SECTION 4

ACCIDENT AVOIDANCE SEMINAR

Part 1 – Introduction

Mr. Francis A. DiLorenzo, *Chairman*

Part 2 – Visibility

Dr. Hermann Bruns, *Germany, BMW*
Mr. R. Schmidt, *Germany, VW*

Part 3 – Vehicle Steering and Handling

Mr. Vor Der Brueck, *Germany, Ford*
Mr. Berlioz, *France, O.N.S.E.R.*
Dr. K. Enke, *Germany, Daimler-Benz*
Mr. Yasuhiko Fujiwara, *Japan, Nissan Motor Co.*
Dr. K. Enke, *Germany, Daimler-Benz*

Part 4 – General Topics Concerning Accident Avoidance

Mr. Iichi Shingu, *Japan, Toyota Motor Co.*
Mr. W. B. Haro Kopus, *USA, Bendix Research Lab Laboratories*
Mr. F. A. DiLorenzo, *USA, Office of ESV/DOT*

Part V – Vehicle Braking

Mr. R. Cochrane, *United Kingdom, Girling*
Mr. F. A. DiLorenzo, *USA, Office of ESV/DOT*

SECTION 4

PART 1 INTRODUCTION

Mr. Francis A. DiLorenzo,

Chairman, Accident Avoidance Seminar

Part I - Introduction

Opening Remarks

Good afternoon Gentlemen, I am Mr. Francis DiLorenzo and it is an honor and pleasure to chair the Accident Avoidance Seminar. We have received an excellent response to our call for discussion papers and therefore as you can see from the agenda we have a very busy afternoon.

I appreciate that most of you had only a short time to review the papers that will be presented here today and regret that we were unable to exchange papers earlier. Perhaps at the next conference more time will be available.

We will undoubtedly be discussing topics this afternoon that are close to many of us and perhaps even controversial in nature. I would hope that through this free exchange of information, we will better understand each other and improve our research efficiency. We are involved in a scientific endeavour that is technically changing very rapidly. This factor alone necessitates that we communicate freely and exchange ideas.

For the discussions of the papers to be presented, I request that all questions be held until the final paper of each major topic has been presented. For relaxation and further informal discussions among yourselves we have scheduled a short break about midway through the seminar. I would also like to add that for those who do not get a chance to discuss everything they would like to today, there will be a second opportunity tomorrow morning during the seminar summations.

Without further delay then, the first topic this afternoon is a paper on visibility by Dr. Bruns from BMW, Dr. Bruns – – –

SECTION 4

PART 2 VISIBILITY

RANGE OF VIEW, STOPPING DISTANCE AND DRIVING SPEED ON CURVED ROADS

Dr. Hermann Bruns
Bayerische Motoren Werke A.G.
Munchen

Accident Avoidance Seminary

Modern road vehicles allow high curvature-speeds, which may be much higher than the velocity, the road was designed for. There is the problem, that the range of view is not relative to the required stopping distance. A model should give some results in this theory.

1. Introduction

Contrary to railroad vehicles, the driver of a road-vehicle has a broad range for choosing his driving speed. Except the driver's qualities, the physical borders are the car itself and the road conditions (rain, snow, mist etc.). Both limitations cannot be separated exactly, because for each driving situation, both, vehicle and road have to be considered together. In most cases, the limitation is given by the form of the road, the traffic-density and other environmental influences.

Each road is designed for a certain velocity. It is designed in such a way, to give a good compromise between the expected traffic and the resulting costs. For normal environmental conditions it may be overstepped without any danger.

Table 1 and 2 show an extract of the German instructions[1] for building main roads, concerning the influence of surroundings and radius of curvature.

It will be the goal of each driver, to reach his designation as fast as possible, so he usually will overstep the designed velocity[2], especially in curves. Therefore it's necessary to give a relation between radius of curvature, stopping distance, range of view and driving speed.

Table 1

Designed velocities (km/h) concerning the difficulties of the landscape [1].

landscape		anticipated load factor (vehicles / day) ≤ 1000	>1000-2000	>2000-3000	>3000
I.	Flat	50	60	80	100
II.	Flat and hilly	40	50	60	80
III.	High-lands	30	40	50	60
IV.	Mountain-Terrain	30	30	40	50

Table 2

Minimum radius of curve for different designed velocities [1].

designed velocity v_e (km/h)	minimum radius of curve R_{min} (m)	lateral acceleration b_q (m/sec^2)
30	35	2,0
40	70	1,78
50	120	1,6
60	180	1,54
80	350	1,41
100	600	1,28

2. Relation Between Driving Speed and Stopping Distance.

At first it would be interesting to know the relations between driving speed and stopping distance, without considering radius of curvature, Figure 1,[3]. For comparison you see the required stopping distance for sufficient range of view s_h for RAL[1].

In a curve it is not possible, to brake with maximum deceleration b±4. The maximum force between a rolling wheel and the street is limited by the wheel load G and

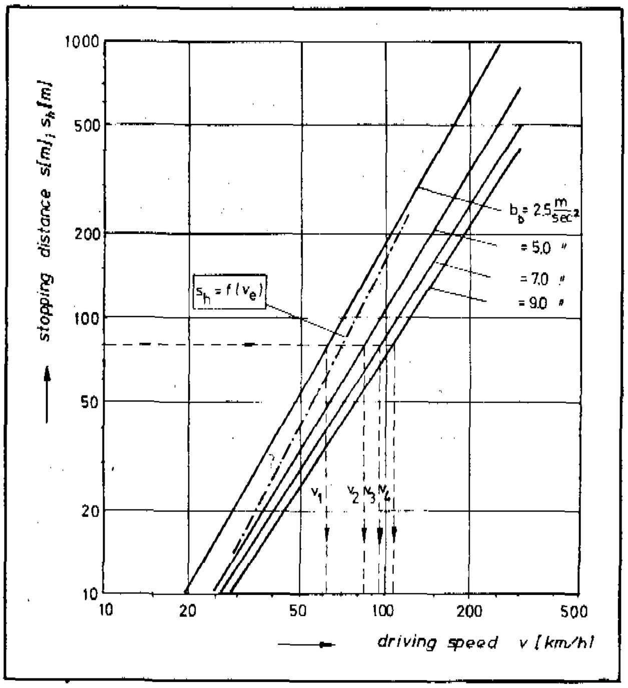


Figure 1 – Relation between driving speed v and stopping distance s, respectively range of view for stopping s_h.

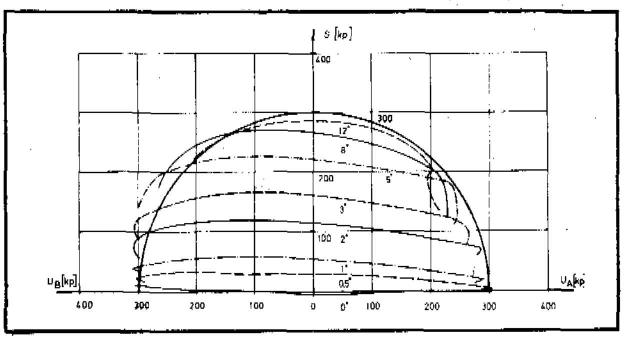


Figure 2 – Lateral force S of a radial – tire as a function of horizontal forces U for different slip angles[5].

the friction coefficient μ. The division in lateral and horizontal forces is given by a special diagram for each tire, Figure 2[5,6]. the relation between maximum lateral acceleration b_q and maximum deceleration b_b can be described by a circle-equation. Supposing the same friction coefficient in radial and axial direction, you get the following formula

$$S = \mu \cdot G = b_q \cdot \frac{G}{g} \qquad (1)$$

$$\frac{S}{G} = \frac{b_q}{g} \qquad (1a)$$

and

$$U_B = \mu \cdot G = b_b \cdot \frac{G}{g} \qquad (2)$$

$$\frac{U_B}{G} = \frac{b_b}{g} \qquad (2a)$$

$$\left[\frac{b_{qmax}}{g}\right]^2 + \left[\frac{b_{bmax}}{g}\right]^2 = 1 \qquad (3)$$

With high lateral accelerations there is a deviation from course[7].

3. Relation Between Driving Speed and Radius of Curve.

If a vehicle is going on a horizontal circular path with radius R having a driving speed v in direction of the tangent to course, there is a lateral acceleration

$$b_q = \frac{v^2}{R} \qquad (4)$$

For maximum lateral acceleration $b_q = 9{,}81\ m/s^2$ there are maximum velocities v_{max} as a function of radius of curve R as shown in Figure 3. For comparison measured driving speeds in curves v(±10%) are added[8]: The resulting lateral accelerations are

$$\overline{V}_m\ (Km/h) = 25 \cdot R^{0.2}\ (m) \qquad (5)$$

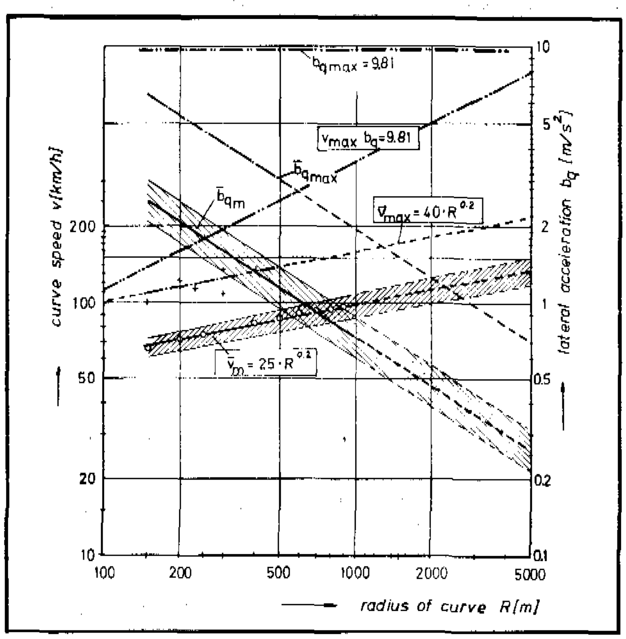


Figure 3 – Relation between driving speed v, radius of curve R and lateral acceleration b_b v_m mean speed [7] for normal drivers, b_{qm} relation lateral acceleration.

v_{max} racing speed, b_{qmax} relating lateral acceleration. V_{max} driving speed for maximum lateral acceleration $9{,}81\ m/s^2$ = constant.

$$\bar{b}_{qm}(m/s^2) = \frac{48.3}{R^{0.6}\,(m)} \qquad (6)$$

A skilled driver is good for higher driving speed in curves, see v_{max} in Figure 3.

$$\bar{V}_{max}(Km/h) = 40 \cdot R^{0.2}\,(m) \qquad (7)$$

$$\bar{b}_{qmax}(m/s^2) = \frac{123.4}{R^{0.6}\,(m)} \qquad (8)$$

(That means 1.6 times the velocity and 2.56 times lateral acceleration of a normal driver.)
According to the utilized lateral acceleration, remains a braking deceleration b_{bmax}

$$\frac{b_{bmax}}{g} = \sqrt{1 - \left[\frac{b_{qmax}}{g}\right]^2} \qquad (9)$$

Supposing an extreme way of driving, you get a relation between the maximum possible braking deceleration and radius of curve by using equation[8] and [9], Figure 4.

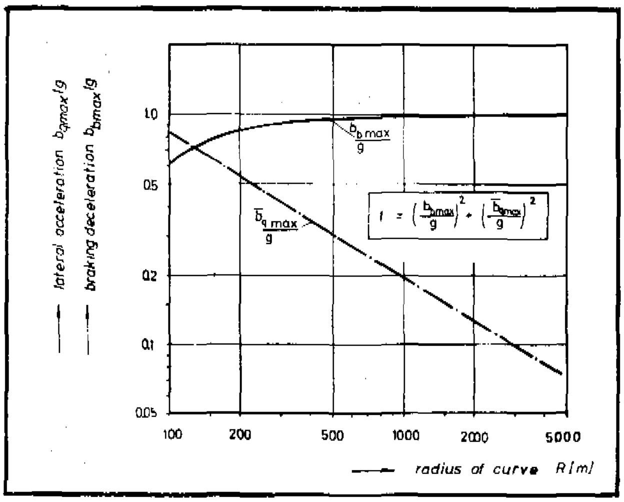


Figure 4 – Functional relation of lateral acceleration and braking deceleration from radius of curve, calculated from equation[8].

$$\frac{b_{bmax}}{g} = \sqrt{1 - \left(\frac{12.5}{R^{0.6}}\right)^2} \qquad (10)$$

Equation[10] equals 0 for $R^{0.6} = 12.5$ that is R = 67.3 m. In the range of R = 0 ÷ 67.3 m the driver used the full lateral acceleration $b_{qmax} = 9{,}81\ m/s^2$.

4. Radius of curve and range of view.

On an even road the range of view only depends on the radius of curve.

Figure 5 shows the model for a vehicle in a curve. The distance between the driver's eyepoint and the inner boundary may be a (AD).

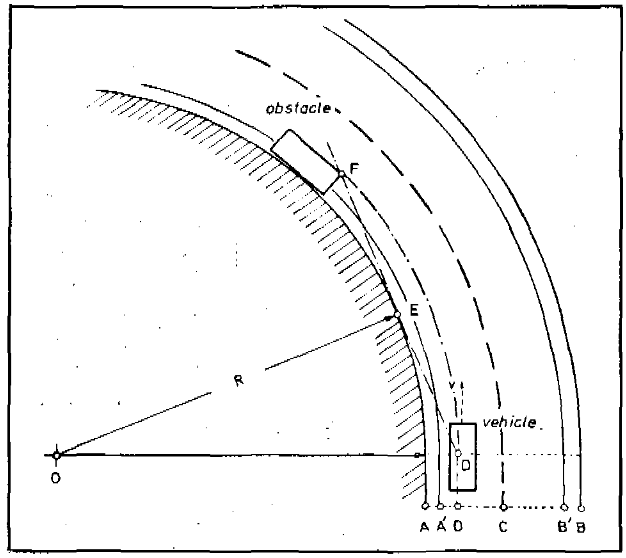


Figure 5 – Vehicle driving a curve:
D: eyepoint of the driver, DF = 2 DE distance d to the obstacle. AD: distance a of driver's eyepoint from the inner border of the road; AB: width b of the road; AA': width of the marginal track; R: radius of curve.

For simplifying, the distance DF equals twice the distance ED with

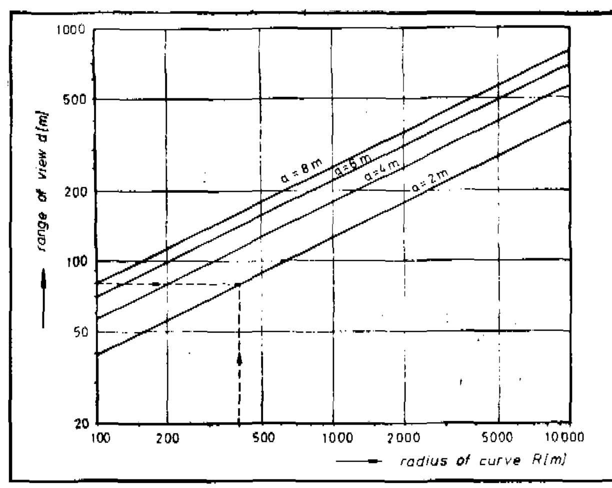


Figure 6 – Dependence of range of view from radius of curve R with eyepoint-distance a as parameter.

with the eyepoint a as a parameter. For example: as-

Figure 6 shows these relations in logarithmic scale, suming a_{min} = 2m and R = 1000 m you get d = 126.5 m, for a_{max} d = 255 m. This influence gets important, if a car overtakes another one in a curve and has to change the lane.

$$\overline{DF} = d = 4a \cdot (2R + a) \qquad (11)$$

5. Correlation Between Driving Speed, Range of View and Stopping Distance in a Curve

The problem exists in finding the right driving speed for the necessary ratio of range of view to stopping distance. Claiming, that the range of view d should be never less than the stopping distances, you get the resulting driving speed for each radius from the shown figures, Figure 7. The range of view for a certain radius of curve, you get from Figure 6. With the condition

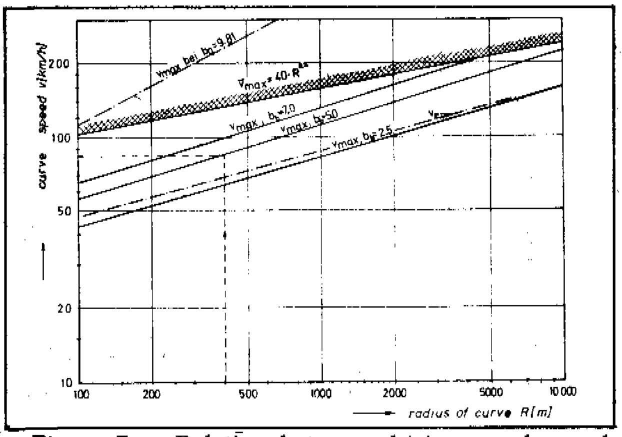


Figure 7 – Relation between driving speed v and radius of curve R, depending on the range of view for different braking decelerations b_b, with constant eye-point distance a_{min} = 2 m.

you get from Figure 1 the concerning driving speed v_{max}. Braking deceleration b_b is the parameter. The limitations in Figure 7 are given by the velocities, driven by skilled drivers. An example may show the way from Figures 1, 6 and 7. For a radius of curve R = 400 m and braking deceleration b_b = 5.0 m/s^2 you get an allowable driving speed of 84 mk/h, if the range of view is limited by for example walls on each side of the curved road.

A BRIEF SURVEY OF REARVIEW SYSTEMS

Dipl.–Ing. R. Schmidt
Volkswagenwerk A.G.
Wolfsburg

Accident Avoidance Seminary

The field of vision which may be obtained through rearview mirrors may be demonstrated by reference to three rearview systems and the advantages or disadvantages of these systems explained. In addition to this these systems are compared with the essential requirements of a rearview system as laid down in the VDA, ESV and docket specifications.

The three systems comprise

1. a conventional rearview system consisting of an interior mirror and an outside mirror on the driver's side,
2. a plane mirror which projects with one half above the roof the other half being within the vehicle body and
3. a periscope system.

Before going into detail on the systems, reference should first be made to some of the optical physiological features of our organs of sight. In the act of seeing, one very important factor is the time which elapses before the eye can again see clearly following a change in the direction of view. In today's dense traffic conditions it is essential to locate the mirror arrangement in such a way that the time which it takes to see a clearly defined picture in the rearview mirror is as brief as possible. The total delay is made up of the diversion of the direction of vision to the mirror and the time taken for accommodation, this term being used to imply the time required to recover full focus. To quote an example, under conditions of high light intensity, the time taken to achieve full clarity of vision when looking away from the instrument panel to a far distant point on the road is in the region of one second. When the light intensity is low this time is vastly increased.

Basically it can be said that the further the rearview mirror is from the normal direction of vision the longer the delay. Because of the focussing properties of the eye, a rearview mirror should be located within a conical space of not more than 30 degrees from the normal line of vision. This not only cuts down on the rearview time, it also means that both the mirror and the normal direction of vision are still inside the peripheral 10% vision area. This then ensures that, even while glancing into the rearview mirror, large obstacles in front of the vehicle can still be recognized. Conversely while looking in the normal visual direction, vehicles approaching from behind and coming into the range of the rearview mirror are noticed *without* concentrating on the mirror.

Because of the physiological optics of the eye an image which is projected on to the periphery of the retina is more likely to be unnoticed the less its movement relative to the retina and the further it is from the normal visual axis. Since the relative movement of following vehicles is frequently very slight, if the rearview mirror is located *far* away, they are relatively easy to lose in the local adaptation function even although they may have quite noticeable features.

Figure 1 – The first illustration diagrammatically shows a conventional rearview system comprised of a plane interior mirror and a plane exterior one. The fields of view have been arranged in accordance with the VDA or ESV specifications. The positions and heights of the fields of view, Q, SR, SL, TR and TL conform with the specifications outlined in Docket No. 72-3a, Notice 1. The size of the interior rearview mirror should be such that as much as possible of the vehicle on which it is mounted – preferably the entire rear window section – should be visible in the interior rearview mirror. This ensures that the relative positions and directions of the vehicle and the vehicle behind may be viewed and makes it much easier to estimate distances.

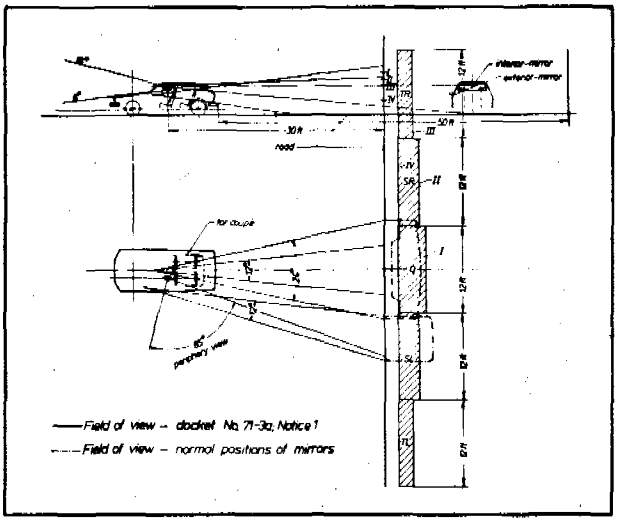


Figure 1

The dimensions of the interior mirror are 195 x 55 mm and those of the outer one 175 x 120 mm. In this case the exterior mirror is not within the 30 degree conical range, the reason being to keep the field of vision as big as possible and to overcome the blind spot with the aid of the peripheral vision.

The main advantages of the conventional rearview system are

1. The good field of view directly behind the vehicle, the surface of the road being viewable 36' (10.90 m) behind the tail-end of the vehicle;

2. the relatively good view to the side when the exterior mirror is appropriately mounted on the driver's side of the vehicle, the road surface then being seen 12' (3.64 m) to the rear of the car, which is a very important feature in being able to see cyclists or motorcyclists in the process of overtaking; and

3. the *low* technical outlay.

The disadvantages experienced are:

1. limitation of the horizontal field of vision by the rear roof pillars; and

2. slight obstruction of view by the rear headrests. These visual obstructions are however not within the range prescribed in the docket.

The requirements laid down in the VDA and ESV specifications in respect of the horizontal and vertical angle ranges are complied with. (VDA and ESV demand a 24 degree horizontal field of vision through the rear window and that the road surface be visible from a point 15 meters behind the rear end of the vehicle to the horizon). The requirements placed on the exterior mirror (12 degrees horizontal field of vision, visibility of the road surface from 35 feet behind the mirror) are fulfilled without difficulty. In the arrangement illustrated the road is already visible at 19 feet.

Figure 2 – The second picture shows the principles of a rearview system involving a plane mirror with one half above the roof and the other within the vehicle.

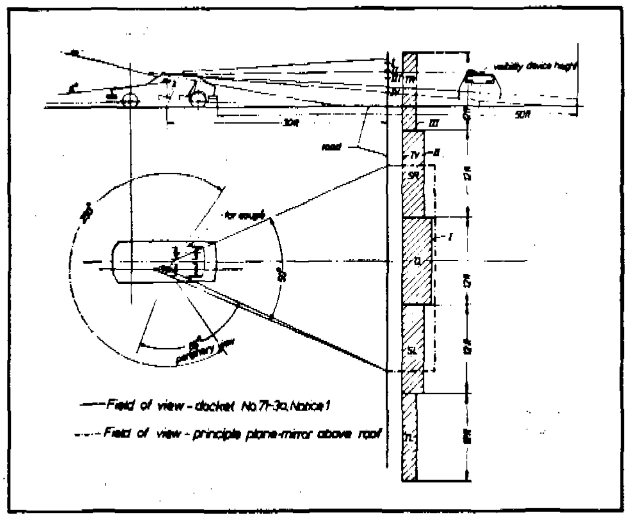


Figure 2

The principal advantages of this system are:

1. Very good visibility directly behind the vehicle (the road can be seen 17 feet beyond the rear limits of the car);

2. The wide horizontal angle of 50 degrees in this case;

3. No obstruction to vision by headrests.

The disadvantages found are:

1. The blind spot behind the car due to the roof shape;

2. The inadequate visibility sidewards;

3. The height of the viewing arrangement in the vehicle resulting in severe impediment of forward vision. It is necessary to change the direction of vision by more than 30 degrees;

4. The *high* technical cost.

Although this system fulfills the requirements in the dockets from 1 Jan. 1974 to 31 Dec. 1976 problems are still presented by the sidewards visibility. The only remedy here is an external mirror. Apart from the important VDA and ESV specification requirements in respect of visibility to the side the remainder of the requirements on rearwards visibility are complied with.

Figure 3 – The third illustration demonstrates a periscopic rearview system.

The essential advantages this offers are:

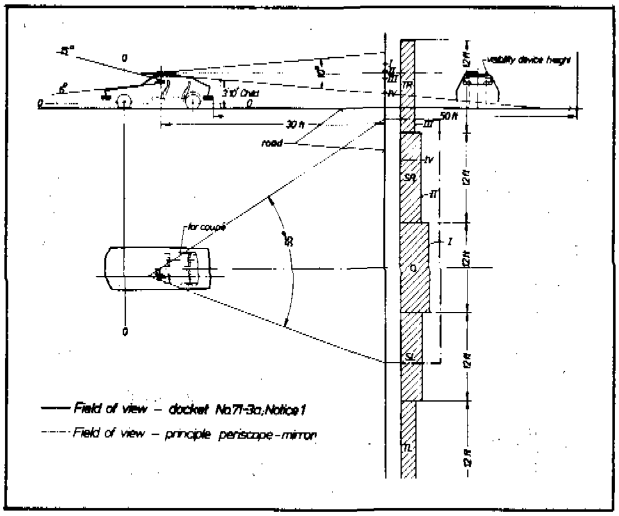


Figure 3

1. The wide horizontal field of vision (55 degrees) and
2. no headrest obstruction.

The disadvantages include:

1. The view directly behind the vehicle is very limited, e.g., a child of 3'9" (1.15 m) immediately behind the car would not be seen at the road, surface only becomes visible at 44 feet (13.2 m). Reversing is safe only if the driver turns round.
2. Inadequate visibility sidewards close to the vehicle. The periscopic system also does not overcome the blind spot on the left side of the vehicle.

Although the horizontal field of vision can be extended through a periscopic system it is necessary to ask the basic question as to whether or not the docket requirements in respect of the horizontal field of vision are really advantageous and the high cost of a periscope system really justified, since the disadvantages of that system (poor visibility directly behind the car, poor sidewards visibility directly adjacent to the vehicle) are not present with the interior mirror combined with an outer mirror on the left: In addition to that, the horizontal field of the interior mirror can be extended by appropriate design of the rear roof pillars.

In light of the features presented, we consider than an interior mirror and a left outside mirror is the right solution on the ESV.

SECTION 4

PART 3 VEHICLE STEERING AND HANDLING

THEORETICAL AND EXPERIMENTAL INVESTIGATIONS FOR EVALUATION OF VEHICLE HANDLING QUALITIES

Mr. R. Vor Der Brueck, Ford of Germany

Objective

In the "technical requirements on experimental safety vehicles" the VDA committee "safety car" has established various requirements on vehicle handling behaviour emphasizing the particular importance of handling qualities for traffic safety.

At present, during the development of a motor car the tuning of the handling performance is based mainly on subjective evaluations although there has been many efforts to establish objective criteria on the vehicle handling qualities. Efforts are made recently to establish test methods on vehicle handling behaviour but the results of at least a part of these test methods are strongly influenced by the skill of the test driver.

Generally valid evaluation methods should be based therefore on the objective measurement of vehicle response for steady state as well as for transient vehicle handling behaviour. The test results gained from these measurements must be related to the subjective assessments in order to obtain results which are evaluated by the human response factors.

Based on criteria for evaluation of vehicle handling behaviour gained in such a way, the influence of different design parameters and of tire characteristics on handling qualities could be investigated already when the first layouts for a new model are available, in order to optimise the basic design.

Method Of Investigation

1. The Mathematical Model Of The Car

For the prediction of vehicle handling behaviour a mathematical model has been developed and a digital computer program for simulation of vehicle handling response has been established. In order to have a realistic mathematical approach the following parameters additionally to the main vehicle data were included in the investigation: the steering elasticity, suspension kinematics, the non-linearity of spring-rates and shock absorber characteristics, and the non-linear relationship between the dynamic wheel loads and tire cornering characteristics.

Either a steering input of arbitrary shape or a side wind gust can be chosen as the exciting function.

2. Test Equipment

For validation of theoretical results road tests have been carried out and the recorded test data have been compared with theoretical results. During the road tests the following vehicle response functions have been recorded:

- the lateral acceleration and the roll angle using a "stabilised platform"
- The yaw velocity with an angular velocity gyroscope
- the yaw angle with a position gyroscope
- the position of the steering idle arm as measure for steering wheel and road wheel angles

3. The Test Car

As test car a medium size sedan has been used which was equipped to this purpose with an adjustable link-controlled rear suspension.

The adjustable solid rear axle allows a considerable change of the understeer character of the test car.

The rear axle consists of two longitudinal links, a torque reaction bar, a Panhard rod, stabilizer and coil springs.

Different hinge point heights both on body and on rear axle for the longitudinal links allow to change the roll steering effect of the rear axle. Roll centre height can be changed by using different hinge point heights for the Panhard rod, roll couple distribution is varied by using different stabilizer bars.

4. The Road Tests

For validation of the computer results the mentioned test data have been recorded for the two following test manoeuvres:

- Entry into circle
- Slalom test with different wave lengths

The road tests have been carried out for a number of modifications on the test car.

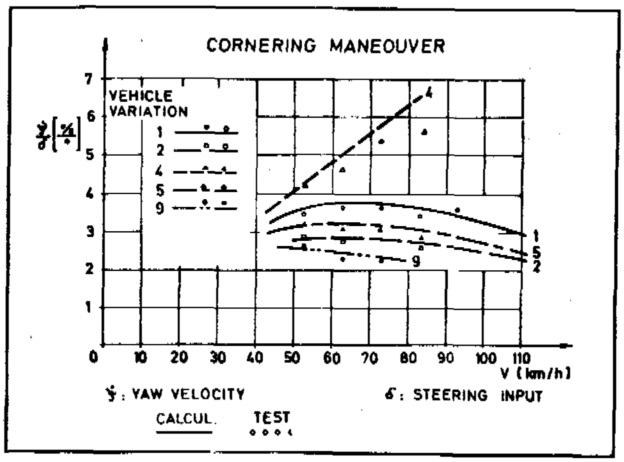
In order to find a relation between the subjective assessments of the driver and the results of calculations and measurements – that means with the response functions of the car – all investigated modifications on the car were evaluated subjectively by a driver team.

Results

1. Entry Into Circle

The test manoeuvre "entry into circle" was carried out as follows: Under the initial conditions of a straight ahead drive with various constant speeds the car was subjected to a constant steering input and the steering wheel was held constant until steady state cornering condition was reached.

The following Slide 1 shows the yaw velocity gain; that is yaw velocity divided by steering input angle,



Slide 1

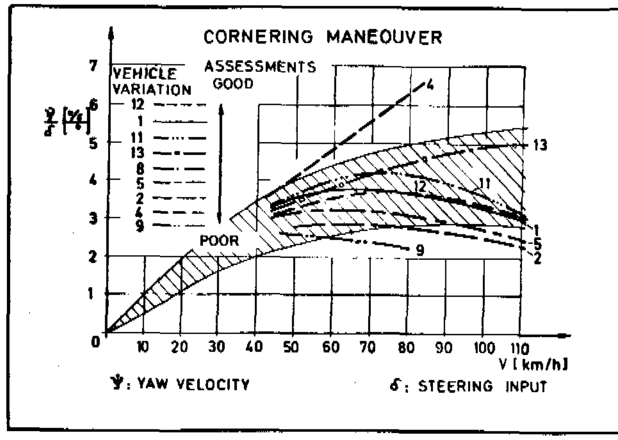
versus vehicle speed for steady state cornering condition.

This diagram is explained on the following example. For steady state cornering the yaw velocity is determined as the ratio of vehicle speed to turn radius. For the same vehicle speed and the same radius the car with a stronger understeering tendency requires a greater steering wheel input, the ratio of yaw velocity to steering angle of road wheel becomes smaller.

With increased understeering tendency of the vehicles the yaw velocity gain, the ratio of yaw velocity to steering input, becomes smaller.

The curves in the diagram are derived from computer calculations, the dots belong to the test results and the figures indicate the various vehicle modifications. The conformity between results of calculations and tests have been found good, only for Vehicle variation No. 4 with more oversteering tendency was the deviation of tests from prediction larger.

In the next Slide 2 all results are compared with the subjective assessments of a driver team.



Slide 2

First vehicle variation 1 with the original condition was rated as the best one, the most oversteering vehicle 4 and especially strongly understeering vehicle 9 have been judged to have the poorest handling qualities.

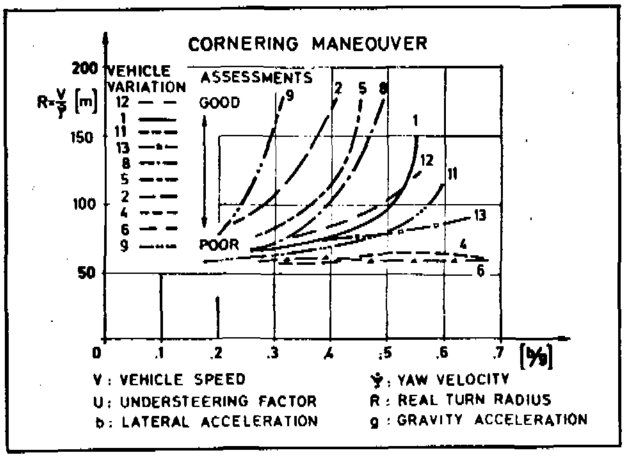
Based on computer calculations it was tried to achieve a further improvement in the subjective rating of handling qualities with small modifications on vehicle 1. This was succeeded with vehicle variation 12.

Due to the correlation of test and calculation results with subjective assessments the hatched area has been settled as the acceptable range.

The following Slide 3 shows the ratio of vehicle speed to yaw velocity, which is corresponding to the turn radius for steady state cornering, versus lateral acceleration divided by gravity acceleration. The steering input was the same for all vehicle variations.

For the same steering input and increasing vehicle speed the turn radius is grown up rapidly with an increase in understeering tendency. From this diagram an "understeering factor" for steady state vehicle cornering can be derived, similar to Hamann's definition.

The understeering factor U is defined as the gradient of the turn radius versus lateral acceleration divided by the Ackermann Radius for the corresponding steering input.



Slide 3

For evaluation, the "understeering factor" at .4 g lateral acceleration has been used. This diagram shows again, that both the more oversteering vehicles 4 and 6 and the strongly understeering vehicles 9, 2 and 5 have the poorest ratings.

The curve of turn radius for constant steering input versus lateral acceleration shows for the vehicle variation 12 – which has been rated as the best one – a flat shape without a sudden increase in turn radius for any lateral acceleration value.

2. Slalom Test

The second type of manoeuvre was a slalom test. The test track, which was marked with pylons, was driven through with step by step increased vehicle speeds. Having reached the highest possible speed for a certain wave length the distance of pylons has been increased for the next test run. The wave length was between 40 and 80 m.

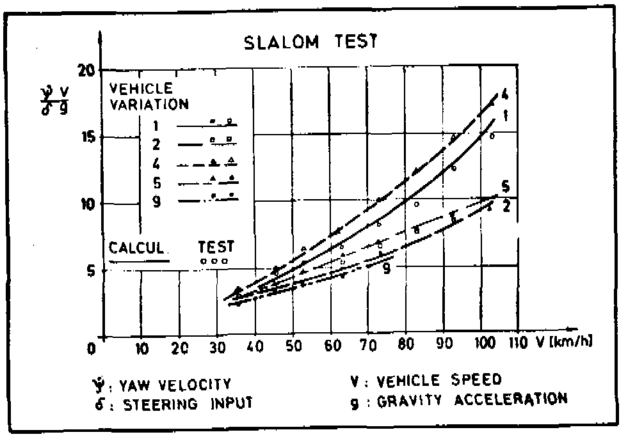
Different possibilities of plotting the results have been investigated, with the aim to get a simple graph which is suitable for a comparison of the test results with subjective assessments and which clearly indicate the difference of the various vehicle modifications at increasing vehicle speed.

From the vehicle response functions, the ratio of lateral acceleration to steering input versus vehicle speed was plotted first. As the slalom track was the same for all vehicle variations, also the lateral acceleration must be nearly the same for same vehicle speeds. Only the necessary steering wheel angles for driving through the slalom track are different, due to the vehicle variations.

The lateral acceleration "b" is equal to sum of yaw velocity "$\dot{\psi}$" and slip velocity "$\dot{\beta}_S$" multiplied with vehicle speed "v."

In order to analyse the handling qualities of the different vehicle variations it seems more convenient to investigate the two components of the lateral acceleration separately.

The ratio of yaw velocity $\dot{\psi}$ multiplied with vehicle speed to steering angle at the front wheel versus vehicle speed is shown in Slide 4. In order to get a coefficient without dimension the value has been divided by gravity acceleration.



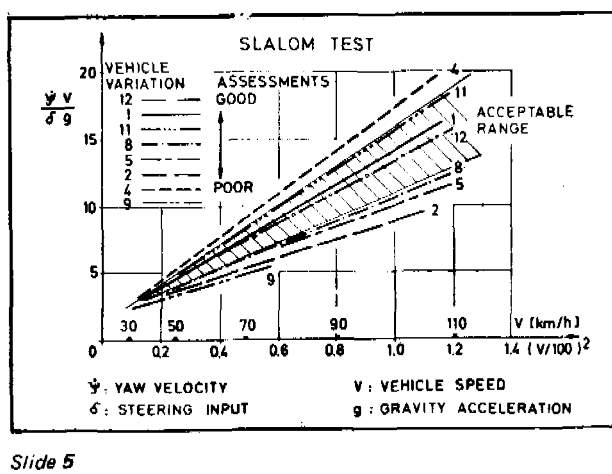
Slide 4

The curves in the diagram are derived from computer calculations, the dots belong to test results which are averaged values from several sinusoidial movements per test and from several test runs.

In conformity between the results of calculations and measurements was found good, the calculated difference between vehicle 5 and 2 was more than that which was obtained from road tests.

The diagram indicates that the upper range of curves corresponds to vehicles with a higher yaw response; that means an increasing tendency of oversteering because for the same vehicle speed and steering input the yaw velocity becomes larger.

These results are compared with the subjective assessments in Slide 5. In this diagram the slalom



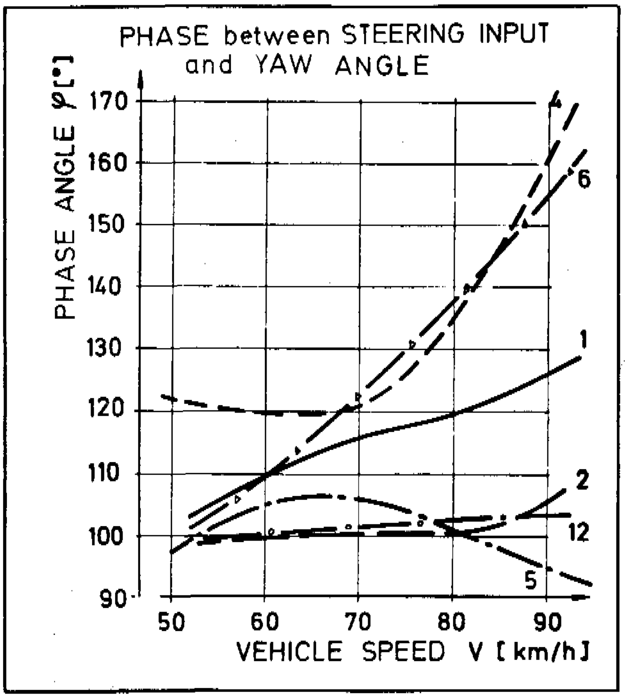
Slide 5

coefficient is plotted over the squared vehicle speed divided by 100 km/h which delivers a dimensionless graph. The parabolic curves of the last diagram have become straight lines in this graph.

The slopes of the different straight lines are used as characteristic values for comparison with the subjective evaluation.

The vehicle No. 8 was accepted as "still acceptable understeering," which draws the limit to acceptable understeering tendency. The limit to oversteering tendency was found by calculations and subjective assessments and vehicle No. 11 has had about the same rating as vehicle 8.

Slide 6 indicates the change of phase relation between steering input and yaw angle versus vehicle speed for the different vehicle variations.



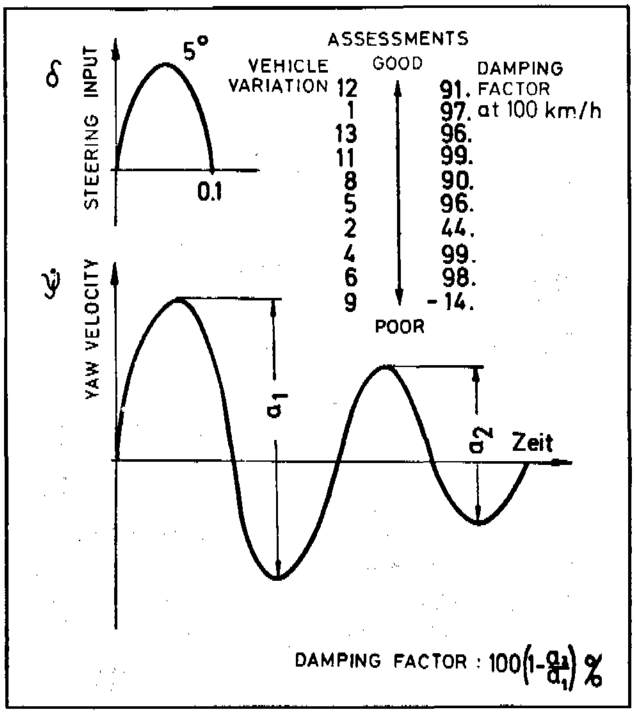
Slide 6

The diagram indicates the increase in lead time for the steering input with an increase in vehicle speed for vehicles with a more oversteering tendency.

It is interesting to note, that the vehicle variation 12 – which was evaluated subjectively to be the best one – has a smaller phase difference between steering input and yaw angle than vehicle variation 1, whereby in this case the phase relation is nearly constant with increasing speed.

The next step was the investigation of the yaw damping as shown in Slide 7.

The factor one minus the ratio of the second to the first double amplitude of the yaw velocity was used as a criterium. Investigations have shown that this factor should not essentially be lower than 80%. The negative value of the damping factor for vehicle 9 indicates that this vehicle is oscillatorily instable. This fact was also identified during road tests.



Slide 7

The yaw time period was between 0.7 and 1.4 sec. for the different vehicle variations.

The vehicle variation 12, rated subjectively as the best one, has a yaw time period of 0.8 seconds.

The objective of this investigation was to develop a suitable mathematical model for the prediction of vehicle handling qualities, to compare the results of road tests with calculations and to correlate these with the subjective assessments of a driver team. A successful correlation between theoretical results and measurements could be found for the investigated test manoeuvres. The investigation has indicated that several criteria should have values in comparatively small ranges for an acceptable handling performance.

These investigations should be considered only as an attempt to define handling qualities with suitable objective criteria.

Further extensive investigations are necessary to get confidential correlation between subjective assessments and vehicle response functions gained from road tests and prediction.

Further targets of these handling investigations are to develop handling measuring procedures, which are satisfactory repeatable and are not complicated. The test engineer should be able to use these methods without complicated hardware and these should be applicable to check the future legal requirements concerning handling performance as well.

THE MEASURING OF THE DYNAMIC BEHAVIOR OF VEHICLES

Mr. Claude Berlioz, O.N.S.E.R.

The dynamic behavior of vehicles is actually appraised, in the great majority of cases, in a subjective manner by professional testers. The general problem that arises is to render this appraisal objective. It is a question of measuring the behavior of a vehicle by guide marks rather than measuring it in an exact sense, since it is not a measurable quantity, the sum of two behaviors seeming difficultly definable.

I. Use of Models

1. Analytic Models

We will call analytic model a model of which the parameters are determined by a knowledge of the characteristics of the make-up of the vehicle: body, suspension system, shock absorbers, tires, direction, etc. Their purpose is to foresee the consequences of certain modifications of a vehicle on its behavior under certain circumstances, and even, in a more ambitious perspective, to foresee the behavior of a vehicle still in a projected state. Such models have been studied by O.N.S.E.R. in collaboration with Charles Deutsch, and are still being studied. It seems difficult to quantitatively represent a vehicle in a precise manner, even with the help of relatively simple models, especially with the differences in phase. One can, on the other hand, obtain valuable qualitative indications.

This type of model, therefore, due to its imprecision and its theoretical character, does not supply a satisfactory answer to the measuring problem; people are often little familiar with the validity of the schemes and simplifications that are adopted.

2. Empirical Models

We will call an empirical model, a model of which the parameters are adjusted due to the knowledge of the real behavior of the vehicle in certain circumstances. Their purpose is to sum up reality by a group of coefficients permitting the reconstitution of this reality in a rather precise manner.

There exist relationships between analytic and empirical models, the choice of the form of the empirical model is guided not only by the rate of the responses of the vehicle at certain entries but also by the form of the aforesaid analytic models. Inversely, the precision with which one arrives at setting up an empirical model of a given form allows one to judge the validity of any hypotheses leading to an analytic model of the same form, independently of measuring errors which might affect the coefficients of the latter, the particulars beginning with the components of the vehicle.

An empirical model is relative to a domain of its usage, defined by limits of speeds, limits of acceleration, limits of the grip of the wheels and limits of the frequency of entries. These entries are the orders of the driver, the steering wheel, the accelerator, the brakes, and the imperfections of the testing area, such as the unevenness of the roadway or aerodynamic disturbances. A vehicle will not, therefore, be represented by an empirical model but by a group of models relative to diverse domains of usage. These domains should be sufficiently vast to limit the number of models, but restricted enough to obtain a precision sufficient for the interior of these domains. In fact, the choice of one of the models depends upon the particularities of the behavior in which one is interested.

3. The Interest of Empirical Models

Independently of any models, one could seek to define the dynamic behavior by an untested result, representative of a certain domain of usage. This result could be:

–A maximum test speed. This presents the serious inconvenience of depending very heavily upon the qualities of the driver. It indicates nothing about the easiness of the drive, and the consequences of an error in appraisal.

–The maneuvering turns of the driver to obtain a definite trajectory beforehand with a certain precision. These maneuvers allow an appraisal of the simplicity of the driving, but not for the consequences of an error in the appraisal. Moreover, the tests could become very long in order to obtain the maneuver adequate for a very precise, given trajectory in difficult conditions.

–Trajectories obtained following automatic entries. This type of test presents the inconvenience of necessitating very large asphalt surfaces, as soon as one operates at realistic speeds, the response of one given entry being able to very enormously from one vehicle to