論文　**Original Paper**

An Investigation on the Design Method of Suspension System for Small Formula Type Vehicle − Especially, Design Method of Suspension and Steering System −

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Abstract: The main purpose of Formula SAE Competition (hereafter called "FSAE", "Formula Society of Automotive Engineering") is to let students learn the basic ability necessary for engineers through design, fabrication and test projects [01]−[11]. Higher running performance of a manufactured vehicle is one of the most important themes that should be studied in Student Formula Japan Competition (hereafter called SFJ Competition) [01]−[11]. Also, SFJ Competition is the series of the FSAE. The purpose of this study, the chassis must be required light weighting and high stiffness. The former can reduce the centrifugal force and the inertial force in the turning and the latter can contribute to demonstrate the suspension performance according to design [12]−[20]. The SFJ Competition has Skid Pad event to compete for steerage responsiveness and high suspension performance on turning. The balance of the highly performed engine and chassis requires to keep high running performance of competition vehicle. Additionally, it is necessary for improvement of the drivability to not only improve the engine torque in the low engine speed range, but also to improve the suspension performance infer the unsteady conditions such as slalom and acceleration at corner exit. This study refers to designing and manufacturing a competition vehicle for SFJ Competition with higher torsional rigid and bending rigid chassis and high speed running performance. I perform light weighting and a design, the production of the chassis of high-stiffness SFJ Competition vehicle and weigh the design level against an actual machine run examination. In this study, we investigated the differences between the design values and the test results by a vehicle with the chassis of light weighting and high-rigidity SFJ Competition. As the summary,

[1] The rigid improvement of the chassis with big driver space reducing chassis rigidity is a chassis concept of this study. The improvement of the torsion rigidity and the bending rigidity of the chassis was confirmed by both results of the numerical computation and the experiments.

[2] The change of the camber angle at every each steering angle under the conditions of the vamping, bounding and a rolling can be controlled by the wheel alignments of suspension geometries decided in this study.

[3] A suspension system, steering system and brake system can be manufactured by the comparison between data obtained from the run experiments and design data in order to aim a high rank in the SFJ Competition.

Key words: Design Method, Suspension System, Steering System, Formula Type Vehicle, Numerical Calculation, Finite Element Method, Experimental, Running Test

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1. Vehicle Dynamics

The vehicle dynamics is used in depth for development of the vehicle suspensions from passenger cars to trucks and buses [01]−[11], [13]−[20]. The suspension and the steering system must be also designed for our Student Formula Japan vehicle to utilize drivability and engine performance enough. For achievement of the purpose, the theory analysis of the vehicle dynamics of vehicle should be carried out as the stage before the design and manufacture of the vehicle. The one side two-wheeled vehicle model, which is called bicycle model, is assumed as the analysis of the suspension for the Student Formula Japan which is a four-wheeled vehicle. The bicycle model's parameters are as follows;

[1] Wheelbase.

[2] Weight distribution.

[3] Steering gear ratio.

Figure 1 shows the external representation of the design vehicle drawn in three-dimensional CAD for planning to manufacture. A three-dimensional coordinate is drawn on this Figure to investigate the behavior of the vehicle. X-axis, Y-axis and Z-axis show cross direction, progress direction, and vertical direction respectively, locating the origin of the coordinate in the gravity center of the vehicle. In addition, the coordinate where dash was marked shows coordinate transformation occurred by some kind of behavior of running vehicle. **Figure 2** shows the change of the yaw rate to the length of the wheelbase. As a matter of course, the decrease of the wheelbase indicates the increase of the yaw rate, but all results indicate that they converge in a certain value. **Figure 3** shows yaw rate to the change of the weight distribution that is the ratio of front weight to rear weight assuming that vehicle weight is constant. It is shown that the yaw rates decrease with decrease of the front weight. These calculation results indicate that the yaw rate greatly influences on easiness for drive at turning and slalom.

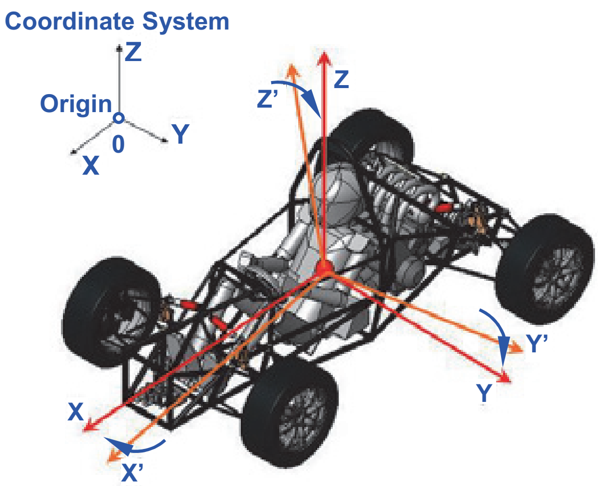


Figure 1 Coordinate System and Origin of Three-Dimensional CAD Solid Model and Numerical Calculation for Student Formula Japan Vehicle

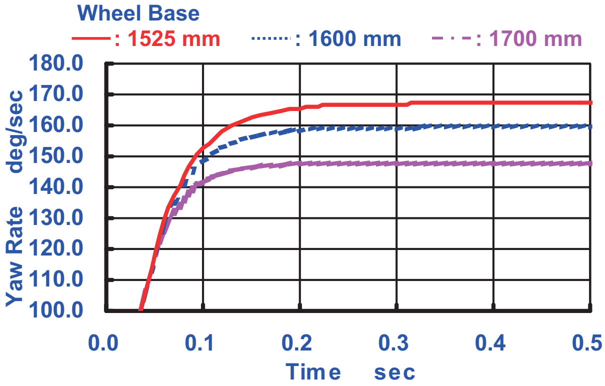


Figure 2 Change of the Yaw Rate [deg/sec] to the Length of Wheelbase [mm]

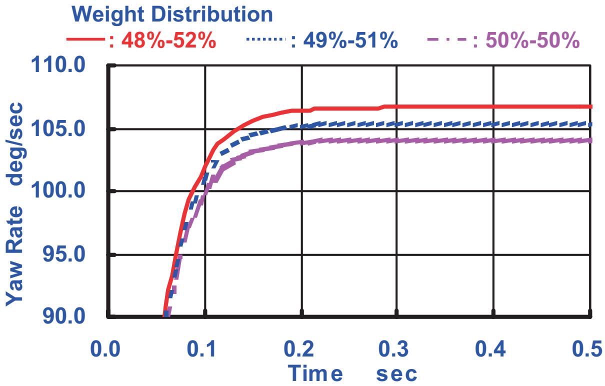


Figure 3 Change of the Yaw Rate [deg/sec] to the Weight Distribution of Front and Rear Weight [%]

However, it cannot be said that only the expansion of the wheelbase influences cornering performance without the discussion of the other influence factors such as tires and stabilizer also on the performances.

2. Design of Suspension System

2.1 Select of Suspension Types

The push rod system is adopted for our vehicle's front and rear suspensions. The function of the push rod is able to expand easily by the increase of the quantity of stroke by the crank and the adjustment of the damping coefficient of the shock absorber. Moreover, the stabilizer was decided to an adjustable stabilizer. A leaf spring is employed as the suspension arm as it is able to change the bending rigidity with the turning of its own. In other words, this leaf spring can change the bending rigidity with quantity of bending displacement of its own by bending moment occurred by the rolling of the vehicle. As a result, it is possible to coordinate the spring constant of the leaf spring to the roll angle of the vehicle. The three-dimensional solid CAD model of the stabilizer for the rear suspensions is shown to **Figure 4**.

Figure 4 Three-Dimensional Solid CAD Model of Rear Stabilizer for Rear Suspension

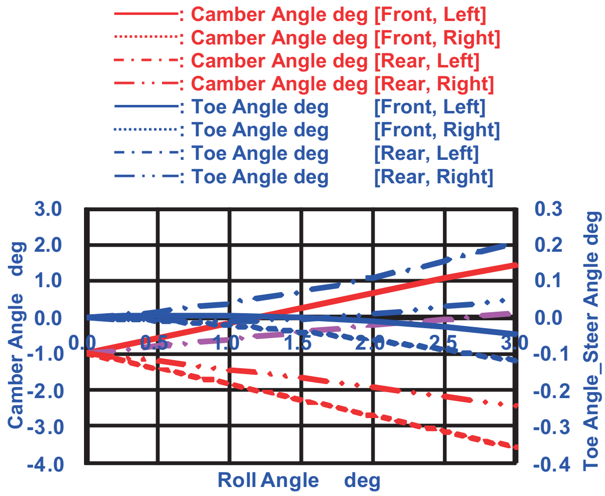


Figure 5 Relationship between Camber Angle [deg], Toe Angle [deg], Steer Angle [deg] and Rolling Angle [deg]

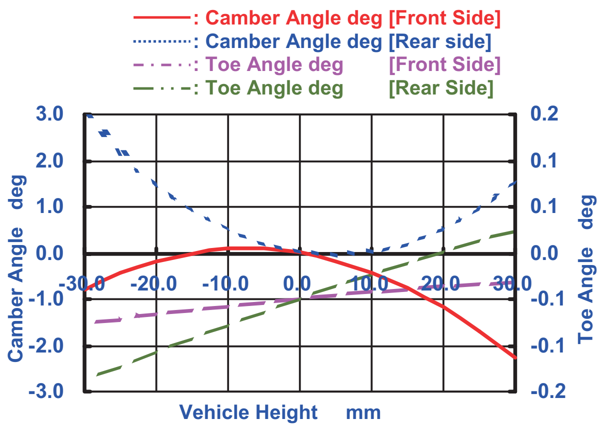


Figure 6 Relationship between Camber Angle [deg], Toe Angle [deg] and Vehicle Height [mm]

2.2 Investigation of Suspension Geometry

The suspension arm adopted A type of double wishbone type. Wheel alignment must control wheels to improve driving stability. Traction force making the run of the vehicle is influenced by the road surface grounding of the tire, but the change of the alignment makes not only bigger resistance force against running but also the instability of the vehicle behaviors. The investigation items that are paid attention to setting in suspension geometry are camber conjugation and toe change at turning in the course run, accelerating and braking. The front suspension must bear many of braking forces as the gravity center position of the vehicle moves forward at rapid speed down such as braking. The change of the camber angle at bump and rebound must be considered to decrease the instability of the vehicle as much as possible. The camber angle change at the bump and rebound is determined by using a difference among swing arm, top arm length and bottom arm length. In other words there are few camber changes in the case of the longer swing arm, and the camber changes increase in the case of shorter arm. Therefore, the swing arm had better be long. The length differences of the upper and lower arms influences on the curve of the camber change curve. If an upper arm is shorter than a lower arm, camber changes increase due to a swing arm shortening with the increase of the bump. The rear suspensions bear all of traction forces to accelerate at the corner escape. It was set for the achievement of these matters to reduce the camber change in the rolling. Thus, it was decided to shorten the swing arm. Furthermore, the lower arm is extended to body center part to match the trace of the upper and lower arms with the trace of the body in the roll. The toe changes of both of the front and the rear were set to shrink as possible in bump, rebound and roll. **Figure 5** shows the relation among of a camber angle and the steer angle against the roll angle. In addition, **Figure 5** indicates the relations of the camber angle and the toe angle against the roll angle of the body. It is found that the change of the camber angle of the rear suspension is bigger in comparison with that of the camber angle of the front suspension. This calculation results show that rear tires cannot convey traction force enough at turning. Furthermore, a vehicle cannot show enough turning characteristics because the sidewall of the outside rear tire is transformed by the minus camber angle. As the toe angle changed by a roll occurring at turning influences on steer characteristics, the drive characteristics at turning turns worse. **Figure 6** shows the relations of a camber angle and the toe angle to the car height changed by suspension stroke.

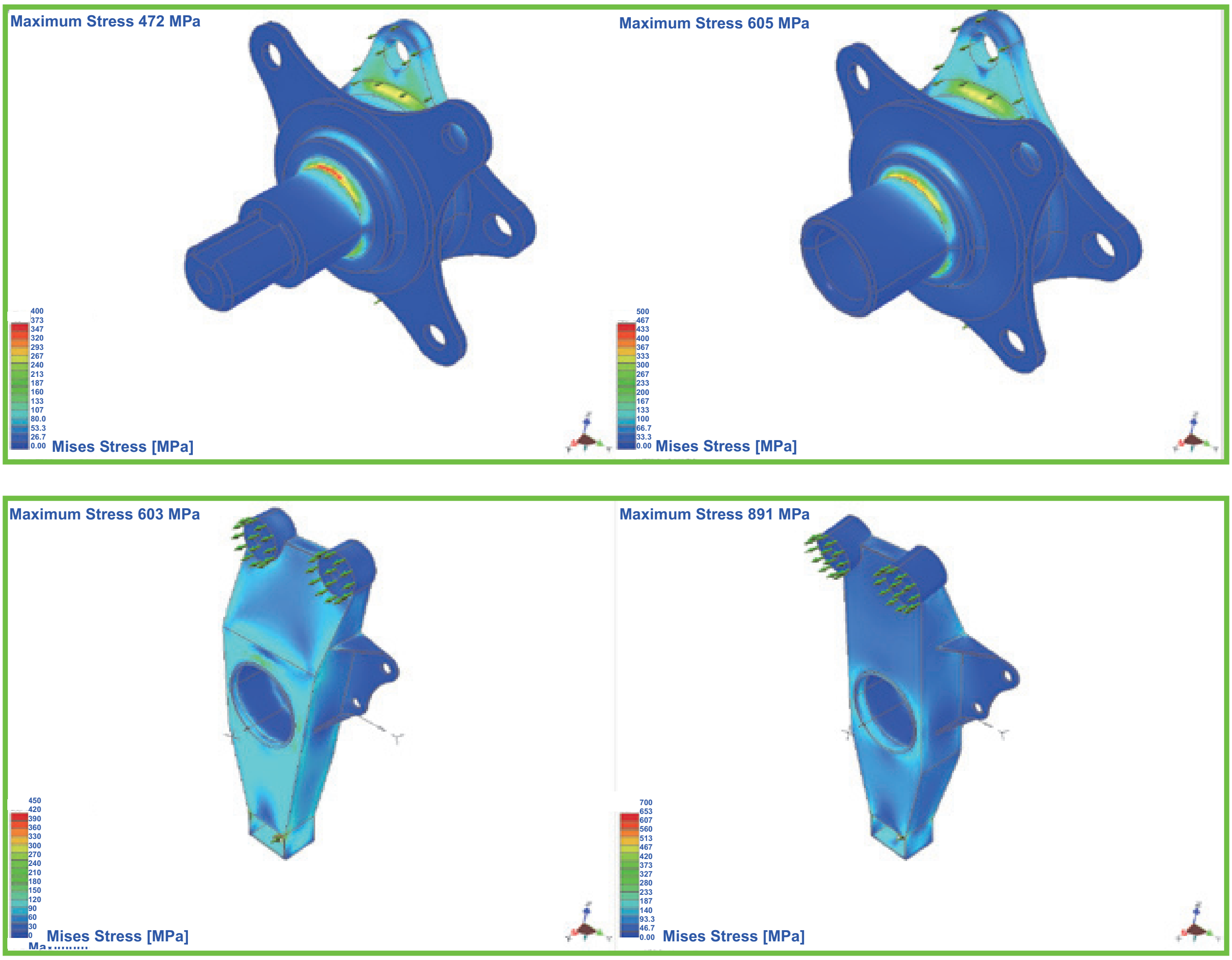


Figure 7 Stress Distribution Maps obtained from the FEM Analysis of the Front Wheel Hubs, Rear Wheel Hubs and Front Upright, Rear Upright (Conventional Design Model)

This Figure also indicates the change of the camber angle and the toe angle to the change of the car height. This Figure shows change of the suspension geometry with the up-and-down motion of a vehicle occurred at bump and rebound. Even if the behavior change of the vehicle occurs, it is desirable that the suspensions have a few changes of camber angle and toe angle. The structure of the suspension must be decided after shape and the length of each part constituting suspension were studied enough. However, an anti-roll bar and a stabilizer may be needed to keep the minimum geometry change of the suspension if suspension structure cannot adequately control change of the suspension geometry.

2.3 Design of Wheel Hubs and Upright

The wheel hubs are designed and manufactured by our own works to reduce the quantity of load under spring. **Figures 7** shows comparison between the shapes of the wheel hubs and uprights in the previous studies and this study. This Figure also shows stress distribution maps obtained from the FEM (Finite Element Method) analysis of the front wheel hubs and rear wheel hubs. The upper Figures show the front wheel hubs and the lower Figures show the rear wheel hubs. The left side Figures show hub and upright designed in our previous studies and the right side Figures show hubs and upright designed in this study similarly. Comparing between these stresses distributions (Mises Stresses) in these front hubs, it is found that the hub in this study does not indicate a red color area displaying stress concentration. In other words, it is shown that the maximum stress value of the hub in this study is lower. The analysis results show that the shape to round edges succeeded in maximum stress reduction while the thickness of each part of the hub was reduced for light weighting. **Figures 8 (a), (b)** shows the stress distributions of the front and rear upright by FEM. It is confirmed that these uprights are worthy of the external forces occurring during vehicle traveling by making the uprights shapes that stress

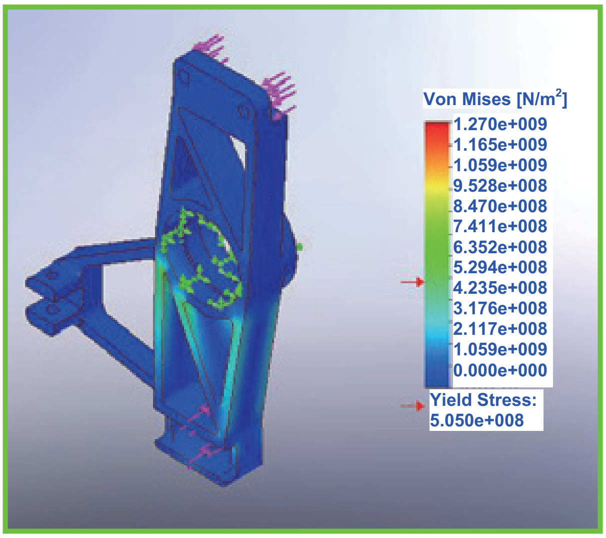


Figure 8 (a) Stress Distributions in the Front Upright by FEM Analysis (New Design Model)

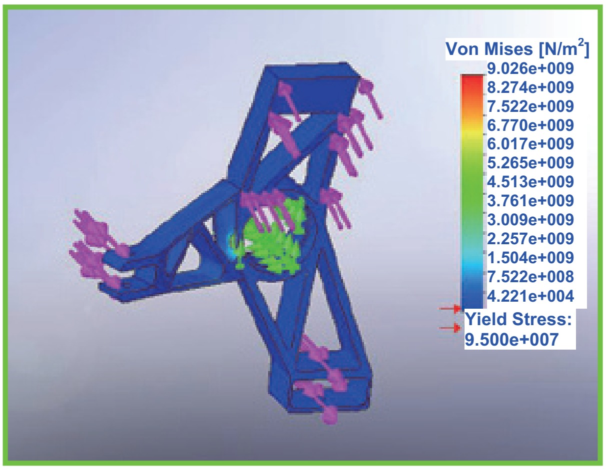


Figure 8 (b) Stress Distributions in the Rear Upright by FEM Analysis (New Design Model)

concentrations are hard to generate in.

3. Design of Steering System

The steering system is classified roughly by Ackermann steering and a parallel steering system. Ackermann steering system requires that the turning center exists on the extension line of each axle in order to harmonize a surface of revolution and the progress direction of each wheel at turning. The parallel steering system is a system making steering angle of the front wheels the same.

3.1 Satisfaction of Demand

It is important for the satisfaction of the demands that extension lines of the right and left front axle always cross in the extension line of the rear axle in one point for the

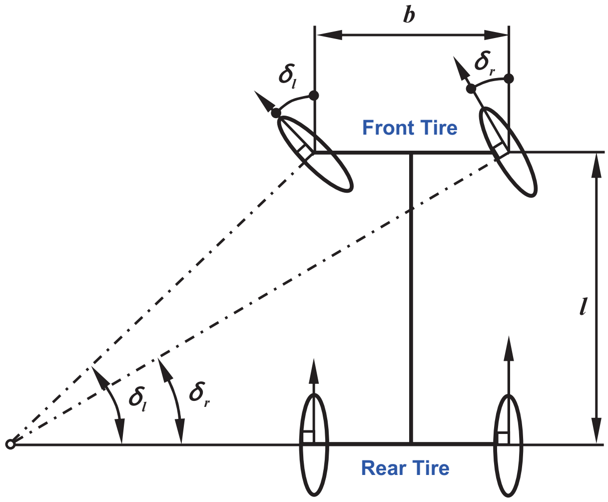


Figure 9 Schematic Diagram of Ackerman Type Steering System Model

decision of the steering wheel angle as shown in **Figure 9**. As shown in this Figure, the relations of each extension line are expressed in the next equation.

$$\cot \delta_r - \cot \delta_l = \frac{b}{l} \qquad (1)$$

Here, δ_r, δ_l : the steering wheel angle [deg] of the right front wheel and the left front wheel respectively, b : tread [mm], l : wheelbase [mm].

This equation means that the inside steering wheel angle must always satisfy to be bigger than the outside steering wheel angle if tread is longer than wheelbase. However, it is difficult to realize the linkwork that always satisfies equation (1). As Ackermann steering is a trapezoid linkwork consisting of a knuckle arm and a tie rod, it cannot realize equation (1) precisely. In addition, the theory of the Ackermann steering is a precondition that all wheels do not generate sideslip. Unlike the above-mentioned Ackermann steering theory, the parallel steering theory assumes that a vehicle turns while either front wheel is slippery without the turning center converging in one point. As results of these investigations, the adoption of a steering system placed midway of Ackermann steering and the parallel steering system were decided to correspond to many kinds of corner in racecourses.

3.2 Design of Steering System

The following two assumptions are decided to design the steering system;

[1] The maximum steering wheel angle that the driver can turn with his holding the steering wheel is 180 [deg].

[2] The movement of a rack determined from a gear ratio of the lack work is 40 [mm].

The steering system is designed with the use of diagram processing (diagram elucidation) shown to **Figure 10**. In addition, the signs of this Figure indicate each angle. Moreover, the tie rod position was brought close to isometric and parallelism to an upper arm as much as possible to reduce the toe change. And the wearing position of the rack was set to several ways in front and back direction without disturbing driver space. **Figure 10** is overview of the steering system being given rudder angle in right direction. A trapezoid linkwork constituting in the knuckle arm and the tie rod produces differences between the rudder angle of the inside tire and that of the outside tire. The difference between the rudder angles of both tires at every angle of the steering wheel was check by diagram processing. This diagram processing is carried out till smooth turning is possible while changing a trapezoid linkwork. The angles of the tie rod were checked from directly overhead of each set wearing position to satisfy an above-mentioned supposition. As a result, the positions of the rack and the tie rod where are the nearest to the quantity of movement of the rack 40 [mm] were determined in consideration of the interference such as frames. Where, α_1, α_2: rudder angle [deg]. **Figure 11** shows the relation of movement quantifies of the rack and steer angle of the tire comparing between the experimental values and the calculation values by using geometry software. Furthermore, this Figure shows this diagram processing results with the difference of the rudder angle of both tires against the quantity of movement of the rack. As much as a turning radius becomes smaller, this difference grows bigger. As the difference of these steering angles correspond with the crossing angle of the normal of outside and inside tires, this intersection angle increases with the decrease of turning radius. This Figure shows quantity of rack movement against the camber angle. The wheel angles of the tire became 29.1 [deg] at inside tire and 32.5 [deg] at outside tire. The difference between wheel angle of the inside and outside wheel almost agreed with the design value in the maximum wheel angle of the tire. The steering system almost satisfied the design values in the driving tests. **Figure 12** shows the relations of the change of the camber angle and the turning angle of the tire was substituted for quantity of movement of the rack. This Figure shows the calculation results assuming the right cornering of the vehicle, and the camber angle of the left side front tire located in outward displays the value of the minus. The cornering speed is in inverse proportion to a turning radius, but the camber angles of the both side tires are calculated to approach zero by every roll angle calculated by assumed cornering speed. **Figure 13** shows measurement results of steering stroke by running test (Refer to **Figure 17**). The steering stroke is quantity of change for an appropriate turning angle of a steering decided at the radius of the corner. It is defined for quantity of movement of the circumference direction of the steering.

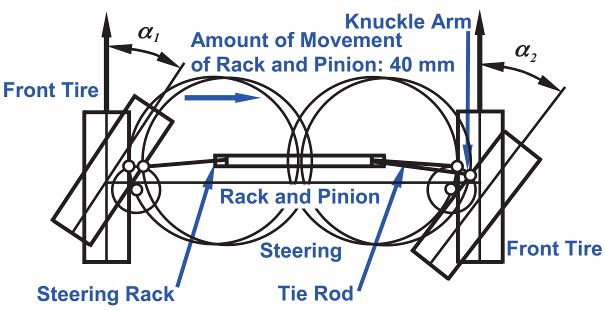


Figure 10 Overview of the Steering System being given Rudder Angle in Right Direction

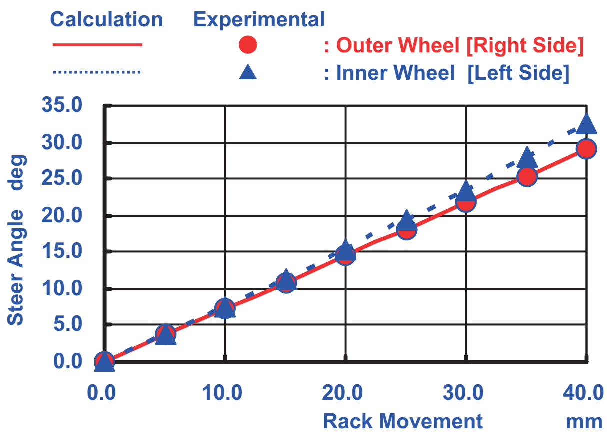


Figure 11 Diagram Processing Results with Difference of Rudder Angle [deg] of both Tires to the Quantity of Movement of the Rack [mm]

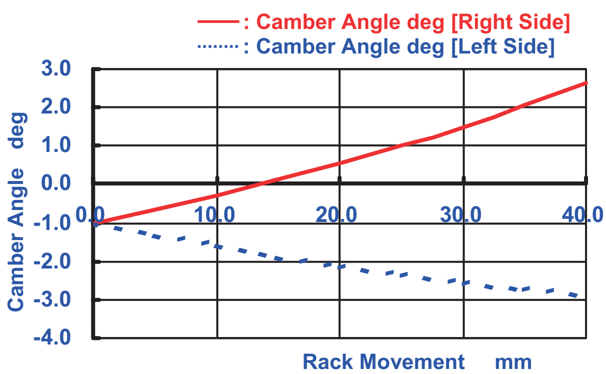
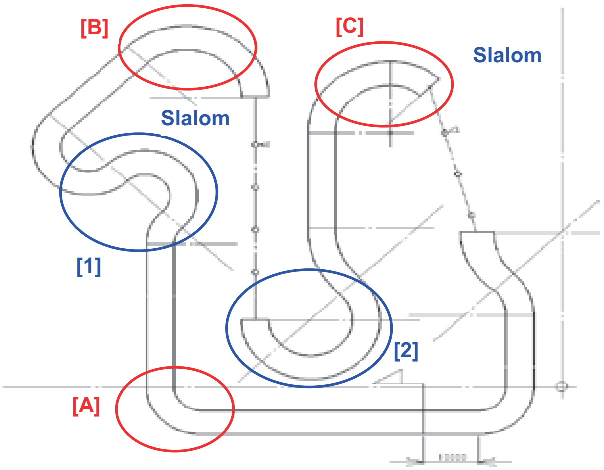


Figure 12 Relations of Change of Camber Angle [deg] and Turning Angle [deg] of Tire was Substituted for Quantity of Movement of Rack [mm]



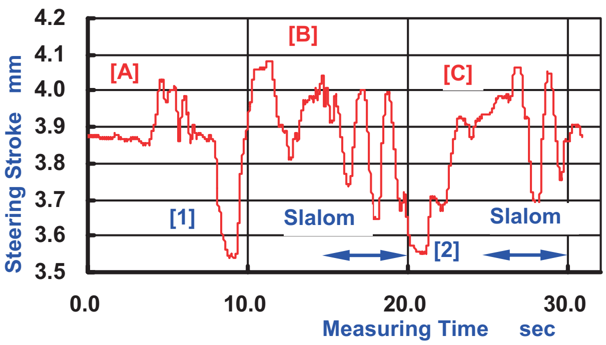


Figure 13 Measurement Results of Steering Stroke by Running Test [Refer to Figure 17]

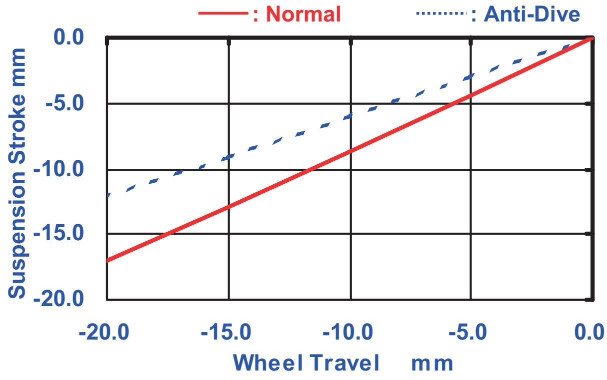


Figure 14 Numerical Computation Results of the Anti-Dive and the Anti-Squat Geometries

4. Vehicle Movement Control

During a vehicle movement, the motion of the vehicle is divided into three items in driving.

[1] Braking.

[2] Acceleration.

[3] Turning.

These items need to require the design of a suspension with high-level functions. The suspension to make tires hold sufficient mechanical grip is necessary. The following systems will be incorporated in the designed suspension to achieve the purpose;

[a] Anti-geometry.

[b] Anti-roll bar.

4.1 Anti-Geometry

Wheel displacements and suspension strokes are investigated by numerical calculation of the anti-dive and anti squat geometry to control the pitching of the body at the vehicle motion. In these calculations, two calculation models are adopted. The first calculation model with anti-geometry can decrease a suspension stroke and the second model in which suspension stroke was 28.4 [%] is employed at maximum displacement 20 [mm] of the wheel of both wheels. The adoption of the anti-dive geometry model made the pitching control of the vehicle possible and contributed to the stability of the vehicle in braking and accelerating conditions as shown in **Figure 14**. This Figure shows numerical computation results of the anti-dive and the anti-squat geometries to control the pitching of a vehicle occurring at brake and acceleration. Suspension strokes on the vertical axis and wheel displacement on the horizontal axle are shown in this Figure. The calculation result means that the change of the suspension stroke is reduced to dynamic load fluctuation occurring during vehicle traveling. The difference between these calculation results and the results obtained from the normal suspension geometry grows clearly bigger with the increase of the suspension stroke. As a result, the stabilities of the dynamic behaviors of the vehicle are effective for the drivability of the driver.

4.2 Stress Distribution of Anti-Roll Bar and Numerical Computation of Quantity of its Displacement

An anti-roll bar is required to control of the roll direction of the body. Using the results of the numerical computation in plural models and the torsional experiment, an adapted anti-roll bar is decided for our vehicle. **Figures 15 (a), (b)** and **(c)** shows the stress distribution maps of the anti-roll bar by FEM. It is found that stress concentrations generate on the ends of the arms that receives the external force indicated in arrows. The stress concentrations are different at angle of a leaf spring installed to the arm of the anti-roll

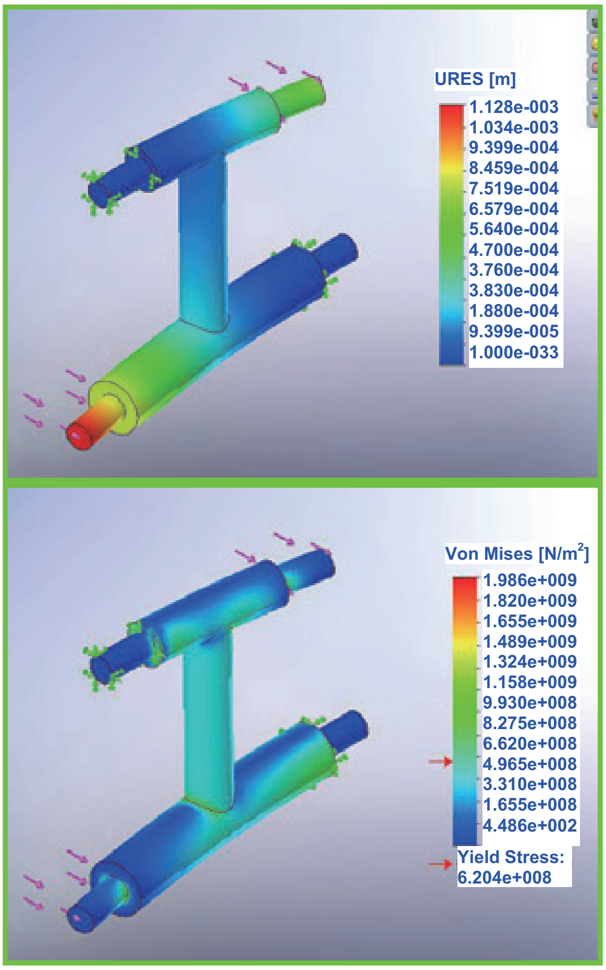


Figure 15 (a) Displacement and Stress Distribution Map of the Anti-Roll Bar by FEM Analysis (Displacement, Mises Stresses, Type: H)

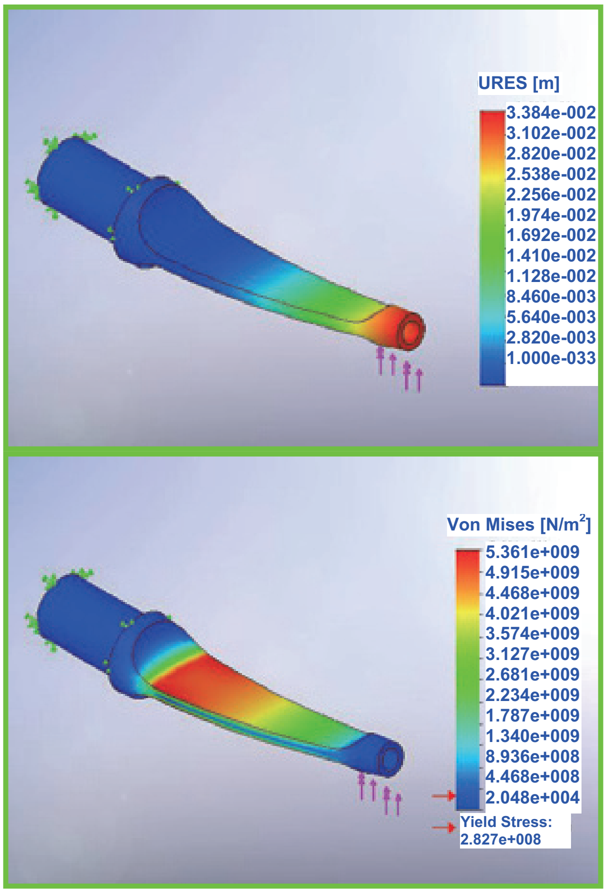


Figure 15 (b) Displacement and Stress Distribution Map of the Anti-Roll Bar by FEM Analysis (Displacement, Mises Stresses, Type: I, Horizontal Direction)

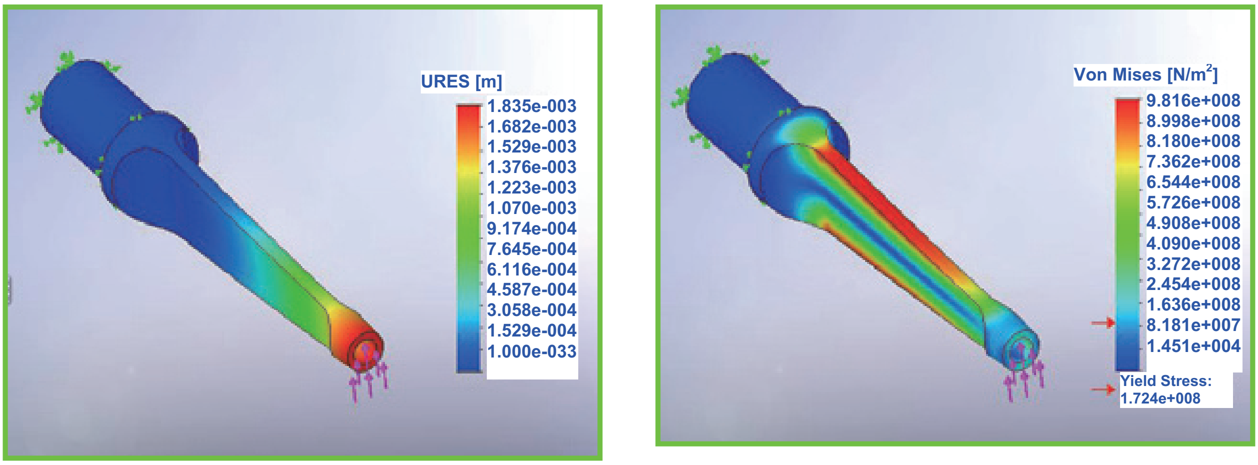


Figure 15 (c) Displacement and Stress Distribution Map of the Anti-Roll Bar by FEM Analysis (Displacement, Mises Stresses, Type: I, Vertical Direction)

bar. The leaf spring controls rolling to link the suspension of right and left to the frame and can change a flexural rigidity by an installation angle to the frame. The root has the difference of the second moment of cross section area by the installation angle. Therefore the offset of the center axis of the arm for the line of action of the external force is set to

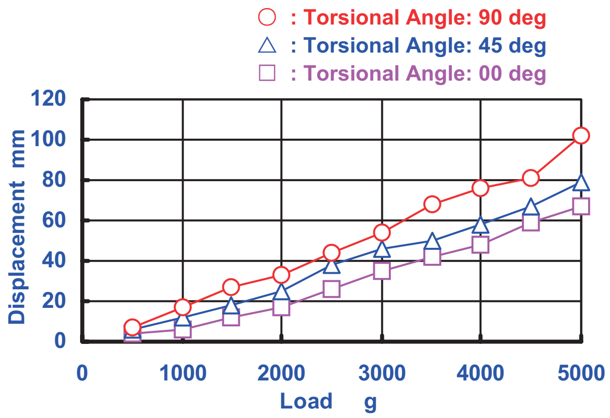


Figure 16 Torsion Test Results of Anti-Roll Bar [Torsional Angle : 0 to 90 [deg]]

twist the arm.

4.3 Twisting Test of Anti-Roll Bar

The measurement of the quantity of twist displacement against the load was carried out by the following experiment to analyze a change of the rigidity because the rigidity of the arm made for an anti-roll bar was variable. By a difference of the rigidity of 30 [%] at the maximum, setting modification of the roll rigidity could be realized easily. **Figure 16** shows the test results. The parameters show the wearing angles of the leaf spring to the anti-roll bar. It is noticed that the wearing angle gives twist rigidity a difference. Therefore, the leaf spring twisted by external force can control the roll angle while changing twist rigidity.

4.4 In-Lift Chilling Effect with Anti-Roll Bar

If the rigidity of the anti-roll bar matches the calculated distribution of the front and back load movement quantity, it will be clear that the in-lift of the rear tire is restrained. The lower Figure shows test results obtained from run tests with the change of the specifications of the above-mentioned anti-roll bar. Because the rigidity of T-type anti-roll bar was too big, the fall of the quantity of load movement to the front suspension caused the in-lift of the front tire. Next, the run test of a vehicle changed to the anti-roll bar of the reverse U letter type was carried out similarly. It was confirmed that the front and rear tires of the vehicle that turned in skid pad of a diameter of 15.25 [m] produced a slight in-lift. It is shown in the lower Figure that the average speed of the vehicle with the in-lift falls 8 [%] in comparison with that of vehicle without the in-lift. **Figure 17** shows the running test courses layout (Refer to **Figure 13**). This course was divided into five sections. Each section time measurement is carried out to examine the effect of the

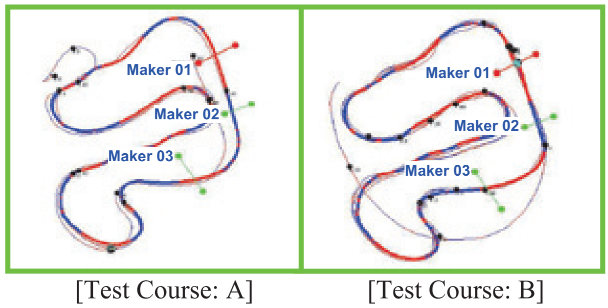
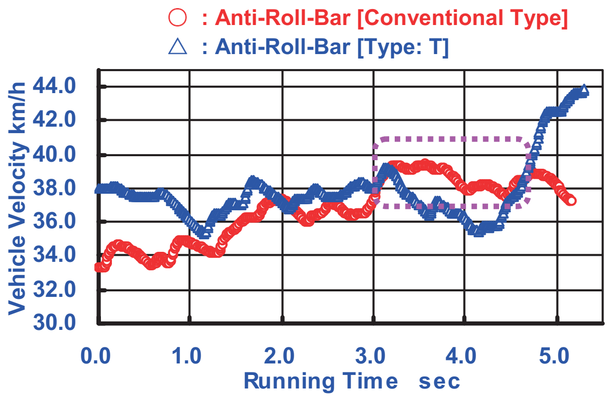


Figure 17 Layout of Running Test Courses [Refer to Figure 13]



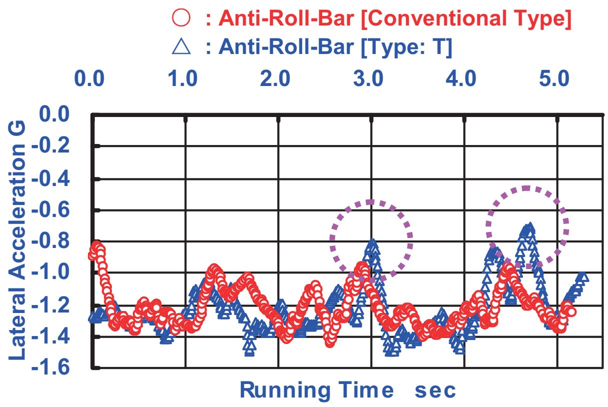


Figure 18 Results of Running Test [Vehicle Velocity [km/h], Lateral Acceleration [G]]

anti-roll bar to the vehicle speed. The results are shown in two graphs of **Figures 18**. Comparison with a conventional anti-roll bar and the anti-roll bar of the T-type is shown in the top Figure of these Figures. The bottom Figure shows the change of lateral acceleration instead of vehicle speed.

5. Running Test of Vehicle

The inspection of the development vehicle is carried out by the run test after comparing the results obtained from the prediction calculation with the results obtained from the bench tests. The theoretical compliance steer characteristics to the external force are calculated by the finite element method previously to inspect the precision of the measurement values of the bench test. The measurement values of the bench test result in the discrepancy of alignment more than 10 [%] against the calculation value. Therefore, the calculation values on the run test occurred in the average discrepancy of 25 [%] against the measurement values of the run test. These facts are one of big problems for design of an excellent suspension that the measurement values obtained from the bench test and the run test results are different from these calculations values. A solution of this problem is to reconsider the structure of the suspension arm and the components. Firstly, the structure and the mechanism of the suspension must be inspected again. Secondly, the reconsideration of the structure and the specification modifications of the component parts from the bolts to the rod end were carried out. Finally, bench tests were carried out repeatedly after the modification or structure change of the parts to bring the measurement values close to the calculation values. As a result of these works, it succeeded in that the differences between the measurement values and the calculation values put in less than 4 [%]. **Figure 19** shows the experimental values of toe angle changed by static roll angles and the calculation values. No. 1 of the experimental value is the toe angle of suspension designed at the initial stage, and No. 2 shows the toe angle of suspension improved after the running tests. It is judged that experiment results of No. 2 agree approximately with the theory values. **Figure 20** indicates changes of lateral acceleration obtained from the run test in the skid pad. No. 1 and No. 2 indicated in this Figure are the same as parameters of **Figure 19**. In comparison with the theoretical value calculated by centrifugal force, it succeeded that experimental values are equal to the theory

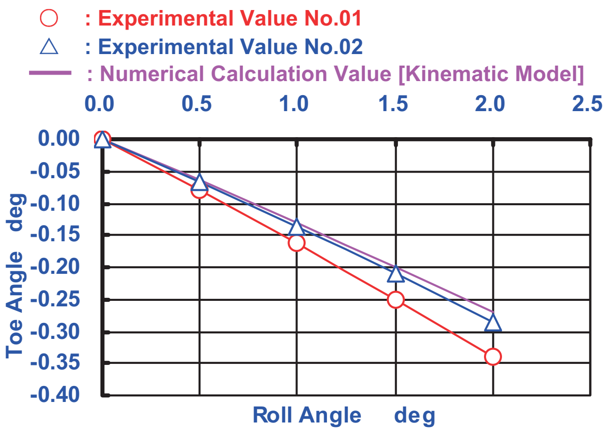


Figure 19 Experimental Values of Toe Angle [deg] Changed by Static Roll Angles [deg] and the Calculation Values

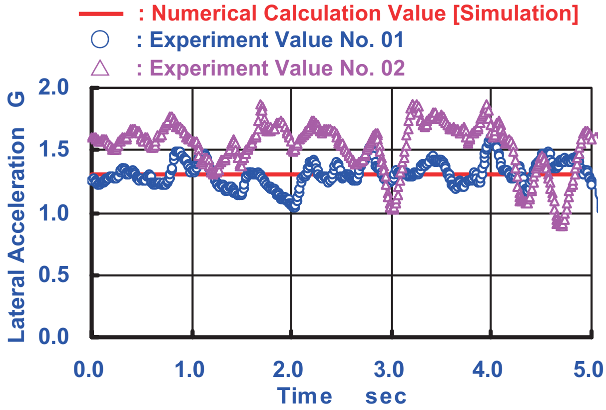


Figure 20 Changes of Lateral Acceleration [G] obtained from the Run Test in the Skid Pad

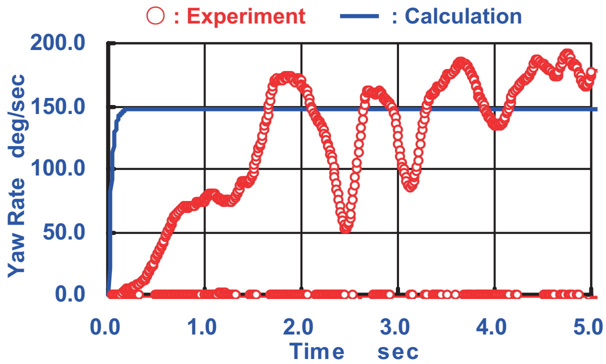


Figure 21 Yaw Rate [deg/sec] occurred at Turning in Skid Pad [Comparison of Experimental and Calculated Values]

value or larger than it. **Figure 21** indicates the yaw rate occurred at turning in skid pad. The yaw rate just after the start of the skid pad is small, but it is found that the driver slows down at yaw rates exceeding the theory value because the tendency of the under steering becomes stronger. As a result, it is judged that the yaw rate tends to be similar to the change of lateral acceleration shown in **Figure 20**.

6. Competitive Examination of Tires

Tires to use in the competitions are examined with logging data in the vehicle running tests. The tires improving the suspension behaviors with pitching and rolling in acceleration and braking generally must be employed by the run test results. In addition, the tire selected according to these tests was more stable than those of our previous adopted tires comparing between the lap times in the durability tests. This fact means that the tires can reduce the differences among the driving techniques of all our drivers. As an example of the measurement results at the run, **Figure 22** shows the vehicle speed changed by the

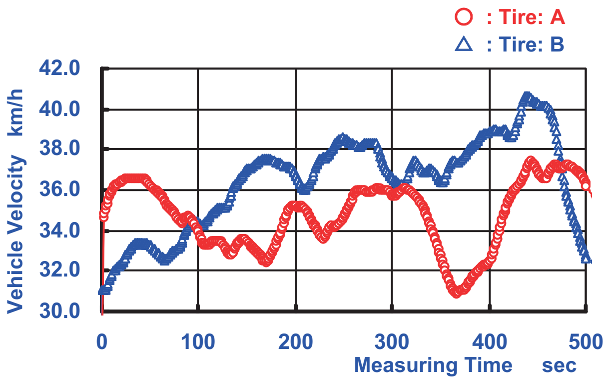


Figure 22 Vehicle Velocity [km/h] Changed by the Difference of the Tire

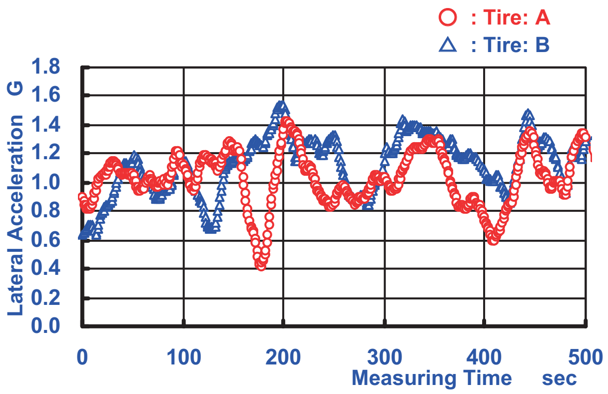


Figure 23 Change of the Acceleration [G] in Side Direction occurred in the Difference of the Tire

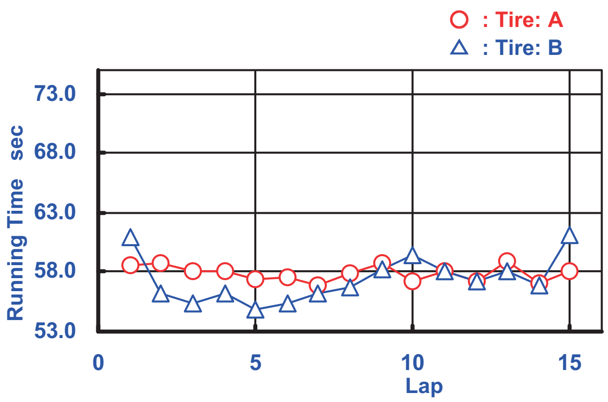


Figure 24 Change of Lap Time [sec] to Lap Numbers by Two Drivers who Drive the Vehicle Attaching Tire A and Tire B [Test Course : A]

difference of the tires. This Figure indicates the change of the lateral accelerations against the elapsed time when each driver run the course shown in **Figure 17** (Refer to **Figure 13**) with two kinds of tires respectively. The run results of a vehicle attached tire B indicate that the tire B has particularly superior performance at cornering. In addition, **Figure 23** shows a change of the acceleration in side

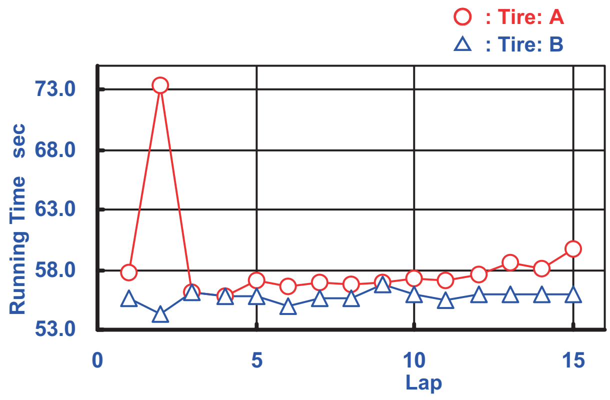


Figure 25 Change of Lap Time [sec] to Lap Numbers by Two Drivers who Drive the Vehicle Attaching Tire A and Tire B [Test Course : B]

direction occurred in the difference of the tire. Furthermore, **Figures 24** and **25** show the results of the durability run test occurred in the difference of the driver. These Figures show that lap times are different by tires used. **Figures 24** and **25** show the change of lap time to lap numbers by two drivers who drive the vehicle attaching tire A and tire B respectively. Tire B indicates turning performance superior to tire A, and the durability more than tire A.

7. Conclusions

Suspension, steering and braking system for the formula competition vehicle are designed and manufactured. Through the design and production, the next knowledge was provided.

[1] The wheelbase expansion and the stabilizer of the competition vehicle were able to control pitching and rolling occurred at the vehicle at the same level.

[2] Suspension geometry, steering geometry and shock layout was designed and manufactured corresponding to the racecourse of the competition. If tires were selected to match the driving characteristics of the vehicle equipped with these systems, the drivability can be sufficiently improved.

We are going to participate in the competition with the vehicle designed and manufactured in this study and to continue this study on the basis of this competition results.

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Appendix

Figure A shows block diagram showing the behavior analytical procedure of the vehicle. According to this procedure, the yaw moment to occur at turning of the vehicle is investigated.

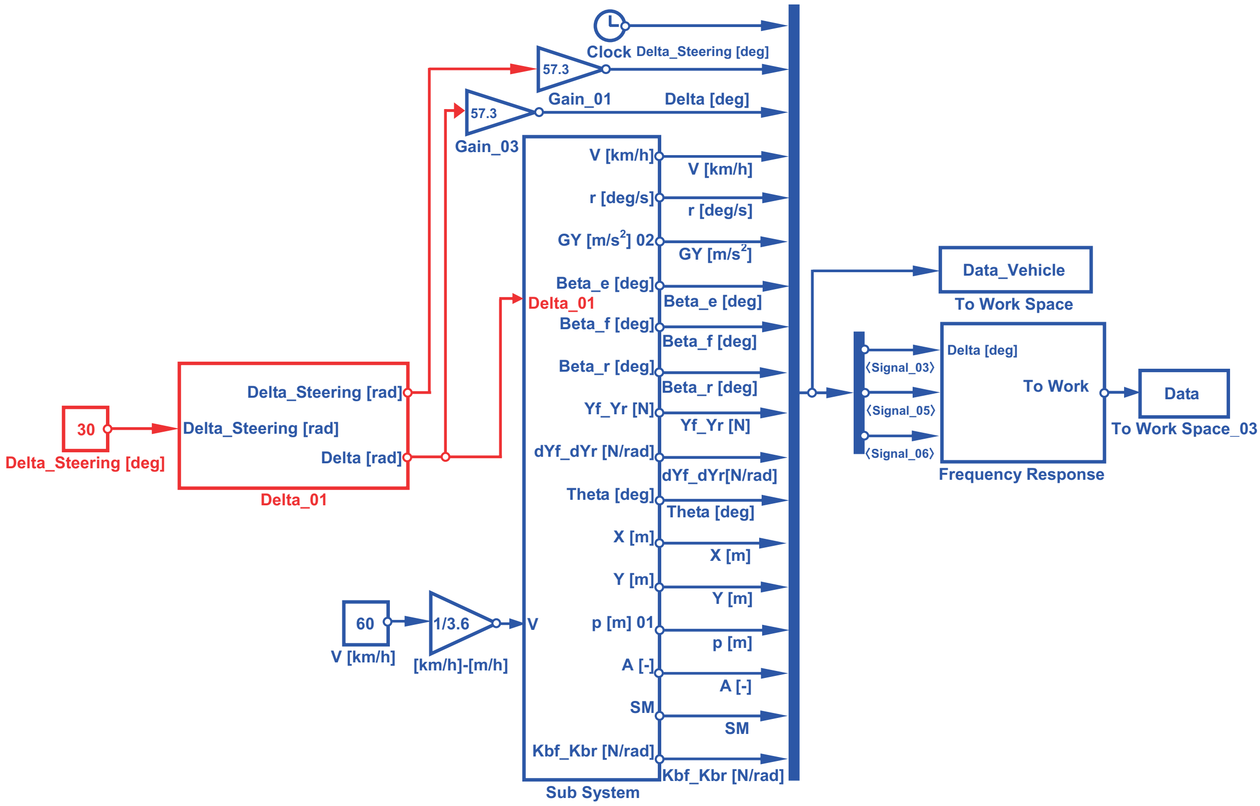

Figure A Block Diagram Showing the Behavior Analytical Procedure of the Vehicle.