arm of the top wishbone. At its worst this led to several chassis breakages under braking

from high speed. The 1960 car, with wide-based double wishbones and a separate anti-roll bar, suffered no chassis failures whatsoever.

Another problem connected with braking is that of nose dip. To avoid—or minimise—this, the suspension must be laid out in such a way that lines drawn through the inboard pick-ups of the linkages reacting braking loads pass—when produced—through the centre of gravity. Other factors may conflict with this, of course, but it should nevertheless be borne in mind in the interests of ultimate over-all performance.

SUSPENSION PRACTICE AND TYPES

Suspension can only be as good as resources will allow. In this context, as with the power unit and transmission, it is obviously cheapest to use mass-produced proprietary components or extremely simple layouts with the minimum number of linkages. Proprietary front suspension assemblies of the double wishbone type are usually too heavy for specialist cars (and may also be geometrically inadequate), but it should be possible to fabricate a tubular double wishbone linkage fairly inexpensively. Failing this, the cheapest alternative would seem to be the use of a beam axle cut in half and modified to give swing axle suspension.

If a live rear axle is considered adequate, the adoption of one of the many proprietary assemblies on the market would seem to be an obvious step, such considerations as weight, track and final drive ratio (or available alternatives) governing the final choice. Of proprietary independent rear suspension layouts, only swing axle systems—with their attendant disadvantages—are available cheaply, and here again it is necessary for the constructor to make his own parts if a more advanced layout is required.

From this it is obvious that, if cost is a major factor, it may be necessary to forgo the ideal in suspension, as in other aspects of design. Fortunately, provided welding facilities are available, it is possible to fabricate most chassis and suspension parts from cheap and readily obtainable material, but work of this nature must be backed up by sound, up-to-date design. Without this, even unlimited resources cannot produce a successful car, as has been shown by a number of contenders for Grand Prix honours in recent years.

On a road car, a live rear axle can provide quite satisfactory roadholding, but for serious National or International class racing independent suspension is now an essential on sports and racing cars. Furthermore, this independent suspension must be of an advanced type for the car to have any chance of success, and thus will generally conform with current practice. Due to the extremely rapid development of suspension in recent years, even suspension principles are liable to change, and thus a layout which is regarded as ideal one year can become obsolete within the course of a single racing season.

Numerous arguments can be put forward for independent suspension in general, and for the more advanced forms in particular. All of them offer reduced unsprung weight by comparison with live axle layouts, with beneficial effects on both roadholding and ride. Both of these factors also gain considerably from the fact that one wheel can move up and down

SUSPENSION PRACTICE AND TYPES

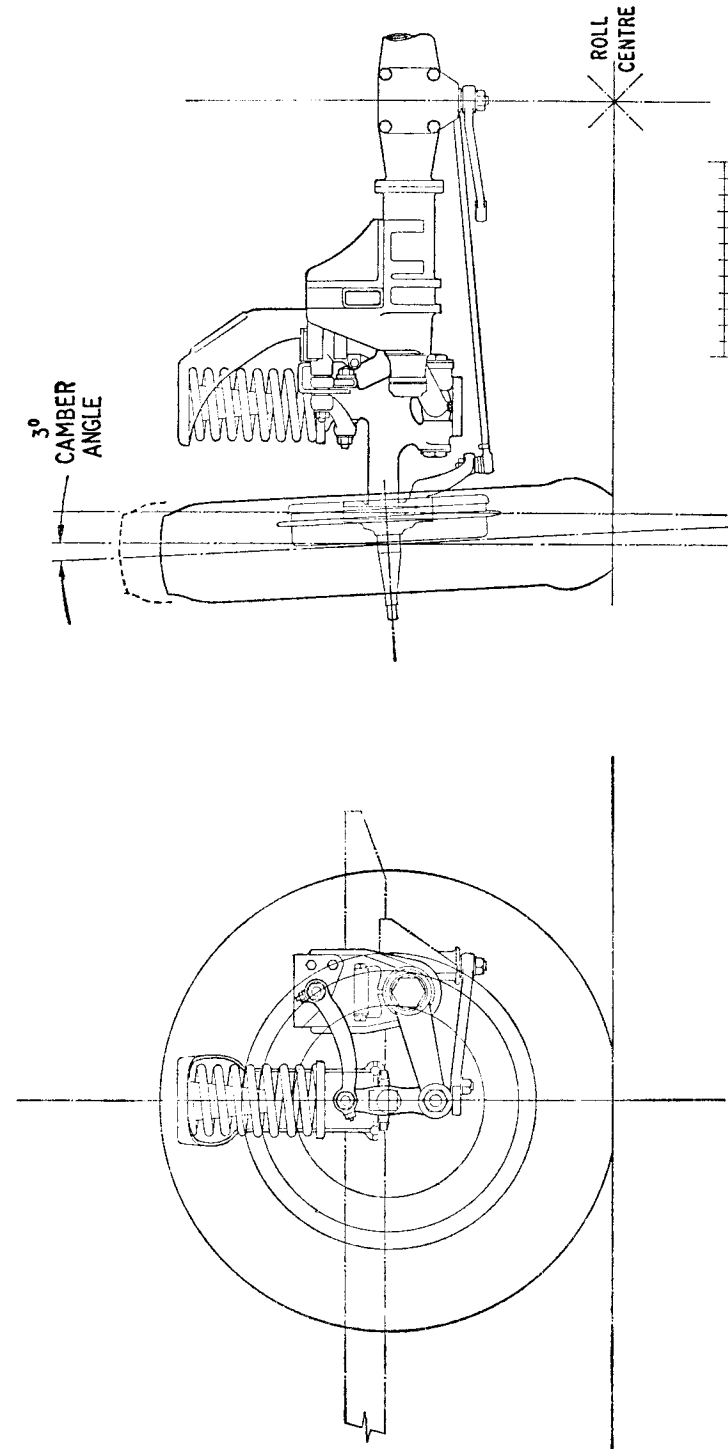
independently of the other and without influencing the other—and possibly without affecting the attitude of the car if advanced springing is employed. The insulation of one wheel from another also prevents the “sideways hop” characteristic of many live-axle cars when one wheel encounters a bump while being subjected to considerable cornering force. The use of a chassis-mounted differential eliminates the effect of propeller shaft torque, which can, in extreme cases, lead to handling characteristics varying from oversteer on right-hand corners to understeer on left-hand turns. In road cars of conventional layout the use of a fixed final drive unit also makes it possible for the luggage boot to be larger than would be the case with a live axle, when provision must be made for movement of the whole axle casing on bump. Because of the need to insulate the power unit from road shock, independent suspension is virtually imperative in conjunction with either a rear engine driving the rear wheels or a front engine driving the front wheels.

Unfortunately, due to the fairly widespread use of the simple swing axle in both racing and production cars, there was a time when independent rear suspension had a bad name from the handling point of view. This is why de Dion rear suspension was retained on the majority of racing and sports/racing cars until the late 1950s. Today, however, the advantages of systems which keep the outside rear wheel virtually upright under all conditions of roll and bump are apparent to all who take an intelligent interest in motor racing (see figs. 29 and 30). The cars which develop the greatest cornering force at the time of writing are those with double wish-bone type suspension at both front and rear, in conjunction with chassis having considerable torsional rigidity. On most of these cars anti-roll bars are required at both front and rear to counteract the effects of using relatively low roll centres. The resultant layout may appear to be fairly complicated, but in its most advanced (yet simplest) forms consists merely of two links taking lateral loads and two taking fore-and-aft loads, with a torsion bar controlling the amount of weight transfer from the inside to the outside wheel. The geometry of the lateral links is all important in regulating camber change and keeping it within the desired limits.

Suspension Types

Trailing link (See fig. 38)

Normally a trailing link layout consists of two short arms, which are effectively parallel, locating a front hub assembly of fairly massive dimensions. Heavy construction is necessary, as the trailing links provide much better wheel location in a fore-and-aft plane than in a vertical plane. Torsion bar springing is commonly used in conjunction with this type of layout, with the lower link acting on a transverse bar located in a chassis cross member. An alternative, as instanced by the Aston Martin DB2-4,



38 The chief advantage of the trailing link type of suspension is that the number of components involved is reduced to a minimum. However, high bearing loads and camber change directly related to body roll more than offset this factor

SUSPENSION PRACTICE AND TYPES

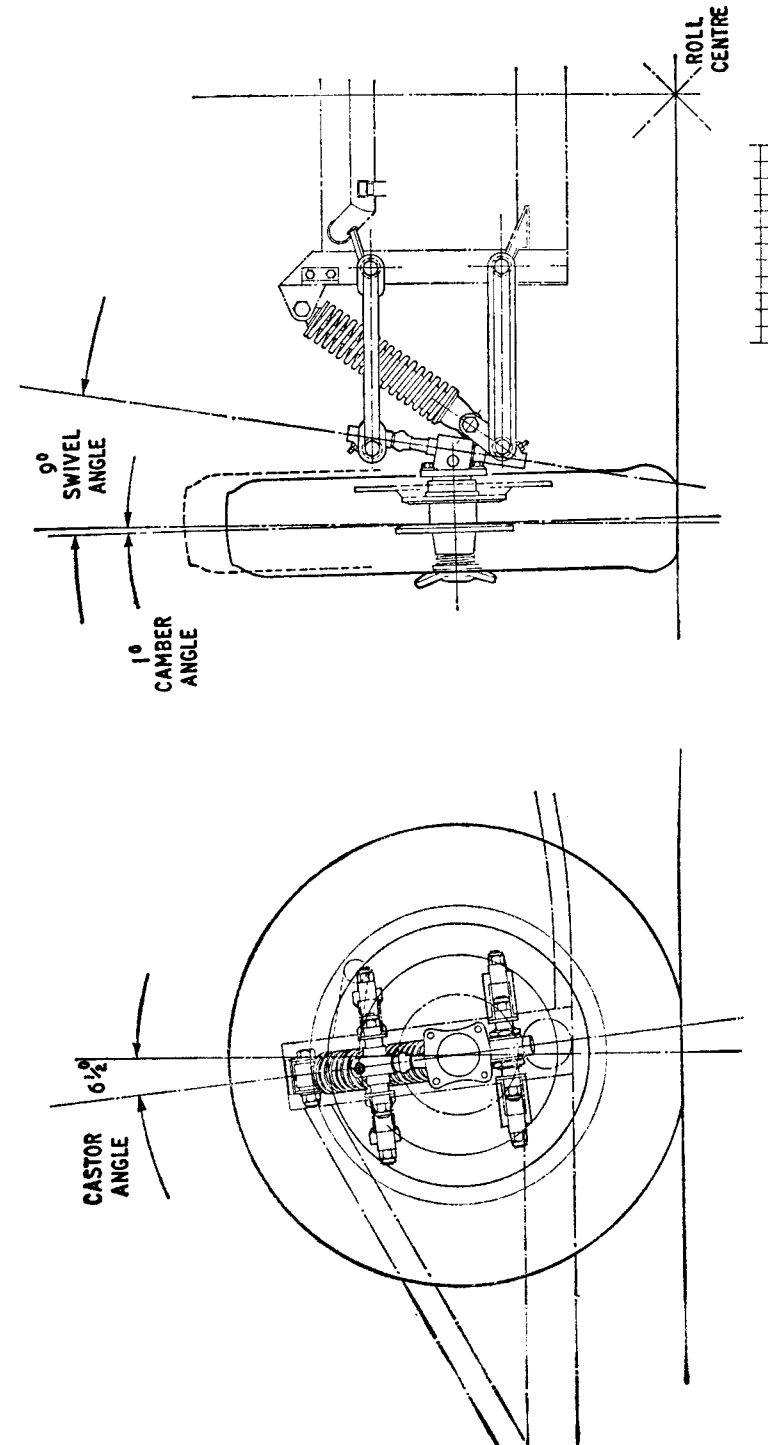
is the use of coil springs acting on the upper link, with the chassis cross member accommodating an anti-roll bar.

Although this type of front suspension has been used until recently on Porsche sports and racing cars it is generally excluded from the current range of ultra-high performance cars because the camber angle is directly dependent on, and changes with, body roll. As the roll centre with this type of layout is at ground level, quite considerable roll angles are likely unless some form of anti-roll bar is used. Structurally the chief disadvantage is that high loads are put through the suspension, leading to bending loads in the links themselves and thus requiring the use of heavy material. Another disadvantage is that, due to the very high bearing loads at the pick-up points, a slight amount of wear promotes very considerable play. One great advantage of this type of suspension, however, is that it can be fitted in a comparatively small vertical space, and this helps to keep frontal area to a minimum.

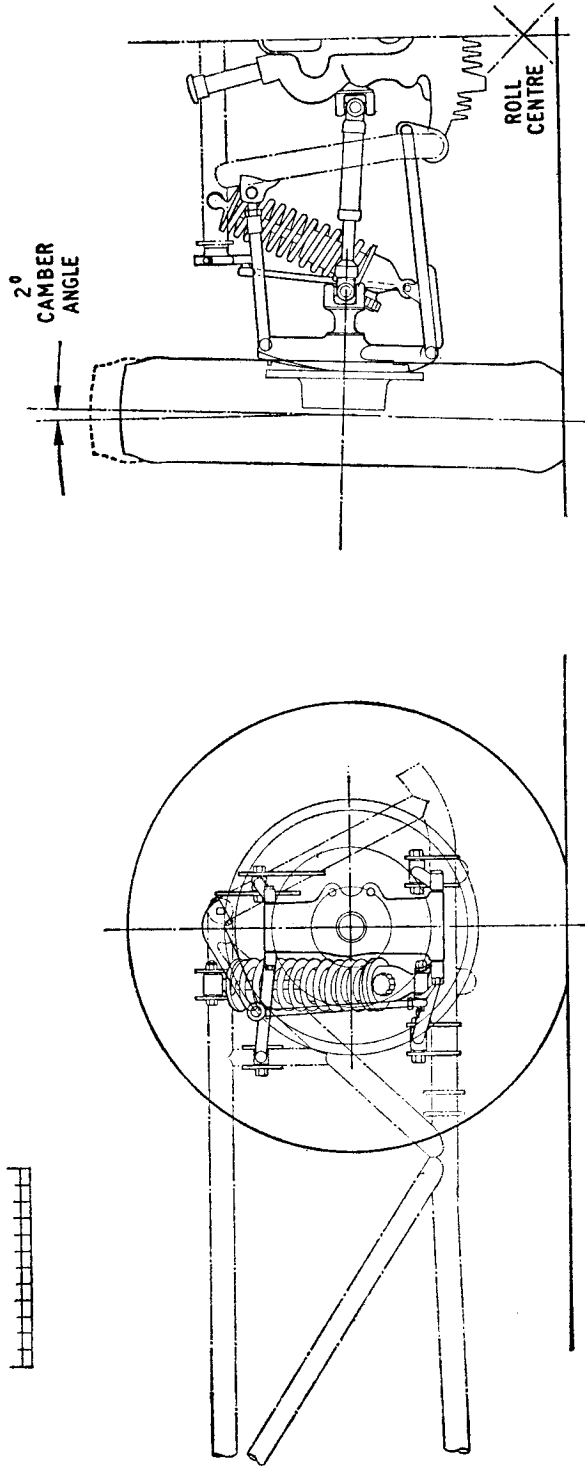
Wishbones (See figs. 39, 40, 41 and 42)

There are so many types of wishbone suspension that it is necessary first of all to consider the progressive development from the original short, equal length parallel wishbones—generally horizontal at normal ride level—to the latest unequal length, non-parallel, asymmetrical layouts. As first introduced, equal length parallel wishbones (fig. 39) represented a considerable advance in that they cut down the disadvantages of interference from wheel to wheel as experienced with beam axles. They also assisted the reduction of unsprung weight, because with the resultant smaller suspension loadings the structure could be made considerably lighter. The next step was to one short (top) wishbone and one long one, still parallel and horizontal. This was a definite step forward and reduced the effect of body roll on wheel camber change. The move to unparallel, unequal length wishbones brought a more pronounced effect on camber change, and for the first time made it possible for the outside wheel to be kept virtually upright under all conditions of body roll. The success of wishbone type suspension depends on a large number of factors—the length and angle of the links, the position of pick-ups—both inboard and outboard—and the maintenance of the desired roll centre under all conditions. On this subject it is necessary to remember that the roll centre directly governs weight transfer; its ideal height can only be worked out as the result of experience.

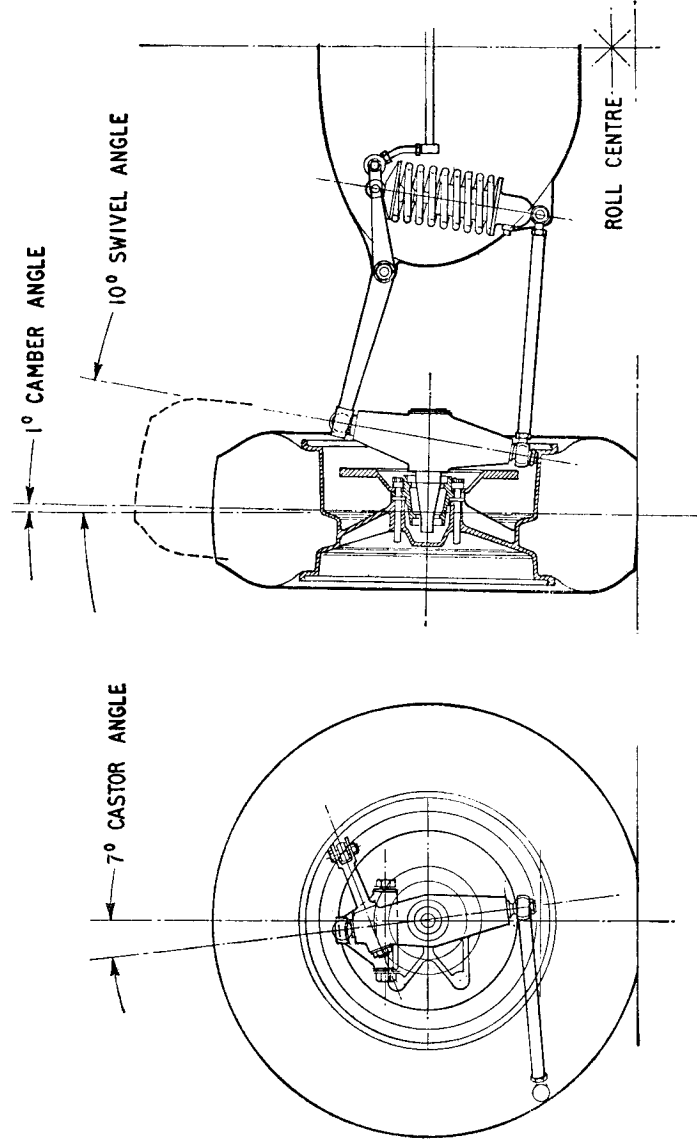
Whereas many wishbone designs have symmetrical linkages and some—particularly those of the pressed type—consist virtually of twin transverse links, making little provision for fore-and-aft location other than in the stiffness of their mountings, the current tendency is towards asymmetrical layouts, with lateral and fore-and-aft loads taken out separately over an extremely wide base. This type of design can be used at both front and rear, and an example of its application in the latter context is shown in fig. 42.



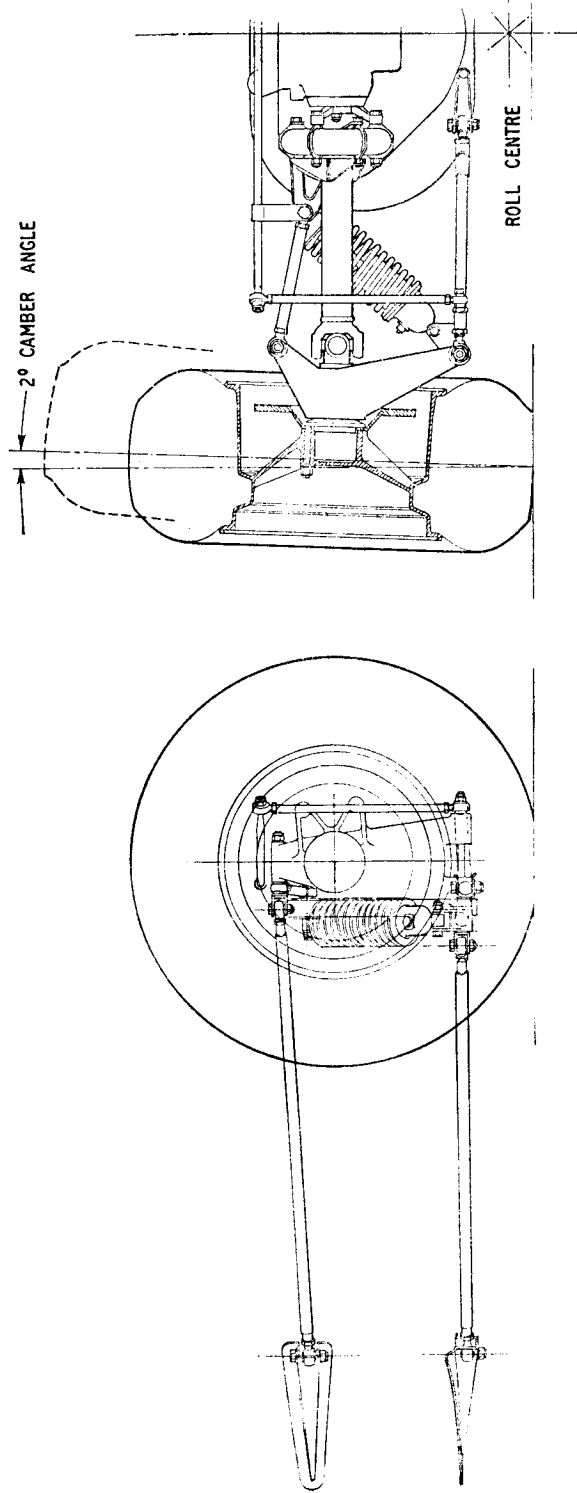
39 Equal length, parallel wishbones, as instanced by the Lister-Jaguar. Dotted line shows position assumed by wheel on suspension bump travel; this does nothing to offset camber change due to body roll, to which, in this case, it is directly related



40 Double wishbone rear suspension exemplified by the 1960 Formula One Cooper. The wishbone links provide both lateral and fore-and-aft location; no suspension loads are taken out by the spined drive shafts. An anti-roll bar, housed in the top transverse chassis member, is connected to the lower wishbone



41 Unequal length, non-parallel wishbones—with the added refinement of inboard suspension units—as seen on the Lotus 25/33. The dotted line shows how wheel movement on suspension bump travel largely compensates for camber change due to body roll; the latter has far less effect on camber change in this case than with less advanced layouts. The suspension unit takes up a lot of space in an area where space is at a premium, but as the damper forms the bulk of it there is little to be gained from the use of longitudinal torsion bars instead of coil springs



42 Wide-based double wishbone rear suspension, as used on the Lotus 33. Lateral location is provided by the two transverse links—the lower one a “reversed wishbone”, with one inboard and two outboard mountings—and fore-and-aft location is by the asymmetrical radius arms. A similar layout, but with parallel radius arms, was used on the Lotus 25

SUSPENSION PRACTICE AND TYPES

The loads involved are far smaller than those associated with the more conventional type of wishbone, and can normally be taken out into the chassis far more easily. Wishbones can be made in a variety of ways, but those of the tubular type are undoubtedly the simplest and cheapest for small-scale production. In this context it is significant that the conventional narrow-base wishbone sometimes needs cross-bracing, whereas the wide-based type, with its reduced loads, can be made in relatively light gauge tube without suffering any adverse effects.

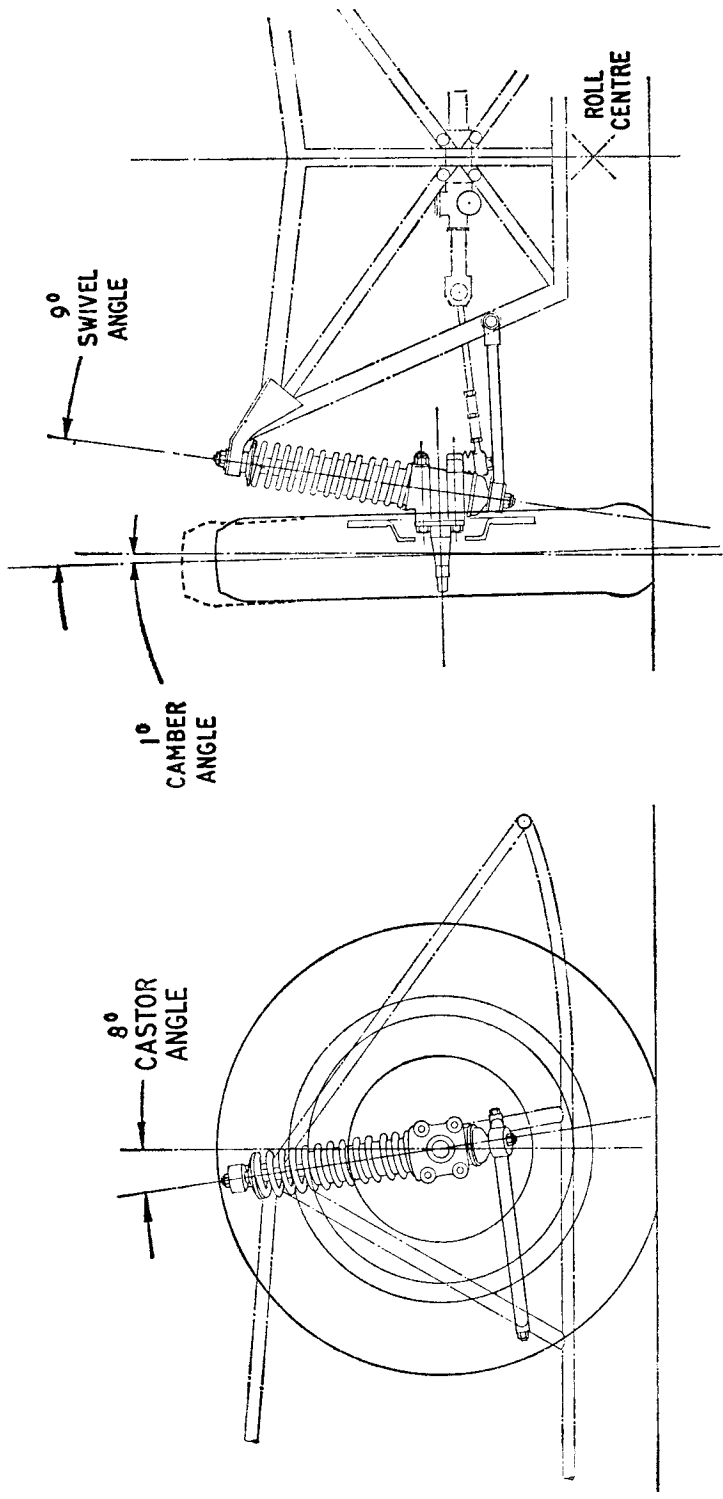
Since 1962 it has become fashionable to mount the front suspension units of single-seaters within the bodywork, and operate them by rocking arms pivoted on the chassis (see fig. 41). The chief purpose of this is to get the suspension units out of the airstream (in most cases the track rods are mounted in line with the top wishbones for the same reason) but the very slight advantage gained in this way is offset by the difficulty of accommodating the suspension units in an area where space is already at a premium, and also by the possibility of overheating the dampers. (Ducts to cool the latter take up yet more space.) In addition, special springs, dampers and bump rubbers are necessary on account of the lever action of the top wishbone (the ratio of which should not exceed 2 to 1) and special bearings are needed where it pivots on the chassis. There have also been several attempts to get rear suspension units inboard, but as the airstream is so broken up by this time there seems little justification for them.

Strut-type suspension (See figs. 43 and 44)

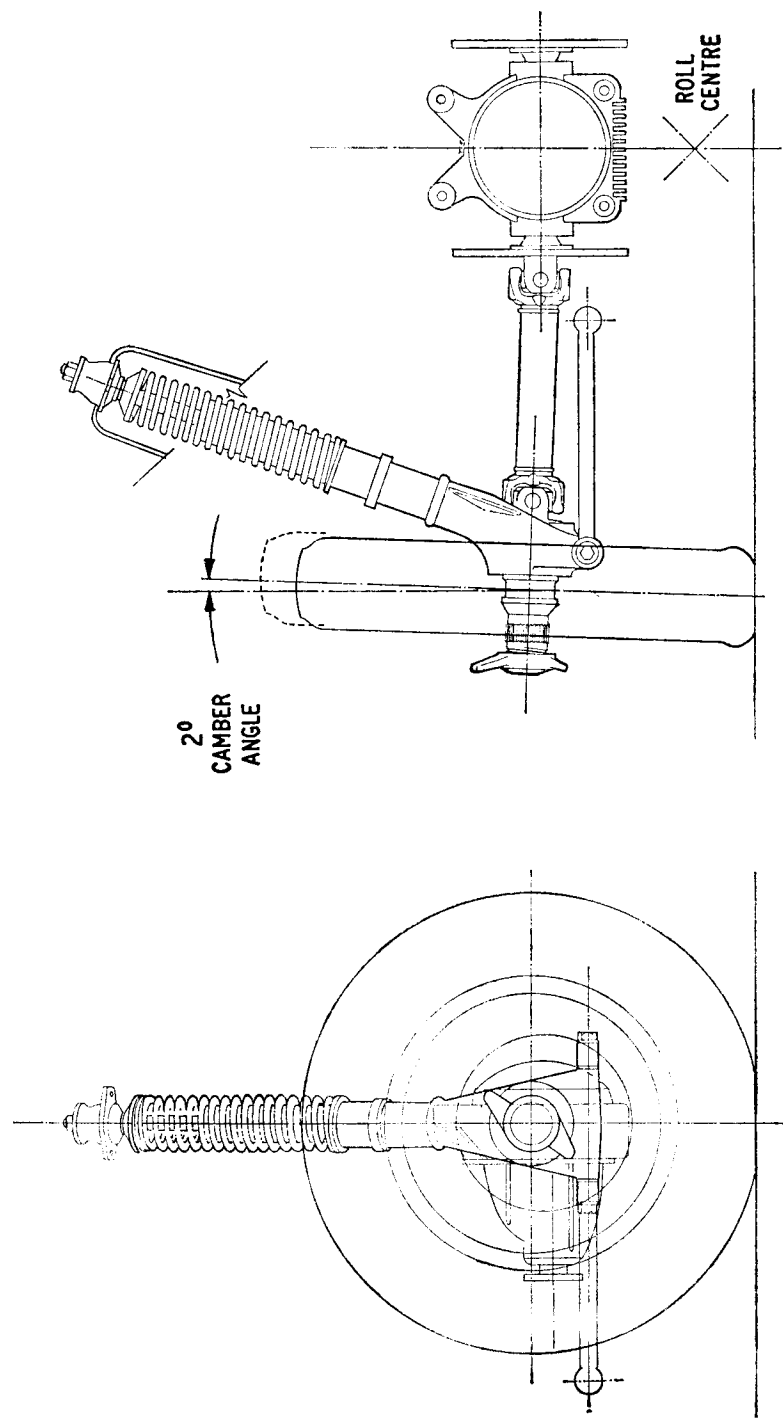
The most common use of the strut layout is for the front suspension of all current English Ford saloon cars. It consists of a coil spring/damper unit, whose lower end also acts as a king post, and a single bottom wishbone. The same basic layout has been successfully applied to the rear suspension of the Lotus Elite and a number of other cars. In this case the lower wishbone consists of the unsplined drive shaft and a radius arm. There have also been instances of strut-type rear suspension with splined drive shafts and separate lower wishbones. In many ways the basic strut system is ideal. It is extremely simple. It ensures the maximum utilisation of all components. Very good suspension geometry can be obtained, but to work satisfactorily the strut requires a large amount of vertical space; this renders it generally unsuitable for cars of low frontal area, especially at the front. A good example of this is the Lotus Seventeen, in which the suspension was designed on the strut principle to fit into a very small vertical space. This reduced the designed wheel travel to a minimum and increased the bearing load on the strut to undesirable proportions. After prolonged experiment this suspension was eventually abandoned in favour of a more conventional wishbone system.

Swing axle (See figs. 45 and 46)

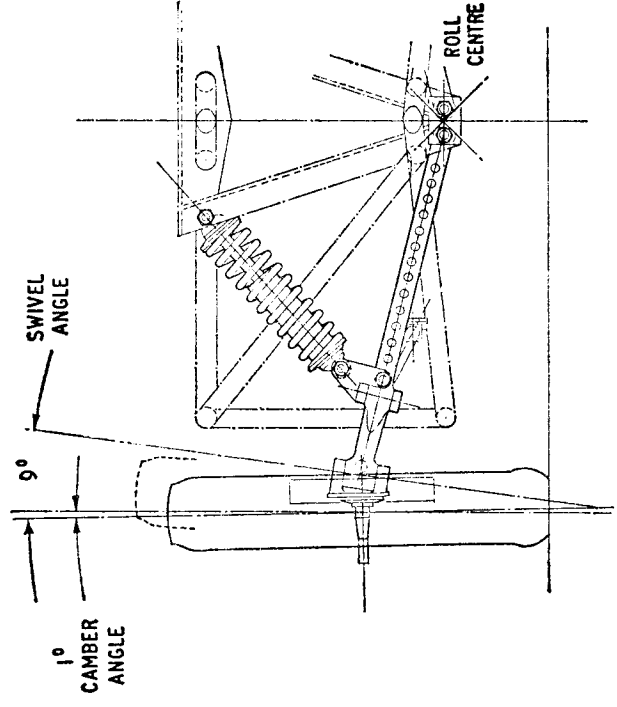
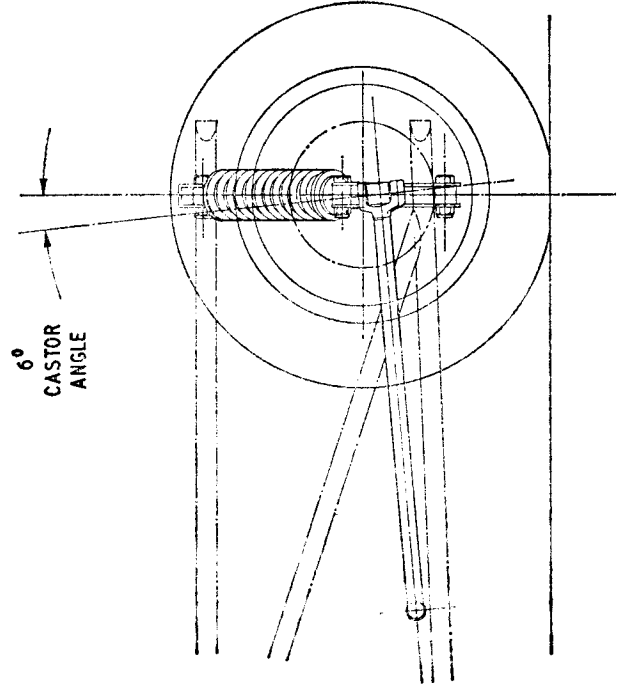
As its name implies, the swing axle suspension system consists of a swinging half axle, pivoted at or near the centre line of the car in front



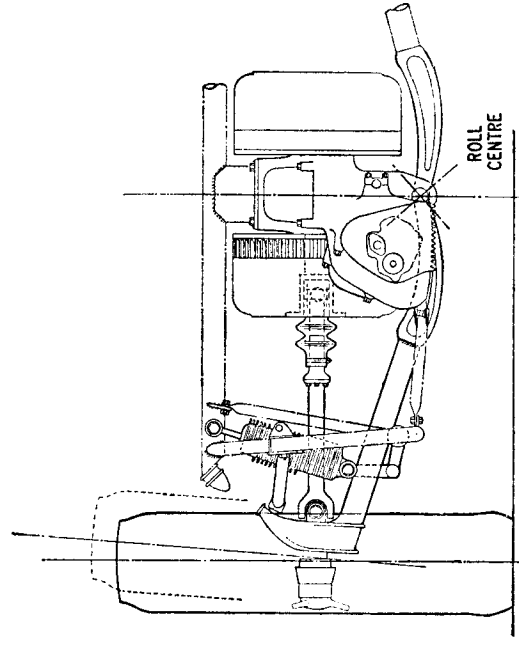
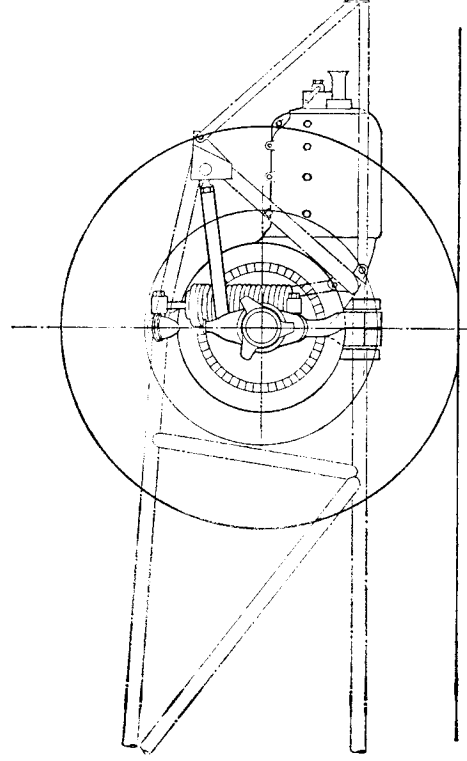
43 Strut-type suspension as applied to a low-built sports car—the Lotus Seventeen. Although one of the simplest forms of suspension, this layout requires considerable vertical space, which puts it at a considerable disadvantage on cars where minimum frontal area is a basic requisite



44 Chapman-strut rear suspension, as used on the Lotus Elite. Lateral location is provided by the fixed-length drive shaft, and fore-and-aft location by the two-piece radius arm. The use of a wide-based outboard mounting reduces toe-in loads to a minimum and gives the Series Two Elite much better straight-running characteristics than the Series One model



45 The simplest form of independent front suspension—the swing axle. The example illustrated (Lotus Mark Eight) consists of a divided Ford beam axle, with each wheel located in a fore-and-aft plane by a leading radius arm. The chief disadvantage of this type of suspension is excessive camber change with roll



46 Low pivot swing axle rear suspension, as used on the Mercedes-Benz W196 (illustrated here) and 300SLR. The transverse links, pivoted beneath the final drive unit, considerably reduce the camber change associated with a basic swing axle layout

elevation and (normally) some form of fore-and-aft linkage. It can be used at either front or rear (in which case the drive shaft forms the swing axle) and is undoubtedly the cheapest form of independent rear suspension—hence its widespread use on low- and medium-priced saloon cars.

This is another simple layout which can be made to work well when it is designed to give good geometry bearing in mind the roll centre limitations of a normal swing axle. For general use its application is limited by the speed of the car and the effect of camber change and gyroscopic precession. When it is used for rear suspension, an additional transverse linkage becomes almost essential to avoid the effects of camber change, and this invariably means that some form of sliding joint must be incorporated in the drive shaft. This, together with the low pivot linkage, immediately increases the complication of the layout and causes it to compare unfavourably with other systems. With a normal swing axle there is far too much camber change for satisfactory handling because the swing arm length is the actual length of the drive shaft. The advantages of the low pivot system are much reduced camber change and a generally better suspension, due to the use of a lower roll centre.

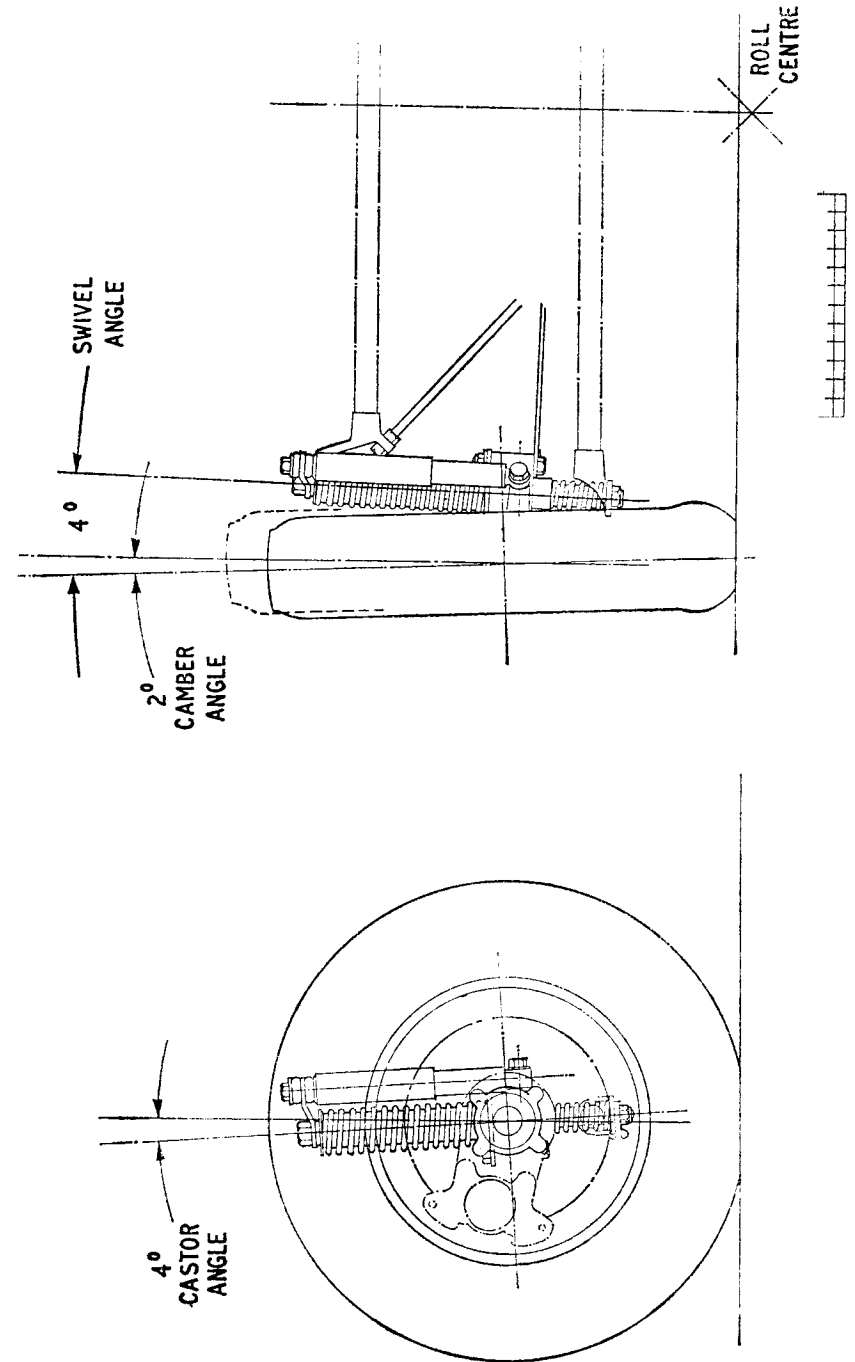
Sliding pillar (See fig. 47)

This type of independent front suspension has been used for many years by both Lancia and Morgan. Basically, each wheel is mounted on a vertical sliding member, which acts on a coil spring and is controlled by a telescopic damper. Geometrically this layout results in the same wheel movement as with twin trailing links or equal length, parallel wishbones. Structurally it has most of the disadvantages of the trailing link system, but it does not involve bending and torsional loads of the same magnitude. It has an advantage over the trailing link system in that loads are taken out in stationary rather than in suspension members. However, with sliding pillars difficulties also arise in connection with the steering geometry. In the light of current developments it is much easier to design a structure to take out suspension loads at the inboard ends of wishbones rather than directly over a wide base, as is the case with the sliding pillar system.

Live axle (See fig. 48)

Working on the assumption that the beam front axle is now completely outmoded, the only application of live axles at the present time is for rear suspension. In view of the complication of the more advanced types of independent rear suspension, and the progressive development of springs and shock absorbers, a live rear axle can still be considered suitable even for reasonably high performance cars, particularly when the ratio of sprung to unsprung weight is fairly high.

High unsprung weight is one of the penalties of this layout, and is further increased by the fact that inboard brakes cannot be used with such a



47 Sliding pillar front suspension, as used on the Morgan, necessitates vertical movement of the wheel on bump and camber change which is directly related to body roll

layout. (With a live axle the only alternative to conventional outboard brakes is some form of transmission brake, and even this is scarcely worthy of mention in this context.)

In recent years there have been numerous developments in connection with live axles. From three-quarter, semi- and quarter-elliptic springs, using the springs as stabilising and locating media, the more advanced manufacturers have now progressed to helical springs with telescopic shock absorbers and methods of positively locating the axle.

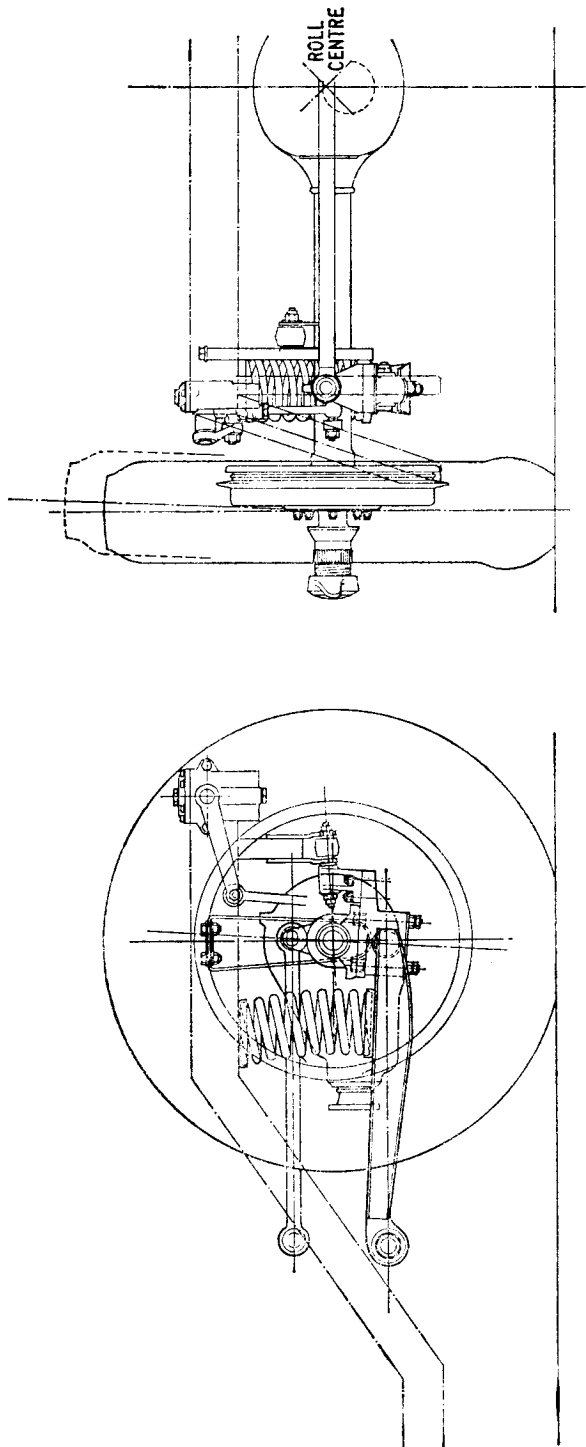
To be entirely satisfactory, even on very smooth surfaces, a live axle transmitting any really considerable amount of power requires positive location in both fore-and-aft and lateral planes. The former purpose is best served by parallel radius arms mounted above and below the axle and running forwards to a chassis bulkhead; leading links can be used, but require either main chassis members well behind the rear wheel centre line, or a special sub-frame, if they are to perform satisfactorily.

There are various means of providing lateral location, and as all have considerable merit the choice of any particular one should depend on its over-all suitability for a specific design. The simplest is a form of transverse link, normally known as a Panhard rod, attached (via bushes) to the axle casing at one side of the car and the chassis at the other. A simple but effective alternative to this is an A-bracket mounted on the differential housing via a ball joint, with diagonal links attached to the chassis at either side. With certain types of chassis it is possible to use a Watts linkage for lateral location. This consists of a centrally pivoted upright member, bolted to the rear of the final drive housing, with transverse links running in diametrically opposite directions from the top and bottom of it to suitable points on the chassis. There is obviously still a considerable future for live axles on light, low-powered cars or on fairly heavy cars, but not on cars with a high power-to-weight ratio.

In the design of a car with a live rear axle, particular care must be taken that no undesirable roll steer effect is built in. Large axle movements must be cut down and small ones controlled to give roll understeer (for instance, by altering the angles of the bottom linkage). It is the tendency towards very low rate springs and large axle movements (the complete reverse of earlier practice) which has led to the increasing use of coil, rather than leaf, springs. In addition, the use of separate locating rods is in itself far better than requiring the leaf spring to act as a stabilising medium, particularly in the lateral plane. One of the chief disadvantages of the live axle is that it cannot provide such a comfortable ride as fully independent suspension, and for this, if for no other reason, it must eventually be superseded.

De Dion (See fig. 49)

This suspension system was evolved at the dawn of motoring and was utilised until the end of the 1950s, due largely to the slow development of



48 A well-located live rear axle—as instanced by the Aston Martin DB Mark III. Fore-and-aft location is by parallel radius arms and lateral movement of the axle is prevented by a Panhard rod

independent rear suspension. In fact independent rear suspension has only just become superior. In the early days of independent—especially wish-bone—suspension it was extremely difficult (in fact, impossible) to keep the outside rear wheel upright, but the de Dion system ensured this. Its chief disadvantage, especially on cars with a high power to weight ratio, is that disturbance of one wheel on bump affects the opposite wheel, which is not the case with independent suspension.

Basically, the de Dion layout consists of an offset tubular axle beam, carrying hub bearing housings and outboard drive shafts, located in a fore-and-aft plane by radius arms and transversely by some means of Panhard rod, Watt's linkage or sliding guide. If inboard brakes are used adequate fore-and-aft location can be provided by single radius arms at either side, but a more satisfactory method is the use of parallel radius arms, mounted above and below the axle tube, which should effectively prevent any "winding up" of this component.

As regards lateral location, the remarks made in connection with live rear axles are generally applicable here. A variety of straight and jointed transverse and diagonal links have been used on sports and racing cars, as also have Watt's linkage systems. Also widely used in this application is the sliding guide, as illustrated in the drawing of the Lister-Jaguar rear suspension (fig. 49).

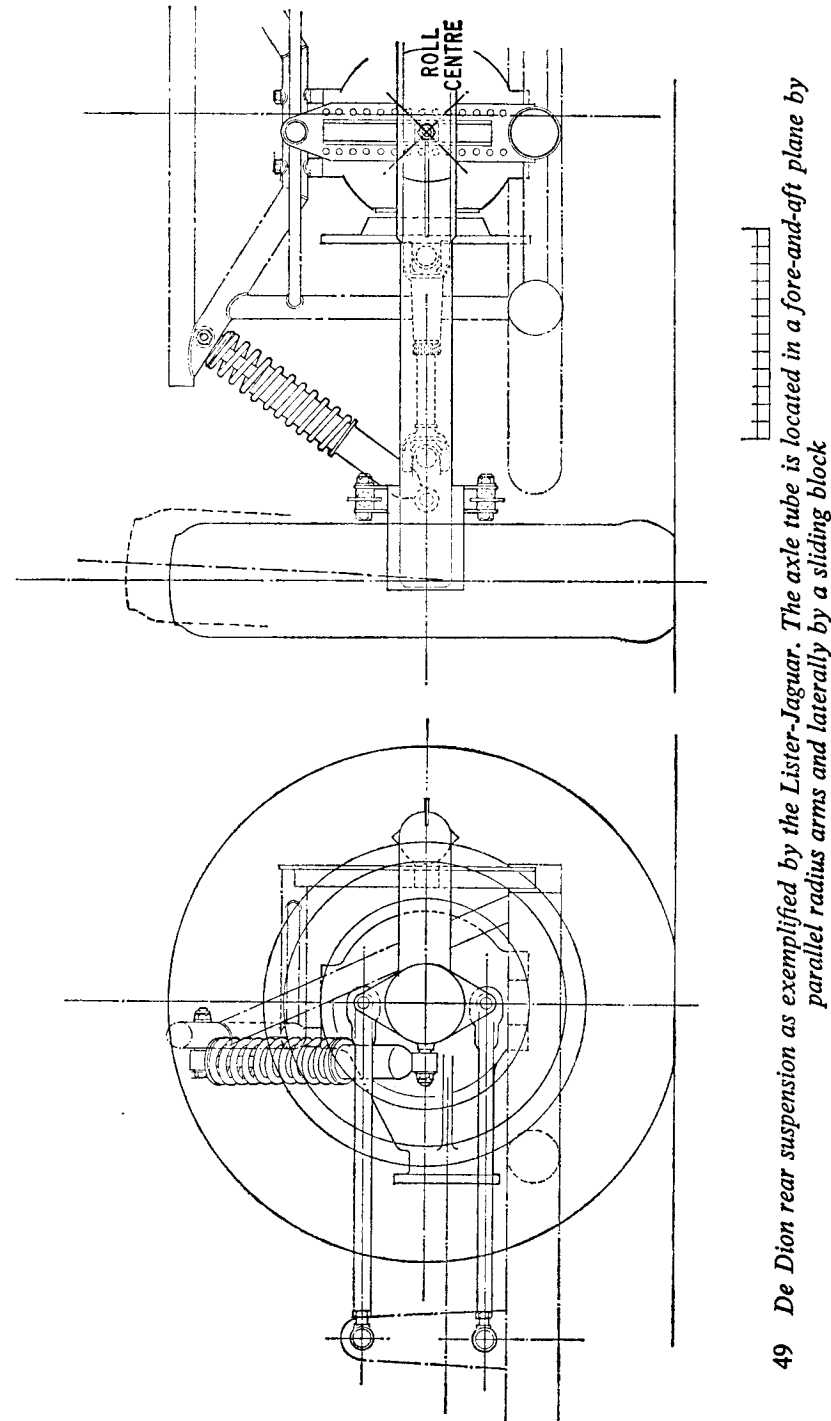
One major advantage of the de Dion layout over the live axle is a considerable reduction of unsprung weight. It also permits the use of inboard brakes, which further decrease unsprung weight and also reduce the loadings on the axle locating linkage by removing the necessity to accommodate braking torque. De Dion suspension also involves the use of a chassis-mounted final drive unit, which—in the case of a front-engined car—eliminates the influence of propeller shaft torque on handling characteristics.

Due to the difficulty of accommodating the de Dion tube, this type of suspension has only rarely been adopted for rear-engined cars. One of the very few examples was the Bugatti type 251, which only raced once—and then most unimpressively. This car had a transversely mounted engine which left room for the de Dion tube behind the transmission. Curiously enough, it also had de Dion front suspension, but its poor handling and straight line instability should be sufficient to discourage anyone from following this precept.

Tyre Pressures

Some remarks have been made earlier on the subject of tyre pressures. At this point, however, it seems appropriate to consider this aspect of road-holding in greater detail.

It is generally accepted that tyre pressures should be increased by some 6 lb above normal for sustained high speeds, and a further, similar increase is normally recommended for racing; the manufacturer's chief concern in



this connection is to avoid temperature rise due to distortion but, as has already been shown, this increase in pressures also increases cornering power. The only exception to this rule is that on light cars with live rear axles it may be preferable to have rather less air in the rear tyres, to control the behaviour of the axle and to reduce the degree of understeer which is often inherent in such designs. Normal racing pressures are in the region of 25–30 lb p.s.i., but on fast circuits the question of heat build-up may make it necessary to use pressures in the region of 40–45 lb, possibly at some cost in terms of ultimate roadholding; inflation to over 60 lb is quite common on the larger, heavier sports cars at Le Mans.

The above pressures apply to dry conditions. In the wet it is normal to decrease pressures by 10–15 per cent, and in some instances to change to a different type of tyre. Such a step should only be taken when there is no chance of the surface drying out, as the performance of “rain” tyres in the dry is vastly inferior to that of the normal racing cover. It is also significant that a new tyre will give a better performance when over-inflated by some 2–3 lb, while with a worn tyre the same degree of under-inflation is desirable.

As has already been mentioned, the pressure differential between front and rear can have an important influence on roadholding, particularly in racing cars. This is the final method of “tuning” handling characteristics to suit a particular driver or a particular circuit, and thus it is hardly surprising that different drivers will choose different pressure differentials, while the same driver may vary his differential from circuit to circuit.

While taking all these factors into consideration, the designer must not lose sight of the fact that good roadholding depends on a good chassis. Even the most advanced suspension layout will not provide really adequate roadholding unless the chassis is torsionally rigid.

DESIGNING A MOTOR CAR

Long before pencil is put to paper, the motor car designer has a fairly clear idea of the projected layout of a new car. Detail considerations at a later stage of design may cause many of his initial ideas to be changed, and he must always be prepared for this. In fact, a flexible mind is one of the essential requirements of a successful designer.

There are many ways of designing a sports car, of which the simplest is probably fitting a proprietary open body on a proprietary saloon car chassis. Such a car is rarely satisfactory, however, and in this context it should be pointed out that even for a road car of modest pretensions a torsionally rigid chassis and advanced independent suspension are very desirable qualities. The twin tube or ladder-type chassis—with independent front suspension and a well-located live rear axle—provides a reasonable compromise between the two, but it seems logical in this final chapter to deal with what is currently considered to be the ultimate type of design.

However experienced the designer, there is a basic sequence in which he goes about his task. Before any detail design work is begun it is necessary to establish the over-all specification of the car. It is also important at an early stage to determine the weight range of the car, which has a considerable bearing both on the structure and on wheel and tyre sizes. This is equally true of a single-seater racing car or a two-seater road/sports car, although in the latter case a much wider range of weights must be considered.

The choice of wheel and tyre sizes fixes the hub height. The next step is to decide upon the preliminary wheelbase and track measurements. In both of the above cases these are likely to be in the region of 7 feet 6 inches and 4 feet respectively, although it may be decided to increase the rear track somewhat in line with current trends. Assuming that it is decided to adhere to current practice in suspension layout, wishbone type suspension will be used at both front and rear, in conjunction with roll centre heights in the region of three inches above the ground at the front and four inches at the rear. These figures are purely arbitrary, but offer a reasonable compromise between the conflicting properties of high and low roll centres and are similar to those of a number of successful cars. And design cannot start until they have been decided.

Before any attempt to work out the suspension geometry can be made, it is necessary to decide on the type of front and rear suspension uprights

which are to be used. In the case of the front, by far the simplest method is to use the proprietary Standard-Triumph upright found on the majority of British sports/racing and racing cars. For the rear, a special casting or fabrication will probably be necessary.

All work at this stage is still in the nature of preliminary investigation rather than detail design. It is, however, possible to begin to commit certain features of the design to paper. For this several drawings will be necessary. The minimum number is four; these are: front elevation of the front suspension; front elevation of the rear suspension; side elevation of the whole car; and plan view of the chassis. For convenience these will most likely be scale drawings, in which case the most practical size will probably be fifth scale. However, the car has to be built from these drawings, so it is imperative to scale them extremely accurately. All dimensions are taken from the drawings, and a small fault becomes exaggerated by as much as five times in the car itself. It is, of course, a bad practice to use scale drawings, but normally this is the only way in which the job can conveniently be done unless full drawing office facilities are available.

The first step is to draw in the front and rear roll centre heights and the positions of front and rear uprights. From these an idea of the layout of the suspension linkages can be obtained. It is also necessary at this stage to decide on the desired amount of wheel travel and suspension unit travel—factors which in turn control the limits of roll centre movement—sketching in the wishbone linkages to give some idea of the pick-up points for front and rear suspension. The next step is to draw in the front and rear axle centre lines, the ground line and the undertray line (to give at least an inch of ground clearance on maximum bump—more on road cars).

While deciding on the basic layout of the car, it is necessary to consider the location of such items as fuel tanks, pedals, gear lever, pipes and cables, in addition to the major components—engine, transmission, radiator, seats. The difficulty comes when two or more of these need to be in the same place.

By this time it is necessary to decide upon the type of chassis to be used and, if current practice is to be followed, a space frame constructed of steel tubes will almost certainly be specified. Even if it is decided to use unitary construction, the method by which the layout of the major structural members is determined is almost identical. The design of the chassis itself starts with the major transverse members or bulkheads, of which one is found at or near the front axle line, one at the rear axle line and two in between. The actual position of the central bulkheads largely depends on whether the car is front- or rear-engined. And if wishbone-type front suspension is used the main bulkhead must be supplemented by a subsidiary bulkhead. The same is true at the rear if narrow base wishbones are used.

The decision as to whether the main front bulkhead is positioned ahead

of or behind the axle centre line depends on two conflicting factors. Ideally it is desirable to take out the main suspension loads through the nearest bulkhead to the axle centre line, but this member must also serve as a mounting for the steering gear and—in the case of a rear-engined car—for pedals and brake and clutch master cylinders.

Although the front and rear bulkheads serve the same basic purposes of location on both front- and rear-engined cars, the functions of the intermediate bulkheads differ on the two types. On a front-engined car the front bay is occupied by the power unit, the central bay by the occupants' legs and the rear bay by the seat or seats. On a rear-engined car (assuming the engine is mounted ahead of the rear wheel centre line) feet and legs are in the front bay, the seats are in the centre bay and the rear bay is occupied by the engine and/or transmission. Even in a rear-engined design it is desirable to bring the driver back as far as possible—bearing in mind the question of weight distribution for ultimate roadholding—if only to give him a better "feel" of the car.

Whatever the layout, the next step is to design a member to support such items as the steering wheel and instruments, and to form a bulkhead between front and centre, or centre and rear, bays. As the occupants' legs pass through this bulkhead it is necessary to incorporate some form of triangulation other than a simple diagonal. The final bulkhead is the one found at the rear of the power unit on a front-engined car or at the front of it on a rear-engined car. With both of these bulkheads it is important to decide upon the angle of layout which will give the greatest advantage from both structural and practical points of view. Whatever the layout, the front mountings for the power unit are generally located on the side frames, and this may involve the use of long mounting bars or beams on a two-seater. The question of rear mountings depends largely on transmission layouts, but when the engine is in unit with the gearbox or clutch bell-housing it is normally possible to arrange a central rear mounting either above or below this unit.

Structurally, the engine and the occupants are by far the most difficult items to accommodate in a car with a space frame chassis, as they effectively prevent the triangulation of bulkheads through which they pass. It is possible, if solid engine mountings are used, to utilise the power unit as a structural member, but in general this is not good practice, for the reasons already stated. In the case of a rear-engined car, where the weight of the engine and gearbox/final drive assembly is supported by the rear bulkhead, it is important for the load to be taken out as directly as possible through the rear suspension springs. Thus the aim should be to design mountings which pick up near the rear axle centre line and react the major suspension loads vertically through this point. This, in fact, is the chief feature controlling the location of the rear bulkhead. As mentioned above, with narrow base wishbones a secondary bulkhead will be required somewhat further forward, but if wide base wishbones are used, consisting of

transverse links and long trailing radius arms, then the latter can pick up on the next main structural bulkhead.

Thus far no reference has been made to longitudinal chassis members. Before these are considered, however, it is essential to deal with all the major transverse load bearing areas. Since one of the primary purposes of the chassis is to take out bending loads, all the transverse bulkheads must be part of a structure which is strong enough to support itself with negligible deflection in bending. This should, in fact, cause little difficulty. The criterion of successful chassis design is to combine this with adequate torsional strength. Basically, tubes providing bending stiffness must also be placed in positions to provide the maximum torsional stiffness. In this connection it should also be remembered that, the larger the chassis section, the stiffer the chassis. The ultimate in this connection is a fully triangulated square structure. Since in front elevation such a layout conflicts with the provision of the lowest possible frontal area, a compromise is once again required to obtain the best results in all circumstances.

While working out the basic layout of all the major components, the designer must also take into consideration such items as the position of the driver's seat, knees and feet, the sight line—particularly in the case of a sports/racing or racing car—the steering wheel position and the general body shape. It must be emphasised, however, that all these are still liable to change when detail design is commenced. Nevertheless, by now the designer should have a reasonable idea of the over-all layout of his chassis as a series of bulkheads with longitudinal members giving support to them and taking support from them. He should also have finalised the shape of the body as a series of transverse and longitudinal sections, into which all the components of the car will fit without any need for external alterations.

At this stage it is possible to start on the actual design of the car in detail. There are so many variables to be considered, particularly in suspension design—which controls chassis layout more than any other single factor—that a number of arbitrary decisions must be made in order to get the work started. Of these, the most important is to establish the front and rear roll centre heights. As has already been suggested, these should be in the region of three and four inches respectively.

The length of suspension linkages is decided by the need to restrict roll centre movement, and the width of the chassis in front elevation—decided by the need to obtain the best possible proportion from the stiffness point of view—controls the limits within which the suspension links can be picked up. A great deal of compromise may be necessary and several different schemes may be drawn before an acceptable suspension layout is obtained.

To obtain the desired wheel movement it is normally necessary to have the top wishbone shorter than the bottom one. While plotting the front suspension it is necessary to draw the wheel at the limits of its travel and

work out the bump and droop roll centres and the consequent camber change. If necessary the wishbone pivots must be moved to control this. Another arbitrary decision which must be taken at this point is to fix the maximum desirable roll angle, and from this the wheel inclination at maximum roll can be established.

A further arbitrary decision to be reached before the suspension geometry can be drawn in is the virtual swing arm length. This is the length of linkage which would be required to produce the desired wheel movements if a simple swing axle was used. With current wishbone layouts (excluding parallel, equal length wishbones, which have no swing axle effect) the virtual swing arm length varies between one and a half and three times the track. A good compromise is for a virtual swing arm length of twice the track; thus a four foot track gives a virtual swing arm length of 8 feet.

Now the work can be begun in earnest. Roll centre height and virtual swing arm length have been decided. The positions of the outboard wishbone pivots have been determined by the type of upright which has been selected. The wheels have been drawn in at the required camber angle—the latter governed by the need to counteract the effects of body roll.

Front Suspension

The first step in establishing the front suspension geometry is to draw a line from the roll centre through the tyre contact patch, produced to the virtual swing arm length. Lines drawn back from this point, through the outboard wishbone pivots, give the layout of the suspension linkages in front elevation. The next step is to decide suitable inboard pick-ups for the wishbones. Having decided on suitable pivots, bearing in mind the relation between the lengths of top and bottom wishbones, it is possible to plot the roll centre height change on bump and droop. As far as the roll centre only is concerned, the ideal is to keep its movement to a minimum, but a desirable feature which conflicts with this is accentuation of the camber change normally produced by a swing axle of the same virtual length. In practice it is advisable to sacrifice some roll centre movement in favour of increased camber change—the latter to offset camber change due to roll. From normal ride level upwards, the camber change should be positive, continued as negative camber change on droop.

There are many other factors to be considered in the design of suspension linkages, but it would be outside the scope of this chapter to consider them all here, particularly as all the necessary calculations for determining suspension characteristics can be found in Appendix III. However, a simple instance should serve to illustrate the importance of this subject.

Because of the unequal angular movements of unequal length wishbones, the distance between the centre of gravity and the roll centre varies, and with this there follows variation in roll weight transfer. Basically, weight transfer decreases as the roll centre rises. It increases on bump, and this will

normally lead to a momentary increase of slip angles—which could cause the car to twitch sideways or even go completely out of control.

Having established the length and the position of the front wishbones it is possible to decide on the location of pick-ups on the chassis. Here there should be no compromise, and all loads should be fed into the structure with the minimum of offsets. For this purpose the chassis rails can be moved up or down and in or out to obtain the most desirable position. As has been mentioned above, however, it is important from a structural viewpoint to avoid the chassis becoming either too wide or too narrow in front elevation. On a single-seater it is also important—from a purely practical angle—to leave adequate room for the driver's feet, knees, shoulders and elbows.

Once the wishbone pivots have been decided it should be fairly easy to fix the shape of the chassis in this area, although there are numerous other factors to be taken into consideration. These include mountings for the suspension units, steering gear, radiator and pedals (in the case of a rear-engined car) or engine (in the case of a front-engined car). All this requires careful design, aimed at accommodating all the various mountings to the best advantage from the structural point of view. An alternative—but one which is not to be recommended—is to decide the location of all the above items in space, design a chassis which goes fairly near to these points and build brackets out from the nearest frame member.

Before determining the location of the wishbone pick-ups in plan it is necessary to decide on the type of mounting and bearings to be used (rubber, uniball joint, ball races, plain bearings, etc.). The four pick-ups can be positioned almost anywhere along a fore-and-aft line if the wishbones are of conventional design. If possible, however, the simplest layout is to have one arm of the top wishbone pivoted alongside the top eye of the suspension unit (if this type of springing is to be used) so that one bolt and bracket can do two jobs. The ideal point for this mounting is at the top corner of the front bulkhead. The use in this connection of a straight-forward transverse link has the advantage that almost all lateral loads will be taken out into this bulkhead, leaving only braking loads to be absorbed by the secondary bulkhead, in the form of tension and compression in the top and bottom members. As a result, the secondary bulkhead can be made considerably lighter than if symmetrical wishbones were used, and will not necessarily require full triangulation.

In practice it is not always possible to put the inboard pick-ups on a direct fore-and-aft axis. Because of the need to accommodate wheel lock, the tendency is for the axis to be in towards the front, which gives the suspension a specified lead angle governed by the amount of chassis taper. This must be borne in mind during design but need not have any adverse effect. It is, however, desirable to have the axes of the front and rear pick-ups on the same straight line; otherwise special bearings will be needed to accommodate the movement of the wishbone ends.

Treatment similar to the above can be adopted for the bottom wishbone. This has to provide an outboard anchorage for the suspension unit, which can be attached in a variety of ways—as can be seen in the paddock at any race meeting. The ideal method of mounting, however, is to use a cross tube or plate welded as near as possible to the apex of the wishbone at the outboard end. The chief essential is to design a wishbone having adequate stiffness against lozenging. As suspension loads are reacted through the bearing at the bottom of the upright, a primary aim should be a minimum offset between the axes of the suspension unit and the bearing. The ideal would be to make a special bottom knuckle joint having a suspension unit mounting incorporated in line with the bearing, but this would be extremely expensive to manufacture. In addition, owing to the need to provide a reasonable amount of suspension travel, it might not be possible to fit the suspension unit into the available space.

Steering gear can be of various types, but rack and pinion will normally be adopted as being the simplest, the easiest to fit to the chassis and to adjust for wear, and among the most efficient in use. As was the case with the suspension layout, the design of steering linkages is dependent on the steering geometry. The chief essentials here are to cut down toe-in and toe-out on bump and droop and to provide slight anti-Ackermann effects. The Ackermann principle states that for any given corner the outside wheel should have less lock because it is running at a larger radius than the inside wheel. On many current high-performance cars this principle is reversed, so that for any given degree of turn the outside wheel has more lock than the inside. Owing to weight transfer the outside wheel always runs at a higher loading than the inside wheel, and therefore at larger slip angles, which necessitate greater lock. During design the steering geometry must be considered in conjunction with suspension geometry, and experiments with various amounts of lock on bump and rebound may be necessary before the correct geometry can be obtained. Having established the correct amount of anti-Ackermann effect and freedom from toe-in, the designer can fix the outboard and inboard ball joints of the track rods, which in turn fix the ball joints on the end of the rack and position the centre line of the rack both fore-and-aft and vertically.

The position in height of the inboard pick-ups is governed by suspension geometry in front elevation—modified slightly by the castor angle in side elevation, which can be fixed after practical experiment to get the desired amount of steering feed-back. Some castor is certainly necessary, and as the limits on most designs are between 3 degrees and 8 degrees, a mean of 5 degrees would seem to be a reasonable compromise.

Rear Suspension

The design of any type of rear suspension starts with the hub height fixed and the wheels drawn in at the correct track and camber angle in

relation to the centre line of the car and the ground line. Assuming that independent suspension of an advanced double wishbone type is to be used, the designer is presented with even more variables than with the front suspension in that no obvious proprietary upright is available—leaving him at liberty to design his own. The outboard pick-ups could be close together or wide apart, symmetrical or asymmetrical—the only controlling factor is the diameter of the wheel. However, placing the pick-ups controlling toe-in close together increases tension and compression loads. As the basis for high-performance chassis design is to keep all loads low, it would appear that the wishbone pick-ups should be as wide apart as possible. The ultimate from this point of view would be to have one at the top of the rim and one at the bottom, but this would involve the disadvantage of a relatively heavy upright. In practice, an upright of more than half the wheel diameter adds unjustifiable weight and lowers suspension loads so far as to make it difficult to design a suspension for them. Thus the practical ultimate is an upright of approximately half the wheel diameter. The ideal in this case is to use a light-alloy casting, but a fabrication—which obviates the cost of making castings—can be quite adequate for small production. The chief snags in this case are increased weight and liability to distortion; in addition, difficulties are sometimes experienced during machining. Holes for pins to locate wishbones must be drilled extremely carefully and extra material must be built in to allow for distortion.

There can be no doubt that the best rear suspension in current use is of the double wishbone type. Various configurations are possible, however. With the narrow-base type of wishbone, toe-in and toe-out loads are taken out into the chassis merely over the wishbone base. In the case of the wide-base type of wishbone, radius rods take out the fore-and-aft loads and transverse links cater for lateral loads—one of the latter also being triangular and dealing with toe-in and toe-out loads. The use of pick-ups which are wide apart on a broad base also decreases the magnitude of the loads involved. It is generally agreed that this is the best type of suspension to use from a performance point of view, but it has the disadvantage that it is space-consuming and not entirely practical for a front-engined road sports car. However, for a rear-engined car with the engine mounted ahead of the wheel centre line it has much to recommend it.

The first step in designing a suspension of this type is to decide on the layout of the upright. The general trend is for one pick-up to be placed fairly low down in the region of the wheel rim (a rather higher pick-up would be desirable in the case of a road car) and the other at or above the axle centre line. Scrutineers and other race officials have tried to bar this type of suspension on engineering grounds, claiming that the upright will touch the road if the tyre deflates and will dig in and turn the car over if a wheel should come off. Both ideas are completely erroneous. If a tyre deflates slowly the driver soon becomes aware of the effect on the handling of the car. And with a burst tyre or a lost wheel—both extremely undesi-

able—the equilibrium of this type of car is likely to change less than that of a car with conventional suspension. The reasons for taking the outboard wishbone pick-ups so low are concerned partly with geometry and partly with engineering. To obtain a four-inch-high roll centre, the top wishbone must be inclined upwards slightly from the outboard end, and the bottom wishbone considerably so. Thus the chassis base over which the loads are reacted is somewhat shorter than the uprights into which they are fed at the outboard end. A relatively deep outboard casting is therefore essential to give an adequate inboard base from the point of view of load dispersal. Compressive loads in the region of four to five hundred pounds may be developed in the bottom wishbone, but the loads in the top wishbone can be extremely low.

The design of the upright itself is fairly straightforward. The loads involved are bending in both vertical planes, and torsion from toe-in and toe-out. The best design to meet these conflicting requirements would be a tube capable of taking all torque loads down to the bottom wishbone pick-ups. To disperse these loads over a wide base the best means is to design a casting or fabrication which is round at the top and flattened towards the bottom. The design of the hub and outboard drive shafts can follow general engineering practice.

Having decided in this way the location of the outboard pivots, it is possible to draw in the suspension geometry, as was the case with the front suspension. Lines drawn back through these pivots from the point at which a line from the roll centre through the tyre contact patch meets the line of virtual swing arm length give the layout of the wishbone in front elevation.

In the design of this type of rear suspension a large number of other factors have to be taken into consideration. A drawing showing the front elevation of the rear suspension should have the final drive unit, drive shaft and brakes drawn in. One of the most important points is to minimise transmission power losses. It is important that wishbones, drive shafts and universal joints should run at small angles at all times and that the friction of sliding splines should be reduced to a minimum, or even eliminated completely, by the use of fixed length drive shafts. In the ideal suspension, however, it may be considered preferable to avoid using the drive shaft as a transverse link, this function being carried out by a member running just above or ahead of it.

It should be possible to position the inboard wishbone pick-ups to give the desired geometry, although small concessions may be necessary to obtain an ideal chassis structure. However, as the chassis is performing few other functions in this area it should not be difficult to arrange for both ideals to be accommodated without conflict. Acceptable limits of compromise in this case would be one bracket mounted slightly inboard and another outboard.

The next step is to plot the movement of the wheel at full bump and

droop, allowing something in the region of $3\frac{1}{2}$ inches bump and $2\frac{1}{2}$ inches droop. These figures could be slightly higher on a road car and rather less on a single-seater racing car.

With this type of suspension the ideal is to take the major lateral loadings out on the same plane as the drive shafts and to carry the fore-and-aft loads well forward. The latter feature is the one which conflicts most with the idea of using this type of suspension in a front-engined road sports car. At best the radius arms would reduce the useful width of the car and also interfere with door openings. In such cases it would probably be more practical to use the narrow base wishbone currently favoured by Cooper and Ferrari. However, by requiring the use of a secondary bulkhead this type of suspension immediately increases chassis complexity, one of the chief problems being to triangulate the secondary bulkhead. Simple diagonals can rarely be used because the engine or transmission intervenes. Furthermore, with each wishbone link taking both transverse and fore-and-aft loads, the complexity of loadings is considerably increased.

Another important factor is the suspension unit and the provision of suitable anchorages for it. Here again it is advantageous if the suspension unit and the transverse wishbone member can pick up on the same axis. However, the suspension unit must be positioned either ahead of or behind the drive shaft, and mounted outboard at one or other of the bottom pivots of the upright.

By this time the designer will have realised that many of his original ideas may require drastic modification. It may even be necessary to complete several design studies before a satisfactory layout is achieved. With increased experience the designer is able to foresee difficulties in advance and to allow for alterations which do not reduce the efficiency of any part of the car.

Once the suspension and engine mountings have been established it should be relatively easy to accommodate all the other components, provided these have been borne in mind throughout the detail design of the car. However, should any of these provide a major snag it may be necessary to do a complete re-design of the whole section concerned. Having decided on the layout of the chassis in the areas of the front and rear suspension it should be possible to fill in the gaps without too much difficulty. Most of this work can be done on a side elevation drawing on which the general body outline is drawn in, together with radiator, front and rear suspension, engine, clutch, transmission, driving position, steering wheel, gear-lever, instruments, fuel and oil tanks, etc. For a racing car, it is important to know the amount of fuel which will be required for given events in order that adequate tankage can be provided in the most advantageous places from the point of view of weight distribution. In a racing car careful consideration must also be given to the position of the windscreen and the driver's sight line, as it is important for the driver to be able to see

over the windscreen without receiving the full blast of the air in his face.

As shown earlier in the analysis of the 1960 Formula One Lotus, finding space for all the pipes and wires on a rear-engined car can be a major problem. This is particularly true when the transmission, as well as the engine, is fitted with a front-mounted oil radiator. Ideally the aim should be to keep hot and cold pipes separate. On a car of the space-frame type it is possible to use chassis tubes as oil and water pipes, but this practice cannot be considered desirable on a road car for everyday use.

Another extremely important point is the angle of the bulkhead in front of the driver. This can be either vertical or inclined forwards, and in the latter case it usually inclines at the angle of the steering wheel. Here again, a factor which may outwardly appear relatively unimportant must be taken into consideration. This is the question of gear lever movement. As rearward travel of the gear lever is limited by the driver's elbow movement it is essential to leave adequate space for forward travel.

Such practical problems are far more important in the design of a car than the mathematical formulae by which the theoretician may arrive at the perfect chassis. It is equally important to ensure that in each bay there is sufficient room for all the components which are mounted there—and that they can be serviced or changed in the minimum time. Nevertheless, the best car will be the one in which all these considerations have been taken into account without detriment to the chassis as a complete structure. In addition, any sub-frame which is built on to the chassis to support such items as radiator, fuel tank or body, must be designed as a complete structure in itself.

The front bulkhead is normally fairly easy to triangulate, but this is not so in the case of the bulkhead through which the driver's legs pass. Such items put to the test the ingenuity of the designer and have led to the development of the pierced hoop type of bulkhead. In addition to being structurally rigid, this member also supports the steering column, the instruments, the gear lever, the handbrake mechanism and the front of the driver's seat. In the case of a rear-engined car the bulkhead immediately behind the driver accommodates the main loads from the front of the engine and the fore-and-aft loads from the rear suspension. It also supports the rear of the seat—and thus the bulk of the driver's weight.

To ensure that the chassis is a complete structure, all bays should be structures in themselves and all frames should be triangulated, but this is extremely difficult. However, with pierced bulkheads or dog-leg diagonals it is possible to get round fairly small obstructions and it is also possible to triangulate a frame externally, as shown in fig. 14. The ultimate is that all six sides of every bay should be fully triangulated, even if this can only be achieved by liberal use of hoop sections. Even with straightforward triangulation, however, it is important to bear in mind the advantages offered by alternative diagonals and to put these to the best possible use,

especially where they may be used for mounting major components. A good example of this can be seen in the location of the front mountings for the engine of the Lotus Nineteen.

Mounting Brackets

General

Whatever the component, and however light its weight, it is important to avoid mounting it in the middle third of a chassis tube without careful consideration of the structural implications involved. Ideally all components should be mounted at the meeting point of three tubes; this is particularly important in the case of major items such as the engine. In this latter case, as in all others, one of the primary considerations is cost. The first essential is to obtain a good structure, but normally it is equally important to obtain a good structure cheaply.

Methods of mounting suspension

As a rule, the most commonly used methods of mounting suspension represent a compromise between good design, expense and fitness for purpose. For instance, although it is bad practice to pick up a wishbone in the middle of a tube—thus automatically putting bending loads into the tube—circumstances sometimes dictate such a move and cause deviations from what could be considered ideal design. On many current sports and racing cars it is common practice to pick up the lower front wishbones part of the way up the vertical members of the front bulkhead. These members could be specially designed, to accommodate tensile and compressive loads, as well as the bending loads involved in such a layout, but they would probably cost far more than a tube strong enough to do the job. In addition, many components are mounted in such a way that they impose offset loads, since they need to pick up some distance away from the chassis tubes concerned.

There are several methods of mounting suspension units, all of which have a variety of advantages and disadvantages from the structural, practical and cost points of view. One of the best methods is the use of a short length of $\frac{3}{4}$ inch 10 gauge tube welded on to the top longitudinal frame member in the region of the front bulkhead. This can locate both the suspension unit and a wishbone link by means of a single bolt, and has the structural advantage that loads are quickly taken out into other chassis members in tension and compression. It is also extremely cheap, very effective and quite acceptable from an engineering standpoint; for over-all practicability, in fact, it is probably the best method which can be used in this context. Much more common are the sheet-metal brackets used by Cooper, Lola and Tojeiro to mention but a few. In the case of the Cooper the loads are normally taken out in bending in the nearest tube. The Lola method is to distribute torsional loads into the vertical member of the bulk-

head concerned as bending loads. In certain designs the brackets mounting the suspension units are extended in the form of hooks, but this practice must be condemned as bad engineering, as it is outside the normal limits of acceptability due to the very considerable offset loads involved.

Wishbones—particularly lower wishbones—are quite often mounted on tubes welded into upright members. The mounting tube is then drilled and tapped in order that a bolt may be screwed in to tighten up the wishbone. The possibility of damage to these threads is the chief weakness of this type of design. A further method of mounting at the junction of three or more tubes is to use a sheet-metal plate welded across the apex of the triangle. In this case, however, bending loads in the tubes are unavoidable.

Engine Mountings

In the majority of cases the engine can be supported by three basic mountings, two forward and at the side and one at the rear. These three mountings have to support the engine vertically and must also supply adequate fore-and-aft location to take out torque reaction and lateral location to prevent movement in cornering. In addition, when the drive-shaft is used as a suspension member in a rear-engined car, suspension loads are transmitted into the engine through the final drive housing. It is possible for the engine to be used as a chassis member, but the high-frequency vibration which usually results from such practice can have adverse effects on carburation, and indeed on most aspects of engine and transmission performance. In addition, it sometimes occurs that slight chassis distortion leads to a bad fit, which in turn causes pre-stressing when the mountings are tightened up. Too much flexibility is an equally bad thing, especially in a car intended for racing, and the most acceptable compromise seems to be the use of a relatively small amount of rubber.

Of the three engine mountings, that at the rear—on the clutch bell-housing in the case of a front-engined car or the gearbox in the case of a rear-engined car—is always the more difficult to design. The front mountings can normally be accommodated fairly easily on a side-frame diagonal supported by a further short diagonal at the most suitable mounting point. It is important that the side-frame members are capable of supporting any sideways engine movement as well as torque reaction. As the mountings are normally at least 20 inches apart the loads involved can be of a fairly low order. On a sports car it is quite possible to use what would appear to be unduly long members without any adverse effects. The rear mounting can either be hung from the top of the relevant bulkhead or supported at its base.

Steering Gear Mountings

Another component whose mountings are extremely important is the steering gear—rack and pinion in most cases. Rack and pinion mountings

are always awkward because the loads transmitted on the central axis of the rack almost invariably have to be reacted into the chassis at some distance offset. When proprietary steering gear is used the normal methods are either to build brackets with flat surfaces on to the chassis structure and mount the rack on these, or to weld or braze tubes through chassis members and bolt the brackets to them. The ideal method, of course, is to design special steering gear, making the mountings an integral part of the casing. Rack and pinion loads can be of a fairly high order, but their worst aspect is that they are reversible and of a very high frequency, which leads to the possibility of fatigue failures in mountings or attachments. A satisfactory alternative is to locate the rack on two castings and bolt the latter to thick-walled chassis tubes capable of dealing with all possible loads.

Body Mountings

Body mountings constitute a major problem on both sports and racing cars. For maximum accessibility the ideal is to hinge the main sections of the body at front and rear, but this method has the disadvantage that the body cannot be made particularly rigid. For road-going cars it may be necessary to build a subsidiary framework into the body itself—be it of aluminium or glass-fibre—and hinge only the engine access panel and the boot lid. Bodywork can be made to stay on racing cars at the highest speeds with the use of simple hinges and catches and Dzus fasteners, but such fastenings are liable to develop too many creaks and rattles for use on road-going vehicles. The conflict is between rigidity on the one hand and lightness and accessibility on the other, and the constructor must make the compromise which most nearly suits his requirements.

Exhaust Pipe Mountings

Even in the fairly early stages of design it is necessary to give thought to the mounting of relatively small items such as coil, fuel pump, voltage control and exhaust pipe. All of these can give trouble unless they are carefully designed and, as mentioned above, they should never be mounted in the middle third of a chassis member unless it is absolutely impossible to locate them elsewhere. The exhaust pipe frequently causes difficulty because, although its mounting must be balanced both with engine vibration and with any engine movement caused by flexible engine mountings, it is normally connected rigidly to the engine. This means that provision must be made for mounting the silencer and tailpipe in such a way as to avoid any torsional loadings. On racing cars a good means of supporting the tailpipe has proved to be a semicircular bracket, with a spring stretched around the other half of the exhaust pipe. This allows the exhaust pipe to move but restrains it from vibrating at its natural frequency when excited by engine vibrations.

Engines and Transmissions

In this survey no attempt will be made to design engines or transmissions. However, it is essential to decide on the type and position of the power unit very early in the draft design stage. Because of the many considerations involved, of which the chief are cost and the use to which the car is to be put, it is virtually impossible even to give guidance on the type or make of engine to employ. Once again, however, current practice should provide a useful guide.

Much the same may be said of the transmission, but in a front-engined car the gearbox must be suitable either for bolting directly on to the engine or for mounting in unit with the final drive assembly. The use of a composite gearbox/final drive unit is generally considered to be preferable, particularly in a front-engined sports car, where such a layout offers the great advantage of extra foot room together with improved weight distribution by putting more weight on the rear wheels.

In the case of a rear-engined car with rear-wheel drive, it is normal for the complete transmission to be mounted in unit with the engine. Several proprietary units are available for this purpose but their detail layout is governed by the design of the car for which the gearbox is made. For instance, several rear-engined passenger cars have the engine mounted behind the axle centre line and the gearbox ahead of it. In a rear-engined sports or Formula Junior car it is generally preferable to mount the engine ahead of the rear wheels and the gearbox behind them, in which case it may be necessary to turn the transmission upside down to obtain the correct layout. In this connection it is important to remember the influence on design of the height of the engine and gearbox in conjunction with the suspension layout. An important question in this context is the distance between the gearbox shafts. It is important to avoid angularity in drive shafts, and for this reason the final drive unit must be mounted at much the same height, relative to the wheels centre line, as in the car from which it was taken. It is also important to remember that the rear suspension layout controls the height of the sump. The foregoing assumes the use of the conventional type of gearbox with manual change. Automatic transmission, apart from being power-consuming, is not generally considered sporting!

A final consideration which must not be overlooked in connection with gearboxes mounted in unit with the final drive assembly is the provision of a satisfactory gearchange linkage. Once again space is an important factor.

Brakes

In braking the basic choice is between drum and disc brakes and inboard and outboard mountings. Despite the current trend towards disc brakes on high-performance cars, drum brakes can be quite adequate for light- and medium-weight cars—as instanced by the current performance of Formula

Junior machines. The chief advantage of the disc brake lies in its anti-fade characteristics. Discs also offer improved heat dissipation and reduced wear. There is not much to choose between drum and disc brakes as far as weight is concerned, and in fact lightweight drum brakes with drilled back-plates may well have the advantage on this score. However, to obtain anti-fade performance comparable with that of disc brakes, drum brakes need extremely hard linings, which require high pedal pressures and may also lead to drum distortion. Racing linings of the sintered metal type normally obtain brake shoe life at the expense of drum life.

In general terms, then, the disc brake is to be preferred. If drums must be used they should be made of light alloy to reduce weight—this is particularly important in the case of outboard mounting—and finned to improve heat dissipation. From the latter point of view, diagonal finning would appear to be ideal, but in practice it often results in cracking due to uneven cooling. It is also possible to use peripheral fins, but hot air is less easily scoured away from the drums in this layout. On the whole, the best plan would appear to be to use simple axial finning.

A variety of lining materials is available for both disc and drum brakes, ranging from very soft material which gives adequate braking with light pedal pressure and beds in very quickly but does not have good anti-fade or wear characteristics, to the high-performance type lining which requires much greater pedal effort but provides consistent, reliable braking from racing speeds. A further factor in favour of disc brakes for racing is that pads can be changed much more quickly than brake shoes.

One of the disadvantages of disc brakes as currently used is that slack bearings allow the disc to move axially and push the pads away from it. This phenomenon, commonly known as “knock-off”, can be avoided by careful attention to bearings. (Some systems incorporate devices to minimise knock-off.)

Whatever braking system is used, the distribution of braking effort between the front and rear wheels is extremely important. Largely because of weight transfer under braking, the ideal ratios are in the region of 60 per cent front, 40 per cent rear on a front-engined car and 55:45 on a rear-engined car. Ultimately this depends on such factors as the height of the centre of gravity, weight distribution, suspension layout, and tyre characteristics, but the above figures give a general guide and show that this question must be borne in mind from the early stages of design. Apart from merely fitting larger brakes at the front than at the rear, it is desirable to have a means of altering the braking ratio to obtain the ideal characteristics for any given conditions. It is possible to achieve this simply by altering the size of wheel cylinders, but the ideal method is to use twin master cylinders in conjunction with an adjuster rod which controls the distribution of braking effort.

Regarding the actual size of brake drums or discs, it can only be said that these should be designed to have a capacity in excess of that envisaged

in normal use. However, the question of increased weight militates against making brakes unnecessarily large, especially where they are mounted outboard. Once again a very fine balance is required between conflicting factors.

The choice between inboard and outboard brakes is a relatively simple one. Inboard rear brakes reduce unsprung weight but are liable to overheating and to oil leaks from the transmission, whereas outboard brakes run coolly at the expense of increased unsprung weight. The use of inboard front brakes involves the complications of drive shafts and increased over-all weight, and can therefore be considered impractical—despite the fact that such brakes were used on the extremely successful Mercedes-Benz 300SLR and W196 cars. A reasonable compromise would appear to be the use of outboard front brakes and inboard rear brakes—providing the latter can be adequately cooled and protected from oil leaks—as is now fairly common practice.

Radiators

Assuming the use of a water-cooled engine, it is necessary to find space in the car for mounting a radiator and also—in the case of rear-engined cars—water pipes. In this context it should be noted that although the radiator is normally mounted at the front of the car, this is not an essential feature of design. The only important factor in radiator location is to ensure that pressure is higher on one side of the radiator than the other. This can be achieved by mounting it somewhere between high and low pressure regions linked by a duct, or alternatively by the use of a fan; the latter may also be considered necessary to ensure adequate air flow through the radiator in heavy traffic, and in this case it can be of the thermostatically-controlled electric type. The conventional crankshaft-driven fan used on the majority of front-engined cars is open to criticism on the grounds that it consumes power, wastes space and generally keeps the engine too cool under normal operating conditions. The most effective method of cooling is to keep the water flowing through the engine rapidly, with the outlet temperature only a little higher than the inlet temperature. This gives the engine an even running temperature and also permits the use of a smaller radiator. The only disadvantage is that the water pump, like the cooling fan, may use a little power—probably not more than 1 b.h.p. in this case.

Disregarding convention, if the designer can locate a radiator elsewhere than at the front of a car, without detriment to its cooling function, he may well be able materially to improve the car's air penetration. Wherever it is located, ducting both ahead of and behind it, together with some means of sealing which ensures that all the intake air passes through the radiator, will promote maximum efficiency and allow the radiator to be made as small as possible. If it suits the over-all design of the car a sloping radiator may be used, in which case it would appear better to lean the radiator

forward. However, it is generally more convenient, especially where space is at a premium, to lean the radiator backwards slightly. As the air velocity depends on the drop in pressure across the radiator matrix this should always be the deciding factor in detail layout.

The actual size of the radiator can only be decided by trial and error, by previous experience or by observation of current practice. Several proprietary radiators of the crossflow type are available, but these are usually larger than necessary and thus may have an adverse effect on body shape. Once again, the use to which the car is to be put must be the deciding factor in the choice of size.

It is, of course, possible to dispense with the radiator altogether by using an air-cooled engine. This practice has much to recommend it. The whole engine can be made lighter and simpler owing to the elimination of water passages. The chief difficulty is to provide adequate cooling without increasing the size of the power unit by the addition of a large, power consuming fan. The critical element is cooling of the cylinder head. Lubricating oil also assists in the work of cooling and thus its temperature must be kept within bounds if it is to perform its primary function adequately. For these reasons—and others—the water-cooled engine remains with us, and is likely to do so for a very long time.

Oil Coolers

In addition to a water radiator it is often necessary to provide a means of cooling the oil on high-performance cars of both the water- and air-cooled varieties. The oil radiator can either be mounted in unit with the water radiator or separately—in which case most of the remarks made concerning water radiators still hold good. It is also possible to use a heat exchanger in order to warm engine oil quickly from a cold start and to keep water and oil at similar temperatures under running conditions.

Electrical Wiring

On the question of electrical wiring it is necessary only to say that it is desirable to use proprietary electrical components, including such items as windscreen wipers and lights, wherever possible—if only from the point of view of cost.

Seats

It is most important to give the driver the greatest possible amount of support, particularly in high-performance and racing cars, and in this case support is required not only for the hips but also for the upper part of the body, as the tendency when driving without support in this region is to obtain it by gripping the steering wheel—to the possible detriment of controllability. A means of seat adjustment is also important, to get the

feet the right distance from the seat, and in conjunction with this the length of the steering column should also be adjustable.

The basic seat can be made in either sheet metal or glass-fibre and should be sufficiently padded to avoid fatigue over long periods. On a racing car it should ideally be moulded to the anatomy of the individual driver, and even when this is not practical it should be possible to provide some means of increasing or decreasing the amount of padding in the sides of the backrest. Whatever the car, the bottom of the seat should not be horizontal, as this tends to cause the driver to slide forward under braking, nor should the lip of the seat be inclined too sharply, as this tends to restrict the blood supply and cause fatigue. In open cars it is also useful if the seat padding material can be made removable or waterproof.

The driver's comfort is an essential feature of racing car design, and for this reason support is normally provided for the knees and for the clutch foot. Similar considerations can also increase the comfort of both driver and passenger in a road sports car, and in this case the best solution is to divide the whole compartment into two separate cockpits which provide lateral support for the knees and a comfortably-angled footrest for the passenger, whose seat, like the driver's, should be adjustable.

Controls

Mention has already been made of steering wheel layout and gear lever position, and thought should also be given to the disposition of switches and instruments. Switches should be easily reached and instantly recognisable, while instrumentation depends largely on the price range of the car. Rev counter, oil pressure gauge, water and oil temperature gauges and ammeter are always useful but are not necessarily essential. In some instances they can even cause the driver unnecessary concern. Even on a racing car, only those instruments which are absolutely essential should be included, and they should be in a position where they can be read at a glance. There is little time to contemplate instruments at racing speeds, and for this reason there is much to be said for mounting them in such a way that their needles are all upright when they are in the operationally correct positions.

Luggage

On a road sports car it is important to have space for a certain amount of luggage—preferably in a separate compartment. Door pockets or cubby holes within the car are also extremely useful and it is an advantage if a lockable compartment can be provided for valuables. Weather protection on open cars should really provide protection against the weather, and soft hoods should be capable of being put up or taken down in a very short time. Proprietary hoods and sliding side-screens now function remarkably well in most respects, and open sports cars *can* provide all the

protection of family saloons—even down to heaters and demisters. Here again it should be possible to fit proprietary units providing sufficient space has been left under the bonnet.

* * *

Whatever the car and whatever the use to which it is to be put, it is important that a complete design study is made before work is begun, to avoid the risk of major changes being necessary in the final stages of construction. And it is even more important that all detail work is completed before the car is considered “finished”—in the case of a one-off—or “passed for production”. Impatience to get the car on the road has been the downfall of many a good design.

APPENDIX I

CHASSIS STRESS CALCULATIONS

In the following pages are extracts from the stress calculations for a sports car chassis. The purpose of these is to work out the loads to be dealt with by the chassis, in order to decide on the diameter and gauge of tubes required. In general, only instances of each design case are given, as the full calculations run to over 60 pages. There are other methods of achieving the same results, but this is the one adopted by a fully-qualified stress engineer.

ITEMS TO BE CHECKED

- (1) Weight distribution.
- (2) Wheel bump loads
 - (a) Front, static 1 g.
 - (b) Front, 4.5 g bump.
 - (c) Rear, static 1 g.
 - (d) Rear, 4.5 g bump.
- (3) Chassis load distribution
 - (a) 4.5 g diagonal bump.
 - (b) 4.5 g symmetrical bump rear.
 - (c) 4.5 g symmetrical bump front.
- (4) Chassis torsional stiffness.
- (5) Detail check on chassis tubes.
- (6) 1 g braking loads
 - (a) front.
 - (b) rear.
- (7) Detail check on suspension pick-ups.
- (8) Front frame detail check. Jacking.
- (9) Rear frame detail check
 - (a) jacking.
 - (b) side load.
- (10) Detail check on front suspension.
- (11) Detail check on rear suspension.
- (12) Detail check brake pedal mtg.
- (13) Detail check clutch pedal mtg.
- (14) Detail check diff. box mtg.
- (15) Rough check on steering gear.
- (16) Detail check diff. box.

APPENDIX I

PRELIMINARY WEIGHT BREAKDOWN AND C.G. ESTIMATE

Datum Line and Front Axle (Wheelbase 87")

<i>Item</i>	<i>y"</i> <i>above</i> <i>road</i>	<i>w</i> <i>lb</i>	<i>x"</i> <i>behind</i> <i>axle</i>	<i>WX</i> <i>lb/in</i>	<i>WY</i> <i>lb/in</i>
Chassis	11	50	42	2,100	550
Body	15	110	42	4,610	1,650
Engine	13.5	220	15	3,300	2,970
Gearbox	11	45	30	1,350	500
F/axle (100 lb unsprung)	12	110	0	0	1,320
F/spring units	14	15	0	0	210
R/axle (100 lb unsprung)	12.0	200	87	17,400	2,400
R/spring units	18.5	15	86	1,290	280
Engine mounting	8	3	4	12	20
Oil	5	8	15	120	40
Exhaust system	11	10	37	370	110
Propshaft	11	14	65	900	155
Oil filter	11	5	11	-25	60
Radiator & water	12	20	-15	-300	240
Fuel tanks & mtgs	13	17	46	780	220
Steering gear	9	10	25	250	90
Track rods, etc.	6.5	7	3	20	45
Brake and clutch linkages	14	7	33	230	98
Instruments	24	10	50	500	240
Lights	20	10	-10	-100	200
Seats	11	10	75	750	110
Spare wheel	20	30	101	3,030	600
Battery	11	35	-5	-175	385
Fire Extinguisher	17	4	50	200	68
Total kerb wt.	13.0	965 (8.6 cwt)	37.93	36,612	12,561
Add: Driver	14	200	68	13,600	2,400
Total + Driver	12.84	1,165 (10.4 cwt)	43.1	50,212	14,961
Add: 18 gall. fuel	13	131	43	5,630	1,700
Total ready to race	12.85	1,296 (11.6 cwt)	43.08	55,842	16,661

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WEIGHED ITEMS

Tail body panels	18 S.W.G.	27 lb.
Nose body panel	18 S.W.G.	21 lb.
Tonneau cover	18 S.W.G.	4½ lb.
Boot lid	18 S.W.G.	4 lb.
De Dion tube + brakes + drive shafts (3" 16 S.W.G.)		45 lb.
Propshaft tunnel	18 S.W.G.	4 lb.
Rear spring and damper		7 lb.
Ford radius arm		4 lb.
Ford swing arm		6 lb.
Diff., brakes, propshaft, oil (steel diff.)		97 lb.
Silencers		10 lb.
Small silencer		3 lb.
1955 Silencer		4 lb.
Steering box and arm		6 lb.
Fire extinguisher		4 lb.
Fuel tank		10 lb.
Oil heat exchanger		9 lb.
Track rod + link		7 lb.
Front wheel and tyre (wide rim)		27 lb.
Rear wheel and tyre		29½ lb.
Front wheel and tyre (narrow rim)		25 lb.
Alfin Drum 9" x 2"		8 lb.
Brake shoes and back plates		6½ lb.
A90 nose piece		31 lb.
Cast diff. for Mk 6 de Dion		10½ lb.
Half shaft		4 lb.
Distance piece M.G. gearbox to 1100		3 lb.
Seat (bucket)		4½ lb.
Wynn type radiator (Mk 8)		14½ lb.
2 Solex carburettors		10 lb.

BALANCE OUT CASES

The main chassis frame will be checked for the following cases:

- (1) *Maximum torsion.* This will be a 3 g bump applied diagonally with an assumed kerb or bump height of 6"; the over-all factor will be 1.5 giving an upward acceleration of 4.5 g at the wheels. The opposite pair of wheels will be assumed at 1 g.

This is probably rather severe but could occur through running off the road on to rough ground.

- (2) *Maximum retardation.* To give this condition I will assume that the car is braking at 1 g when the front wheel hit a bump 6" high, a kerb

APPENDIX I

for instance. This will produce a vertical reaction of 3 g on the front wheels so that factored reactions will be 1.5 g and 4.5 g on the rear and front wheels respectively.

Local attachment of the suspension and the rear frame will be dealt with as follows:

Front suspension

The 3 g bump on braking should fix most of the reactions but as a check we will do a 3 g bump while cornering at full lock with 1 g sideways.

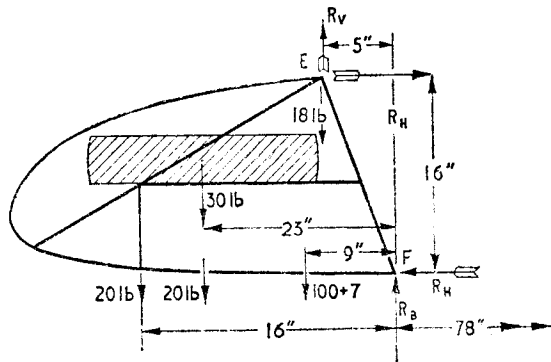
Rear suspension

The radius arms will be designed by a 3 g bump on braking. Maximum side load will occur in the cornering case when the car hits a bump although if this is not greater than 1.5 g applied to the rear of the car sideways then this case will be considered. A possible design case for the radius arms may be 1 g braking in reverse without any bump load.

CHASSIS

Torsion case (3 g × 1.5 up at diagonals).

Weight distribution. This should give approximately the right wheel loads, so first of all the items in the preliminary breakdown will be split up and applied at the appropriate points on the frame, then reactions are worked out to balance.



Rear frame

Total vert. load = 30 + 20 + 20 + 18 = 88 lb.

R_v per side = 44 lb.

R_B per side = 53.5 lb.

Moment of diff. about F = 900 lb in.

∴ Reactions at E and F = 56.2 lb.

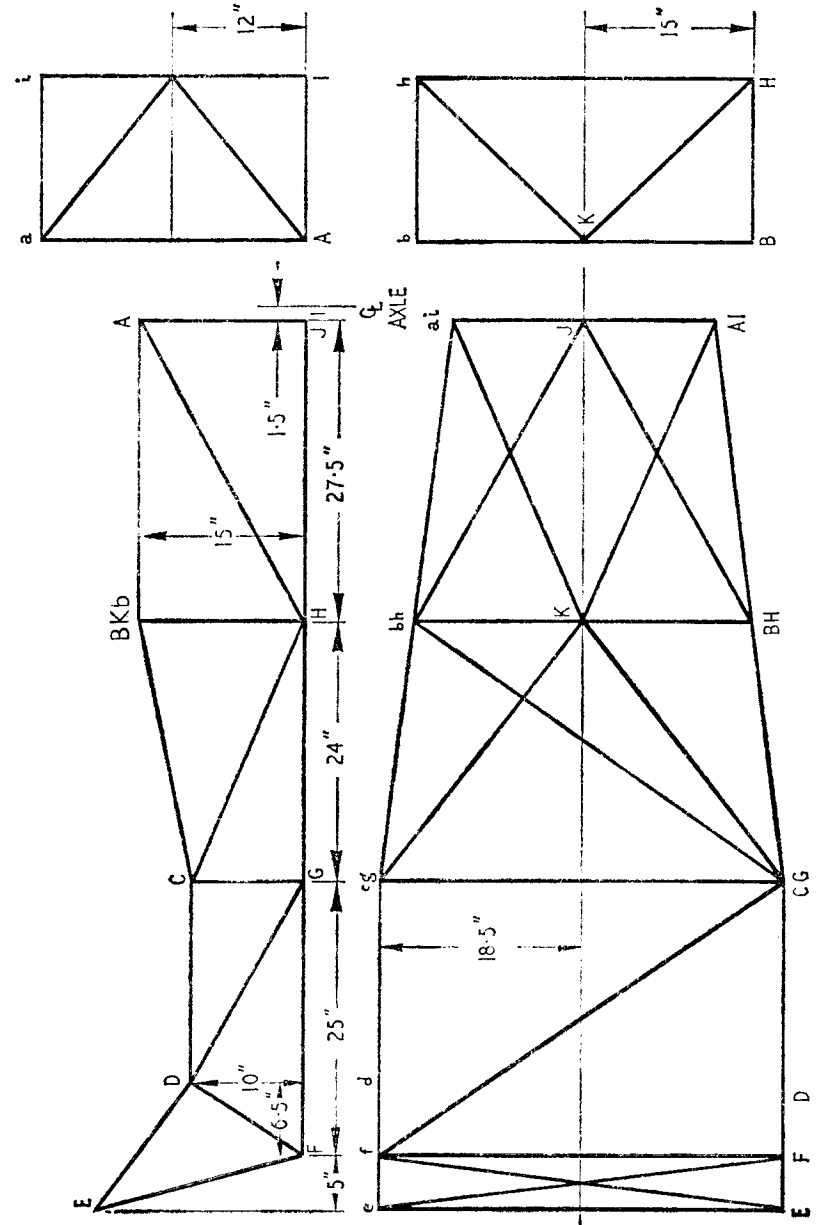


Diagram of chassis tubes in front elevation, side elevation and plan

Moment of tail about E = $(30 \times 18) + (40 \times 11) = 980$ lb in.

$$\therefore R_H = \left(\frac{980}{16 \times 2} \right) + \left(\frac{56 \cdot 2}{2} \right) = 58 \cdot 8 \text{ lb/side.}$$

Driver + seating:

Wt. of driver + seats = $200 + 15 = 215$ lb.

$$\text{Wt. on Ff} = \frac{(200 \times 15) + (15 \times 21)}{24} = 138 \text{ lb.}$$

Wt. on Gg = $215 - 138 = 77$ lb.

$$\text{Reaction at F} = 138 \times \frac{28}{37} = 105 \text{ lb.}$$

Reaction at f = $138 - 105 = 33$ lb.

$$\text{Reaction at G} = 77 \times \frac{28}{37} = 58 \text{ lb.}$$

Reaction at g = $77 - 58 = 19$ lb.

Engine + gearbox + propshaft + exhaust:

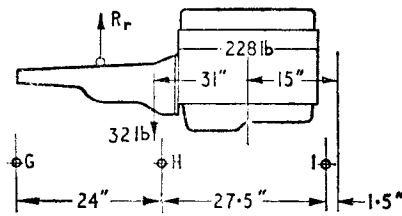
Total wt = $32 + 228 = 260$ lb.

Moment about I = $(39 \times 29 \cdot 5) + (228 \times 13 \cdot 5) = 4250$ lb in.

$$\text{Reaction at rear pick-up} = \frac{4250}{39 \cdot 5} = 107 \cdot 5 \text{ lb.}$$

$$\text{Front reaction} = \frac{260 - 107 \cdot 5}{2} = 80 \text{ lb/side.}$$

$$\text{Reaction at Gg, H and h} = \frac{107 \cdot 5}{4} = 27 \text{ lb.}$$



Fuel tanks, mountings, fuel, radiator:

Total wt = $15 + 200 = 215$ lb.

$$\text{Reaction at g} = 215 \times \frac{14}{24} = 125 \text{ lb.}$$

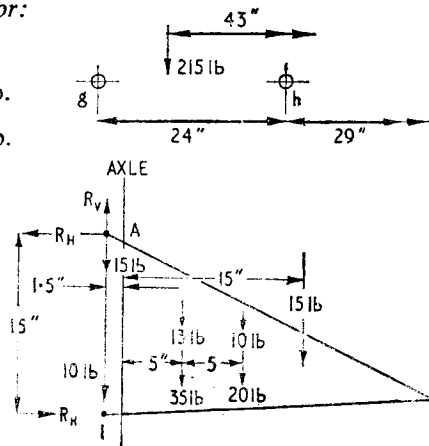
Reaction at h = $215 - 125 = 90$ lb.

Total load = $15 + 10 + 13 + 35 + 10 + 20 + 15 = 118$ lb.

$$\text{Moment about A} = (48 \times 6 \cdot 5) + (30 \times 11 \cdot 5) + (15 \times 16 \cdot 5) = 904 \text{ lb. in.}$$

$$R_H \text{ per side} = \frac{904}{2 \times 15} = 30 \text{ lb.}$$

$$R_V \text{ per side} = \frac{118}{2} = 59 \text{ lb.}$$



Chassis and body:

Total wt = $86 + 120 = 206$ lb.

So far we have taken 15 lb of this at $-15''$ on the front and 40 lb at $94''$.

Moment of chassis + body = $3710 + 5030 = 8740$ lb in.

Moment already catered for = $-225 + 3750 = 3525$ lb in.

Moment to be catered for = $8740 - 3525 = 5215$ lb in.

Weight to be catered for = $206 - 55 = 151$ lb.

$$\text{Centroid of residual wt} = \frac{5215}{151} = 34 \cdot 5''.$$

$$\text{Reaction at Cc} = 151 \times \frac{5 \cdot 5}{24} = 35 \text{ lb.}$$

Reaction at Bb = $151 - 35 = 116$ lb.

Reaction at C and c = $17 \cdot 5$ lb/side.

Reaction at B and b = 58 lb/side.

Instruments, exhaust, fire ext.:

These will be reacted at C and c.

Total load = $10 + 10 + 4 = 24$ lb.

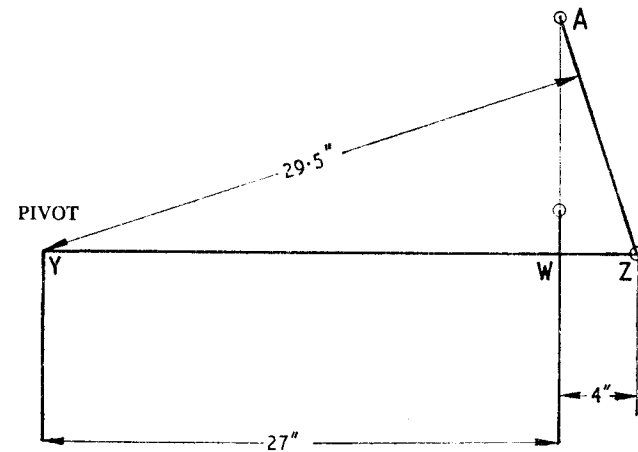
Load/side = 12 lb.

Brake linkage, pedals, etc.:

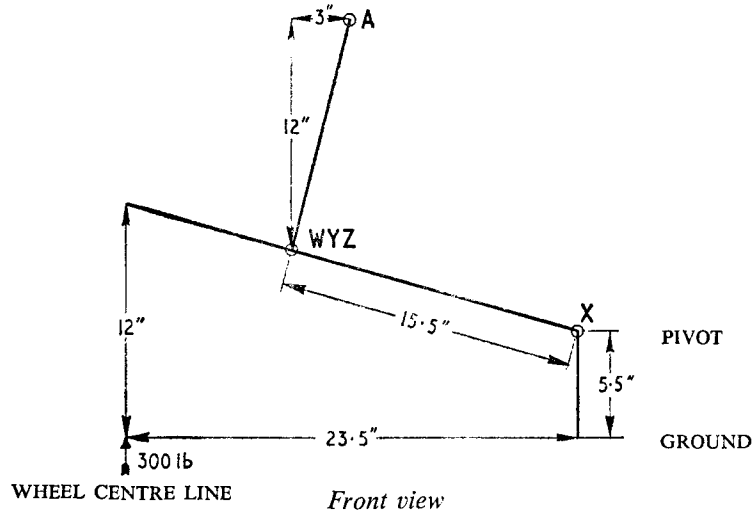
These will be reacted at B, total = 10 lb.

FRONT SUSPENSION

1 g Static loads



Side view



AW = 12.35", AZ = 13.0".

Load along AW = $300 \times \frac{23.5}{15.3} = 455 \text{ lb.}$

In plane AWZ:

Load at V = $455 \times \frac{31}{4} = 58.7 \text{ lb.}$

End load in spring = $445 \times \frac{27}{29.5} = 416 \text{ lb.}$

Component || to AW = $416 \times \frac{12.35}{13.0} = 394 \text{ lb.}$

Vert. component at A = $394 \times \frac{12}{12.35} = 383 \text{ lb.}$

Lateral component at A = $394 \times \frac{3}{12.35} = 95.5 \text{ lb.}$

Hor. component at A = hor. component at Y = $4.6 \times \frac{4}{13} = 128 \text{ lb.}$

Lateral component at Y = $58.7 \times \frac{3}{13} = 13.5 \text{ lb.}$

Lateral component at X = $95.5 + 13.5 = 109 \text{ lb.}$

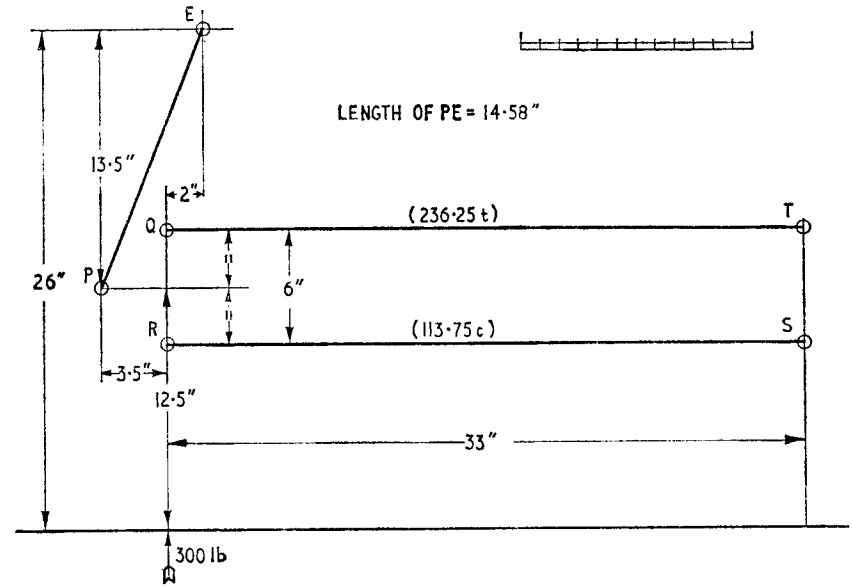
Vertical component at X = $38.3 + 58 - 300 = 141 \text{ lb.}$

REAR SUSPENSION

1 g Static loads

Vert. component at P = 300 lb.

\therefore End load in spring PE = $300 \times \frac{14.58}{13.5} = 324 \text{ lb.}$



Vert. component at E = 300 lb.

Hor. component at E = $300 \times \frac{5.5}{13.5} = 122.5 \text{ lb.}$

Couple about hub due to spring offset = $300 \times 3.5 = 1050 \text{ lb in.}$

\therefore Differential end loads in rad. arms = $\frac{1050}{6} = \pm 175 \text{ lb.}$

Hor. component due to spring = $\frac{122.5}{2} = 61.25 \text{ lb.}$

\therefore Load in QT = $61.25 + 175 = 236.25 \text{ lb.}$

Load in RS = $61.25 - 175 = 113.75 \text{ lb.}$

CHASSIS BALANCE OUT, DIAGONAL BUMP

The diagonal bump will be worst when we get a $3 \text{ g} \times 1.5$ bump up at the corners opposite to the main masses such as driver and fuel. That means that A and e will be the worst cases with full fuel. We will assume a 4.5 g bump on the nominal 300 lb load/wheel.

\therefore Up load at A and e = 1350 lb.

Nett up load on car = $600 + 2700 = 3300 \text{ lb.}$

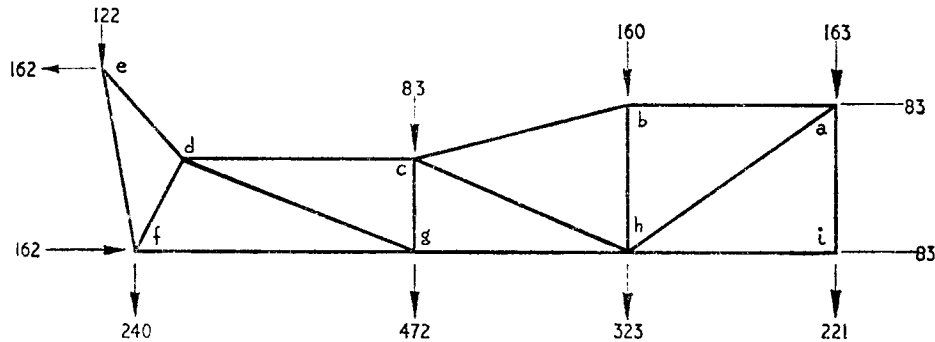
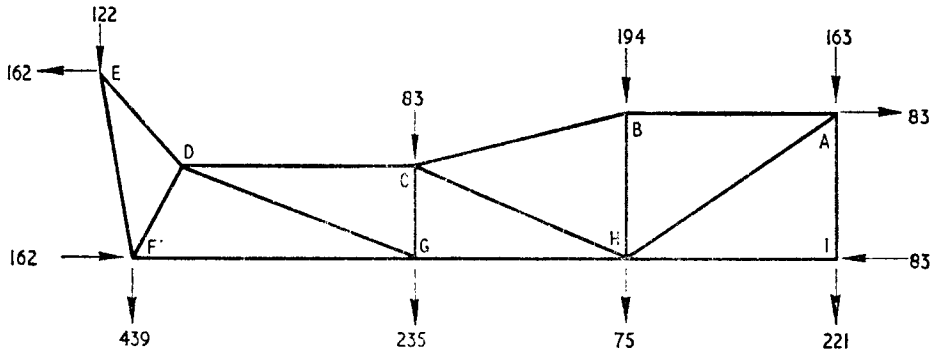
\therefore Normal acceleration at chassis = $\frac{3300}{1198} = 2.76 \text{ g.}$

Lateral couple = $1429 \times 2.76 = 3940 \text{ lb in.}$

S load/wheel at front = $\frac{3940}{47} = 83.8 \text{ lb.}$

∴ Up load at A = 1350 - 83.8 = 1266.2 lb.
 Up load at a = 300 + 83.8 = 383.8 lb.
 Up load at E = 300 lb.
 Up load at e = 1350 lb.

Summary of inertia loads

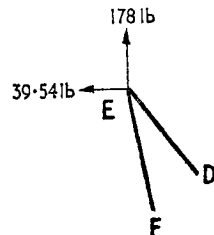


LOADS IN CHASSIS, TORSION CASE (USING TENSION COEFFICIENTS)

Examples

Joint E:

Vert. load = 300 - 122 = 178 lb.
 Hor. load = 162 - 122.5 = 39.5 lb.
 $19EF + 9ED = 178$
 $5EF = 11.5ED = 39.5$
 $3.91EF + 9ED = 31$
 $EF = \frac{147}{15.09} = 9.75$

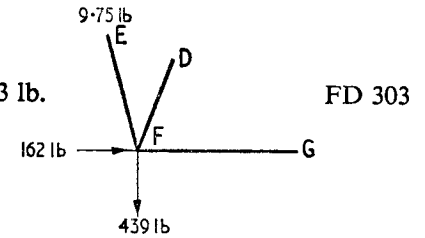


$ED = \left(\frac{178}{9}\right) - \left(\frac{19 \times 9.75}{9}\right) = -0.7$
 End load in EF = $9.75 \times 19.65 = 191$ lb.
 End load in ED = $-0.7 \times 14.59 = -10$ lb.

End Load, lb
 EF 191
 ED -10

Joint F:

$10FD = 439 - (19 \times 9.75) = 254$
 $FD = 25.4$
 End load in FD = $25.4 \times 11.92 = 303$ lb.
 $25FG = -162 + (9.75 \times 5.0) - (25.4 \times 6.5)$
 $FG = -\frac{278.3}{25} = -11.15$

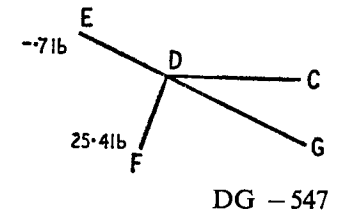


Load in FG = $-11.15 \times 25 = -278$

FG -278

Joint D:

$10DG = -(0.7 \times 9) - 254$
 $DG = -26$
 End load in DG = $-26 \times 21.05 = -547$ lb.
 $18.5 DC = (6.5 \times 25.4) - (0.7 \times 11.5) + (26 \times 18.5)$
 $DC = \frac{639}{18.5} = 34.5$
 End load in DC = 639 lb.



Check at CG

Moment about G = $-(178 \times 30) + (39.5 \times 19) + (439 \times 25) = 6400$ lb in.
 End load in DG = 640 check!
 Res. hor. at G = $(18.5 \times 26) - 278 = -759$ lb.
 Hor. shear due to radius arms = 122 lb.
 Movement due to radius arms = 1050 lb in.
 ∴ Load applied to front at C = $640 + 105 + 61 = 806$ lb. ←
 Load applied to front at G = $-759 - 105 + 61 + 803$ lb. →
 Vert. shear applied at G = $439 + 122 + 235 - 300 = 496$ lb. ↓

SUMMARY OF LOADS AND STRESSES

The following table gives the tube sizes, loads and stresses in the main members. Where the stress is given as an asterisk see the following cases for the resultant load with bending included. All members are 18 S.W.G. grade B2 cold-drawn tube.

Member	Max. Load lb	Size of Tube	Area in ²	Stress lb/in ²
EF, ef	± 2,000	$\frac{3}{4}$	0.106	19,000
ED, ed	± 1,152	$\frac{5}{8}$	0.087	13,300
FD, fd	± 2,020	$\frac{5}{8}$	0.087	23,000
DC, dc	± 3,828	1	0.144	26,500
DG, dg	± 2,080	$\frac{5}{8}$	0.087	24,000
FG, fg	± 1,448	1	0.144	*
CG, cg	± 513	$\frac{5}{8}$	0.087	6,000
GH, gh	± 3,100	1	0.144	22,000
HC, hc	± 630	$\frac{5}{8}$	0.087	7,300
HB, hb	± 562	$\frac{5}{8}$	0.087	6,400
HA, ha	± 990	$\frac{3}{4}$	0.106	9,400
HI, hi	± 678	1	0.144	4,700
BC, bc	± 1,615	1	0.144	11,400
BA, ba	± 3,480	1	0.144	24,000
AI, ai	± 392	$\frac{3}{4}$	0.106	3,700
Cc	*	$\frac{3}{4}$	*	*
Ee	*	$\frac{5}{8}$	*	*
Ef, Fe	*	$\frac{5}{8}$	*	*
Ff	± 221	$\frac{3}{4}$	0.106	2,100
Gf	± 395	$\frac{5}{8}$	0.087	4,550
Gg	± 449	1	0.144	*
Gh	± 777	$\frac{5}{8}$	0.087	9,000
Hh	± 411	1	0.144	*
IJ, iJ	± 120	1	0.144	830
BK, Kb	± 35	*	*	*
Aa	± 1,316	$\frac{5}{8}$	0.087	15,000
KA, Ka	-3,130 + 4,000	1	0.144	27,800
HJ, hJ	± 3,800	1	0.144	26,300
AJ, aJ	± 3,100	1	0.144	21,600
KC, Kc	± 910	$\frac{5}{8}$	0.087	10,400
KH, Kh	± 10	$\frac{5}{8}$	0.087	100

* Designed by local bending.

LOADS ON REAR FRAME

There are two cases to consider on this structure, both well over the normal 3 g bump loading by quite a large margin since there is not much weight in the back.

- (1) *Jacking case.* This is pretty straightforward since there won't be any great acceleration factor and the max. loaded state will be at 1400 lb. i.e. driver and full fuel. The normal 1.5 factor will be applied.
- (2) *Side load on rear axle.* This is not so simple as the 1 g sideways will most likely be much less than a sideways bump applied at the wheel, say over-sliding and hitting a kerb. A reasonable figure seems to be 2 g with the usual 1.5 factor.

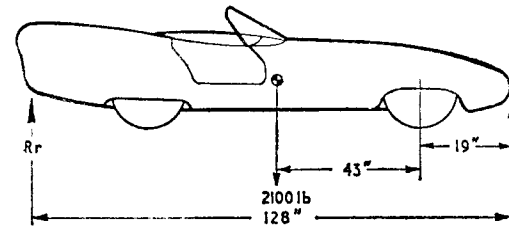
Jacking Case

Factored weight = 1.5 × 1400 = 2100 lb.

With car up on both jacks

$$R_r = \frac{2100 \times 62}{128} = 1020 \text{ lb.}$$

$$R_f = 2100 - 1020 = 1080 \text{ lb.}$$



REAR FRAME, SIDE BUMP CASE

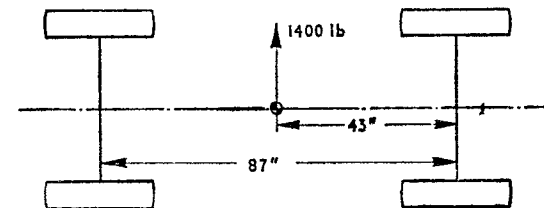
Static 1 g side load at rear wheel

$$= 1400 \times \frac{43}{87} = 690 \text{ lb.}$$

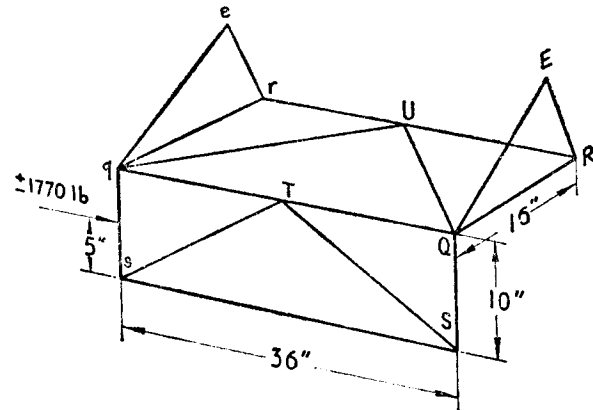
Unsprung mass at rear = 100 lb.

∴ side load on sprung mass = 590 lb.

Applying a factor of 1.5 × 2 g side load = 1770 lb factored.

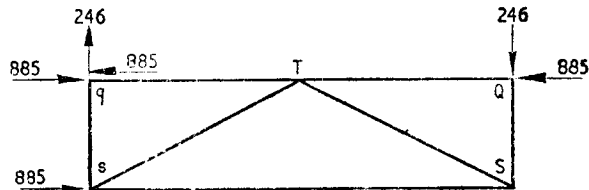


The side load applied halfway up qs will be divided equally between q and s. An imaginary member RUr will be inserted with side load being reacted at U and movements being reacted at ER and er.



Consider frame qsQS:

The loading system will be as follows if we assume the load to be reacted equally at q and Q.

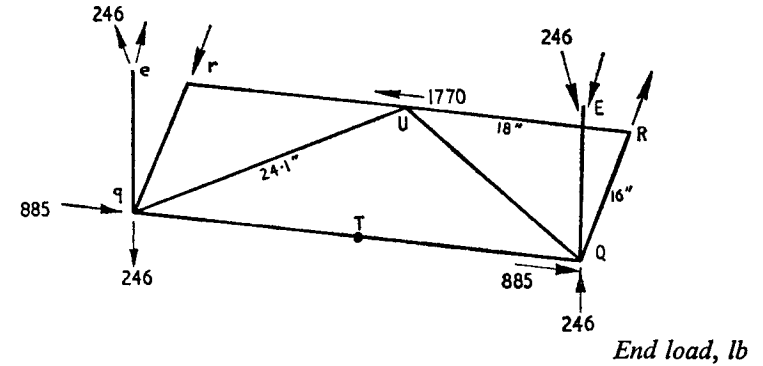


End load in QS = 246 lb c.
 End load in TQ = 885 lb c.
 End load in sS = $246 \times \frac{18}{10}$
 = 442 lb c.
 Load in TS = $2442 \times \frac{20.6}{18} = 504$ lb t.
 Load in Ts = load in TS = 504 lb c.
 Load in qs = load in QS = 246 lb t.
 Load in qT = 0.

End Load, lb
 QS - 246
 TQ - 885
 Ss - 442
 TS + 504
 Ts - 504
 qs + 246
 qT 0

Consider top frame EQRUEqr:

In this frame we will neglect QTqr as the loads in this have already been found.

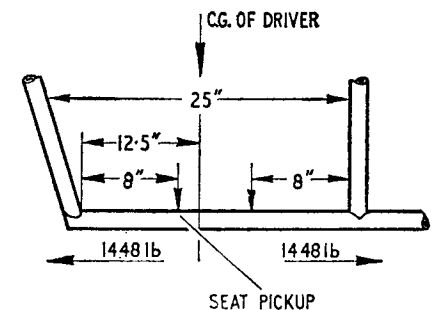


End load, lb
 End load in Uq = $885 \times \frac{24.1}{18} = 1182$ lb c. Uq - 1182
 End load in UQ = 1182 lb t. UQ 1182
 End load in eq = $246 \times \frac{1}{0.5} = 492$ lb t. eq 492
 End load in EQ = 2492 lb c. EQ - 492
 Hor. reaction at e = $492 \times 0.867 = 427$ lb.
 End load in qr = $(1182 \times \frac{16}{24.1}) - 427 = 363$ lb t. qr 363
 End load in QR = 363 lb c. QR - 363

BOTTOM MAIN TUBE FG
 (18 S.W.G. x 1" O/D)

We will assume that the tube is built in at the ends and that we have a 12 stone driver under 2.76 g.

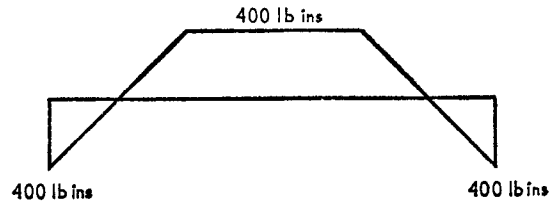
Fully factored wt of driver = $12 \times 14 = 2.76 = 460$ lb.
 Deducting 60 lb for the reaction at his feet load/side = $\frac{400}{2} = 200$ lb.
 ∴ Load at each pick-up = 100 lb.



Fixing movement at ends of beam = $100 \left(\frac{8 \times 17^2}{25^2} + \frac{64 \times 17}{25^2} \right) = 545$ lb in.

Free BM at seat pick-up = 800 lb in.

If we allow for less fixation we get a BM diagram as shown.



$$\text{Ft due to end load} = \frac{1448}{0.144} = 10,000 \text{ lb/in}^2$$

$$\text{Ft due to Bending} = \frac{400}{0.0326} = 12,300 \text{ lb/in}^2$$

$$\text{Nett ft} = 22,300 \text{ lb/in}^2$$

$$\text{Allowable ft} = 25,000 \text{ lb/in}^2$$

Reserve Factor
1.12

LOADING ON REAR RADIUS ARMS DUE TO TWIST

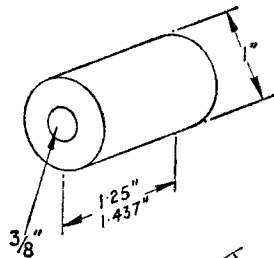
The worst case will be full up and down bump at opposite ends giving a total deflection of 6" on a 38" base.

$$\therefore \text{Slope of axle} = \frac{6}{38} = 0.158 \text{ radians.}$$

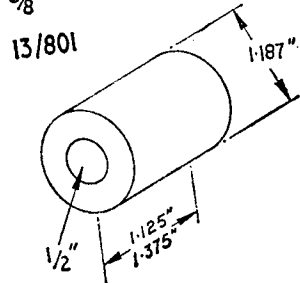
Assume half twist on each bush = 0.079 rad./end.

13/850

The figures are quoted for Silentbloc bushes but for similar Metalastik bushes we have the following figures.

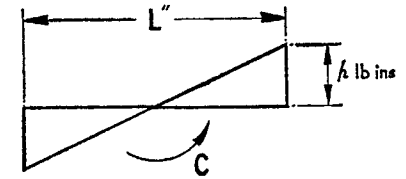


	Torsion	Axial	Radial
rate	200 lb/in	1900 lb/in	15,000 lb/in
max. defn.	16°	0.061"	0.02"
load	57 lb.	115 lb.	300 lb.



	Torsion	Axial	Radial
rate	560 lb/in	2600 lb/in	36,000 lb/in
max. defn.	12°	0.062"	0.015"
load	120 lb.	160 lb.	540 lb.

Using the figures for 13/850 I will now find the torque required in the radius arm for a given bush deflection.



$$\text{Deflection at end of bush} = \delta = \frac{P}{r}$$

$$\text{Couple} = \frac{pl^2}{6} = \frac{\delta rl^2}{6}, \quad r = \text{rate in lb/in/in.}$$

$$\therefore \text{For } 0.02" \text{ defn. couple} = 1200 \times 0.02 \times \frac{1.56}{6} = 62.3 \text{ lb in.}$$

$$\text{Deflection at a slope of } 0.079 \text{ rad} = 0.049"$$

$$\therefore \text{Torque} = 153 \text{ lb in.}$$

$$\text{With a } 1" \text{ O/D} \times 18 \text{ S.W.G. arm, twist} = \frac{153 \times 32.5}{0.0326 \times 12,000,000} = 0.0127 \text{ rad.}$$

Relative to the bush, twist of the tube is small—about 16%—but since the value of the stiffness of the bush is a bit doubtful it will be safer if the arm is checked on the assumption that the arm will have to take up any twist in excess of the allowable deflection of the bushes. The 13/850 bush is rather tight on allowable load so the 13/801 bush will be used instead.

With 13/801 bush:

$$\text{Allowable deflection radially} = 0.015"$$

$$\text{Rate radially} = \frac{36,000}{1.125} = 32,000 \text{ lb/in/in.}$$

$$\therefore \text{for } 0.015" \text{ deflection couple} = 32,000 \times 0.015 \times \frac{1.27}{6} = 102 \text{ lb in.}$$

$$\text{Deflection in rods for } 0.015" \text{ movement} = \frac{0.015 \times 2}{0.635} = 0.047 \text{ rad.}$$

$$\therefore \text{Stiffness of bushes} = \frac{102}{0.047} = 2170 \text{ lb/rad.}$$

$$J \text{ for } 1" \text{ O/D} \times 18 \text{ S.W.G. tube} = 0.0326 \text{ in}^4$$

$$\text{Stiffness of tube} = \frac{11,500,000 \times 0.0326}{32.5} = 11,500 \text{ lb in/rad.}$$

$$\therefore \text{Deflection in bushes} = \frac{11,500}{13,670} = 0.850$$

$$\text{Torque produced by this deflection} = 2170 \times 0.85 \times 0.158 = 292 \text{ lb in.}$$

$$\text{Deflection at ends of bush} = 0.015 \times \frac{292}{102} = 0.043"$$

$$\text{Max. overload on bush} = \frac{2095}{540} = 3.9 \text{ times.}$$

$$\text{Stress in tube due to torsion} = \frac{292 \times 0.5}{0.0326} = 4500 \text{ lb/in}^2$$

The following table shows the minimum bending radius for this range of tubes.

APPENDIX II
MATERIALS

The following details are intended to supplement the general information on materials in Chapter III.

CHASSIS MATERIALS

The material of which space frame chassis are most commonly constructed is tubular steel, of round or square section. Tubes used for this purpose generally range in size from ½ inch to 1½ inch diameter and from 20 to 14 gauge. For most purposes electric resistance welded tube is quite satisfactory, and its price is less than half of solid drawn tube. It is normally readily obtainable from the manufacturers of seamed, welded tubing.

Listed below are the Electric Resistance Welded tubes particularly suitable for space frame chassis. The general specification is approximately 30 ton mild steel.

BS 980	ERW 1	CEW 1
	ERW 2	CEW 2
	ERW 3	CEW 3
		CEW 4

The following table shows the properties of this range of tubes in 18 gauge.

Outside diameter inches	Weight (theoretical)		Area A sq in.	Moment of inertia 1 inch ⁴	Section modulus Z inch ³	Radius of gyration k inches
	lb. per ft.	ft. per ton				
½	0.232	9655	0.0681	0.0018	0.0070	0.1607
⅝	0.264	8485	0.0776	0.0026	0.0092	0.1827
⅜	0.296	7568	0.0870	0.0036	0.0117	0.2047
1	0.328	6829	0.0964	0.0050	0.0144	0.2267
1 ⅛	0.360	6222	0.1059	0.0066	0.0175	0.2488
1 ¼	0.392	5714	0.1153	0.0085	0.0208	0.2708
1 ⅝	0.424	5283	0.1247	0.0107	0.0245	0.2929
1 ¾	0.456	4912	0.1341	0.0133	0.0284	0.3150
2	0.488	4590	0.1436	0.0163	0.0326	0.3370
2 ⅛	0.520	4308	0.1530	0.0197	0.0371	0.3591
2 ¼	0.552	4058	0.1624	0.0236	0.0420	0.3812
2 ⅝	0.584	3836	0.1718	0.0279	0.0470	0.4032
2 ¾	0.616	3636	0.1812	0.0328	0.0525	0.4253

Outside Diameter	S.W.G.	Suggested minimum centre line radius		
		Loaded	Mandrel	Empty
½ in.	18	1	1 ⅛	1 ⅜
	20	1	1 ¼	1 ½
	16	1	1 ⅝	1 ¾
⅝ in.	18	1 ⅛	1 ¼	1 ⅝
	20	1 ⅝	1 ⅝	1 ¾
	16	1 ⅝	1 ¾	1 ⅞
¾ in.	18	1 ¼	1 ⅝	1 ⅞
	20	1 ¼	1 ⅞	2
	16	1 ¼	1 ⅞	1 ⅞
1 ⅛ in.	18	1 ⅝	1 ⅞	2 ⅛
	20	1 ½	1 ¾	2 ¼
	16	1 ¼	1 ⅞	2 ⅝
1 ¼ in.	18	1 ½	1 ¾	2 ¼
	20	1 ⅝	2	2 ½
	16	1 ½	1 ⅞	2 ½
1 ⅝ in.	18	1 ¾	2 ⅝	2 ¾
	20	1 ⅞	2 ⅞	3 ⅛
	16	1 ¾	2 ¾	3
1 in.	18	2	2 ½	3 ¼
	20	2 ¼	2 ⅞	3 ⅝
	16	2	2 ⅞	3 ⅞
1 ⅜ in.	18	2 ¼	3	3 ⅞
	20	2 ⅝	3 ¼	4 ¼
	16	2 ⅝	3	3 ⅞
1 ½ in.	18	2 ½	3 ⅝	4 ⅜
	20	2 ⅞	3 ⅞	4 ⅞

Square tubes. For certain specialised items—and chassis members to which components are bolted or riveted—it is sometimes advantageous to use square section tubing. This can also be obtained quite easily from the manufacturers of seamed, welded tubing, and should be either good quality mild steel or manganese alloy steel of a type particularly suitable for welding.

Other tubing found in chassis of this type is thick wall tube, used as bushes for mounting suspension members, etc; tubing used for this purpose should have a slightly smaller hole than the nominal bolt or pin size, allowing for the hole to be reamed after welding.

Tubing used for wishbones and other suspension linkages will generally be of similar material to that suggested for space frame chassis.

APPENDIX II

For other types of chassis—twin tube and multi-tubular—the diameter of the main members is generally larger; the normal range for twin tube chassis is 3–3½ inch, 16 or 14 gauge. In this case—as torsional stiffness is derived from the stiffness of individual tubes, and as arc welding is normally employed—the material should be of a higher quality than that which is necessary for space frames. In this instance a good specification is manganese alloy steel, which is suitable for both gas and electric welding.

OTHER CHASSIS COMPONENTS

A large variety of steels is available for such items as steering arms, stub axles, nuts and bolts, etc.

The table below shows the most suitable material for each purpose.

<i>Description and application</i>	<i>Type of steel</i>	<i>En. No.</i>	<i>U.T.S. tons/ sq. in.</i>	<i>Elong. %</i>
Free-cutting steel bars for machining for low duty bolts, nuts, studs, etc.	Carbon	En. 1A	32 min.	10 min.
35/45 ton Bright Carbon steel for machined details	Carbon	En. 6	35/45	15 min.
40 Carbon steel, for bolt and machined details in general. Very widely used	Carbon	En. 8	35 min.	20 min.
		En. 8	40 min.	22 min.
		Range Q En. 8 Range R	45 min.	20 min.
50 Carbon steel for general machined parts	Carbon	En. 43A	40 min.	18 min.
		En. 43A	45/60	12 min.
		En. 43A Range S	50min.	18 min.
45/55 ton Alloy Steel for similar duty to the 40/50 ton steel	Mn-Mo	En. 16 Range R	45 min.	22 min.
	1% Cr-Mo	En. 19 Range R	45 min.	22 min.
	1% Cr-Mo	En. 19A Range R	45 min.	22 min.
	Mn-Ni-Cr-Mo	En. 100 Range R	45 min.	22 min.
	Low Ni-Cr-Mo	En. 110 Range R	45 min.	22 min.
50/60 ton Alloy Steel for high tensile bolts, axle shafts, etc.	Mn-Mo	En. 16 Range S	50 min.	20 min.
	1% Cr-Mo	En. 19 Range S	50 min.	20 min.
	1% Cr-Mo	En. 19A Range S	50 min.	20 min.

APPENDIX II

<i>Description and application</i>	<i>Type of steel</i>	<i>En. No.</i>	<i>U.T.S. tons/ sq. in.</i>	<i>Elong. %</i>
50/60 ton Alloy Steel for high tensile bolts, axle shafts, etc.— <i>continued</i>	1½% Ni-Cr-Mo	En. 24 Range S	50 min.	20 min.
	Mn-Ni-Cr-Mo	En. 100 Range S	50 min.	20 min.
	Low Ni-Cr-Mo	En. 100 Range S	50 min.	20 min.
55/65 ton Alloy Steel for similar duty to the 50/60 ton steel	Steering arm. Mn-Mo	En. 16 Range T	55 min.	18 min.
	1% Cr-Mo	En. 19 Range T	55 min.	18 min.
	1½% Ni-Cr-Mo	En. 24 Range T	55 min.	18 min.
	Mn-Ni-Cr-Mo	En. 100 Range T	55 min.	18 min.
	Low Ni-Cr-Mo	En. 110 Range T	55 min.	18 min.
60/70 ton Alloy Steel for similar duty to the 50/60 ton steel, but where higher tensile strength is demanded	1½% Ni-Cr-Mo	En. 24 Range U	60 min.	17 min.
	2½% Ni-Cr-Mo (medium C)	En. 25 Range U	60 min.	17 min.
	Mn-Ni-Cr-Mo	En. 100 Range U	60 min.	17 min.
	Low Ni-Cr-Mo	En. 110 Range U	60 min.	17 min.
70/80 ton Alloy Steel for similar duty to the 50/60 ton steel but where higher tensile strength is demanded	1% Cr-Mo	En. 19 Range W	70 min.	15 min.
	1½% Ni-Cr-Mo	En. 24 Range W	70 min.	15 min.
	2½% Ni-Cr-Mo (medium C)	En. 25 Range W	70 min.	15 min. (under 4 in.)
Low Ni-Cr-Mo	En. 110 Range W	70 min.	15 min.	
100 ton min. Alloy Steel for gears, high duty bolts, axle shafts, high duty spindles and other parts demanding strength, toughness or wear resistance	1½% Ni-Cr-Mo	En. 24 Range Z	100 min.	8 min.
	2½% Ni-Cr-Mo (medium C)	En. 25 Range Z	100 min.	10 min.
	4½% Ni-Cr	En. 30A	100 min.	10 min.
	4½% Ni-Cr-Mo	En. 30B	100 min.	10 min.
Ball Race Steel	1% C-Cr	En. 31	—	—
Carbon C.H. Steel for general details requiring high wear hardness	Carbon	En. 32A	32 min.	20 min.
	Carbon	En. 32B	32 min.	20 min.
	Carbon (semi-free cutting)	En. 32M	32 min.	20 min.
55 tons min. tensile C.H. Steel for similar parts to the 45 ton steel but requiring a stronger and tougher core	3% Ni-Cr	En. 36 Range T	55/75	15 min.

APPENDIX II

Description and application	Type of steel	En. No.	U.T.S. tons/sq. in.	Elong. %
85 tons min. tensile C.H. Steel for high duty C.H. details, transmission gears, reduction gears, etc.	4½% Ni-Cr	En. 39A	85 min.	12 min.
Vehicle spring steels for laminated and coil springs	Si-Mn	En. 45A	—	—
Engine Valve Spring Steel	Cr-V	En. 50	90/110	—

It can be seen from the above table that many steels are repeated; the only difference is the range of heat-treatment and this can be varied to suit a particular application. A wide knowledge of the various types of steel commercially available and the uses to which they can be put—particularly as alternatives to the design specification—is invaluable to the designer in these days of steel shortage.

Forgings. The above table includes details of steels used to produce forgings, for such parts as steering arms, suspension uprights, stub axles etc.

CASTINGS

The basic casting materials used in high-performance cars are of the light alloy types—particularly aluminium and magnesium alloys.

Although there are literally hundreds of aluminium specifications, each having its own particular physical properties, only the more commonly used are listed here.

For lightly stressed or unstressed castings the most suitable material is LM 4. This is one of the most widely used of all aluminium casting specifications and can normally be cast from existing patternwork at very short notice; all foundries use this material almost continuously. For stressed items such as suspension uprights, rack and pinion mountings, differential housings and steering column bosses, a higher performance material is required. The most commonly used in this context is LM 11. LM 6 is a fairly good alternative, but with the basic snag that it is difficult to machine accurately; in addition, the machined finish is not normally very good. For long production runs LM 22 is ideal, but cannot be considered for small-scale production as it is specifically for use in gravity die castings.

Aluminium Casting Alloys

Alloy	Sand Cast		Chill Cast	
	U.T.S. tons/sq. in.	Elongation %	U.T.S. tons/sq. in.	Elongation %
B.S. 1490 ¹ 1955	10	2	12	3
LM 4-M	11	8	14	10
LM 6-M	15	10	19	15
LM 11-W	17	9	17	9
LM 22-W	Gravity Die Casting			

APPENDIX II

Magnesium alloy (Elektron) castings can be used in nearly all casting applications as an alternative to aluminium, and are normally superior as castings, as well as being lighter (by one third) and generally easier to machine. Ultimate tensile stress is only 10–16 tons/sq. in, by comparison with 10–20 tons/sq. in for aluminium alloys, the range depending on heat treatment. And elongation is not as good as that of aluminium, particularly in the as-cast condition. However, it is widely used for wheels, suspension uprights, king posts, clutch bell housings, transmission casings and rack and pinion casings—for all of which applications the most suitable material is Elektron C. The weight reduction it permits is particularly important, and this encourages the use of sand-cast magnesium alloy even in very limited production, its weight being only two-ninths that of mild steel suitable for fabrication. Even allowing for the greater ultimate strength of mild steel, this still allows a very considerable saving in weight.

It is widely thought that Elektron is a dangerous material to use, owing to fire risk. However, although magnesium will burn, it is difficult to ignite. The only real danger occurs during machining, when the fine turnings are liable to ignite. In such circumstances it is important to remember that water will not extinguish a magnesium fire, but only help to spread it.

SHEET MATERIALS

A variety of sheet materials is available for use in both stressed and unstressed panels. Those most commonly used range in thickness from 20 gauge to 14 gauge, and sheets are normally available in sizes 6 ft x 3 ft and 8 ft x 4 ft.

The materials themselves include mild steel, various grades of soft (for shape panels), half-hard and hard (for stressed panels), aluminium, aluminium-clad alloys and sheet Elektron (DTD 120).

The use of steel panelling is restricted almost entirely to series produced cars, although it is sometimes used on limited production cars for such items as the undertray. In general, however, aluminium or clad materials should be adequate for this purpose, and effect a considerable weight saving. The lightest of all metal body materials is Elektron, but the one-third saving in weight (by comparison with aluminium) is offset by the cost and complexity of manufacture. This is due to difficulties experienced in shaping the material, together with the fact that it has to be specially treated before welding can be commenced. Furthermore, a great deal of skill is required in welding thin gauge sheet Elektron.

Glass-fibre. Resin-bonded glass-fibre, to give it its correct name, is becoming more and more widely used both for unstressed body panels and for load-carrying members.

It normally consists of glass-fibre mat and polyester resin, with hardener and accelerator, which is used to laminate as many layers of mat as are

APPENDIX II

required for any particular purpose. No real indication of the strength, weight or performance of glass-fibre can be given owing to great variation of quality in laminates.

Nevertheless, it is a very good material for the small-scale constructor, because tooling charges are relatively low, modifications—even of major parts—can be made with the minimum of cost and the use of pigmented resins obviates the need for painting. In addition, accident damage can be repaired both quickly and cheaply. The weight of a durable glass-fibre body is much the same as that of an aluminium one, and, except for one-offs, the glass-fibre body is always cheaper to make. Materials are readily obtainable, and there are also several firms which will make glass-fibre bodies to order.

Wood. A final possibility for the small constructor is the use of marine plywood, either for body panels or for complete chassis/body structures. The chief disadvantage is the cost of producing shape panels in this material but this can be overcome by using glass-fibre for the more complicated panels.

WEIGHTS OF MATERIALS

Sheet Weight lb/in²

S.W.G.	Duralumin	Magnesium Alloy	Steel
10	0.0139	0.0087	0.0376
12	0.0113	0.0070	0.0305
14	0.0088	0.0055	0.0240
16	0.0070	0.0044	0.0195
18	0.0054	0.0034	0.0144
20	0.0041	0.0025	0.0110
22	0.0032	0.0019	0.0085

Metal lb/in³

Duralumin	0.103
Magnesium alloy	0.066
Steel	0.283

Other Materials lb/in³

Glass (Safety)	0.092
Perspex	0.043
Glass-fibre	0.061

Liquids

	lb/gallon	lb/in ³
Petrol	7.2	0.026
Oil	9.0	0.032
Water	10.0	0.036

APPENDIX III

SUSPENSION CALCULATIONS

DETERMINATION OF ROLL ANGLES AND WHEEL LOADS DURING CORNERING

Calculation of roll angles and wheel loads is obviously a mathematical possibility, and consideration of these factors is essential in order to ensure stability of the vehicle when subjected to lateral loads.

Below is set out a method of determining these roll angles and loads, as applied to a double wishbone type of suspension used for front and rear axles.

The basic approach is to establish the formula for the rolling moment of the sprung mass, and the expressions for the moments of resistance produced by the suspension system on the sprung mass. By equating these the roll angle may be found.

By using the roll angle thus found, and considering moments in the planes of the front and rear axles separately, the weight transfer of each axle may be found.

The means of determining roll centre heights has already been shown in Chapter VII. The mean roll centre height in the transverse plane containing the centre of gravity of the sprung mass is also required, and this may be found as follows:

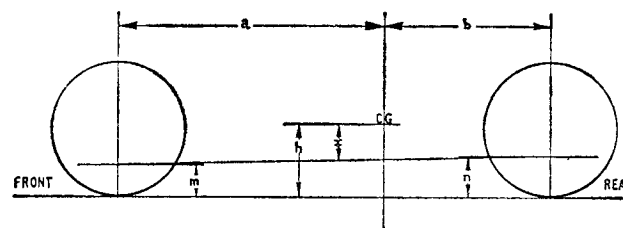


FIG. A

- In Fig. A m = front roll centre height.
- n = rear roll centre height.
- h = height of C.G.
- a = horizontal distance from front axle C/L to transverse plane through C.G.
- b = horizontal distance from rear axle C/L to transverse plane through C.G.

Then by similar triangles

$$\text{height of mean roll centre is } \frac{bm + an}{a + b}$$

Hence
$$x = h - \frac{bm + an}{a + b}$$

We will assume that the lateral force is produced by cornering and equals one "g" although this figure is most unlikely to be attained.

Let W = weight of the sprung mass
then horizontal force through C.G. also equals W .

Hence resulting rolling moment of sprung mass about the roll axis
$$= Wx = W \left(h - \frac{bm + an}{a + b} \right)$$

Additional rolling moments are added in the case of independent wishbone suspensions by the tilting loads on the wheels.

These can be shown to equal

$$W_1(r_1 - m) \text{ for the front wheels.}$$

$$W_2(r_2 - n) \text{ for the rear wheels.}$$

Cornering force is again assumed to be g .

- where W_1 = front unsprung weight.
 W_2 = rear unsprung weight.
 r_1 = rolling radius of front wheel.
 r_2 = rolling radius of rear wheel.

The heights of the centres of gravities of the unsprung weights are assumed to be at the height of the rolling radius of the appropriate wheel, an approximation which is sufficiently accurate in most cases.

Similarly it can be seen that an additional small rolling moment will result from the displacement of the C.G. laterally from above the roll axis when the chassis rolls; this again is small enough to be neglected.

Total roll moment therefore becomes

$$W \left(h - \frac{bm + an}{a + b} \right) + W_1(r_1 - m) + W_2(r_2 - n)$$

MOMENTS OF RESISTANCE

- In Fig. B k_s = spring rate.
 k_p = rate at lower pivot if a spring acted at right angles to lower wishbone.
 k_w = rate at tyre contact patch, as if a spring acted at this point normal to the line joining the tyre contact patch to the roll centre.

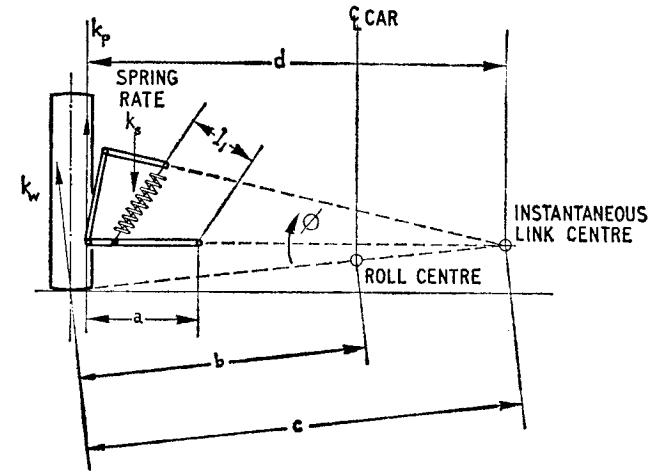


FIG. B

- a = length of lower wishbone.
 b = distance from tyre contact patch to roll centre.
 c = distance from tyre contact patch to instantaneous centre.
 d = distance from lower pivot to instantaneous link centre.
 l_1 = distance from the line of action of the spring to the inboard pivot.

By moments about inboard pivot point A.

$$k_p a^2 = k_s l^2$$

distances squared as k_p and k_s are rates.

Similarly, by moments about the instantaneous centre

$$k_w c^2 = k_p d^2$$

$$\therefore k_w = k_s \frac{l^2 d^2}{a^2 c^2}$$

Let ϕ = roll angle (about roll axis) in radians

If body rolls ϕ radians

$$\text{effective wheel movement} = \phi b$$

$$\text{corresponding force at wheel} = \phi b k_w$$

$$\therefore \text{Moment about roll centre} = \phi b k_w b = \phi k_s \frac{l^2 d^2 b^2}{a^2 c^2}$$

Hence, due to the suspension spring roll angle ϕ , a restoring moment on the spring mass is produced. The other spring produces a similar moment.

\therefore Restoring moment due to front suspension

$$M_1 = 2 \frac{b^2 d^2 l^2}{c^2 a^2} k_s \phi$$

A similar moment M_2 will be produced by the rear suspension.

Additionally, if an anti-roll bar is fitted its effect may be calculated similarly, giving a restoring moment M_3 .

Thus total moment of resistance to sprung mass =

$$M_1 + M_2 + M_3$$

For equilibrium, rolling moments on sprung mass must equal the moments of resistance produced by springs, anti-roll bars, etc.

Hence,

$$W\left(h - \frac{bm + an}{a + b}\right) + W_1(r_1 - m) + W_2(r_2 - n) = M_1 + M_2 + M_3$$

M_1 , M_2 and M_3 all contain ϕ , the roll angle, which can now therefore be evaluated.

The roll angle may be used to calculate the weight transfers for the front and rear axles.

CONSIDER THE PLANE OF THE FRONT AXLE

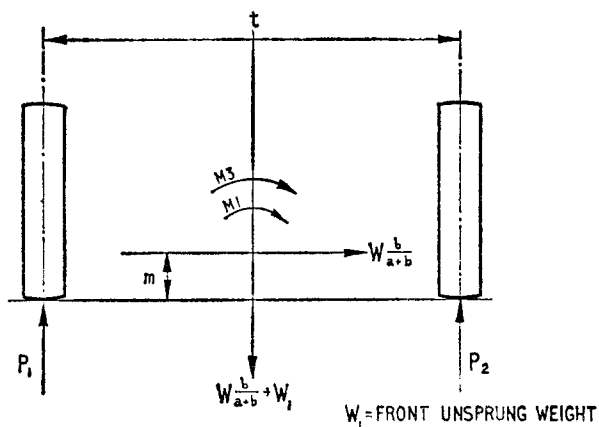


FIG. C

The weight on the front axle = $W \frac{b}{a + b} + W_1$, moments about rear wheel centre in Fig. A.

The effect of centrifugal force on the centre of gravity of the sprung mass has been considered as a horizontal force, with a rolling moment about the roll axis. In the same way the effect of centrifugal force on the unsprung masses, W_1 has been transferred to the roll centre height m and a tilting moment about roll axis of the same sign. These are accounted for by the moments of resistance.

Taking moments about O

$$(P_2 - P_1) \frac{t}{2} = M_3 + M_1 + \left(W \frac{b}{a + b} + W_1\right) m$$

$$P_1 + P_2 = W \frac{b}{a + b} + W_1 = \text{total front axle weight.}$$

$$\frac{1}{2}(P_2 - P_1) = \text{weight transfer from inside to outside wheel.}$$

$$\frac{1}{2}(P_2 - P_1) = \frac{1}{t} \left(W \frac{b}{a + b} m + W_1 m + M_1 + M_3 \right)$$

Thus, using the values of M calculated from previous equations (those from which the roll angle was found) the front wheel weight transfer can be found.

A similar equation will give the rear wheel weight transfer.

In these calculations it has been presumed that the roll angle for the plane containing the front axle is equal to that for the plane containing the rear axle. This in general presupposes a high figure for chassis torsional rigidity.

It is interesting to consider the effect of a chassis having negligible torsional rigidity. In this case the general result would be that front and rear roll angles would be different, and it would be possible to calculate them separately and also the weight transfers. In this case, although the fitting of an anti-roll bar would alter the roll angle of the end of the car to which it was fitted, it would not affect the weight transfer.

It should always be remembered that the total weight transfer from the inside wheels to the outside wheels of a vehicle is determined by the total weight of the vehicle, its track, the height of its centre of gravity and the cornering force. It is only possible by means of anti-roll bars, etc. to alter the fore-and-aft distribution of the weight transfer.

JACKING EFFECT

When cornering hard, cars with conventional swing axle rear suspension can be observed to thrust their tails upwards—hence the term “jacking effect”. This causes the rear wheels to take on considerable positive camber, with correspondingly reduced adhesion.

It affects all independently sprung cars, and the higher the roll centre the greater effect. It is this factor more than any other which has caused roll centre heights to be lowered and this, in conjunction with soft springs, has caused trouble with high roll angles, which have necessitated strong anti-roll bars—sometimes at both front and rear.

The jacking effect can easily be demonstrated by considering a swing-axle set-up.

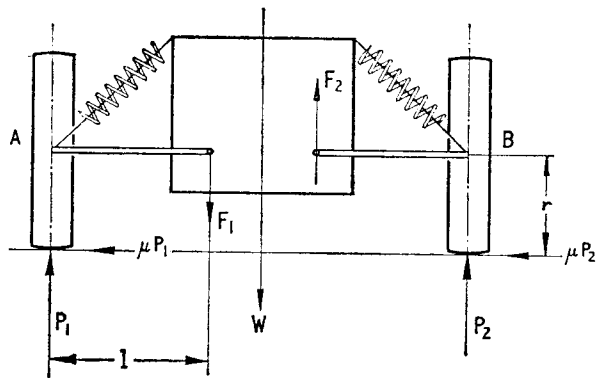


FIG. D

Taking moments about A $\mu P_1 r = F_1 l$

Taking moments about B $\mu P_2 r = F_2 l$

neglecting the couple caused by F_2 and F_1 and considering the vertical forces on body.

$$\text{Jacking effect} = F_2 - F_1$$

$$= \frac{r\mu}{l} (P_2 - \mu P_1)$$

in the worst case of one wheel becoming airborne $P_2 = W$, $P_1 = 0$, and possible values are $r = 11''$; $l = 22''$; $\mu = 0.5$; $W = 400$ lb.

$$F_2 - F_1 = \frac{11 \times 0.5}{22} (400)$$

$$= 100 \text{ lb.}$$

which could easily raise the chassis $1\frac{1}{2}''$, leading to correspondingly high camber change.

GLOSSARY

Bending	Tending to curve or bow.
Camber angle	Angle between centre line of tread and a vertical line drawn through intersection of centre line and ground line. N.B. Camber angle can be either positive or negative; in the latter case the top of the wheel is nearer the centre line of the vehicle.
Castor angle	The angle between a line drawn through the kingpost top and bottom bearings, or equivalents, and a vertical line through the wheel centre in side elevation.
Compression	Pressing together, caused by a load which tends to push in from both ends, or from one end against a fixed end.
Gauge	Throughout this book refers to the Standard Wire Gauge, the most commonly used means of measuring the wall thickness of tubular or sheet metal.
Glass-fibre	Throughout the text, this implies resin-bonded glass-fibre.
Gyroscopic Precession	The tendency for the polar axis of a rotating mass to deflect at 90 degrees to the direction of the applied force.
Load	= Force, and is used when discussing forces in a structure or members of a structure.
Lozenging	Becoming a mechanism, that is being subject to changes of shape, rather than being a rigid structure.
Shear	A strain where compression is answered by elongation at right-angles.
Stiffness	Resistance to deflection.
Strain	Deformation of a structure or individual members of a structure. N.B. This term is often used erroneously in an attempt to describe load.
Stress	Load per unit area—usually expressed in lb per square inch. N.B. This term is very often used erroneously instead of load.
Tension	Tending to stretch.

GLOSSARY

- Toe-in, toe-out** The angle between a pair of wheels—generally not quoted as an angle but as the difference in measurements between the tyre treads at the front and rear of the wheels at hub height. When the distance is greater at the rear, the wheel is said to “toe-in”—and vice versa. It is normal to allow toe-in of about $\frac{1}{8}$ inch to counteract the effect of wheel deflection.
- Torsion** Twisting.
- Track** The distance between the tread centres of the tyres on one axle.
- Wheelbase** The distance between front and rear axle centres.

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