PAPER

delivered by

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The Mercedes-Benz 300 SEL - 6.3
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At the International Motor Show in Geneva in March 1968 the Daimler-Benz Co. introduced a new car

the type 300 SEL - 6.3,

thus extending its new range of cars - the so called new generation - which had been shown to the general public for the first time at the Motor Show in Brussels in January 1968.

The 6.3 is intended to fill a gap in the Mercedes-Benz range by providing a car combining maximum comfort and outstanding performance.

This car - we confidently hope - will meet the requirements of exacting connoisseurs who still enjoy driving and who are prepared to pay a relatively high price for qualities which normally cannot be obtained in costconsciously designed and massproduced cars.

The enthusiastic acceptance in all European countries where the car has already been demonstrated proves that there is a market for a car of this type, and we are convinced that the 6.3 will also gain a respected place among the specialist cars in the United States.

Naturally at the proposed price the number of possible sales will be limited but already now sales in Europe by far exceed our production facilities and unfortunately long waiting lists are the consequence.

This new car is supposed to combine the comfort of a luxurious touring car with the performance, handling and roadholding of a sports car. At the same time very exacting demands concerning safety have to be met a prerequisite to the acceptability of high performance on public roads. This goal is certainly not easy to achieve, and some compromises in one or the other direction inevitably had to be accepted. But on the whole we believe to have been successful.

How far this actually is the case, you will be able to find out for yourselves by testing the car on the Laguna Seca racetrack where high speed, fast cornering and heavy braking is possible.

You will receive a detailed description of the 6.3 in our press information. Therefore there is no need to repeat that here, and I would rather spend our limited time on pointing out some of the measures we took to give the car the desired characteristics.

Luxurious comfort

First of all the interior of the body must be large enough to be able to provide ample and unrestricted room for the passengers on the front and rear seats.

Fig. 1 shows the main dimensions of the 6.3 in relation to the 1968 Cadillac Fleetwood sedan. It will be noted that although the shoulder width is narrower on the 6.3 by 2,3", it neverless is quite sufficient for two large persons. The distance between the front and rear seats is only slightly shorter (- 1,5") on the 6.3. This dimension contributes more to comfort in a fully occupied car than extreme width. The head room over the rear seats is sufficient for tall people.

The overall width of the Cadillac exceeds that of the 6.3 by 8,7" certainly a disadvantage for fast driving on narrow roads.

Fig. 2 shows the relative dimensions of the 6.3 and the Camaro, a competitor on the sporting side of the issue. Here the main difference lies in the accomodation for the rear seat passengers in which the 6.3 excels.

Needless to say, the passenger compartment of the 6.3 is luxuriously equipped, offering rich upholstery, air conditioning, central door and boot locking system etc.

The suspension of a car contributes even more to the comfort of the passengers than does the interior of the body. Here we have tried to achieve an optimum solution between ride and roadholding. For this sake independent suspension of all wheels is - as we feel - an absolute necessity. Although the normal solid beam rear axle functions quite well on smooth roads it's limits show up very soon when a car is travelling fast on corrugated and especially rippled surfaces. Then the relatively soft shock absorber settings which give a good boulevard ride cannot suppress shuddering of the rear axle. This shuddering is not only transmitted to the whole car and the passenger seats but it also causes the wheels to loose contact with the road, thereby impairing the directional stability of the car, and causing break away in cornering.

The smaller unsprung masses of independent suspension systems - in the case of the 6.3 our well known low pivot point swing axle - and the freedom of disturbance of a smoothly running wheel by the deflection of the other wheel on the same axle is of a great advantage in traversing bad roads. The wheels follow the irregularities of the surface more precisely and the reduced variation of the dynamic tire load provides more uniform adhesion in difficult conditions.

Incidentally the whole range of Mercedes-Benz passenger cars has been equipped with independent suspension for all 4 wheels for many years.

The functional limitations of a normal steel spring are well known. If a soft spring giving a lightly laden car, a good boulevard ride is selected then the behaviour of the car in fully laden state becomes unfavourable. The car sinks down over the rear axle and the amount of travel from the static position to the beginning of the rubber buffer is very small. Frequent bottoming especially on wavy roads and on putt holes is the consequence. This can be clearly seen in Fig. 3, the data of which were obtained on a 1968 U.S. sedan. With only the driver in the car the rear wheel has 4" travel to the buffer, thereby increasing the static load by 66 %. This is more than necessary to avoid noticeable bottoming. In contrast to this in the fully laden state there is only 0.9" to the buffer with an increase of static load of 9 %. A bumpy and uncomfortable ride on anything but smooth roads is inevitable. It will be noted that the spring rate is constant during bounce and rebound and is uneffected by the load of the car. Simplicity and low costs are the advantages of this widely used design. But a weighty compromise, usually favouring the lightly laden condition, has to be made. Heavy duty springs have to be installed for more difficult conditions.

In order to avoid these necessary compromises and in order to obtain the best possible car behaviour in all circumstances we have adopted our well proven air suspension system which we have been using on the type 300 SE since 1961 and on the type 600 since 1964. Fig. 4 shows the characteristics of the air springs of the rear axle. The spring rate is not constant as with the steel spring but becomes progressively stiffer with deflection. Further more the spring rate increases with the load of the car thus providing more uniform spring frequency. The travel of the wheel from the static position to the buffer is 4", irrespective of the load, owing to the selfleveling valves and the increase of wheel load is 27 % for light load and 23,5 % for full load which is sufficient to avoid bottoming except on very severe bumps.

In consequence of the automatic adaptation of the spring rate to the load and the selfleveling system, far softer air springs can be used than it would be possible with steel springs (+ 85 % unladen, + 52 % laden)

The possibility of increasing the ground clearance by 2" for traversing backland roads or deep snow is another advantage of the air spring system. Raising the car is triggered by moving a lever under the instrument panel.

Fig. 5 and Fig. 6 show the spring characteristics for the front axles. Here the advantages are not quite so significant as the greater part of the additional load is concentrated on the rear axle. Neverless, the spring rate on the 6.3 is far lower than on the U.S. car (56 lb/in against 14o lb/in).

In order to stabilize the car when driven at high speeds over wavy roads, the damper settings are quite a lot firmer than on the U.S. car. This counteracts the effect of the soft springs to some extent while driving slowly on good roads. This is one of the compromises which had to be made.

Tire pressure is one of the items which have a profound effect on comfort. As I will deal with later in some detail, low pressure, necessary for a soft boulevard ride, will not provide tire reliability at continuous high speed. This part of the problem of high speed and comfort has not been completely solved yet.

Performance

Acceleration

Taking into account that the 6.3 is not a sports car but a luxurious touring car with ample body dimensions, the acceleration capabilities are certainly outstanding. The acceleration is quite comparable to that of current U.S. production sporty cars.

Table No. 7 shows this in figures. The data of the American cars was taken out of a test report in "Car and Driver" March 1968.

It should be noted that the final drive ratio of the U.S. cars is somewhat higher than on the 6.3 indicating that they are aimed at maximum acceleration instead of high top speed. Against this the 6.3 has a final drive ratio more suitable for higher speed, thereby accepting a slight reduction in acceleration. With a ratio of 3,5 instead of 2,85 the time for accelerating from 0-100 mph would be improved from 15 s to about 14,7 s. Diagram No. 8 shows speed against time in acceleration for the 300 SEL - 6.3 and the U.S. sporty cars.

Top speed

Very high maximum speed is of no importance in countries with speed limits. But in countries without limits on open roads out of town and villages as in Germany, France, Italy Spain, some people like to drive fast, traffic and road conditions permitting. We believe that a relatively moderate top speed of about 14o mph is sufficient for a car of this type. Consequently we have chosen a suitable valve timing, which causes the power of the engine to taper off sharply at speeds over 52oo.

Safety

Adhesion

Now in a time when safety has become the most prominent and most advertised property in car design and driving, the question arises immediately

"is this high acceleration capacity dangerous"

"do these powerfull cars demand excessive qualification from the driver"

We believe this is not the case and we think we can prove it.

Adhesion of the car on the road and freedom of skidding is a necessity for safe handling and consequently the conditions existing during acceleration in regard to adhesion should be investigated.

Adhesion of the tire is a complex product of

the design of the tire and the tread pattern

the properties of the rubber used for the tread

the tire load and

the road surface

The adhesion of the driven tire can either be utilized completely for pushing the car straight ahead. Then the tire cannot produce any side thrust and the car is prone to skidding side ways. If none of the adhesion is used for driving then maximum side thrust can be produced. Any mixture of driving thrust and side thrust is possible, as indicated in Diagram No. 9.When the 6.3 is accelerating on full throttle in 1st gear on a dry, rough, concrete surface, then the rear wheels are on the verge of slipping. Calculations and measurements indicate a coefficient of friction = 1,0 or even a little more. Torque of the rear wheels and adhesion are in balance. Theoretically no or very little side thrust can be generated and side skidding could occur. In actual driving this is rarely noticed, and could be stopped by counter steering or by slightly closing the throttle.

In order to avoid getting negligently into the area of marginal adhesion (1st gear) especially on wet and slippery surfaces the automatic gearbox will normally start in 2nd gear on full throttle in the lever position 3 and 4. First gear can either be selected by kick-down or by placing the gear lever in position 2, in which the car will start in 1st gear on part throttle.

As the engine speed rises during acceleration torque of the rear wheels dropes off and only a part of adhesion is used for driving the car. The other part of the adhesion is applicable to producing side thrust if necessary. The car then becomes stable. The higher the speed of the car the larger the amount of adhesion becomes available. Diagram No. 10 shows the actual power of the 6.3 in relation to speed and the power which could be transmitted by the driven wheels on the limit of adhesion assuming the coefficient of friction is 1,0 irrespective of speed. It will be noted that the adhesion is only marginal for the power of the 6.3 engine in 1st gear. In all other gears power far in excess could be transmitted. The diagram also shows that a hypothetical 300 SEL - 6.3 with front wheel drive and a favourable weight distribution of F 60 % / R 40% would not be able to handle the full power of the 6.3 engine in 1st gear. But that in all other gears ample reserve would be available. Diagram No. 11 finally indicates the maximum speeds of a theoretical 300 SEL with tremendous power. The speeds are limited by adhesion of the driving wheels on one side and the rolling and air resistance on the other side.

It should be realized that although by present standards the 6.3 is a very powerfull car, it has nothing like the power adhesion would tolerate, thus providing ample safety against skidding. A wide field for further development is waiting for the enthusiast!

Diagram No. 12 shows the actual acceleration (g) of the 300 SEL - 6.3 in relation to the theoretically obtainable at the limit of adhesion.

Diagram No. 13 shows the percentage of adhesion used for accelerating the 300 SEL - 6.3.

As pointed out above the part of adhesion not used for acceleration can be applied to generate side thrust in cornering. Tests on the skid pad have proven that the coefficient of friction of the tires in combination with the 300 SEL - 6.3 in cornering amounts to about 0,8 against 1,0 in straight line driving thrust.

Diagram No. 14 shows the percentage of the 0,8 cornering friction coefficient, that can be applied during acceleration under full power in relation to speed. The minimum radius of a bend which can be negotiated under these conditions is also indicated.

Diagram No. 15 shows distances during acceleration.

The adhesion of the front wheels for steering the car is ample under all conditions as the effect of rear wheel torque and aerodynamic lift are unsignificant, see Diagram No. 16.

Naturally on wet surfaces more cautions driving is advisable. But it is quite surprising how well the car behaves under such conditions. This must be mainly attributed to the type of tire used. Broad tread radial tires as used on the 6.3 are inherently better on wet surfaces than conventional cross ply tires. Favourable wear characteristics of the radial tire permit the use of a well cut up tread giving superior drainage of water and the choice of relatively soft rubber with a good coefficient of friction.

Brakes

A car is as safe as its brakes. For this reason we have strived to make the brakes of the 300 SEL 6.3 as safe and efficient as the present technical know-how will possibly permit.

If a vehicle is easily capable of reaching a speed of 140 mph, then it must also be possible to brake safely from this speed, and the braking distance must, of course, be as short as is physically possible.

In our vehicles the brakes are therefore so designed that they remain fully efficient, even when applied several times in quick succession. The length of the braking distance is thus solely dependent on the friction value of the tires to the road. Naturally the tires themselves are also an important factor.

Diagram No. 17 shows the dependency of the braking distance to speed, with the road surface and tire quality as parameters. There was no difference, whether the brakes were cold or whether they had already been warmed up.

As can also be seen from the diagram, the mean deceleration was largely independent of the speed. This can be understood from the above.

Under favourable conditions, in relation to load, road surface and tires, it may be possible to improve on these values with some vehicles. However these would only be more or less freak values, whereas the data, given in this paper, may be reproduced any time.

The great stability of the brakes of the 6.3 and the practically non-existent fading, obviously presupposes intensive cooperation with the lining manufacturers. In close collaboration with Daimler-Benz, the firm of Textar succeeded in developing a lining which is distinguished by an extraordinary constancy of the friction value, even at high temperatures.

In Fig. 18 the friction values of this lining are illustrated, which were measured on a test stand specifically constructed for this purpose. The temperature and friction value is plotted against the time. Measuring is carried out as follows:

The brake disc rotates at a constant speed. Via a time switch a fixed brake pressure is applied for 5 secs. Then the brake pressure is released for 10 secs. Subsequently the brakes ware applied again for 5 secs. etc.

The brake torque is measured electrically with a hub-mounted torque meter, and is recorded as friction value in a Moseley recorder. The same happens with the temperature which is determined in the centre of the disc by means of a thermocouple element.

It can be seen from the diagram that the friction value of approx. 0.4, when cold, increases slightly as the temperature rises, reaching its maximum with $\mu = 0.45$ at approx. 750^{o}F disc temperature. It then drops slightly to $\mu = 0.37$. This value is maintained up to the highest temperatures.

These highly favourable characteristics are, of course, not obtained without some drawbacks. Under certain conditions the lining tends to squeak, which does not impair its efficiency but is unpleasant. Braking at high disc temperatures obviously also greatly affects the wear on the brake linings. In Fig. 19 we have compared the ratio of the wear of the pads on discs with various temperatures at the beginning of the brake test to the wear on disc which was initially cold. Braking was in each case carried out from approx. 75 mph with a brake line pressure of approx. 860 psi. The maximum disc temperatures were naturally far higher at the end of the brake test than is indicated here. This will be dealt with later.

It can be seen from the figure, that the wear at a disc starting temperature of 932^{o}F,which is only encountered in fast driving, reaches about 13 times the wear at 212^{o}F, which corresponds to a more normal driving style.

Some of the usual American linings, which are developed mainly with smooth operation and quiet running in mind, could not withstand the stress imposed in our test, and showed extreme fading at temperatures above 752^{o}F, so that our normal test programs could not be carried out.

Naturally we did not limit our investigations to linings which are made of organic materials but we also investigated metal-sintered linings. These linings are certainly quite insensitive to temperature, but have the disadvantage of being good conductors of heat, which means that a large amount of the heat brought about by the friction is taken up by the brake lining and not the brake disc. Even with intermediate layers of heat insulating material, it has so far not been possible to prevent the rubber seals and brake fluid from becoming overheated.

Fig. 20 shows that at a disc temperature of 1112°F and with sintered linings, brake fluid temperatures are already at a dangerous level, which is not the case with organic linings. For this reason and according to the present-day state of technology, sintered linings cannot be used for disc brakes with normal calipers. However, for drum brakes in which the temperature of the brake shoes cannot be directly transferred to the brake fluid, they represent considerable progress.

In order to obtain optimum braking effect, we have tried to distribute the brake force on the front and rear axle, so that they conform as much as possible to the so-called "ideal curve". The centre of gravity in the longitudinal direction of the vehicle, the height of the centre of gravity over the road and the wheel base have to be into consideration. We use a rear brake line pressure regulator which, in case of the 6.3, is designed for a change-over pressure of 418 psi. After the check point the brake pressure for the rear brakes increases with a ratio of only 0.45. From Fig. 21 can be seen to which extent the brake line pressures approximate the ideal distribution. For comparison purposes we have also recorded how the distribution would be without a pressure regulator. We find that the vehicle would remain stable up to deceleration of approx. 0.6 only. At higher deceleration the vehicle is likely to veer violently off course owing to locking of the rear brakes. On the other hand, when decelerating in the region between 0.3 and 0.4 a considerable amount of usable brake pressure is lost. Fixed brake tuning for this vehicle is therefore out of the question.

We have built our vehicle in such a way that all its parts are comfortable and easy to operate. This also applies to the brakes. Diagram 22 shows the relation of application effort to braking effect for the service brake and parking brake. It can be seen from the diagram that the adhesion limit was already reached with a deceleration of 0.83 and at an operating force of 48 lbs. During this test the vehicle was loaded to the permissible gross vehicle weight.

These exceptionally favourable values were obtained by the installation of a 9" Double-diaphragm brake booster, the characteristics of which are shown in diagram 23. The brake line pressure is presented, in one case, as being dependent on the foot pressure and in the other, on the travel of the pedal.

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One can see from the diagram that the adhesion limit, which is reached at a foot pressure of 48 lbs only requires an oil pressure of approx. 1,530 psi, whereas the limit of the booster is approx. 2,000 psi. There is therefore a large reserve available, which is advantageous at low vacuum conditions in the intake manifold at high altitudes.

In a fast vehicle the brakes must respond quickly, for only then precise matching of brake power to adhesion of the tires is possible. For this reason we have endeavoured to cut down the response times of the booster. The extent to which we have been successful is shown in Fig. 24.

After pointing out the prerequisites for the correct operation of the brake, the following shows how the driving manner affects the stress on the brakes and their service life. Here it is also plain, as always, that high performance must be paid for in some way. In order to give an idea of the brake power produced, we have compiled the way in which the brake power is connected with the speed and deceleration, when there is one person in the vehicle Diagramm 25. With greater total weight (therefore heavier load) these values must be increased in proportion to the weight. The unbroken lines refer to one front wheel, the broken lines to one rear wheel.

It can be seen immediately that the momentary brake power shows linear increase with the speed, and that it greatly increases with mounting deceleration. In actual brake tests the limit of adhesion of the wheels is reached at a deceleration rate of approx. 0.8. If the brakes are applied for example at 125 mph with this deceleration, then the surface of the front wheel brake disc (at this speed)will produce a brake power of approx. 370 HP. This naturally results in a sharp temperature increase on the surface and great thermal stress.

In the course of our experiments it became evident that thermal stresses were at their greatest, when the brakes are briefly applied at high deceleration. The surface of the disc then becomes extremely hot, whilst the centre of the disc still remains cold. The surface is therefore actually chilled by the inner disc, which in time may result in heat cracks.

The conditions are somewhat different when braking to a halt. In fig. 26 the temperature curve is plotted, when stopping from 125 mph. The temperature profiles at 113 mph and at 50 mph are also recorded in the figure. At the same time these correspond approximately to the stress pattern.

The curve of the mean disc temperature is indicated as a continuous line, the surface temperature as a dot-dash line and the temperature in the disc centre as a broken line. From this it is possible to see that at the start of deceleration the surface temperature rises rapidly, whereas the disc centre is still unaffected. In this range the thermal stresses in the disc increase rapidly. As speed decreases the temperature differences decline and disappear completely when the vehicle comes to a halt. At the prevailing temperature of approx. 1162° F heat stresses are quickly assimilated so that the disc is not endangered.

In my paper in March 1966, I already showed how difficult it is to construct ventilated brake discs which do not fracture. In the meantime we have carried out extensive investigations which have led to improvements in the design and to less distortion of the disc. The stresses occuring have consequently been considerably reduced. Several seemingly insignificant construction details had important effects. The material has also been improved, and it is interesting to note that a special cast iron with a low coefficient of elasticity and good heat conduction ability produced the best results.

Development was made possible by the construction of a measuring instrument which enabled the distortion of a disc to be measured continuously during braking. In this experiment it is particularly interesting to note to which extent the disc forms a conical curvature. Fig. 27 shows the test readings of two discs, Fig. 28 their cross section, and Fig. 29 the temperatures.

It can be seen from the figures that the distortion differential of disc 178 only amounts to 0.34 mm, whilst disc 177 reaches 0.6 mm. It was finally possible to increase the crack resistance of the disc sufficiently so that installation could be justify.

Extensive endurance tests have proven that now a mean disc temperature of 1562°F is tolerable. The temperature of the brake fluid should not exceed 340°F.

The cooling of a brake is of great importance for its working capacity. Diagram 30 shows that the heat emission capacity is a specific function not only of the disc temperature, but also of the vehicle speed. At 1112°F and 125 mph approx. 13 hp can be dissipated continuously. These values refer to closed wheels as used for our vehicles in serial production. The cooling can be further improved by the use wheels with a ventilation effect.

Investigations of brakes on special test equipment naturally can only produce relative data and are valuable for comparing different devices. However it is most important to know what actually happens during practical application of the brakes in a car.

Diagram 31 shows the temperature behaviour of the 300 SEL 6.3 at fast speed on the autobahn. At intervals of approx. every 2 min. the vehicle was braked from 112 mph to 37.5 mph with a deceleration of 0.4 g. Subsequently it was accelerated to 112 mph, and then the brakes were applied again. A "state of equilibrium" was evident after the third deceleration. The maximum disc temperature was well below 752° F. The brake fluid temperatures attained a maximum of 185° F. This manner of brake application would certainly exceed that encountered during hard but sensible driving on express highways with unrestricted speed. However, it must be appreciated that whereas a crash stop from 140 mph would not overstress the brakes successive tests of this sort would definitely cause overheating and destruction.

In Germany there are various down-hill routes, where particularly high brake stress occurs, if the car is being driven hard. The names of these descents namely "Zuflucht" ("Refuge") and "Notschrei" ("Cry of distress") indicate their severeness. On both these down-hill runs sharp bends alternate with clear, straight stretches which simply ask for swift acceleration.

Diagram 32 shows the temperature behaviour in the brakes of a 300 SEL 6.3, when driving down the "Zuflucht" ("Refuge"). The temperature of the right, front wheel brake disc rises to approx. 1232°F, whereas the highest brake oil temperature is 230° F. Even when travelling down this hill, completely safe temperatures were not exceeded.

Finally Diagram 33 shows the pattern of the run down "Notschrei" ("Cry of distress"), which was carried out under the most severe driving conditions - almost like racing. (At only slightly slower speeds the temperatures are considerably lower.) Here the temperature of the right front brake disc rises to a maximum of 1352° F, thus remaining well below the permissible temperature of 1562° F. The brake fluid temperature is beginning to become marginal at 292° F, only approx. 77° F below the permissible value.

So although the brakes of the 300 SEL 6.3 are extraordinarily efficient on mountainous roads, they naturally have their limitations.

In order to determine the brake torque and brake power occuring during normal operation of the car, a brake torque indicator was attached to a front wheel brake. Readings of torque and accompanying speed were recorded on tape.

The test took place on relatively well constructed, but winding and mountainous roads through the Black Forest. The car was driven enthusiastically and fast, but not dangerously. The average speed on these roads lay between 37.5 mph and 49 mph. Fig. 34 shows the analysis of the brake torque. By far the most braking took place with a brake torque of less than 145 ft lb and only very little braking was carried out with more than 650 ft lb, corresponding to approx. 0.5 g deceleration.

The curve of the brake horse power frequency is particularly interesting. Fig. 35 indicates that a clear frequency maximum of 23 % of all brake horse power peaks lies in the range of approx. 30 hp brake power. According to our observations so far - which at the moment are being further extended by large scale testing - it appears that a particular driver normally brakes with a certain rate of deceleration and brake horse power as long as the traffic does not impose another course of action.

The number of times the brakes were applied per mile of road was 2.94. Total braking time amounted to 9.1 % of the driving time.

The results given here are most assuring, since they furnish clear evidence of the occuring brake stresses and the brake performance. The measurements have proven even during very hard driving, the brakes still have sufficient reserve. At any time and at all speeds a full stop utilizing the entire adhesion of the tires is possible thus affording upmost safety.

Tires

As on all cars the tires have a very pronounced influence on the behaviour of the 6.3. Owing to the high speed of the car and the desire for comfort the demands on the tire are very exacting, and it is here where the most compromises had to be made.

Putting safety at the top of all requirements the qualities of the tire in the order of their importance should be arranged as follows

- durability at high speed
- adhesion
- handling
- comfort
- tread life
- price

Durability.

Long tests have shown that a radial ply tire is more suitable for continuous high speeds than a conventional cross ply tire as it generates less heat and provides more comfort at the necessary high pressures. Fig. 36 shows the build up of the selected tire which at the present time are supplied by Dunlop and Continental.

In contrast to conventional cross ply tires radial tires do not have more than 2 carcass plys. These plys are made of nylon cord and are crossed slightly at 76° on this particular tire instead of 90° as on pure radials. This angle provides better lateral stability without materially reducing radial flexibility. At the present state of development polyester cords cannot be used as the adhesion of rubber to polyester is not quite adequate at the tire temperatures that can be reached during prolonged high speed operation under heavy load and high ambient temperature. The 4 breaker plys are made of rayon cord and have an angle to the crown line of 14°.

The tires have a super low cross section and are mounted on 6 1/2" rims.

Fig. 37 shows the tire pressures which should be adhered to religiously. It will be noted that the pressure build up owing to speed is moderate, s sign of low heat generation.

If overheating of the tire, due to driving at speed with low air pressure, should occur then the design guarantees that the tread will be thrown off without the tire loosing air and disintrigating. Wear resistance of the naked cords is sufficient for the car to be safely stopped from top speed. Violant breaking should naturally be avoided.

Adhesion.

The rigidity of the breaker plys of the radial tires eliminates scuffing of the tread whilst in rolling contact with the roadsurface thus reducing wear very considerately. This is a well known and generally accepted fact.

Consequently a well chopped up tread design can be adopted offering good drainage of water. And in consequence of this relatively superior grip of the tire on wet surfaces and freedom of aqua planing is obtained.

Furthermore a comparatively soft rubber compound,having good natural adhesion, can be used without unduly increasing wear. The wide 6 1/2" rim and the broad tread of the tire contribute materially to adhesion.

Handling.

High cornering force at relatively low slip angles and lateral stability is necessary to provide sports car type handling and feel.

The characteristic side force curves were obtained on a new type of test rig shown in fig. 38. Here the tire runs on the inside of a very large drum with a more natural surface contact than it is the case on the more normal peripheral run on a smaller drum.

Fig. 39 shows the cornering force in relation to the slip angle of the radial tire against an equivalent conventional tire. It is significant that at extreme slip angles the conventional tire provides higher cornering force. But at lower slip angles which occur in normal fast driving, a required side force is provided at a smaller slip angle by the radial tire.

Fig. 40 shows the selfaligning torque of the 2 types of tires.

Comfort.

Using the tire pressures necessary for high speed work, the radial tires have greater flexibility than conventional tires, and consequently they provide a higher degree of comfort.

If the car is not going to be driven fast, the tire pressures can be reduced according to the recommendations in the drivers manual. This is advisable in countries with speed limits. Where no limits exist and long express highways are available, low tire pressures are of rather dubious value.

Unfortunately nylon cord tires still develop flat spots during rest after prolonged running. In this respect radials are superior to conventional tires as the flat spot is shallower and disappears faster as may be taken from Fig. 41.

The test procedure is as follows

1) 3o min. run on drum under load at 93 mph.
2) 16 houres cool down under load on flat surface. Measuring flat spot.
3) Run under load at 31 mph on drum. Measuring flat spot after 5, 15, 3o and 6o min.

In practical driving of the car a few minutes of unconveniance after starting is unavoidable.

Bad tire uniformity (Fig. 42) can have a very unfavourable effect on comfort as vibration of the whole car may be exited at certain critical speeds. This malfunction cannot be eliminated by balancing. Great care in manufacturing the tires must be taken and undesirable tires exceeding the acknowledged limits of uniformity must be eliminated by rigorous inspection.

Price.

It is unavoidable that tires meeting the required qualifications are comparatively expensive.

The chassis of the 300 SEL which was introduced to the press in USA in March 1966 on the Riverside Race Course has been taken over unaltered for the 6.3. It proved to be able to handle the increased torque of the engine and the higher speed with ease. A detailed description of the chassis doesn't appear to be necessary here as all interesting items have already been published. Neverless, I would like to quote the main characteristics of the chassis. The 6.3 is trimmed to have slight understeer thus avoiding rear end break away and plowing out during fast cornering under power. A relatively direct steering ratio (2 1/2 turns from lock to lock) and responsive tires provide high manoevrability. The behaviour of the car is predictable under all conditions thus imposing no high demands on the drivers ability.

Needless to say, the 6.3 meets all U.S. safety and exhaust emission laws.

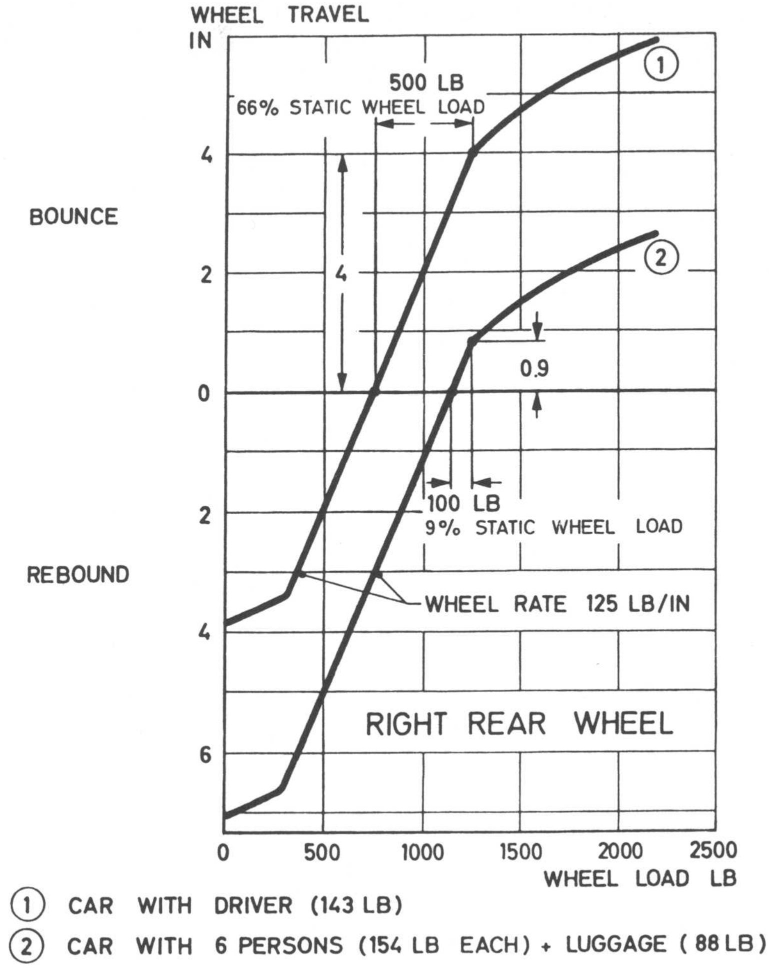


FIG. 3

U.S. CAR 1968
STEEL SPRING

6.68/5603

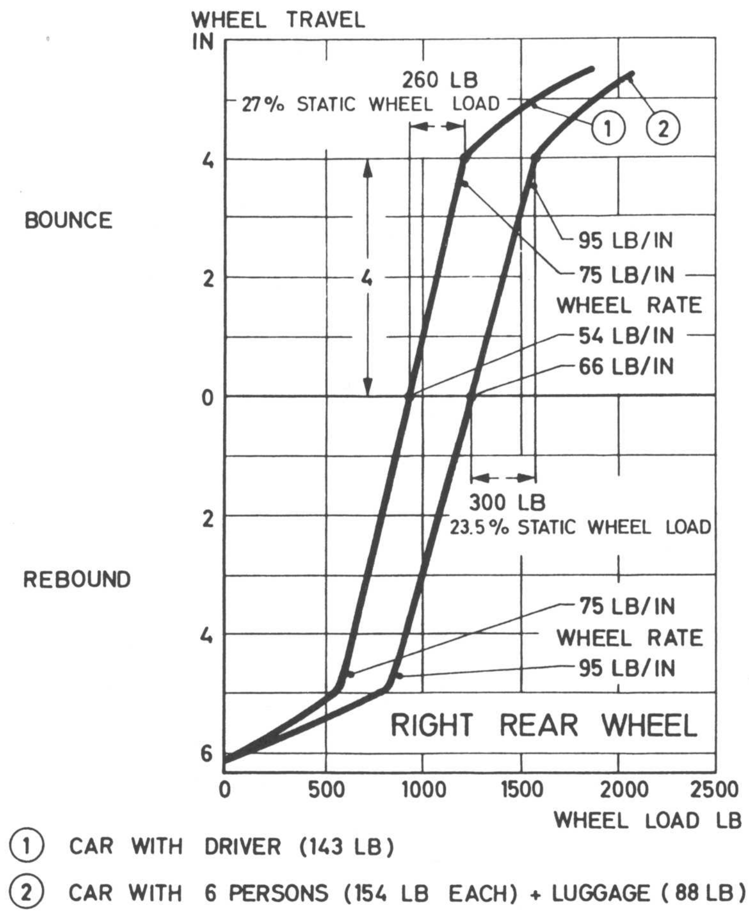


FIG. 4

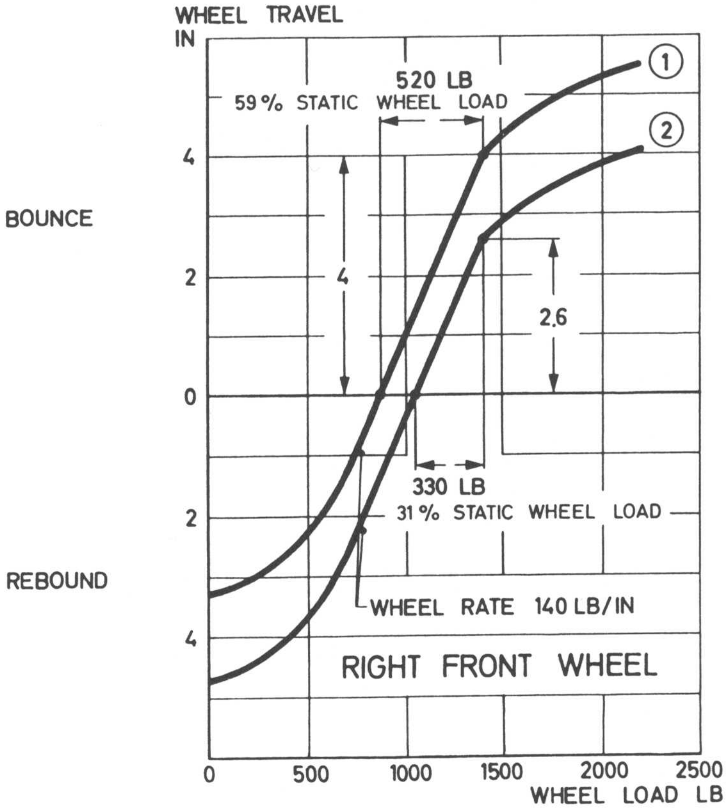
300 SEL-6.3
AIR SPRING

6.68/5601

FIG. 5

U.S. CAR 1968
STEEL SPRING

6.68/5602



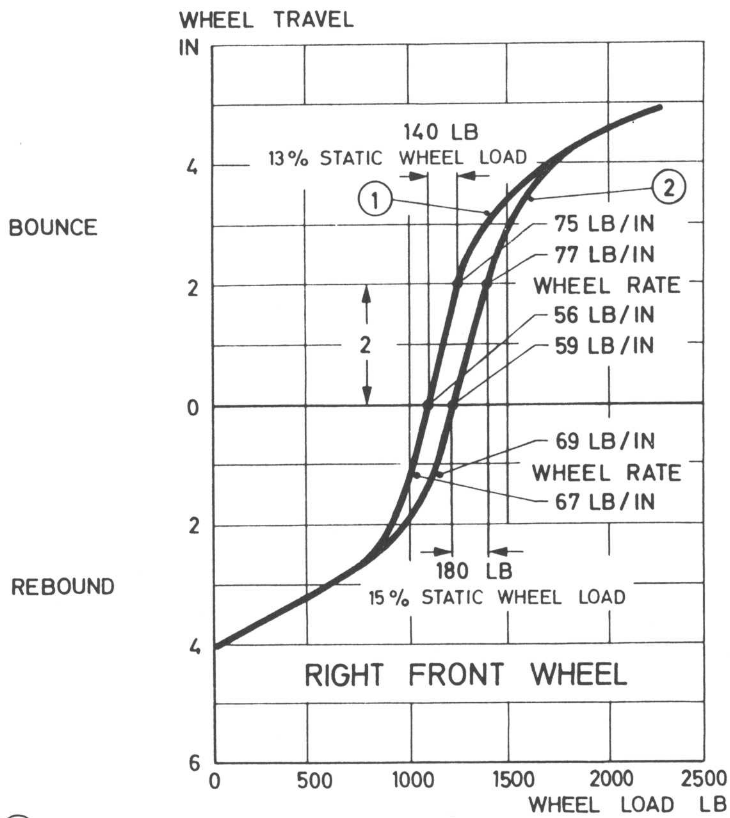
(1) CAR WITH DRIVER (143 LB)

(2) CAR WITH 6 PERSONS (154 LB EACH) + LUGGAGE (88 LB)

FIG. 6

300 SEL-6.3
AIR SPRING

6.68/5600

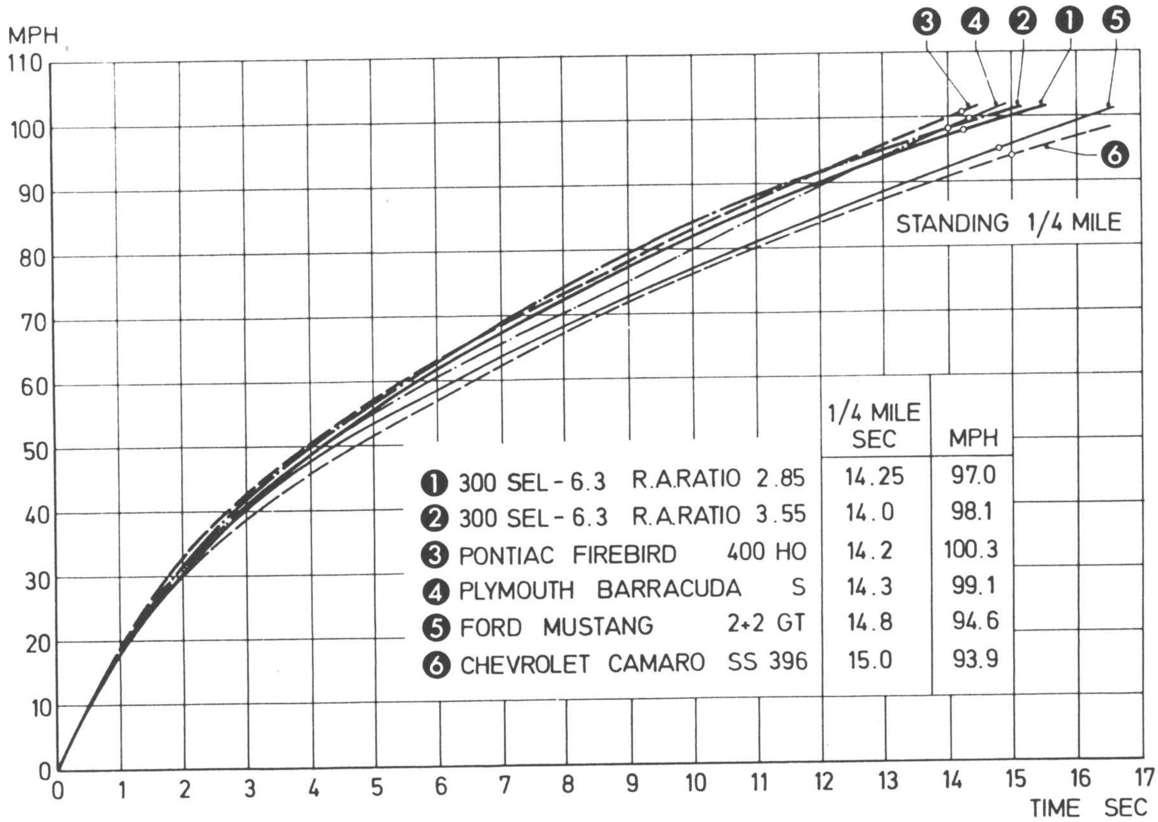


(1) CAR WITH DRIVER (143 LB)

(2) CAR WITH 6 PERSONS (154 LB EACH) + LUGGAGE (88 LB)

	300SEL-6.3 No. 1	300SEL-6.3 No. 2	PONTIAC FIREBIRD 400 HO	PLYMOUTH BARRACUDA FORMULA S	FORD MUSTANG 2+2 GT
ENGINE CAPACITY	6.3	6.3	6.6	5.6	6.5
FINAL DRIVE RATIO	2.85	2.85	3.55	3.55	3.25
CURB WEIGHT LBS	4070	4070	3550	3330	3546
TIME FOR 1/4 MILE SEC	14.4	14.25	14.2	14.3	14.8
TERMINAL SPEED AFTER 1/4 MILE MPH	93.8	97	100.3	99.1	94.6
TIME 0-60 MILES SEC	5.8	5.7	5.5	5.9	6.3
TIME 0-100 MILES SEC	15.9	15.0	14.1	14.5	16.3
TOP SPEED MPH	141 OBSERVED	144 OBSERVED	110 OBSERVED	125 ESTIMATED	127 ESTIMATED
ENGINE SPEED AT TOP CAR SPEED RPM	5410	5530	5180	6060	5460

FIG.7 — PERFORMANCE OF 300 SEL-6.3 AND U.S. SPORTY CARS — 6.68/5581



		1/4 MILE SEC	MPH
1	300 SEL-6.3 R.A.RATIO 2.85	14.25	97.0
2	300 SEL-6.3 R.A.RATIO 3.55	14.0	98.1
3	PONTIAC FIREBIRD 400 HO	14.2	100.3
4	PLYMOUTH BARRACUDA S	14.3	99.1
5	FORD MUSTANG 2+2 GT	14.8	94.6
6	CHEVROLET CAMARO SS 396	15.0	93.9

FIG. 8 — ACCELERATION THROUGH THE GEARS — 6.68/5608

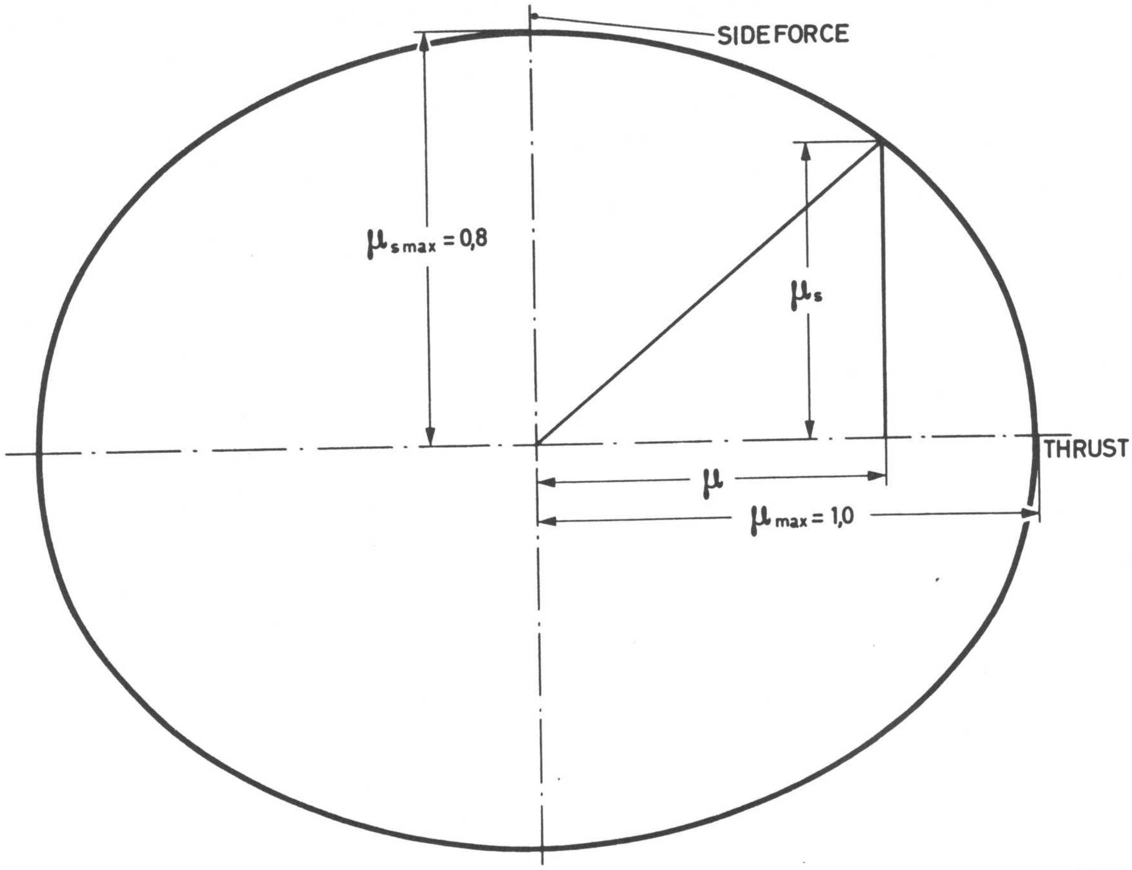


FIG.9	RELATION BETWEEN THRUST AND SIDEFORCE	6.68/5610

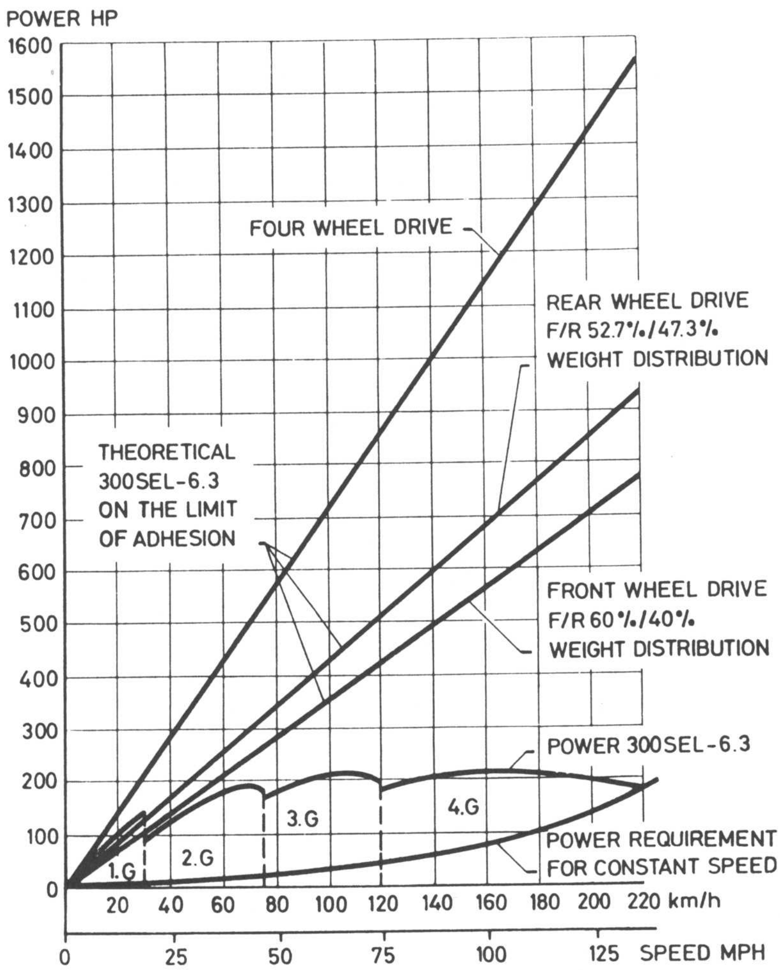


FIG. 10

POWER REQUIREMENT FOR ACCELERATION

6.68/5586

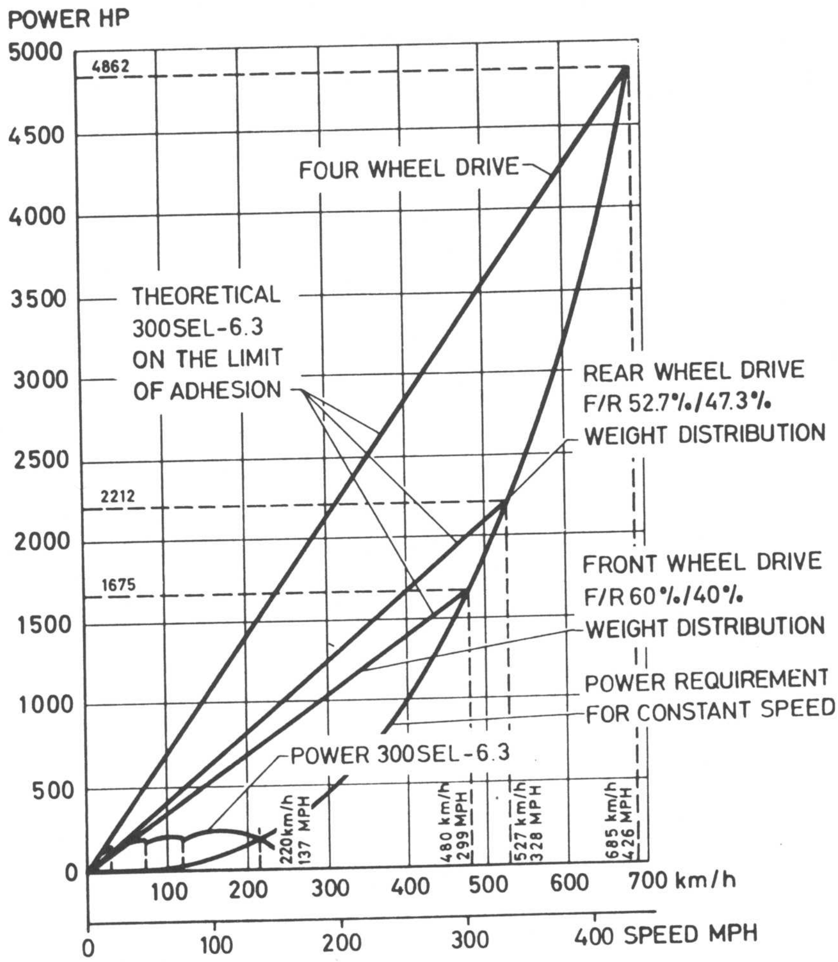


FIG. 11

POWER REQUIREMENT FOR ACCELERATION

6.68/5587

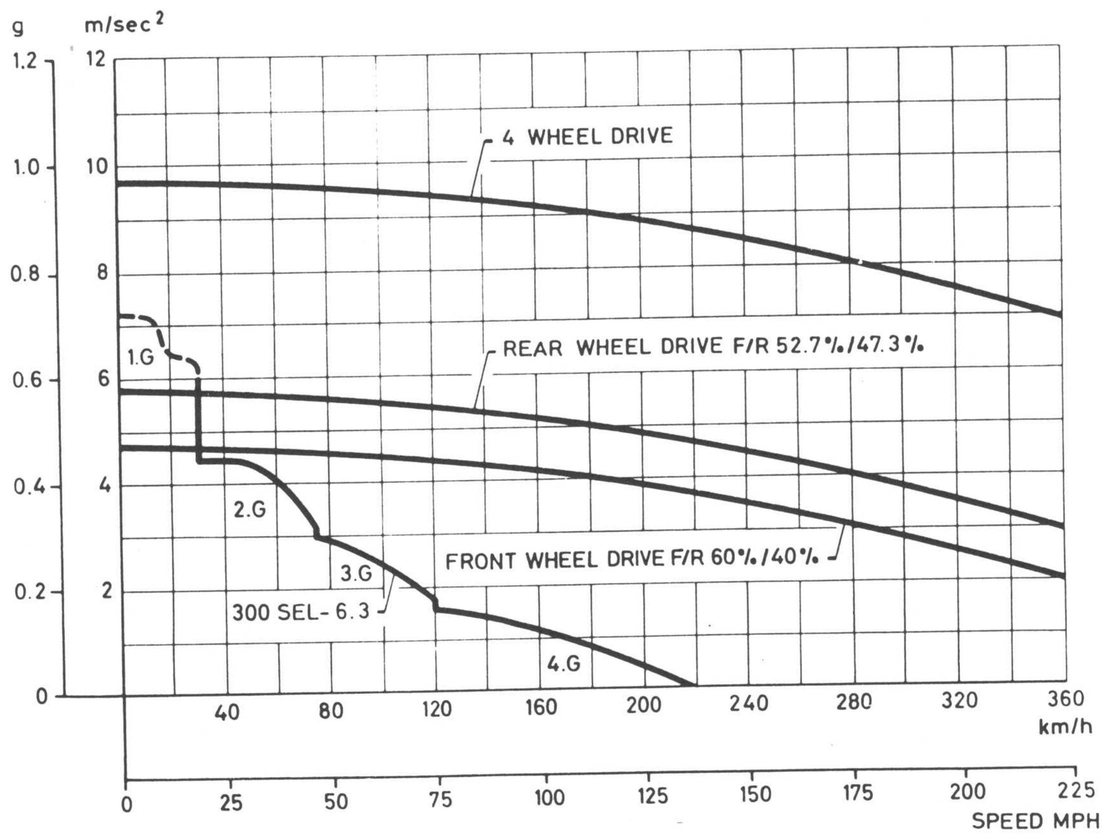


FIG. 12

ACTUAL ACCELERATION (g) OF THE 300 SEL-6.3 IN RELATION TO THE OBTAINABLE ACCELERATION AT THE LIMIT OF ADHESION

6.68/5588

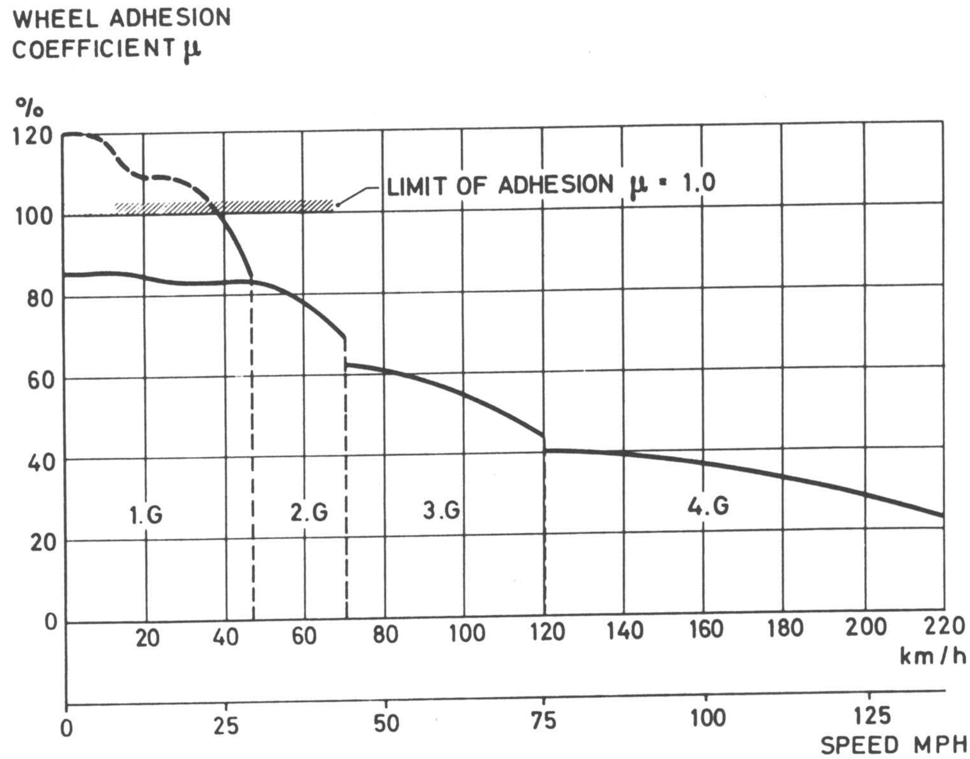


FIG. 13	300 SEL-6.3 EXPLAITATION OF WHEEL ADHESION μ DURING ACCELERATION (DRY ROAD)	6.68/5609

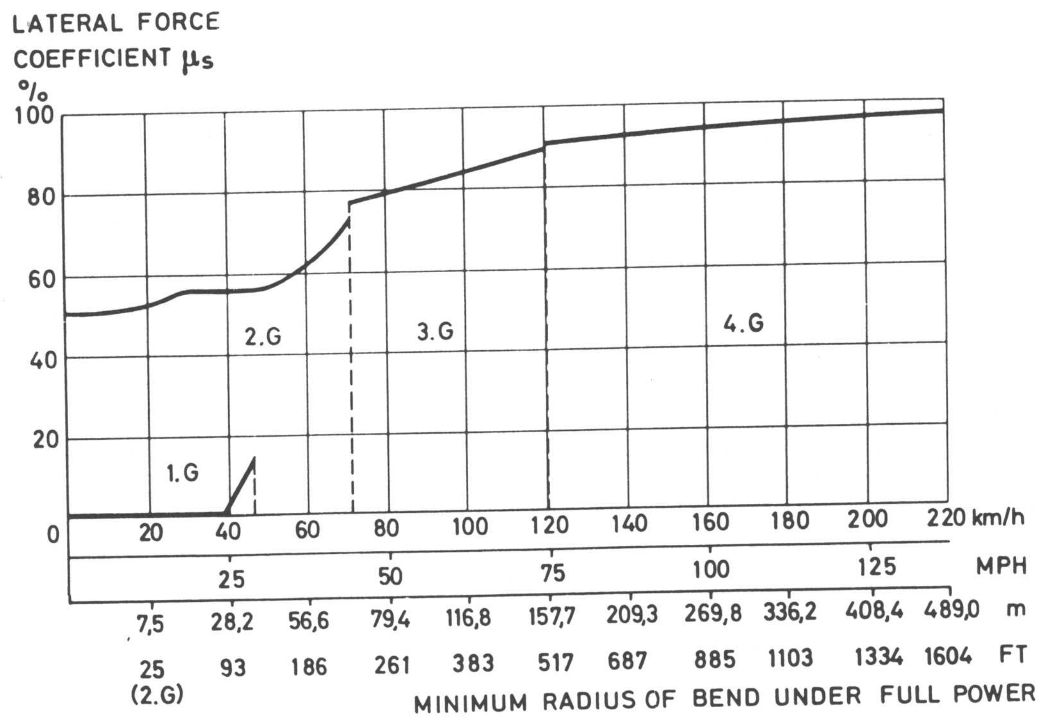


FIG. 14	300 SEL-6.3 EXPLOITABLE LATERAL FORCE μs DURING ACCELERATION (DRY ROAD)	6.68/5612

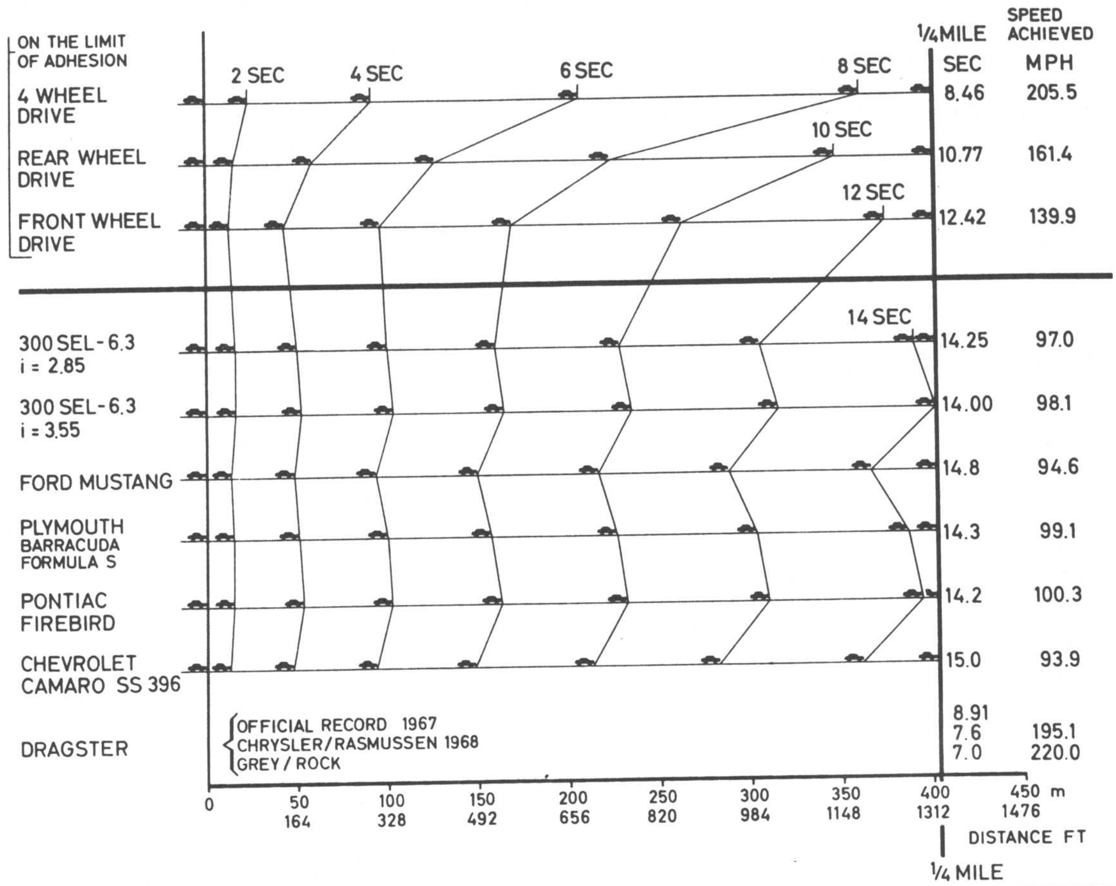


FIG. 15	DISTANCES DURING ACCELERATION	6.68/5611

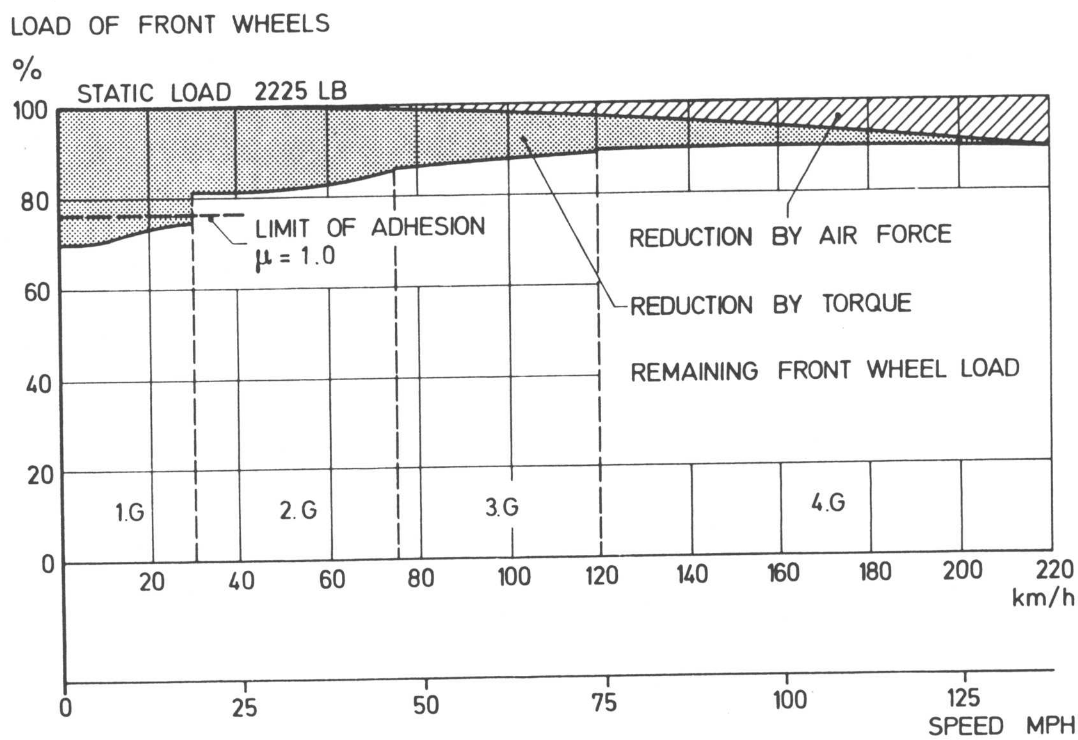


FIG. 16	300 SEL-6.3 (REDUCTION OF ADHESION OF THE FRONT WHEELS BY TORQUE AND AIRFORCE)	6.68/5607

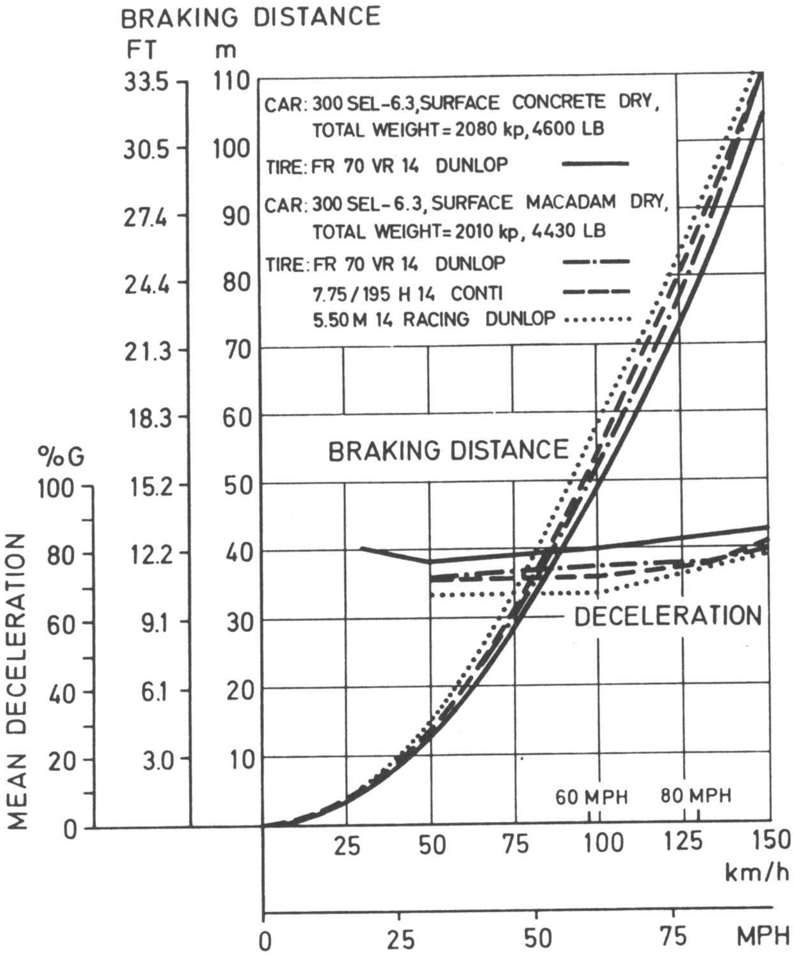


FIG.17

BRAKING DISTANCE IN RELATION TO SPEED WITH DIFFERENT TYPES OF TIRES AND ROAD SURFACES

6.68/5571

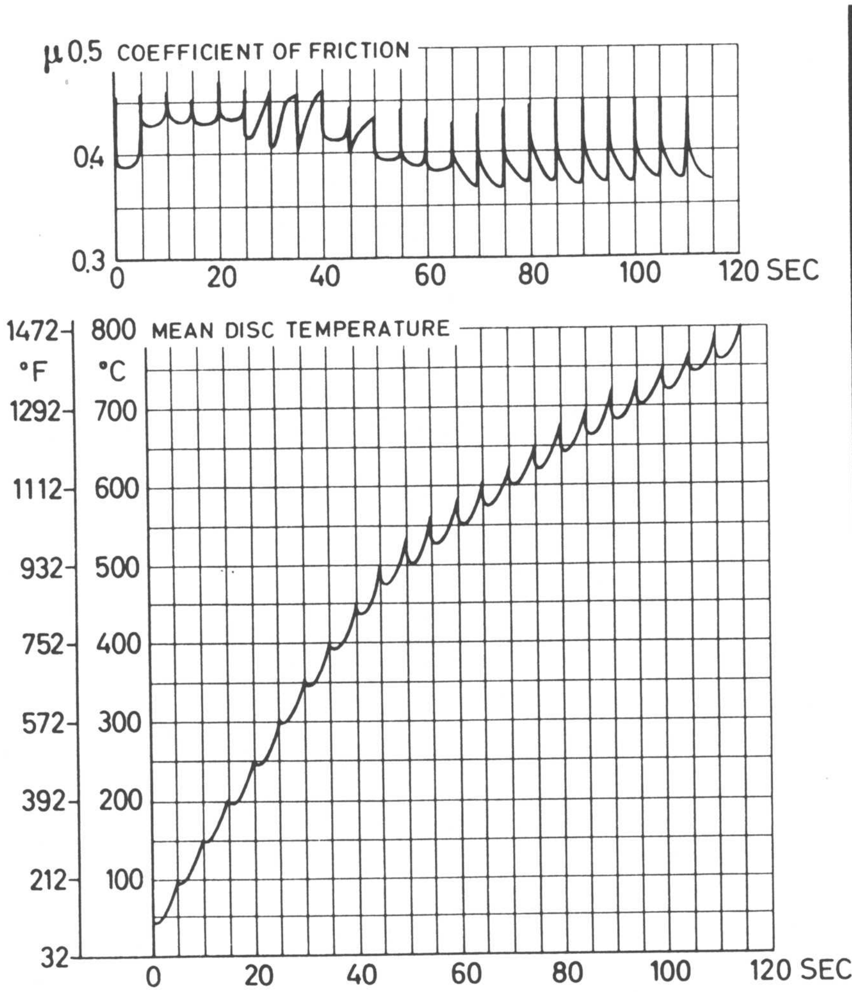


FIG. 18

TEST OF COEFFICIENT OF FRICTION OF BRAKE PAD MATERIAL

6.68/5583

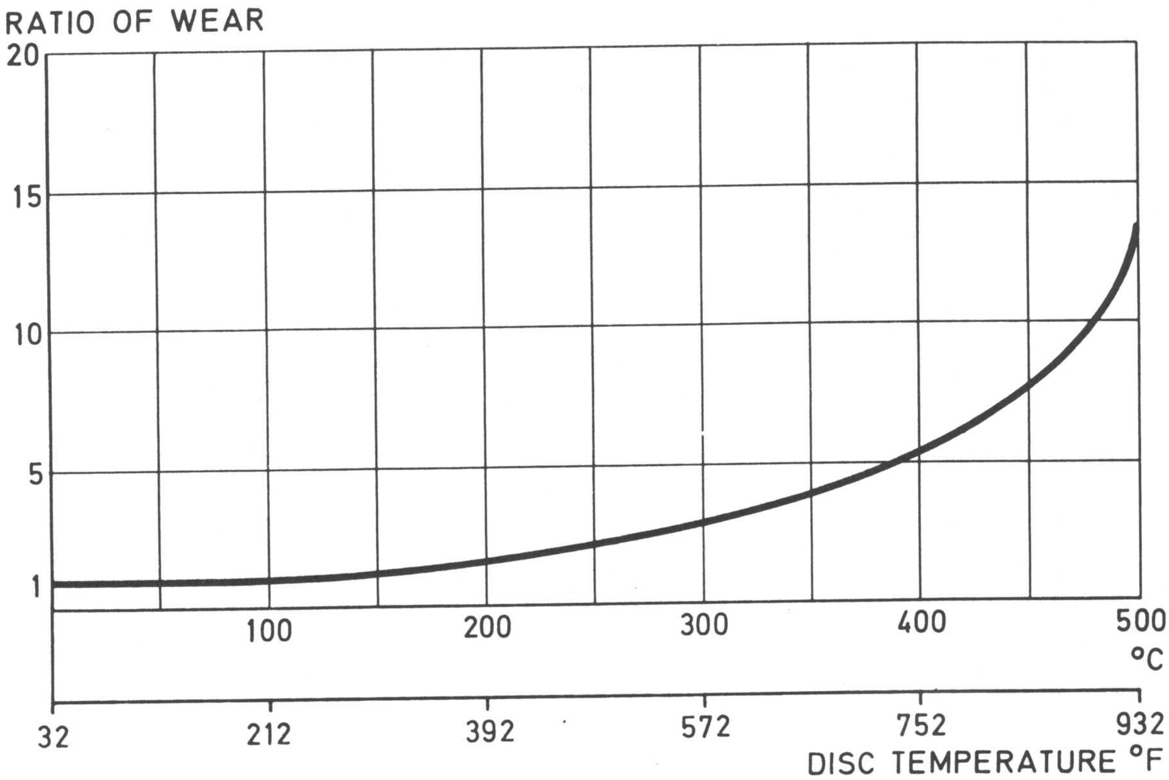


FIG. 19	WEAR OF THE BRAKE PAD AT HIGHER TEMPERATURES OF THE DISC IN RELATION TO THE WEAR WITH A COLD DISC	6.68/5574

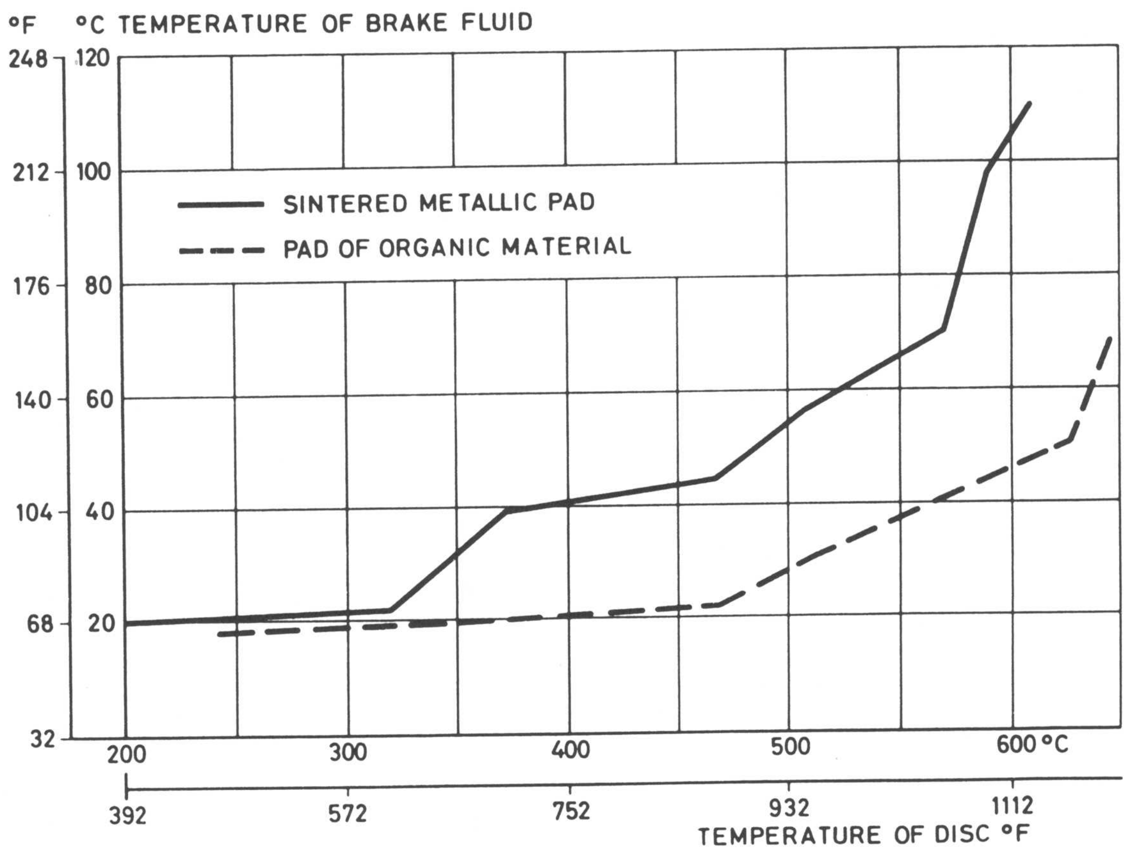


FIG. 20	TEMPERATURE OF THE BRAKE FLUID IN THE FRONT WHEEL CALIPER DURING FADING TESTS	6.68/5584

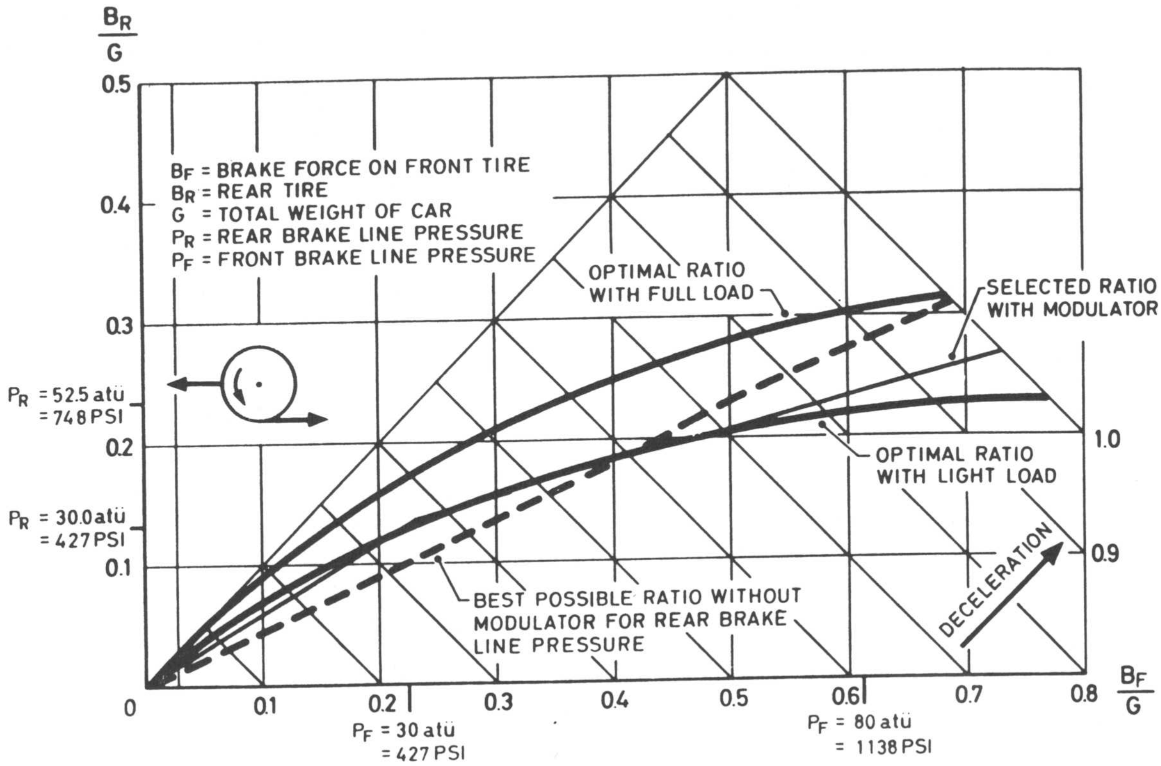


FIG.21 | DISTRIBUTION OF BRAKE FORCE 300 SEL-6.3 | 6.68/5572

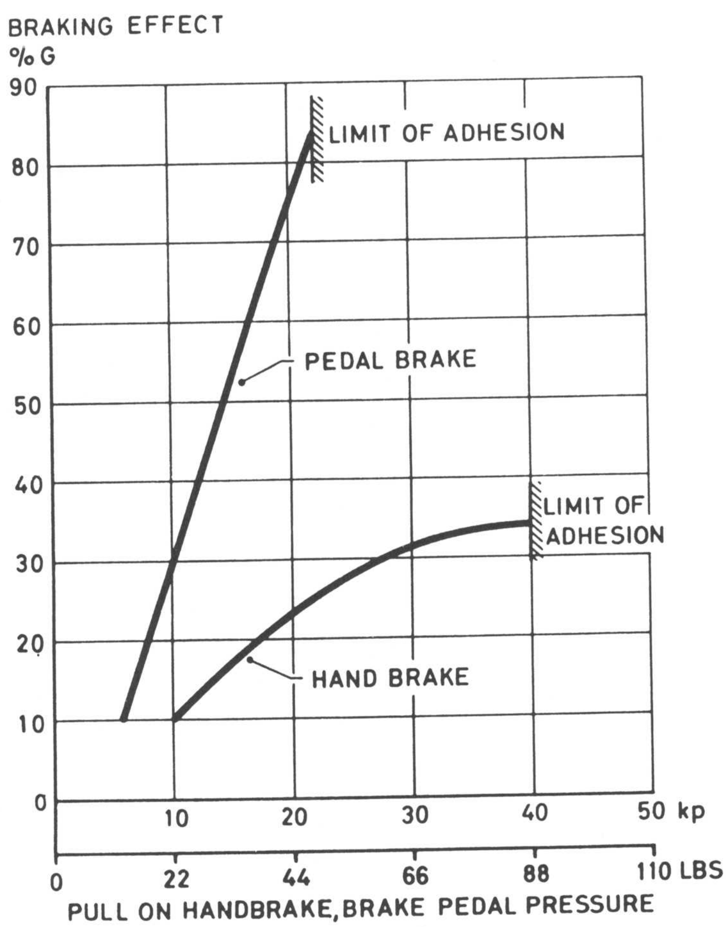
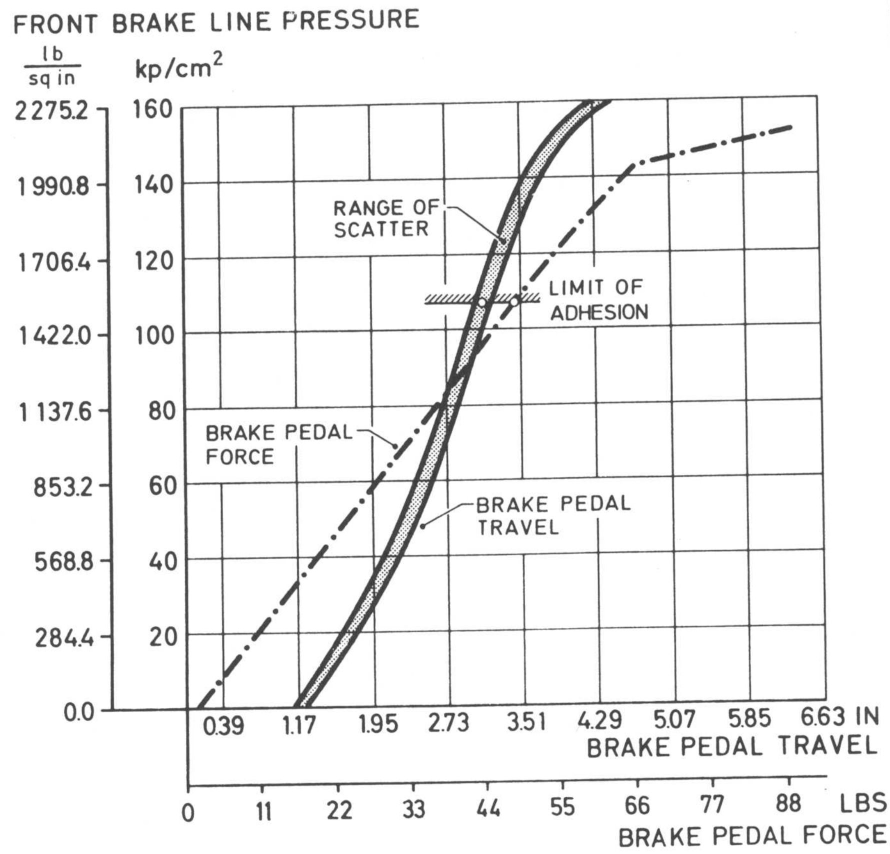


FIG. 22

BRAKE EFFECT IN RELATION TO APPLIED FORCE

6.68/5573

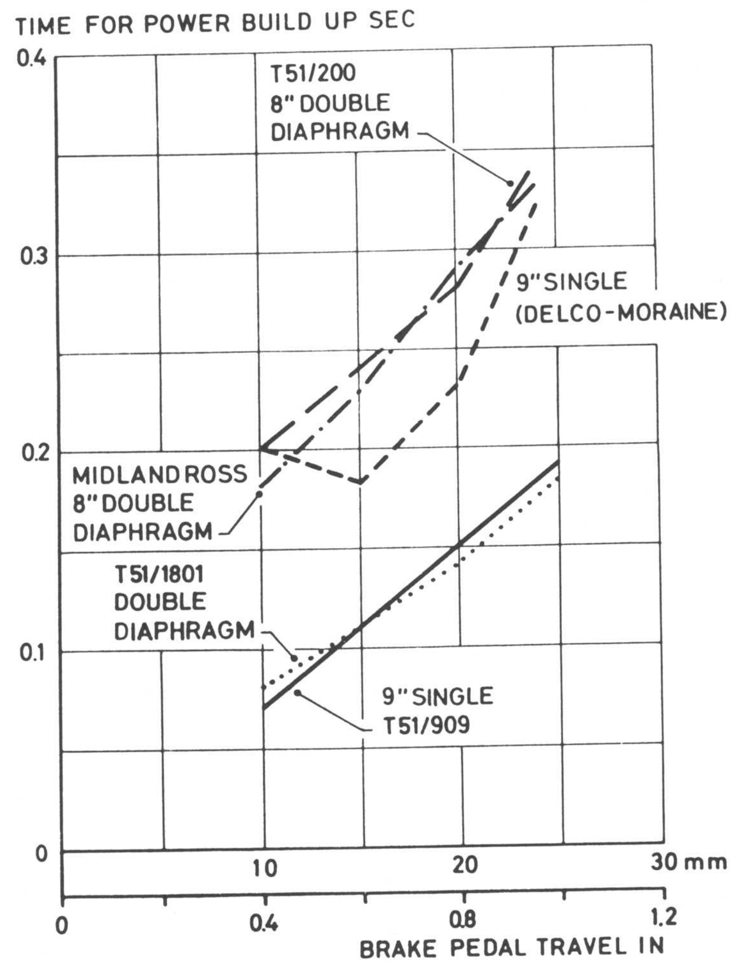
FIG. 23



BRAKE LINE PRESSURE IN RELATION TO PEDAL FORCE AND PEDAL TRAVEL

6.68/5578

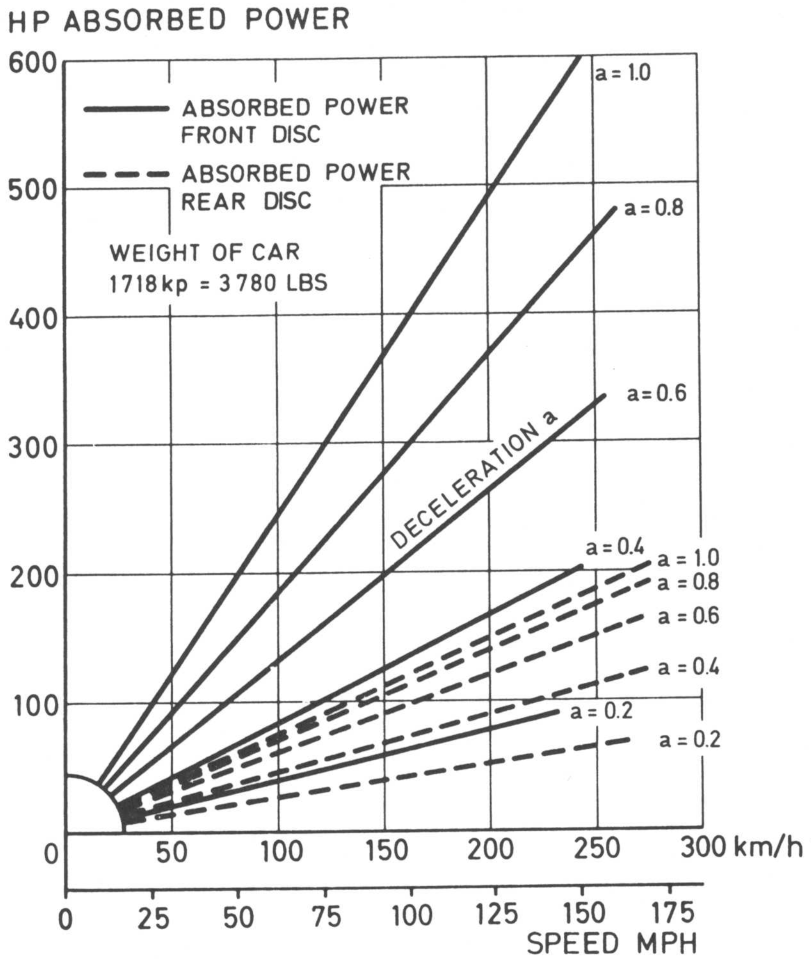
FIG. 24



TIME FOR BUILD UP OF BOOSTER FORCE

6.68/5589

FIG. 25



ABSORBED POWER BY DISC IN RELATION TO SPEED AND DECELERATION

6.68/5577

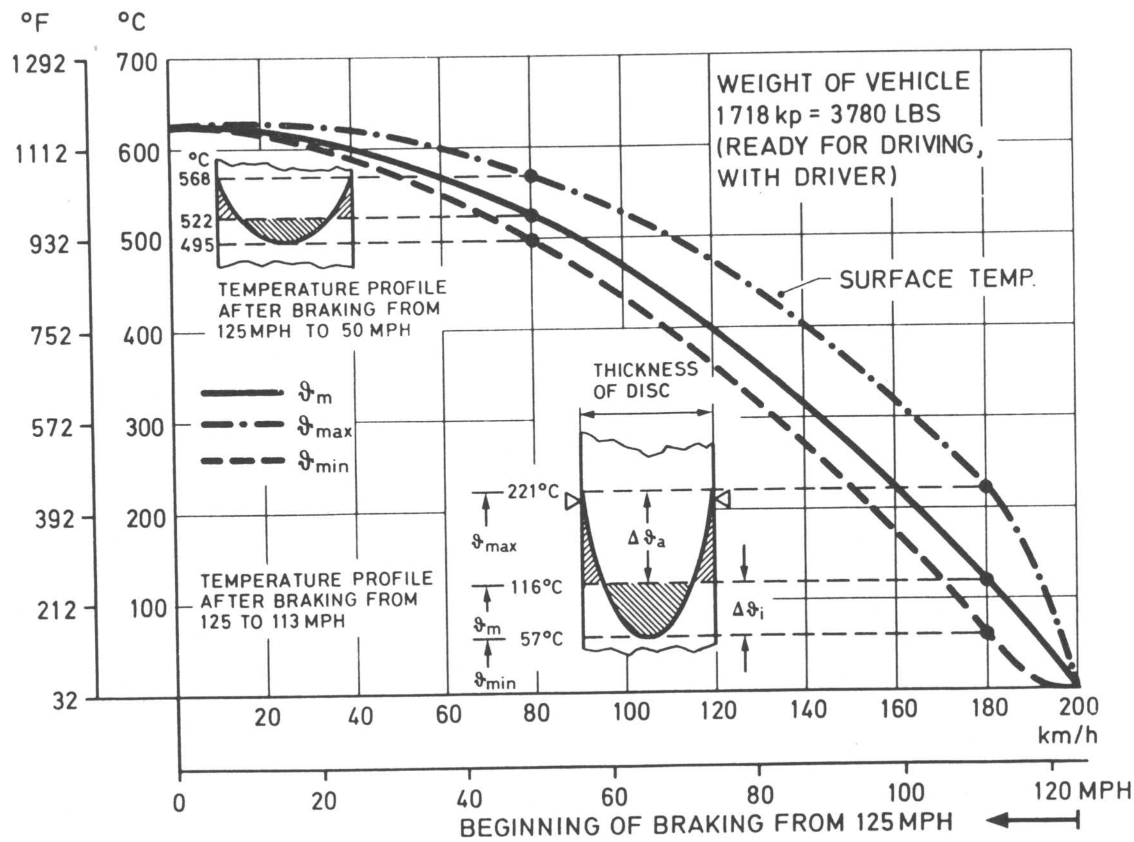


FIG. 26 | TEMPERATURE PROFILE IN THE FRONT WHEEL DISC OF THE 300 SEL-6.3 DURING A BRAKE TEST FROM 125 MPH | 6.68/5570

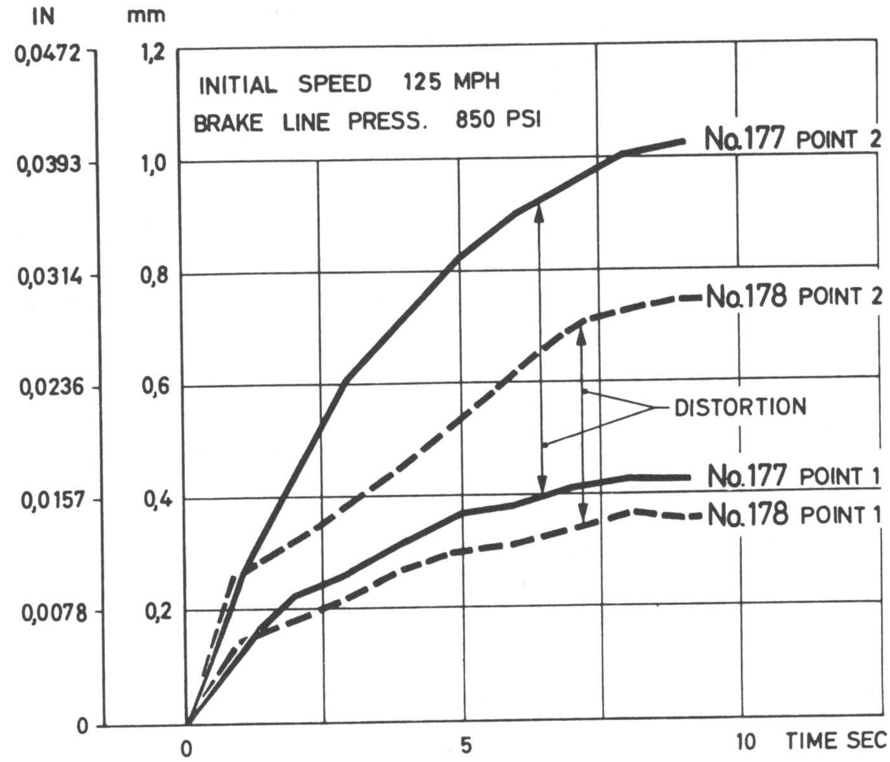


FIG. 27	DISTORTION OF DISC DURING BRAKE TEST	6.68/5580

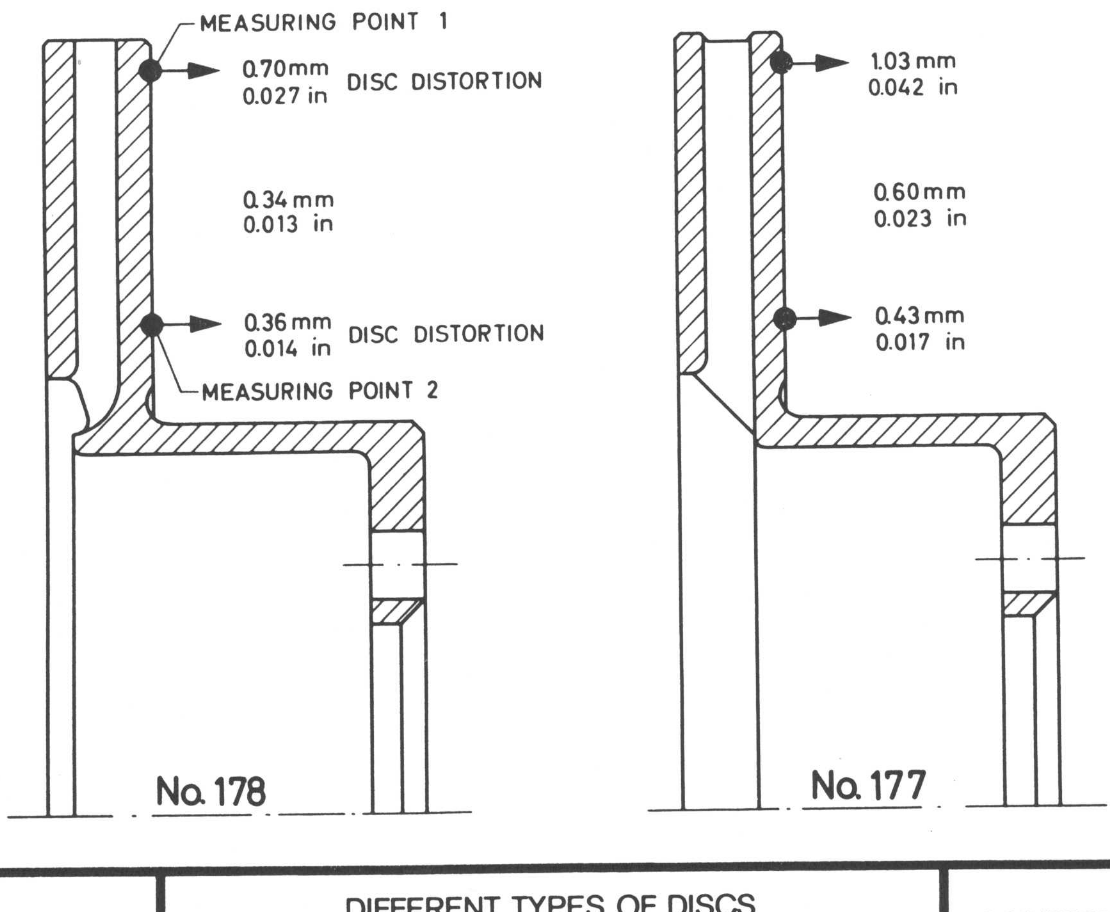


FIG. 28	DIFFERENT TYPES OF DISCS FOR TYPE 300 SEL-6.3	6.68/5585

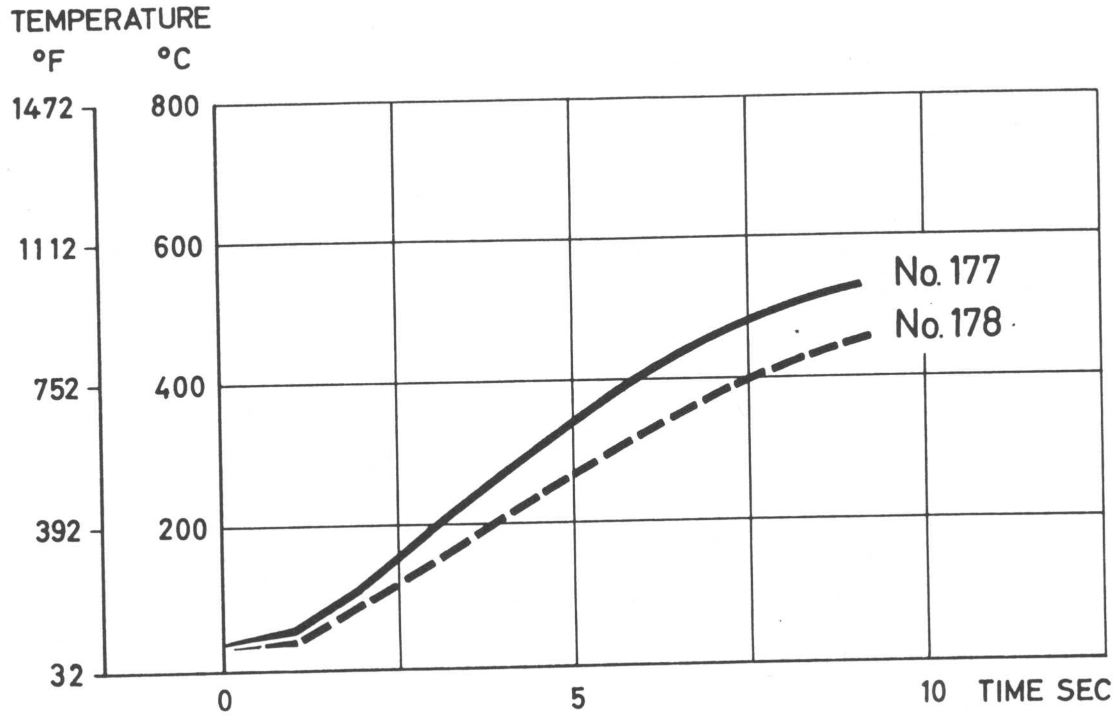


FIG. 29 | RISE OF TEMPERATURE OF DISC DURING BRAKE TEST FROM 125 MPH WITH 1430 PSI BRAKE LINE PRESSURE | 6.68/5579

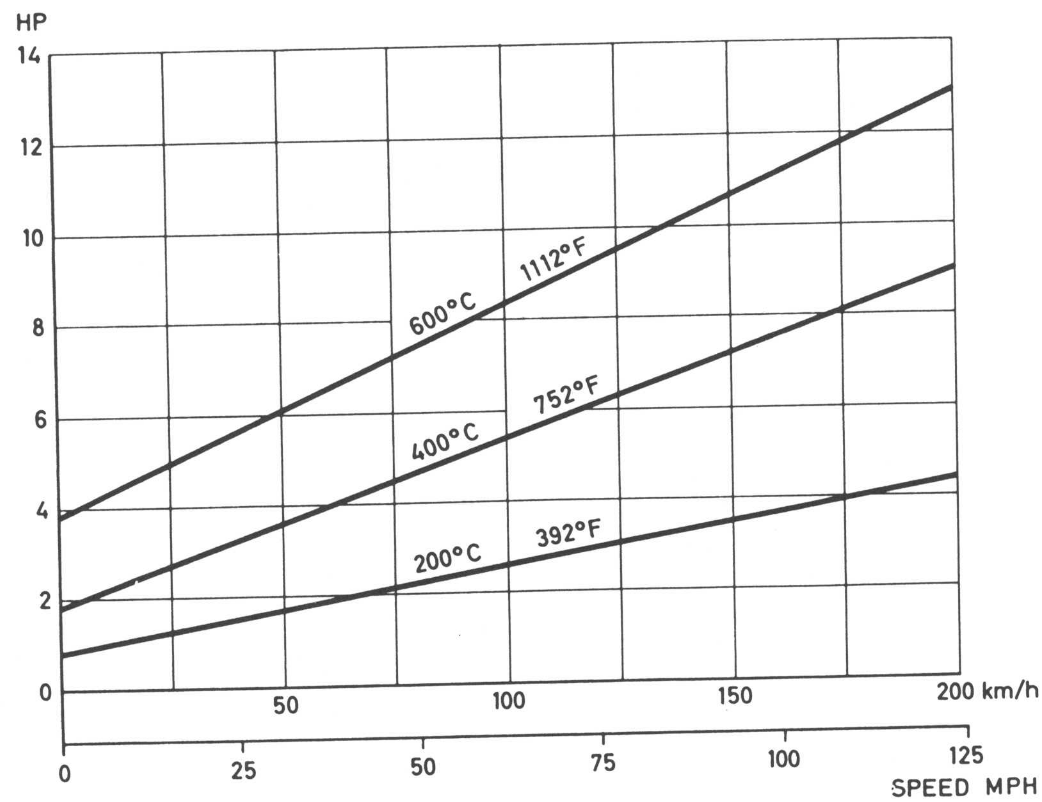


FIG. 30 | HEAT DISSIPATION OF FRONT WHEEL DISC
DISC VENTILATED, CLOSED WHEEL | 6.68/5606

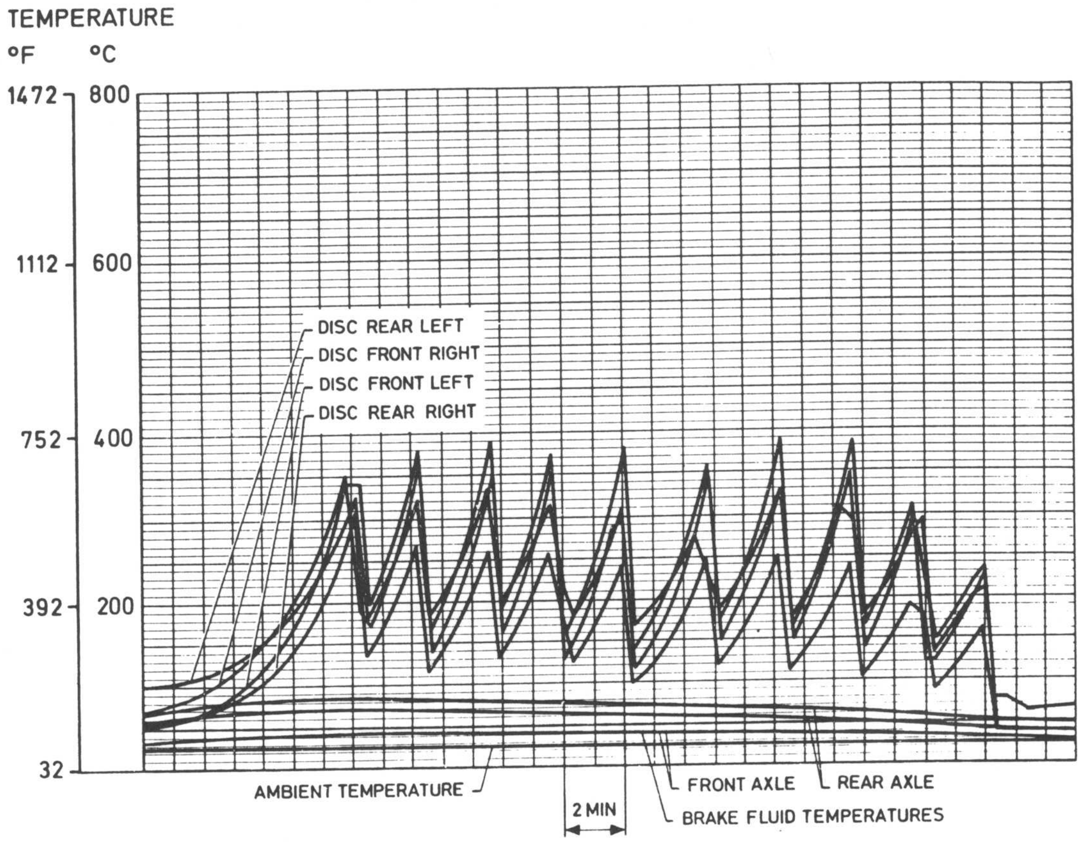


FIG.31	DISC TEMPERATURE DURING TEST WITH 0.4 G DECELERATION FROM 112 MPH TO 38 MPH EVERY 2 MINUTES	6.68/5591

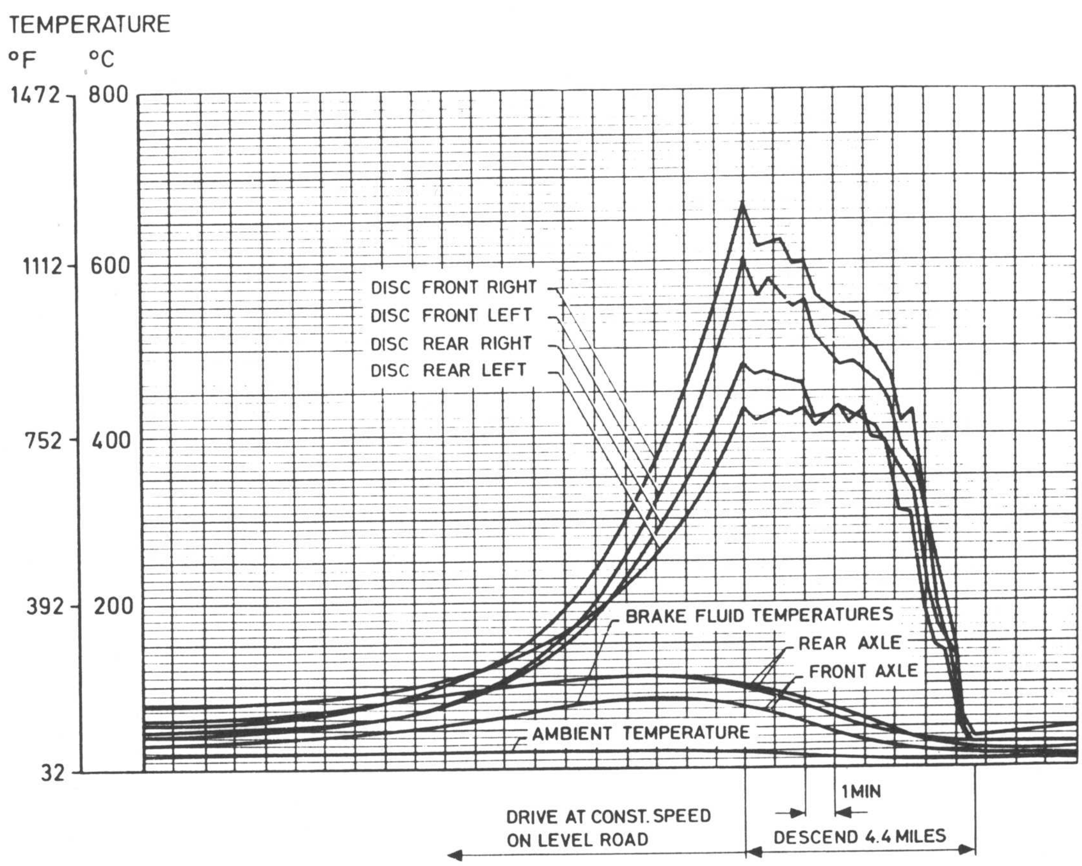


FIG. 32	DISC AND BRAKE FLUID TEMPERATURE DURING MOUNTAIN DESCEND MEAN GRADE 17% LENGTH 4.4 MILES	6.68/5590

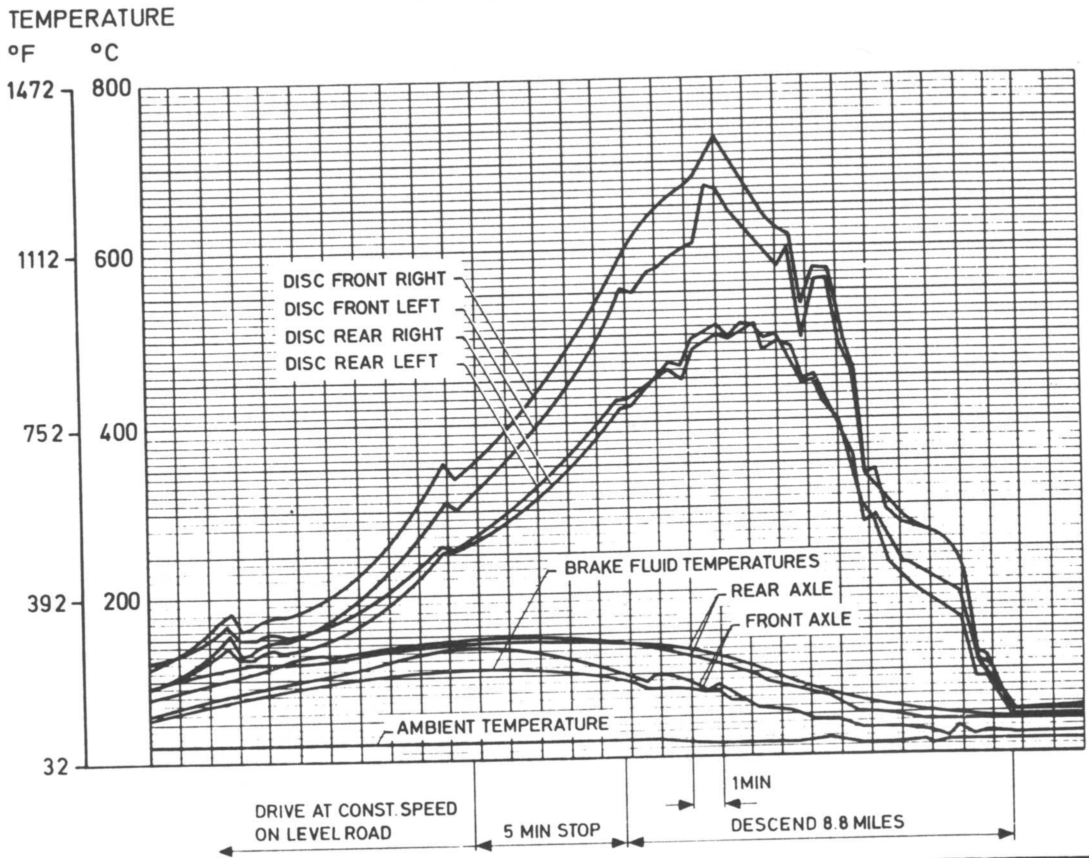


FIG. 33	DISC AND BRAKE FLUID TEMPERATURE DURING RAPID MOUNTAIN DESCEND MEAN GRADE 14% LENGTH 8.8 MILES	6.68/5592

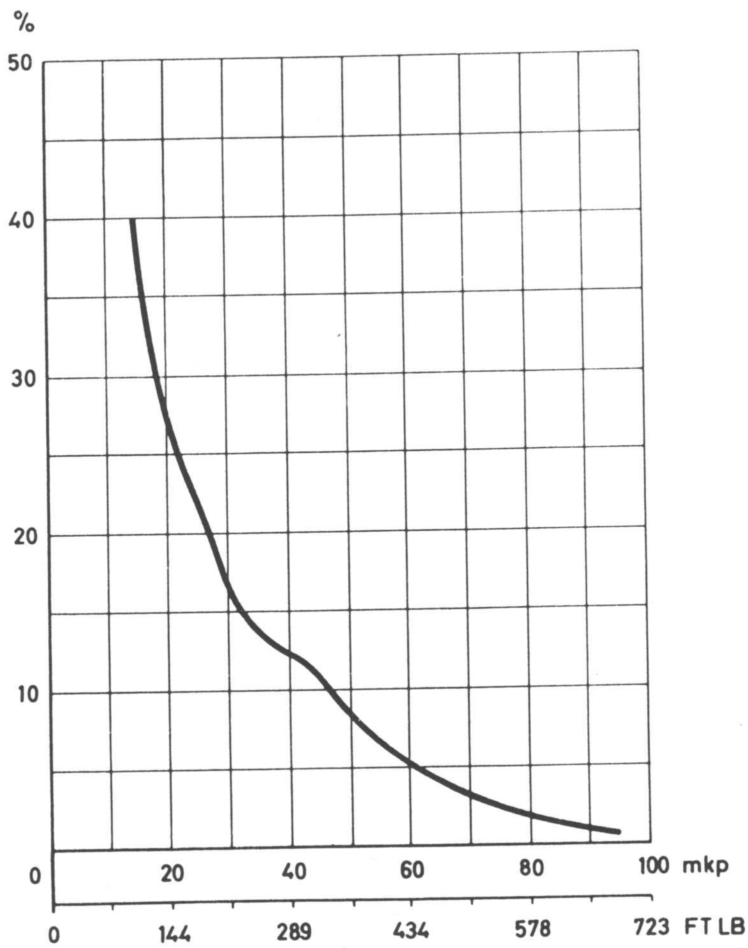


FIG. 34

BRAKE TORQUE OCCURENCES ON MOUNTAINOUS COURSE IN THE BLACK FOREST FAST DRIVING

6.68/5575

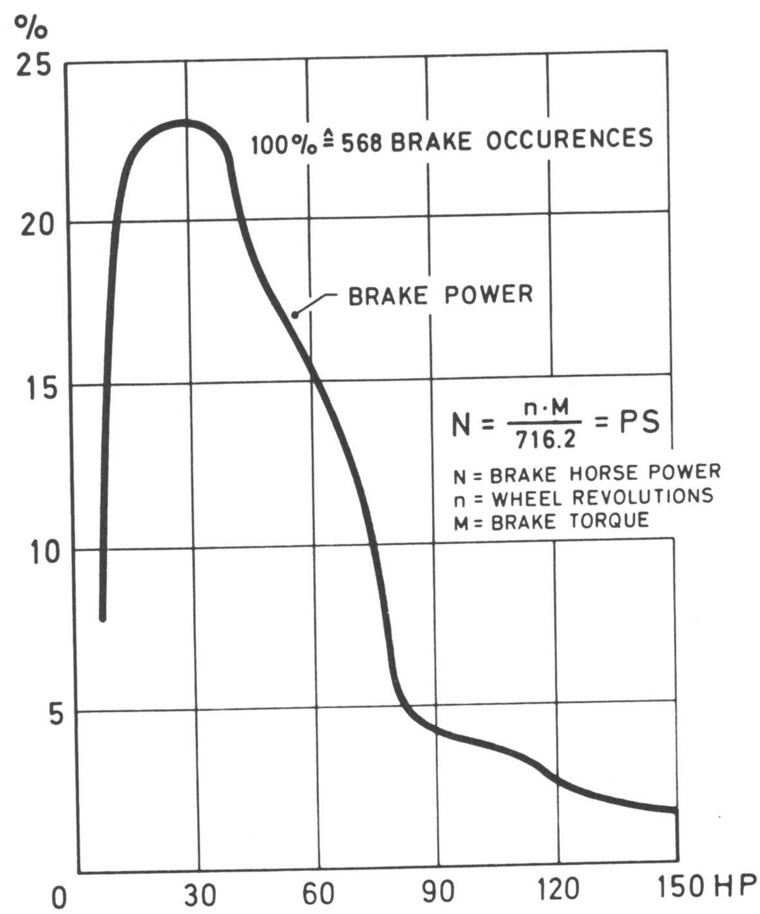


FIG. 35

BRAKE OCCURENCES ON MOUNTAINOUS COURSE IN THE BLACK FOREST

6.68/5576

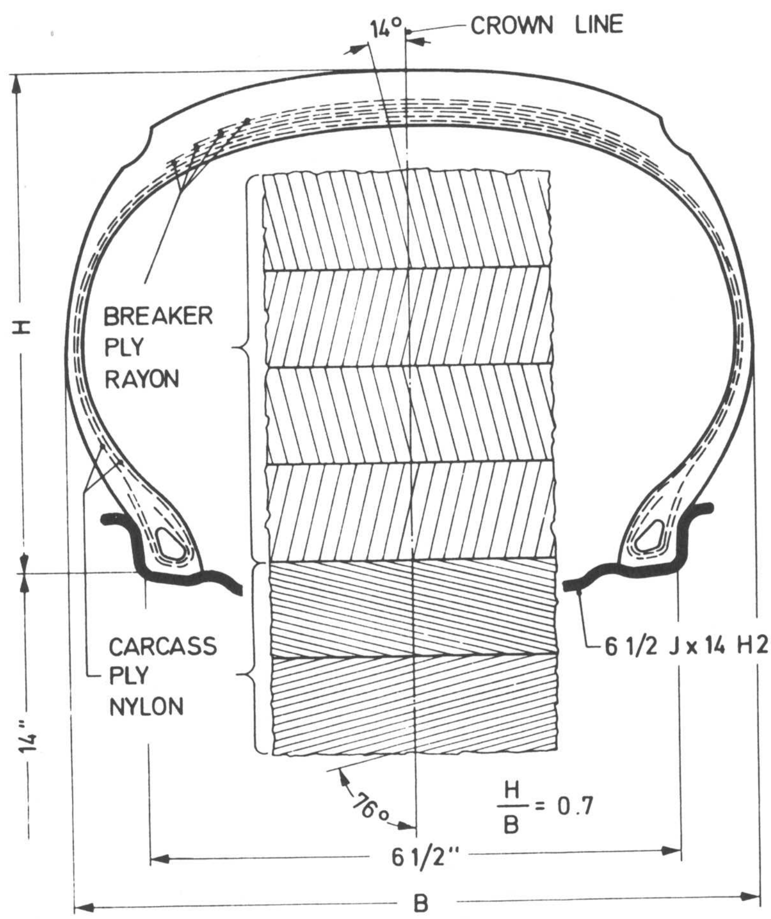


FIG. 36

RADIAL TIRE
FR 70 VR 14
(EUROPEANE DESIGNATION 195 VR 14)

6.68/5594

CORRECT TIRE PRESSURE IN PSI ONLY FOR RADIAL TIRE

SPEED	COLD TIRES		WARM TIRES			
			AFTER PROLONGED CITY DRIVING OR MODERATE OVERLAND TRIP		AFTER FAST DRIVING ON HIGHWAYS	
	FRONT	REAR	FRONT	REAR	FRONT	REAR
MORE THAN 125 MPH	40	42	42	46	46	50
UP TO 125 MPH	32	34	36	38	40	42
SPARE WHEEL	WINTER TIRES					
42	30	36	32	40	36	42

PLEASE REFER TO OWNER'S MANUAL AS WELL!

FIG. 37 TIRE PRESSURE RECOMMENDATION 6.68/5613

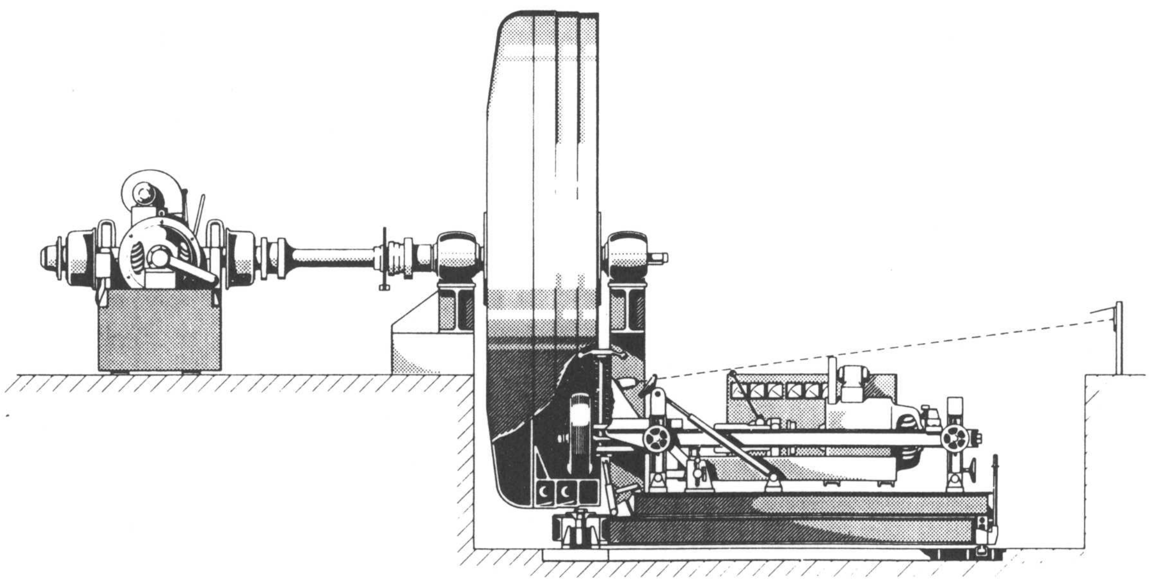
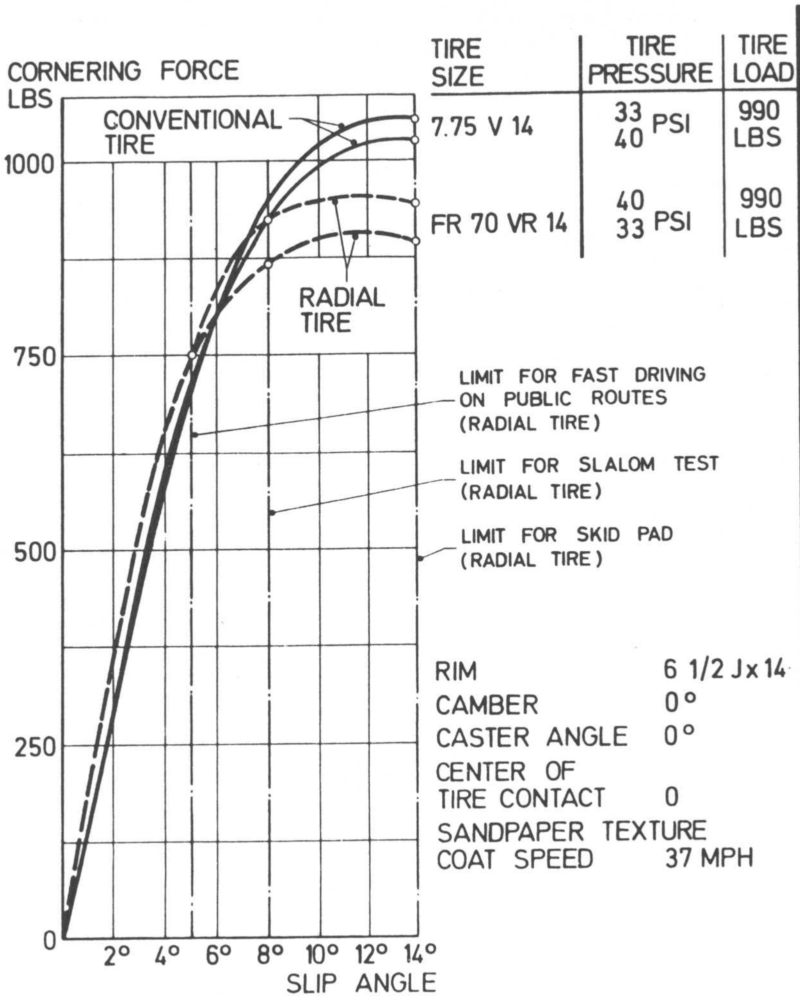


FIG. 38 TIRE TESTING STAND
TECHNISCHE UNIVERSITÄT KARLSRUHE, GERMANY 6.68/5605

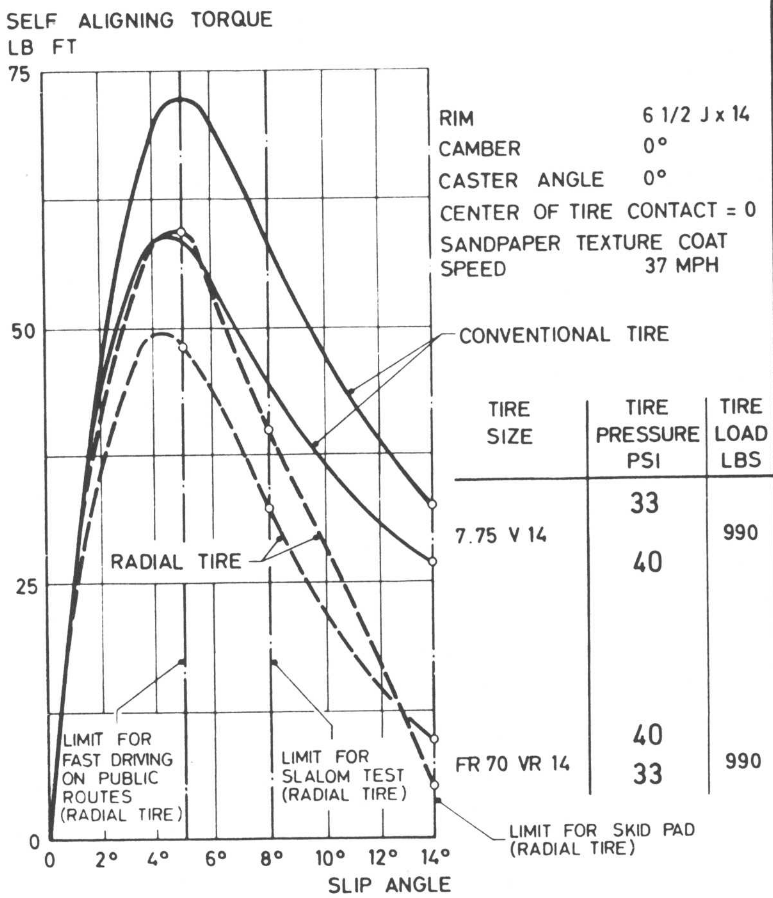
FIG. 39



RADIAL TIRE
FR 70 VR 14
(EUROPEANE DESIGNATION
195 VR 14) AND
CONVENTIONAL TIRE
7.75 V 14

6.68/5597

FIG. 40



RADIAL TIRE
FR 70 VR 14
(EUROPEANE DESIGNATION
195 VR 14) AND
CONVENTIONAL TIRE
7.75 V 14

6.68/5596

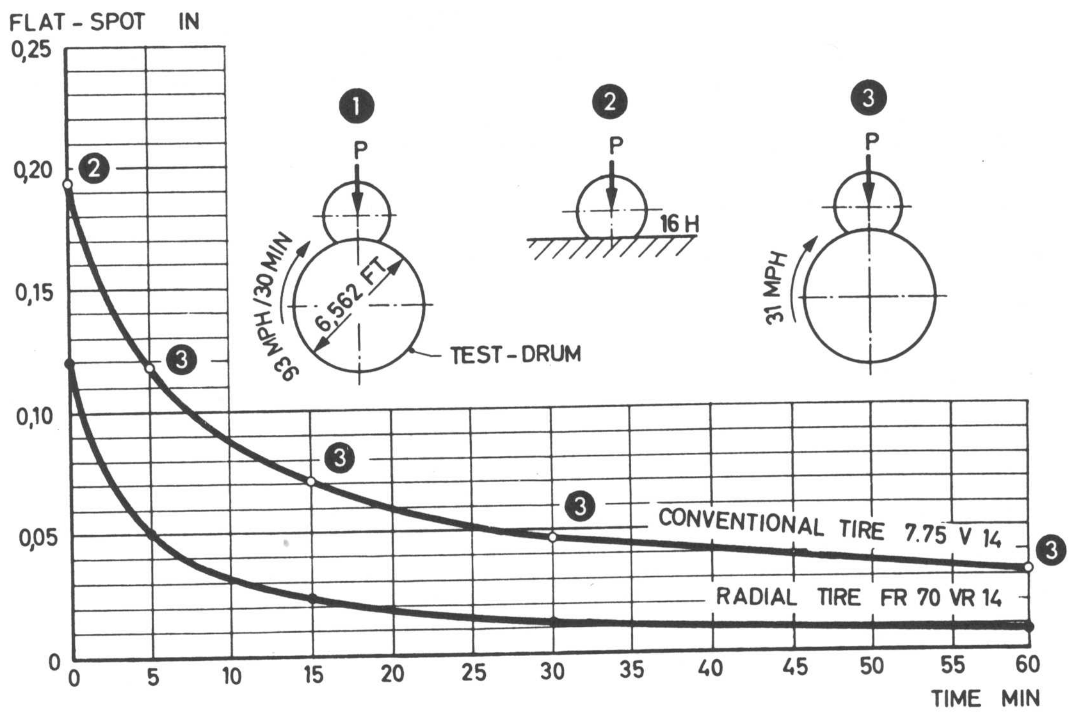


FIG. 41	FLAT-SPOT-TEST	6.68/5595

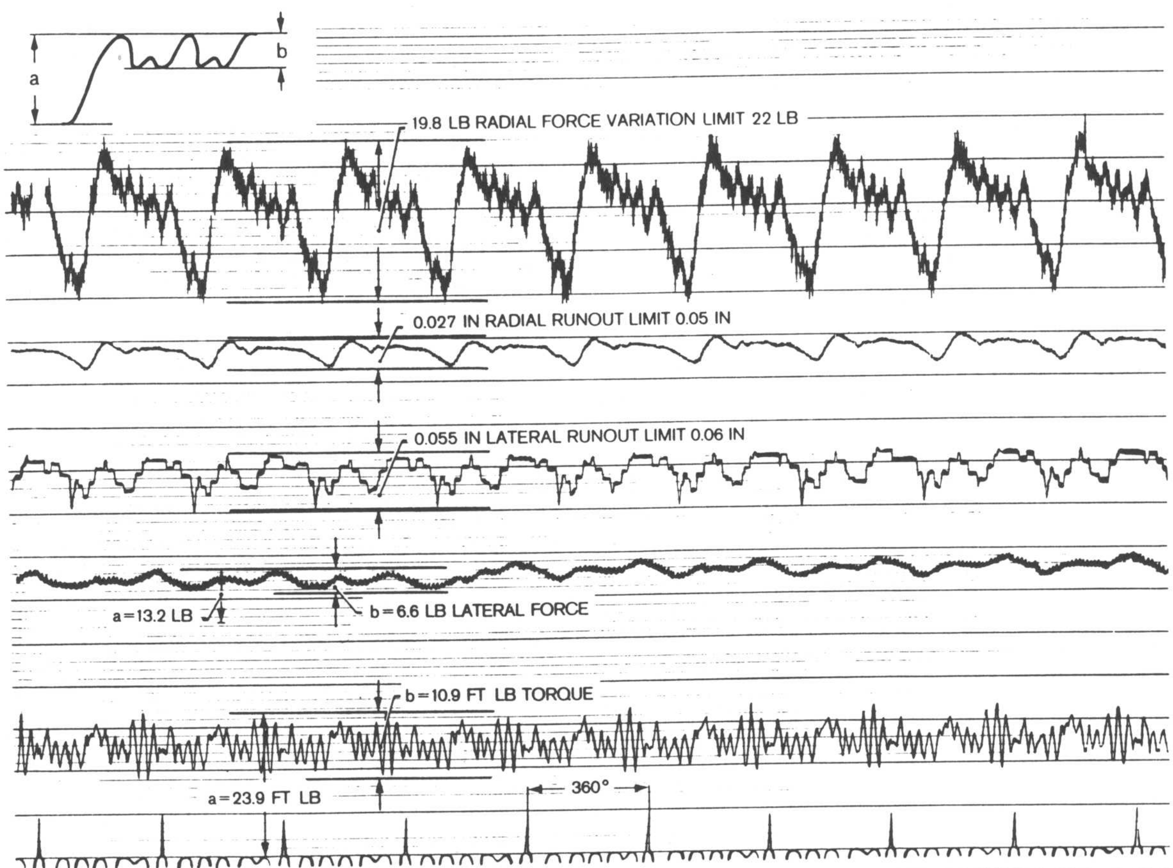


FIG. 42	TIRE UNIFORMITY TIRE SIZE FR 70 VR 14; EUROPEANE DESIGNATION 195 VR 14	6.68/5593