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<td>学位授与年月日</td>
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STUDY OF ACTIVE CONTROL FOR VEHICLE DYNAMICS

1994

SHUN'ICHI DOI
ACKNOWLEDGEMENT

The author is indebted to Dr. Tomio Matsubara, Professor of Engineering, Nagoya Institute of Technology, for his generous guidance and encouragement in the preparation of this thesis. Grateful acknowledgement is made to Dr. Teiichi Ohkouchi, Professor of Engineering, Nagoya Institute of Technology, for his kind guidance and helpful advice. The author sincerely acknowledges his indebtedness to Dr. Yasuyuki Funahashi, Professor of Engineering, Nagoya Institute of Technology, for his kind suggestion in elaboration of the thesis. The author wishes to express his sincere thanks to Dr. Yoshitaka Togari, Professor of Engineering, Nagoya Institute of Technology, for his kind suggestions and kind opinions.

The author is deeply grateful to Toyota Central Research and Development Laboratory Ltd. for giving the opportunity to carry out this research project. Gratitude is also expressed to Dr. Osami Kamigaito, President of the Laboratory. Appreciations are also expressed to Dr. Yasutaka Hayashi, Director of the Laboratory, and Mr. Junzo Hasegawa, Adviser of the Laboratory, for the generous guidance and suggestion throughout the research project. The author is indebted to the staff of the Mechanical Divisions of the Laboratory, particularly to the members of Vehicle Dynamics Laboratory, for their generous helps and experimental supports.

The author thanks very much to Dr. David Crolla, Professor of Vehicle Dynamics, The University of Leeds in United Kingdom, for his helpful advice for tidy up of this thesis.

The author greatly appreciates Late Dr. Kenkichi Hayashi, Assistant Professor of Engineering, Nagoya Institute of Technology, for his first guidance to the engineering research fields.
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CHAPTER 1 INTRODUCTION

1.1. Background and Purpose of the Study

With the recent progress of electronic technology, remarkable improvements have been rapidly introduced in the performance control of vehicle maneuvers and ride comfort. Above all, the problem of vibration and ride comfort has been investigated across such broad fields as mechanical engineering, precision engineering, control engineering and sensory engineering in regard not only to controlling the functions and performance of the vehicle but also to producing vehicle that is a controlled machine with high comfort as well as safety and reliability.

Various methods have been tackled to solve the problem of vibration during vehicle running and continuous attention has been made to achieving better man-vehicle control systems in the fields of vehicle design and characteristic modulation\(^1\)\(^2\).

The improvements in vehicle motion and ride comfort until now have mainly depended on the improvements in various structural elements, measurements of high accuracy and good tuning technique, and it has been considered of secondary importance to adopt positive vibration control using additional control equipment. On the contrary, nowadays these former methods can no longer be expected to obtain rapid improvements in performance, and research and development of various vibration control methods have been proposed along with optimal structural designing. There is now great expectation of progress in active control methods as well as in the usual semi-active methods of passive vibration adjustment\(^3\)\(^4\).

Achieving the active control of the vehicle would make require to deal with various systems such as suspension, steering, traction and braking. Adaptation of active control for the vehicle would make possible the following:
a) Improvement of comfort by controlling the attitude and vibration both in the lower frequency and higher frequency.

b) Realization of high quality for vehicle dynamics and critical performance by optimally adjusting tire friction forces against road surface.

c) Increased controllability in the closed loop between driver and vehicle systems.

When taking fundamental performance such as straight running and cornering into consideration, suspension and steering systems are important controlling elements.

In the suspension system of vehicles, vibration adjusting equipment using metal-springs and fluid absorbers is common, but it cannot adapt to instantaneous changes of running conditions. Therefore, it is necessary to develop a small, compact and efficient controlling equipment.

Steering systems include newly developed methods such as four wheel steering as well as the ordinary front-wheel steering. Four-wheel steering methods \(^5\)
\(^6\) are known to give better performance because of large degrees of freedom of control design and greater efficiency in critical maneuvers of vehicle motions. However, the increased weight of equipment for controlling rear suspension and the increased number of parts cause many difficulties for four-wheel steering. It is necessary to develop simpler and more effective control equipment.

From the point of view mentioned above, a feasibility study of an actively controlled vehicle was carried out to improve performances of vehicle safety and comfort significantly. The purposes of this study are to develop a basic technology for realizing an actively controlled vehicle and to predict its effect on performance. The proposed methods have been verified experimentally and shown to be applicable to vehicle design methods.

Using these design methods, the vehicle is constructed with independent hy-
dropneumatic suspension systems that integrate control elements and hydraulic power sources for an individual wheel. As to the steering systems, the vehicle is constructed with a front active steering system that enables to add a few degrees of actual steering angle in practical motion. A design procedure applying modern control theories, and a sophisticated element technique using actuator and multi-processor are also applied to the vehicle.

The characteristic of the suspension system is functional integration in each suspension unit. These systems are expected to improve the response of each control element, reduce expended energy to compress the oil and the generated noise from the oil pipes. The control strategy for suspension uses combined methods of feedforward and feedback controls corresponding to various running conditions.

On the other hand, the active steering control is based on the optimal control theory including a driver-vehicle closed control system. It is possible to improve the vehicle controllability and stability at high speed by a slight front steering angle control.

With the simultaneous control of both suspension and steering made possible by these methods, the ease of vehicle handling maneuvers is derived both predictably and experimentally.

1.2 Survey of the Literature on Active Control for Vehicle

The study on vehicle motions and vibrations was begun from the beginning of this century, and the vehicle dynamics' theories discussing dynamic stability for vehicle lateral motions were systematized during the early 70s8). Research on riding comfort of the vehicle made alongside progress for airplanes and railway vehicles, and in the 60s many studies discussed human sensitivity to vibration.

Research on active control of the vehicle was begun with theoretical consid-
erations during the 1960s and was based on the traditional studies in passive vibration control and the characteristic control technologies. In the middle of the 70s, theoretical studies on active control suspension by A.G. Thompson\textsuperscript{9)} were presented. Since then with the recent progress of electronics and the promotion of vehicle functions for comfort, many studies have continued to be offered in the basic and practical fields\textsuperscript{10)-12).}

The first fabrication of a practical active suspension was the Lotus Active Formula 1 model in the early 1980s\textsuperscript{13}). Japanese automobile industries introduced these active suspension systems to commercial car fields in the late 80s\textsuperscript{14)-15}). These active suspension vehicles are well compromized with the better performances and energy consumptions.

On the other hand, semi-active suspension was introduced more rapidly in these fields\textsuperscript{16}). In the beginning of the 80s, there were many car-models equipped with semi-active suspension control, which were selected damping controls for suspensions.

Apart from suspension control, there were many methods for vehicle dynamics' controls. Basic research on steering and suspension control was developed in the fields of chassis control design. Positive control elements were designed such as four-wheel steering systems, which enabled to provide more freedom of motion control to vehicles. The first generation of these controls to the fabrication of commercial car is separate four-wheel steering devices. This technology became better system with both steering and suspension control by the second generation of fabrication in the beginning of the 90s\textsuperscript{17}).

These studies did not depend only on the evaluation studies of vehicle performance. Many studies\textsuperscript{18)-20}) were also offered to apply to the control designs. Some of these are considering the trade off between human drivers and controlled vehicles. Such man-vehicle system studies are necessary for evaluation and design of the controlled vehicle. The more detailed researches for drivers' maneuvers have been continued as well as for improving active safety performances of the vehicles.
1.3. Scope of the Dissertation

The overall objective of this dissertation is to provide a fundamental understanding of active control applications in automobile suspension and steering systems, with particular attention being given to their effective practice for a vehicle.

The construction of the dissertation is as follows.

In CHAPTER 1, "INTRODUCTION", the background and the significance of the study are presented and the survey of the literature on active control for vehicle is reviewed. The standpoint and contents of the dissertation are also shown.

In CHAPTER 2, "STUDY ON EVALUATION AND ANALYSIS FOR VEHICLE VIBRATION ", the evaluation of vehicle vibration on floor acceleration is measured for ordinary passive vehicles. The aim of this study is concerned with an optimal performance index for the design procedure discussed in the following chapters. As for the vibration input to the vehicle, tire transfer forces and surface irregularities of the road are measured and evaluated in relation to vibration levels during practical running. The tire is rotated and deformed according to vehicle running, and excitation forces to the car body are affected by the anisotropic deformation behavior of the tires. Road irregularity as fundamental inputs to tire deformation must be evaluated quantitatively for the vehicle vibration analysis. Measuring apparatus of tire uniformity according to irregular structural stiffness of a rotating tire is newly developed. The measurement of road irregularities from running vehicle is also developed. Applying these measurements, a quantitative evaluation of vehicle vibration is measured for the vehicle floor vibration acceleration. A performance index for vibration control is deduced from these results.

In CHAPTER 3, "BASIC THEORIES OF ACTIVE CONTROL FOR VEHICLE DYNAMICS ", active control theories are discussed for realizing the optimal performance
mentioned in chapter 2. First, vibration control for a quarter car model is designed considering the characteristics of delays of the controlling elements. The control methods are selected for optimal regulator with feedback control of state variables. Next, considering the trade off between human sensitivity and vehicle motions, vibration control using frequency-shaped cost functions is applied, aimed at improvements in both ride comfort and vehicle attitude characteristics. Moreover, vehicle vibration control for half and whole car models, and vehicle motion control by steering maneuver are also discussed. Then, these ideal control theories are applied to the real control systems in the following chapters.

In CHAPTER 4, "STUDY OF SEMI-ACTIVE VIBRATION CONTROL ", to realize the optimal control mentioned in chapter 3, a system configuration of semi-active vibration control using a newly designed, continuously controlled damper is proposed. In these methods the damping valve is controlled successively according to the simultaneous changes due to state variables that is different from ordinary multi-stage damping control. The analysis and optimization of the damper valve are discussed here, and predictive analysis is also shown to be effective for vibration suppression of the vehicle. Moreover, the effect of the system is verified experimentally using a test apparatus with two degrees of freedom.

In CHAPTER 5, "STUDY OF ACTIVE VIBRATION CONTROL ", aiming at the low frequency vibration control of the vehicle that is realized by adopting the above-mentioned methods, the control for vehicle attitude is designed using compensating methods for the delay of the active control elements. The system configurations for active attitude control and control system with a combination of feedforward and feedback loops are shown. After showing the predictive analysis for the control effects, experimental verification of this system is offered with practical running tests using the experimental vehicle.
In CHAPTER 6, "STUDY OF ACTIVE MOTION CONTROL", the effect of active steering control for vehicle handling is discussed in relation to the vehicle motion. The control for stability and controllability is often evaluated by the closed loop between driver and vehicle system. The control system is therefore designed including the dynamic characteristics of the driver as mentioned in chapter 3. Control methods are shown and the predictive analysis is also used to prove the effectiveness of stability and steerability of the vehicle during handling maneuvers. Moreover, the effect of the control is verified by the real running tests. The performance of controlled experimental vehicle is preferred to that of non-controlled vehicle and the system is easy to construct with simple attached active control element to ordinary steering systems.

In CHAPTER 7, "EXPERIMENTAL STUDY OF ACTIVE CONTROL", the adaptive control system is proposed with hierarchical active control systems using the three kinds of controls discussed in chapters 4 to 6. The system combines active control of steering, suspension and semi-active control of suspension damper considering the energy dissipation. It is constructed by giving priority to each control according to vehicle running status. The system superiority is confirmed by practical running experiments using the experimental vehicle and the comfort and ease of steering are also derived from these results.

In CHAPTER 8, the results obtained in this study are summarized and conclusions are drawn to highlight the major findings of this study.

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<th>Symbol</th>
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<tr>
<td>( A, B, C )</td>
<td>System matrix</td>
</tr>
<tr>
<td>( A_{C} )</td>
<td>Sectional area of cylinder</td>
</tr>
<tr>
<td>( A_{S} )</td>
<td>Cross-sectional area of orifice</td>
</tr>
<tr>
<td>( A_{f}, A_{r} )</td>
<td>Longitudinal distances from sprung mass to front and rear axles</td>
</tr>
<tr>
<td>( B_{S} )</td>
<td>Width of orifice</td>
</tr>
<tr>
<td>( C_{f}, C_{r} )</td>
<td>Cornering power of front and rear tires</td>
</tr>
<tr>
<td>( C_{sf}, sf )</td>
<td>Cornering stiffness of front and rear tires</td>
</tr>
<tr>
<td>( c_{0}, c_{C}, c_{S} )</td>
<td>Coefficient of orifice, flow and equivalent flow</td>
</tr>
<tr>
<td>( c_{1}, c_{2} )</td>
<td>Damping coefficient of unsprung and sprung masses</td>
</tr>
<tr>
<td>( F_{C}, F_{g}, f_{C} )</td>
<td>Generated active and damping force</td>
</tr>
<tr>
<td>( F_{sf}, f_{sr} )</td>
<td>Side force of front and rear axles combining left and right tires</td>
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<td>( f )</td>
<td>Ideal control force for suspension</td>
</tr>
<tr>
<td>( f_{0} )</td>
<td>Resonance frequency of spool</td>
</tr>
<tr>
<td>( f_{r}, f_{r} )</td>
<td>Front and rear damping force</td>
</tr>
<tr>
<td>( f_{K} )</td>
<td>Control gain for active steering</td>
</tr>
<tr>
<td>( f_{S} )</td>
<td>Flow force of spool</td>
</tr>
<tr>
<td>( g )</td>
<td>Gravitational acceleration</td>
</tr>
<tr>
<td>( g_{i} )</td>
<td>Weighting coefficient for active steering</td>
</tr>
<tr>
<td>( H )</td>
<td>Rolling radius of tire</td>
</tr>
<tr>
<td>( h_{i} )</td>
<td>Weight coefficients of performance</td>
</tr>
<tr>
<td>( I_{x} )</td>
<td>Roll moment of inertia of sprung mass</td>
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<td>( I_{y} )</td>
<td>Pitch moment of inertia of sprung mass</td>
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<td>( I_{z} )</td>
<td>Yaw moment of inertia of sprung mass</td>
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<td>( J )</td>
<td>Performance index</td>
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<tr>
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<tr>
<td>( K_{fi}, K_{ri} )</td>
<td>Front and rear semi-active suspension feedback coefficients (i=1-5)</td>
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<td>( K_{g} )</td>
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<tr>
<td>( K_{h}, K_{ii} )</td>
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<tr>
<td>( K_{uf}, K_{ur} )</td>
<td>Matrix of feedback coefficients</td>
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<tr>
<td>( K )</td>
<td>Front and rear tire spring stiffness</td>
</tr>
<tr>
<td>( l_{1}, l_{2} )</td>
<td>Coefficient of roll moment</td>
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<tr>
<td>( k )</td>
<td>Spring coefficient of unsprung and sprung masses</td>
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<tr>
<td>( k_{C} )</td>
<td>Coefficient of valve delay</td>
</tr>
<tr>
<td>( k_{r} )</td>
<td>Yaw rate gain against steering in active steering</td>
</tr>
<tr>
<td>( k_{s} )</td>
<td>Control gain for active roll</td>
</tr>
<tr>
<td>( L_{i} )</td>
<td>Spring coefficient of spool</td>
</tr>
<tr>
<td>( L_{x} )</td>
<td>Equivalent length of orifice chamber</td>
</tr>
<tr>
<td>( M, m )</td>
<td>Distance of the front view point of driver</td>
</tr>
<tr>
<td>( M_{S}, M_{sr} )</td>
<td>Vehicle mass</td>
</tr>
<tr>
<td>( M_{sf}, M_{sr} )</td>
<td>Front and rear sprung mass in quarter car model</td>
</tr>
<tr>
<td>( m_{f}, m_{r} )</td>
<td>Mass of spool</td>
</tr>
<tr>
<td>( M_{Ur} )</td>
<td>Front and rear unsprung mass</td>
</tr>
<tr>
<td>( N_{s} )</td>
<td>Total gear ratio of steering</td>
</tr>
<tr>
<td>( N_{p} )</td>
<td>Revolutions of oil pump or motor</td>
</tr>
<tr>
<td>( P_{0} )</td>
<td>Pressure fluctuation in the cylinder</td>
</tr>
<tr>
<td>( P_{ci} )</td>
<td>Initial pressure of gas spring</td>
</tr>
<tr>
<td>( P_{p} )</td>
<td>Cylinder pressure (i=1-4)</td>
</tr>
<tr>
<td>( P_{g} )</td>
<td>Pressure fluctuation of gas spring</td>
</tr>
<tr>
<td>( P_{S} )</td>
<td>Pressure in the valve orifice chamber</td>
</tr>
<tr>
<td>( p )</td>
<td>Roll rate</td>
</tr>
<tr>
<td>( Q_{i} )</td>
<td>Flow volume</td>
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The subscripts f and r denote front wheel and rear wheel.
CHAPTER 2 STUDY ON EVALUATION AND ANALYSIS FOR VEHICLE VIBRATION

2.1. Introduction

The subjects of research for vehicle comfort are many items concerning the ease of driving, but hereunder "vibration and riding comfort" is chosen as the most important problems during practical running of the vehicle. The factors influencing vehicle comfort are classified as shown in Figure 2-1. This figure shows both a narrow classification and a broad one with and without vibrating phenomena.

![Ride Comfort Diagram](image)

**Fig. 2-1 Factors influencing vehicle comfort**

The changes of vehicle attitude, motion and vibration are classified as riding comfort characteristics, and are thought to be those which most affect the vehicle comfort. The vibration composed of several frequency bands due to vehicle running has a closed relationship with the sensitivity of motion and vibration of human occupants. These evaluations, therefore, should be discussed as part of the so-called "Road-Vehicle-Human System" concerning vehicle vibration and motion dynamics as well as road disturbance characteristics.

On the other hand, besides mechanical vibrations there are many other feel-
ings like visibility, hearing, touching and so on which influence human response. For example, these are visibility of instrumental panels and frontal scene, audibility for interior or exterior noise, ease of pedal motions of accelerating or braking and support of occupant seats relating to fatigue with a long term driving.

In this chapter, the evaluation of riding comfort of human occupant during vehicle running is discussed for one of the most important subjects of vehicle comfort. These results provide basic background for deciding the desired level of control in the design of vehicle vibration.

It is necessary to consider various phenomena from the motion and vibration of human body to the sound and vibration of vehicle compartment including high frequency components. These situations of sounds and vibrations are changed due to the environment like running speed, surface geometry of road, suspension characteristics and tire selection. In these evaluations, therefore, it is necessary to consider the amounts of contribution of individual components during each transfer process.

In section 2.2., the evaluation of riding comfort is executed by measurements of vehicle vibration. Considering various states of running vehicle, the target levels for improving riding comfort are offered according to the survey results of experiments using ordinary commercial cars.

In section 2.3., the influence of tire generated force on vehicle vibration is discussed from the point of view that it is one of most important critical paths to vehicle body from vibration inputs. It is also shown how to develop the tester for measurement of tire uniformity, with which the unevenness of tire structural stiffness is evaluated during running conditions.

Finally in section 2.4., evaluation of road surface irregularities as input to vehicle is measured and vehicle vibrations on various roads are observed on ordinary commercial vehicles. It is also shown how to construct a newly developed road profile measuring apparatus on a real running vehicle.
2.2. Evaluation of Vehicle Vibration

2.2.1. Measurement of Vibration and Riding-comfort

In the evaluations of riding comfort during running in the vehicle, there are two cases in the standpoints: one is passenger's seat vibration from the point of view of subjective judgement of comfort, and the other is vehicle floor vibration from that of suspension control design. In the former case, it is necessary to take into account the dynamic sensitivity of human occupants including the differences of seated posture and individual body shape. In the latter case, it is also necessary to consider the individual vibrations of both the vehicle and the components. Hereunder, targeting the vehicle suspension which is part of the transfer process to human occupants as comfort evaluator, the measurements and evaluations of vehicle vibration are discussed from the latter viewpoint.

(1) Vibration phenomena as evaluation subjects for ride comfort

The evaluating terms for ride comfort are often expressed in Japanese as onomatopoetic words. These words are classified by the individual phenomena divided with the difference of emerged frequency regions. It also includes "flat-ride" as describing the effect of road unevenness and "harshness" as effect of road joints and pot-holes.

Table 2-1 Classification of ride evaluations

<table>
<thead>
<tr>
<th>frequency</th>
<th>0.4</th>
<th>1</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>8</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>60</th>
<th>100</th>
<th>150(Hz)</th>
</tr>
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<tbody>
<tr>
<td>feeling</td>
<td>flat ride</td>
<td>rugged wheel hop</td>
<td>harshness</td>
<td>road noise</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>origin</td>
<td>sprung mass resonance</td>
<td>unsprung mass resonance</td>
<td>body structural resonance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>roll</td>
<td>suspension friction</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>pitch</td>
<td>front/rear wheel phase</td>
<td>seat/occupant resonance</td>
<td>E/G shake</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>tire/uniformity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>tire eccentricity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Table 2-1 shows the classification of the words of evaluations and the phenomena of perceived vibration. This table describes these phenomena by frequency region. The classifications are divided into several frequency bands from human sensitivity: "flat ride" feelings for 0.25-3Hz, upward shock feeling for 2-4 Hz, the most sensitive feeling by human stomach resonance for 4-8Hz, "wheel hop" feeling of unsprung mass resonance for 8-15 Hz, small busy vibration to ankle and calf for 10-20Hz, harshness for 20-60 Hz and so-called road noise from the coupling of tire vibration and interior noise for the beyond range of frequency. These classifications are not precise for every occupant so that the feeling evaluation is quite individual.

(2) Classification of vehicle vibration

The causes for generating these phenomena are divided into the following:
First the vibration of vehicle is composed with various resonant frequencies of the structures and components. From the composition of sprung and unsprung masses there are primary and secondary resonant frequency regions corresponding to slow motion feelings and busy vibration feelings. These sprung mass vibrations are concerned with the spring and damping characteristics of the suspensions. The friction generation in the movement of the suspension is also influenced by the vibration between primary resonance and the secondary one. Moreover various vibrations are induced from the corresponding phenomena between vehicle velocity and the length of the wheel base. The forced vibration from tire also emerges including the tire vibration itself and other interactions with the road surface.

These factors are interact with each other and the final vibration to human occupant consists of all of them with various transfer processes. Transfer elements of vibration are body and floor structural members as distribution properties for vibration system and have much influence to high frequency vibration and acoustic phenomena. Moreover related to the internal vibrating parts like power train, engine and transmission lines, they are also concerned with the vibration sources as well as transfer components.
(3) Human sensitivity and human dynamics on the vehicle

As for the criteria of vibration riding comfort, the regulation was determined by ISO standard. According to the ISO-2631\textsuperscript{1)}, the acceleration level for vertical and horizontal direction is decided as shown in Figure 2-2(a) and (b). These are given in root-mean-squared value by 1/3 octave band frequency, and describe the criteria for one direction. They show the human sensitivity is high between 4-8Hz for the vertical and 1-2Hz for the horizontal direction. They are depending on the human body sensitivities concerning to the local resonance of stomach and other parts of the body.

There are many researches for the human sensitiveness toward vibration, for examples, like Janeway\textsuperscript{2)}, Goldmann\textsuperscript{3)}, Lee, Pradko\textsuperscript{4)}, Dieckmann\textsuperscript{5)} and Miwa\textsuperscript{6)}.

These are all evaluation reports and the results are almost same as shown in Figure 2-3.
The basic studies for human sensitiveness of vibration were reported in detail by Dupuis and others\textsuperscript{7\textendash }10 concerning the occupants on the seats. There is an example shown in Figure 2-4. The stimuli for seated occupants are larger in the frequency region of 3-4 Hz with vertical vibration input and the local stimuli are composed of combining movement between the vertical and fore-and-aft displacement of human body as shown in Figure 2-5.

![Fig. 2-4 Vibration transfer](image1)

![Fig. 2-5 Vibration stimuli](image2)

On the other hand, there are many reports\textsuperscript{11\textendash }16 concerning the multi-axial vibration to human occupants' sensitivity of vibration. One of the evaluation criteria offered by Griffin\textsuperscript{11} is shown in Figure 2-6. This is one of the results of researches for examinations for ride comfort during vertical and fore-and-aft vibrations. Thus, the human on the vehicle evaluates the vibration and ride comfort from the mixed phenomena of the resultant vibrations.

The evaluation of ride comfort is expressed by the representative properties for physical values concerning the above-mentioned human sensitive range of the frequency or some weighted value of the imposed energy to the human body.

![Fig. 2-6 Human occupants sensitivity](image3)
(4) Sampling methods for the vibration phenomena concerning riding comfort

There are many physical values for assessment of comfort and sampling methods for the vibration phenomena related to riding comfort. These are also representing the characteristics for individual feeling of the phenomena. Here is the explanation about adopting measurements and the representing values for the evaluation.

The measured value of the floor acceleration is used for the evaluation and the values for representing each vibration phenomenon are divided into three kinds of terms: these are all integral value for band passed power spectrum density of the acceleration measured on the central floor of the vehicle. The frequency bands are 0.4-2 Hz, 4-7 Hz and 8-20 Hz for representing sprung resonance, the most sensitive region for human body and unsprung mass resonance, respectively. For higher frequency ranges, the frequency bands are chosen to 8-15 or 20 Hz and 20-60 Hz. The latter case is adapted to the evaluation for the transient vibration when passing by the road joints on highway.

2.2.2. Target Level for Improving Riding-comfort

To obtain the basic data for assessing the target properties for riding comfort of vehicle, evaluation tests are executed by real running of the commercial cars. The above mentioned classifications and sampling methods are adopted in this study and the vibrating accelerations are measured on the central floor of the experimental vehicle. The method for these tests and the results for the evaluations are as follows.

(I) Tests for measuring vehicle vibrations

The practical running tests were carried out for 22 commercial vehicles. Table 2-2 shows the list of the cars with traction type, the engine volume, the wheelbase, the weight and suspension type for front and rear, respectively.

According to the difference of traction type there are seven front-engine and front-driven types and others are all front-engine and rear-driven types.
According to suspension type there are two types, namely strut or wishbone type for front one and independent or dependent type for rear one. The independent suspensions are divided into 3 types as strut, semi-trailing arm and trailing arm type. The dependent suspensions are also divided into 4 link and leaf type. As shown in this table, front-mounted engine and front-driven cars are equipped with independent suspensions for the rear wheels.

<table>
<thead>
<tr>
<th>Car Class and Name</th>
<th>Traction</th>
<th>Engine (l)</th>
<th>Wheel base (m)</th>
<th>Weight (kg)</th>
<th>Suspension type Front</th>
<th>Rear</th>
<th>Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bentz 300 Medium Car</td>
<td>FR</td>
<td>3.025</td>
<td>2.79</td>
<td>1505</td>
<td>Wishbone</td>
<td>Strut</td>
<td>Semi-trail</td>
</tr>
<tr>
<td>Bentz 450 Medium Car</td>
<td>FR</td>
<td>4.520</td>
<td>2.86</td>
<td>1730</td>
<td>Wishbone</td>
<td>Strut</td>
<td>Semi-trail</td>
</tr>
<tr>
<td>Civic Medium Car</td>
<td>FF</td>
<td>1.488</td>
<td>2.32</td>
<td>780</td>
<td>Strut</td>
<td>Strut</td>
<td>Strut</td>
</tr>
<tr>
<td>Leohne Medium Car</td>
<td>FF</td>
<td>1.595</td>
<td>2.45</td>
<td>870</td>
<td>Strut</td>
<td>Strut</td>
<td>Semi-trail</td>
</tr>
<tr>
<td>Tercel Medium Car</td>
<td>FF</td>
<td>1.452</td>
<td>2.50</td>
<td>800</td>
<td>Strut</td>
<td>Strut</td>
<td>Trailing</td>
</tr>
<tr>
<td>Mirague Medium Car</td>
<td>FF</td>
<td>1.244</td>
<td>2.30</td>
<td>775</td>
<td>Strut</td>
<td>Strut</td>
<td>Trailing</td>
</tr>
<tr>
<td>Palser Medium Car</td>
<td>FF</td>
<td>1.397</td>
<td>2.39</td>
<td>800</td>
<td>Strut</td>
<td>Strut</td>
<td>Trailing</td>
</tr>
<tr>
<td>Fiesta Medium Car</td>
<td>FF</td>
<td>1.595</td>
<td>2.28</td>
<td>850</td>
<td>Strut</td>
<td>Strut</td>
<td>Trailing</td>
</tr>
<tr>
<td>Audiel Medium Car</td>
<td>FF</td>
<td>2.144</td>
<td>2.67</td>
<td>1170</td>
<td>Strut</td>
<td>Strut</td>
<td>Trailing</td>
</tr>
<tr>
<td>BMW 320 Small Compact Car</td>
<td>FR</td>
<td>1.990</td>
<td>2.56</td>
<td>1115</td>
<td>Strut</td>
<td>Strut</td>
<td>Semi-trail</td>
</tr>
<tr>
<td>Taupus Small Compact Car</td>
<td>FR</td>
<td>1.593</td>
<td>2.58</td>
<td>1020</td>
<td>Strut</td>
<td>Strut</td>
<td>Semi-trail</td>
</tr>
<tr>
<td>Runcer Small Compact Car</td>
<td>FR</td>
<td>1.411</td>
<td>2.44</td>
<td>895</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Callora Small Compact Car</td>
<td>FR</td>
<td>1.407</td>
<td>2.40</td>
<td>840</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Sunny Small Compact Car</td>
<td>FR</td>
<td>1.711</td>
<td>2.34</td>
<td>840</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Starlet Small Compact Car</td>
<td>FR</td>
<td>1.290</td>
<td>2.30</td>
<td>715</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Corona Small Compact Car</td>
<td>FR</td>
<td>1.588</td>
<td>2.53</td>
<td>970</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Sunny Small Compact Car</td>
<td>FR</td>
<td>1.237</td>
<td>2.34</td>
<td>815</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
<tr>
<td>Violet Small Compact Car</td>
<td>FR</td>
<td>1.595</td>
<td>2.40</td>
<td>930</td>
<td>Strut</td>
<td>Strut</td>
<td>4 link</td>
</tr>
</tbody>
</table>

Fig. 2-7 Weight and wheelbase for the experimented cars
Figure 2-7 shows a comparison of the experimented cars for the total weight and the wheelbase graphically. Each notation coincides with the classification of these cars as shown in the figure. The small ordinary cars and the medium size luxurious cars are nearly distributed in one line toward the right-upper side of the figure. In the small cars the suspension types are the same for either FF or FR as strut for front suspension and wishbone for rear suspension.

Subjective road classification is shown in Figure 2-8 and they are all common and ordinary roads. The road surfaces and running conditions are also shown in Table 2-3. Objective vibration phenomena are distinguished on each test road as shown in the table.

The measurement sensor in the tests was an accelerometer for vertical and fore-and-aft located in the central flat part of each experimental car.

![Fig. 2-8 Classification of tested roads](image)

<table>
<thead>
<tr>
<th>Test road</th>
<th>Surface geometry</th>
<th>Speed</th>
<th>Objectives</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Paved road (inside city)</td>
<td>60km/h</td>
<td>rugged feeling</td>
</tr>
<tr>
<td>B</td>
<td>Mended paved rough road</td>
<td>50km/h</td>
<td>and</td>
</tr>
<tr>
<td>C</td>
<td>Berugian road</td>
<td>60km/h</td>
<td>flat ride</td>
</tr>
<tr>
<td>D</td>
<td>Mended paved road (suburbs)</td>
<td>60km/h</td>
<td></td>
</tr>
<tr>
<td>E</td>
<td>Paved road (suburbs)</td>
<td>60km/h</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>Freeway for automobile</td>
<td>50km/h</td>
<td>flat ride</td>
</tr>
<tr>
<td>G</td>
<td>Paved smooth road</td>
<td>60km/h</td>
<td>rugged and flat</td>
</tr>
<tr>
<td>H</td>
<td>Highway</td>
<td>60km/h</td>
<td></td>
</tr>
<tr>
<td>I</td>
<td>Paved smooth: Obstacle N</td>
<td>60km/h</td>
<td>harshness</td>
</tr>
<tr>
<td>J</td>
<td>Paved smooth: Obstacle W</td>
<td>60km/h</td>
<td>feeling</td>
</tr>
</tbody>
</table>
(2) Results of the tests

Figure 2-9 shows vertical vibration acceleration of the vehicle floor in the form of power spectrum density to vibration frequencies. These are the averaged results for three representative roads from smooth to rough surface. From the frequency distribution of measured vibration, there are marked resonant frequencies for the sprung mass in the 1-2Hz region and for the unsprung mass around 10 Hz. The first peak coincides with a soft and flat subjective feeling and the bottom part between first and second peak concerns the sensitive characteristics for human body. According to the unevenness of the road surface, the acceleration levels of two kinds of the vibration resonance are increased significantly.

Next, as a quantitative evaluation for each vibration resonance representative, every integrated value for the power spectrum density of floor vertical acceleration is calculated. Figure 2-10 shows a comparison of the two major components for evaluation on the various test roads. In this figure, the integral value in the frequency range 0.4-5Hz for soft riding feeling and the one
with frequency range 5-20Hz for busy riding feeling are compared. The range of generated acceleration in steady state running on smooth and rough road is shown in the area domain indicated by two dotted lines. Hereunder, representing road conditions for the survey, the paved ordinary road E in the suburbs and the repaired paved road B are chosen for subsequent experiments.

Various kinds of vibrations for classified riding comfort are examined. The input level for vehicle vibration is affected by the combination of road surface and the vehicle velocity in the stationary cruising conditions. On the contrary, as for the high frequency vibrations that are not revealed at the steady running conditions, we can survey this phenomenon by passing over the obstacle as the response for a transient impulsive input. The response concerning the riding comfort is examined by both the stationary running test and obstacle passing test here.

First the results of the former tests are described.

**Vibration in the range of 0.4-2 Hz**: Figure 2-11 shows that vibration of small commercial cars (FF, FR) is bigger than that of medium luxurious cars. There is a tendency that vibration levels of FF car with front strut type and rear independent type suspension are higher than those of FR car with front wishbone type and rear independent type.

![Fig. 2-11 0.4-2 Hz range](image1)

![Fig. 2-12 4-7 Hz range](image2)
Vibration in the range of 4-7 Hz: Figure 2-12 shows that vibration level of this range is most notable for small compact cars. The frequency components in this area are much dependent on the frictional characteristics of the composed suspensions.

Vibration in the range of 8-15 Hz: Figure 2-13 shows the vibration level of the small compact cars is larger than that of the medium cars in this vibration range. Among small compact cars, the level of FF car is rather small and the level of the medium has also rather small levels of the vibration. In this area of the vibration the vibrating component depends on the unsprung mass resonance and other factors depend on the engine shake and the tire uniformity.

Figure 2-14 shows the comparison with fore-and-aft vibrations for these vehicles. The plotting points are scattered as shown in the figure but there is some tendency that small FF and medium independent exhibit small vibration levels.

![Figure 2-13 8-15 Hz range (vertical)](image)

![Figure 2-14 8-15 Hz range (fore-&-aft)](image)
Hereunder the results are conducted from the transient response survey of passing over an obstacle on the road.

**Vibration in the range of 5-20 Hz:** Figure 2-15 shows the vertical acceleration components' distribution under the condition of narrow width and length of obstacle and broad width and length of obstacle at the vehicle speed 40km/h. For both conditions of obstacle, there is a linear tendency and the medium cars exhibit lower response values than the small compact cars. It is the same tendency for fore-and-aft horizontal vibration shown in Figure 2-16.

![Graphs showing vibration response](image)

**Fig. 2-15 5-20 Hz range (vertical)**

**Fig. 2-16 5-20 Hz range (horizontal)**

**Fig. 2-17 20-60 Hz range (vertical)**

**Fig. 2-18 20-60 Hz range (horizontal)**
Vibration in the range of 20-60 Hz: Figure 2-17 and 2-18 show the vertical component and the horizontal fore-and-aft component, respectively. Both of the tendencies are similar but the medium cars' level is rather smaller. In this area of the vibration there would be an influence of body size on vibrations and the sensitivity for the vehicle structure.

(3) The vibration phenomena for individual drive train type and suspension type

The above-mentioned results of comparisons between the steady and transient running data induced another classification for both "drive train type" and "suspension type".

Figure 2-19 shows the different lines of plotting with FF and FR cars in the vibration components of 8-15 Hz of the vertical acceleration in both steady and transient running conditions. The vibration level for FF cars is lower than that for FR cars by 50%.

Next, considering the type of the suspension, the characteristic values are plotted into one figure shown in Figure 2-20. It is shown by the axis of vertical acceleration and fore-and-after accelerations. From the figure, the results for the medium cars are small compared with those for the small com-
compact cars. Moreover, in the small compact range, the responses for FF cars are smaller than those for FR cars. For the medium cars, the results for link suspension type are smaller than those for independent suspension. They are shown in the circle as shown in this figure.

As the final classification example for this approach, according to the floor vertical acceleration during ordinary road running conditions, the two kinds of the most sensitive components for human occupants are compared as shown in Figure 2-21. These components are selected for the vibration level of sprung mass resonance and the most sensitive human body resonance as evaluation factors. As shown in the figure, the medium cars exhibit the lowest values.

As the conclusion of this section, the target characteristic for ride comfort is induced from these results that it is desirable that the vibration quantities for both frequency ranges should be lower for realizing better ride comfort as shown in the left bottom of Fig.2-21. These vibration components should be changeable according to the various running conditions. These results are useful to determine the target values of the suspension characteristics tuning, the active force control and semi-active control of the spring and damping factors.

Fig.2-21 Target floor vibration on ordinary paved road
2.3. Influence of Tire Generated Force on Vehicle Vibration

2.3.1. Measurement of Tire Uniformity

With an expansion of the highway network and increased performance of passenger cars, there have arisen frequent occasions of driving a high-class passenger car on a flat smooth road for long hours at constant speed. Problems concerning the stability and controllability and vibration riding comfort have attracted attention due to the reduction of vehicle weight to save the fuel consumption. Many of these problems have been believed to come not only from unbalanced tire weight but from the non-uniformity of tire dimensions and its rigidity.

There are three directional variations in the reaction forces caused by the non-uniformity of rotating tire. Those which give large influences on the vibration riding comfort are the magnitude of force variation in radial direction of tire, i.e. the radial force variation (RFV) and the magnitude of force variation in lateral direction of tire, i.e. the lateral force variation (LFV). That which gives large influence on the stability and controllability is the constant component of force in lateral direction, i.e. the lateral force deviation (LFD). Various studies have been made on these problems by A. Lloyd Nedly et al\textsuperscript{17}-23. Above all, the most influential factor is the RFV value which concerns vibration frequencies from several to ten-several Hz.

Conventionally, the measurement of tire uniformity has been made by automobile makers using large tire testers conforming to the measuring method specified in SAE J332\textsuperscript{24}. These testing equipments\textsuperscript{25} of multi-function, though large-scale and expensive, are very useful to tire research. However, the installation space and price are practical disadvantages for use here. For this reason, a compact light-weight and low-cost tester has been developed, which enables the measurement of tire uniformity of passenger car under an assembled state of tire and wheel\textsuperscript{26}. Hereunder, the influence of tire uniformity on the riding comfort is examined noticing the value of tire RFV.
The construction and appearance of the newly designed tester are shown in Figure 2-22(a) and (b). Table 2-4 shows also the specification of the developed tester. As shown in these figures, the developed tester is constructed of a tire fitting, a drum up and down mechanism, a load detective mechanism and a measuring system. As to the tire fitting, to adapt to a variation of central hole diameters depending on the tire size, centering sleeves of various sizes are provided. Achieving a compact and low-cost design, the drum up and down mechanism applying a setting load on the tire is designed into a manually operated construction.

The compact load detective mechanism permitting no interference between the measured RFV and LFV values was devised. The force variation in radial direction is detected by load cell located at an end of the measuring beam.

### Measuring system

<table>
<thead>
<tr>
<th>Measurable items</th>
<th>Measurable units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter of tire</td>
<td>545 ~ 680 mm</td>
</tr>
<tr>
<td>Width of tire</td>
<td>135 ~ 215 mm</td>
</tr>
<tr>
<td>Diameter of central hole</td>
<td>50 ~ 70 mm</td>
</tr>
<tr>
<td>Offset</td>
<td>20 ~ 50 mm</td>
</tr>
<tr>
<td>Applicable load</td>
<td>0 ~ 4900 N</td>
</tr>
<tr>
<td>Drum size</td>
<td>D 300 X W 250 mm</td>
</tr>
<tr>
<td>Rotating speed of drum</td>
<td>11 rpm</td>
</tr>
<tr>
<td>Total accuracy</td>
<td>RFV, LFV 9.8 N</td>
</tr>
<tr>
<td>Measuring items</td>
<td></td>
</tr>
<tr>
<td>Radial force variation</td>
<td>0 ~ 245 N</td>
</tr>
<tr>
<td>Lateral force variation</td>
<td>0 ~ 245 N</td>
</tr>
<tr>
<td>Lateral force deviation</td>
<td>± 980 N</td>
</tr>
<tr>
<td>Major dimensions</td>
<td>W 830X L 620 X H 1500 mm</td>
</tr>
<tr>
<td>Mass</td>
<td>About 400 kg</td>
</tr>
</tbody>
</table>
The measuring system is constructed of the sensors, measuring circuits and a recorder as shown in Figure 2-23(a). Signals detected by the RF and LF load cells are amplified by each input amplifier. The DC components of these signals are removed by the variable voltage adjuster, and RFV and LFV components are amplified by each amplifier. Respective output signals are monitored in meters and displayed in recorder. The value indicated on the radial force meter denotes the setting load on the tire, and an operator adjusts the load on the tire while watching this meter.

Figure 2-23(b) shows examples of waveform of each output for one rotation of a tire. Peak-to-peak values of the RFV waveform and the LFV wave-form denote the RFV value and the LFV value, and an area mean value of the LF waveform denotes the LFD value, respectively. Incidentally, the measuring circuit in the system is connected with the recorder by using a cable in order to allow other instruments to carry out data processing according to the purpose, for example, the revolution order ratio analysis and so on.
2.3.2. The Effect of Tire Derived Force on Riding-comfort

The measurement and evaluation tests of vehicle vibration are executed either on practical road running tests and on drum running tests which imitate the real running on the road in the laboratory. The road tests were carried out on the proving ground and two-axle drum tester\textsuperscript{27}) installed in the laboratory were used in these experiments. The drum tester is equipped with front and rear drum, which is 1.59 m in diameter and 3.2 m in width.

The tested vehicle is a small passenger car and the tested tire is a radial tire. The RFV value of the tire was measured before the running test by the developed tester. The measurements of the vibration were on the cabin floor of the car for vertical and horizontal accelerations and these data were processed by using Fast Fourier Transform Processing Analyzer.

The examples of the experiment are as follows.

Figure 2-24 (a), (b) and (c) show the examples of vibration power spectrums for floor vertical and fore-and-aft acceleration for paved road, highway and flat surface of proving ground, respectively. These results are all in the cases for fitting tires with acceptable level of the tire uniformity.

In the figure (a), there are the peak for 5-8 Hz component as well as the both resonance frequencies of sprung and unsprung masses. In the both figures (b) and (c), there are remarkable peak around 14-15 Hz component and other higher harmonic components.

Fig. 2-24 Examples of floor PSD
Figure 2-25 (a) and (b) show the results of the drum running tests. Vertical acceleration of the floor obtained from two cases: (a) Non-uniformity tire with RFV values of 100-150 N is mounted on the left front wheel and (b) Uniform tires with RFV value of 50-80 N are mounted to all wheels.

The vibration caused by RFV was generated at the floor as a vibration synchronized with the rotation of the tires. It is obvious that the vibration level is increased when one tire with large RFV value is mounted to the wheel.

(a) Non-uniform tire mounted  
(b) Uniform tire mounted  
Fig. 2-25 Floor vertical vibration in drum test

Figure 2-26 (a) and (b) show also the results of the tire revolution order ratio analysis for the above-mentioned cases. From these figures, the influences of the first and second order component on the overall value are remarkable in both cases. That overall value of the non-uniform tire mounted case is higher than that of uniform tire mounted one.

(a) Non-uniform tire mounted  
(b) Uniform tire mounted  
Fig. 2-26 Tire revolution order ratio analysis
According to the vibration level change due to increase of the speed, as shown in the figure (b), there are the first peak at 80-90 km/h and second one at 110-130 km/h. The reason is considered to be the resonances between the generated excitation by the tire and vehicle components. In this case the resonances with engine components and unsprung masses would be generated. The floor vibration level is much affected by these generated forces. Consequently, it was found that the floor vertical acceleration coincided with the component of revolution order ratio of RFV in the drum running test.

The component due to the tire could be discriminated from the component induced by the road when the spectrum waveform obtained from the drum running test was superimposed on and compared with that obtained from the road running test.

Therefore, power spectrums of the floor vibrations obtained from the drum running test and the road running test for the vehicle of improper tire mounted case were calculated by the FFT analysis as shown in Figure 2-27 (a) and (b). It is clear that there exists peaks due to the RFV in the vibration of road running test for both vehicle speeds of (a) 40 and (b) 100 km/h. Thus the primary vibration component or its harmonic vibration component of tire rotation exerted an effect on the floor vibration in case of the tire having large RFV value.

Fig. 2-27 Comparison of acceleration
2.4. Evaluation of Road Surface as Input to Vehicle

2.4.1. Measurement of Road Surface Geometry

The observation of the surface geometry of road as the fundamental input concerning tire deformation during vehicle running is indispensable subject for the quantitative evaluation of vehicle vibration. The measurement of the road surface is mainly due to measuring technology of civil engineering field and it is necessary for these measurements to need a large man power, complicated instruments and equipments. Recently the special measurement systems using laser (light amplification by stimulated emission of radiation) applications\textsuperscript{28}-\textsuperscript{30} are available. However these systems are limited to the measurements of dug holes and large cracks on the road and need large scale measurements and special instrumented vehicle. Moreover in these systems it is uncertain how to cancel the influences of vehicle movements like pitching and rolling. Therefore it is rather doubtful to use in practical applications. From these reasons, to clarify the correlation between road profile and vehicle vibration at running condition, the simple measuring system of road surface is proposed from the viewpoint of easy mounting on passenger car.

The feature of the measuring system of the road surface in the running is to compensate the vehicle sprung mass location. The road profile is calculated from the vertical relative distance between the reference point and the road surface. Here the reference point of sprung mass is movable in the vehicle running. Then to obtain the correct road profile \(R\) during running, it is necessary to compensate the given vehicle height \(H\) with subting of the absolute movement \(Z\) of the sprung mass. As the results from these compensations, the road profile is expressed by running distance \(L\), which is detected by speed detector.

The fundamental measuring equation for road profile \(R\) is as follows.

\[
R = H - Z
\]

(2-1)

where,

\[
Z = \int \alpha \, dt
\]

(2-2)

Here the vehicle height \(H\) is obtained from optical sensing probe using
semi-conductor laser devices. And the absolute mass movement $Z$ is obtained by the double integration of the sprung mass acceleration $\alpha$ and the running distance $L$ is measured by the rotational angle of the wheel.

The system configuration is shown in Figure 2-28. The measurement vehicle installs the height measuring devices for relative distance between vehicle floor and the road surface, the accelerometer for absolute sprung mass movement, the travel distance detector and data recorder for storing these signals. The stored data is converted to digital signals by analog/digital converter scanner and calculated by personal computer.

Fig. 2-28 Measuring system for road profile

Here the way of treatment of vehicle sprung mass movement is shown in more details. The sprung mass displacement is calculated from double integration of the acceleration in this system. There are some DC components due to the drift in measured acceleration signals. Then, the direct calculated value from the measured data is superimposed by the drift component and the correct value could not obtain from these methods. Therefore to avoid the drift component, the high-pass filter is used before the integral calculations. In the case that the maximum detectable wave length is 50 m (spatial frequency is 0.02 cycle/m), the high-pass filter of 0.2 Hz is applied for running speed above 40 km/h. Moreover in the acceleration signals there are more high frequency components like vehicle local vibrations and unsprung mass vibrations. Therefore, choosing the sprung mass vibration, another low pass filter is needed for the calculation system. For the decision of the characteristics
of the filter, the acceleration of the floor which installed vehicle height detecting probe is measured during real running.

Figure 2-29 shows the measured data and the shape of the filter. Although frequency characteristics are depended on the running conditions such as speed and payload of the vehicle, the component of frequency region 0.2-2 Hz including sprung mass resonance is remarkable and the spectrum shape is stable. As the results, the higher cut-off frequency is chosen at 3 Hz as B and the lower cut-off frequency is at 0.2 Hz as A shown in the figure. From these operations of setting band pass filter before integral calculations, the vehicle sprung mass displacement can be obtained.

Next the signal processing flow is shown in Figure 2-30. As mentioned above, two kinds of filter is used in the measured acceleration signal. It becomes to be a phase-shift between the calculated value of absolute movement and measured one of relative movement by height sensor. Therefore obtaining the correct road profile, it is necessary to compensate the phase mismatching by phase shifter. In the integration of acceleration signal, the arithmetic methods and the Runge-Kutta methods are also used.

![Diagram of filter shape for acceleration](image1)

![Signal flow chart](image2)

Fig. 2-29 Filter shape for acceleration  Fig. 2-30 Signal flow chart

2.4.2. Observation of Vehicle Vibration on Various Road

The measuring performance is verified for the practical running on the imitated and the special road. The imitated road is made in stepwise shape by wood and settled on the road. In the cases of the various speeds of vehicle
the measured data is detected. One of the results for the vehicle speed of 40 km/h is shown in Figure 2-31 comparing with the actual stepwise cross-section of the road.

Next, the comparison of the measured data between profile meter and this method is tried for the road which geometry is already measured. In this case it is difficult to pass away on the same course along the road, the road is selected that the cross-sectional geometry change is very small. The results are shown in Figure 2-32 (a) and (b). Figure (a) shows the profile by the cross sectional profile meter and (b) shows the one using this system, respectively.

Moreover, the statistical evaluation is tried for the same results as shown in Figure 2-33. This is the comparison with power spectrum density of road surface displacement against spatial frequency. As the result of the comparison, the tendency of the road profile characteristics is almost similar to each other, but in the low frequency regions it is in the lack of coincidence because of the shortage of data length in this experiment.

![Fig. 2-31 Measured data for imitated road](image1)

![Fig. 2-32 Comparison of data](image2)

Finally the measured road profile and the subjective judgement of the ride comfort are compared. The examples are shown for flat and rough road in Figure 2-34 (a) and (b). In figure (a) it is shown for the flat road and the waviness of the surface geometry is rather long and the superimposed fluctuation is also small. On the other hand, in figure (b) there are some long waves and the superimposed fluctuation is rather large and revealed frequently.
From these results and discussions, the effects of the running road profile to the vehicle vibration are evaluated from the point of view of exciting input for the vehicle.

2.5. Summary

In this chapter, the vibration generated situations are surveyed and evaluation of ride comfort is tried in the real commercial vehicles.

First, the vibration of the vehicle is examined for the acceleration on the floor of the practical running state of vehicle on various roads. The vibrating phenomenon of the vehicle is classified from the quantitative evaluation of the individual frequency regions, and the generation levels for small compact cars and medium luxurious cars are compared with each other and the representative features are introduced. As the results of this survey, for the target characteristics of the vehicle comfort control, both the frequency regions of 0.4-2Hz components for flat feelings and 4-7Hz components for rugged feelings are important for the considerations. For improving the vehicle comfort, it must be realized that the both of these two components are to be lower to some extent.

Next, noticing the rotating tire for one of the factors of vibration generations during vehicle running, the uniformity of the tire structural stiffness is tried to measure. The relation between the tire uniformity and the...
vehicle vibration is discussed for the influence on the ride comfort characteristics. Here the new apparatus is developed for the measurement of the tire uniformity. Examination of the tire induced vibrations are shown to clarify the influences of the tire generated force of the various frequency regions on the vehicle vibration.

Finally in the last section of the chapter, the road surface disturbance is measured to clarify the influence of the road geometry on the vehicle vibration. The measuring equipment for the road profile is also developed here. It is composed with laser detector and other apparatus. From the experimental results of using these equipments, it is clarified that both of the evaluation and classification of the road surface become possible.

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CHAPTER 3 BASIC THEORIES OF ACTIVE VIBRATION CONTROL FOR VEHICLES

3.1. Introduction

In the field of research on active control for vehicle, many studies are presented for compromising both performances in vehicle motion and ride comfort.\textsuperscript{1-12} There is still existing the trade-off between the vehicle stability, controllability and ride comfort. On the other hand, the way of thinking for application of active control technology to aeronautic space craft and airplane is about to make utilized to automobile design of controlling elements. Besides there are significant progresses in the application studies according to the recent developments in the field of electronic control.

In vehicle suspension, vibration suppression mechanism is made of bearing spring, absorbing damper and their connecting linkage to decrease the harmful movements in vehicle attitude and vibration. In this technological field, there are many analytical studies for the active suspension and basic studies for application of recent modern control theories\textsuperscript{13-24}. They are dealing with theoretical approaches by numerical simulations and practical applied one by experimental apparatus.

Adapting active control to car chassis is expected to improve the vehicle stability, controllability and ride comfort remarkably. In the applications of the control technology, it would be accomplished for better safety and comfort by achieving the simultaneous adequate adjustments for each control element\textsuperscript{25-26}.

In this chapter, the basic theories for realizing the characteristics in vehicle performance are presented. They are all intended to achieve the desirable vehicle performances mentioned in the former chapter.
First in section 3.2., supposing a single wheel suspension, the design for vibration control is attempted for two degrees of freedom vibration model. In this study, the subject to be taken into account is a response characteristic of the control actuator. Then the control system is constructed by state variable feedback control, in which the linearization for the response property of the each actuator is intended. Moreover in this control, the frequency shaped performance index is also tried to introduce for the control target.

Next in section 3.3., another vibration control method is attempted in the low frequency range for controlling the vehicle attitude. It is closely related to the vehicle motion stability due to the steering maneuvers especially in the body response characteristics. Here the control system for pitching motion of the body due to the road irregularities is constructed for front and rear suspensions. Moreover to decrease body rolling and diagonal motions, the control system for four wheels is also discussed for reducing the body attitude fluctuations in the case of running on uneven road with different phase road irregularities.

Finally in section 3.4., the control system for applying the motion control due to steering maneuvers is introduced for achieving excellent vehicle controllability. Here, in the case of vehicle attitude control, the feedforward control methods are tried to compromise the feedback control of state variables using the information of preceding the delay of the control. Moreover considering the steering control for vehicle motion, the active steering control system is also induced for the ordinary steering system. It is additional control for ordinary front steering for compensating the more effective motion control according to the driver maneuver model.

These theoretical considerations are based on the ideal states of assumptions and intended for applications to the practical control systems which would be described in the following chapters.
3.2. Vibration Control for Quarter Car Model

3.2.1. Quarter Car Model with Controlled Elements

The quarter car model with active controlled suspension and its controlling system is shown schematically in Figure 3-1. This system is composed of an unsprung mass \( m_1 \) and a sprung mass \( m_2 \) with a bearing spring and a force generator as a controlling element. First of all, supposing the whole system to be a linear optimal feedback control one, optimal active system control laws are generated. The equations of motion for this system are shown as follows. Here the actual control force \( f \) is supposed to have a relation as shown in eq. (3-3).

\[
\begin{align*}
\dot{m}_1x_1 &= k_1(x_4-x_1)-k_2(x_1-x_2)-f \quad (3-1) \\
\dot{m}_2x_2 &= k_2(x_1-x_2)+f \quad (3-2) \\
\dot{f} &= -f/T + k'u/T \quad (3-3)
\end{align*}
\]

Fig. 3-1 Model of active suspension

Rearranging above equations, the following differential equations are obtained as a state space form.

\[
\begin{align*}
\dot{X} &= AX + Bu + Du \\
Y &= CX
\end{align*}
\]

where, \( X = [y, \dot{y}, x_2, \dot{x}_2, f] \) \( y=x_1-x_3 \)

\[
A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ a_1 & 0 & a_2 & 0 & a_3 \\ 0 & 0 & 0 & 1 & 0 \\ a_4 & 0 & 0 & 0 & a_5 \\ 0 & 0 & 0 & 0 & a_6 \end{bmatrix} \quad (3-7) \\
B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ b_1 \end{bmatrix} \quad D = \begin{bmatrix} 0 \\ 0 \\ a_2 \end{bmatrix} \quad (3-8)
\]

\[
\begin{align*}
a_1 &= -(k_1/m_1 + k_2/m_1 + k_2/m_2) \\
a_2 &= -k_1/m_1 \\
a_3 &= -(1/m_1 + 1/m_2) \\
a_4 &= k_2/m_2 \\
a_5 &= 1/m_2 \\
a_6 &= -1/T \\
b_1 &= k'/T \end{align*}
\]

C: Unit Matrix
Using these equations, the controlling gain for each variable is calculated by supposing the performance index $J$. In the performance index, the weighting parameters are selected from the points of view that the sprung mass vertical acceleration is concerned with ride-comfort of the vehicle and the sprung mass vertical displacement is also concerned with vehicle motion. The other weighting parameter is for energy used in this system. The control system is designed to minimize the following performance index.

$$
J = \int (X^T(t)QX(t)+U^T(t)RU(t))\,dt
= \int \left( \alpha_1 \dot{x}_z^2 + \alpha_2 x_z^2 + Ru^2 \right) \,dt \quad (3-9)
$$

From these equations, solving the linear quadratic optimum regulator for this system, the optimal control force $u$ is obtained as the function of the state variables as follows.

$$
u = -KX \quad (3-10)
$$

where each coefficient is expressed as:

$$K = [K_1, K_2, K_3, K_4, K_5] \quad (3-11)$$

The optimal control theory presents the feedback gain for each state variable as follows. These gains are numerically obtained by Runge-Kutta methods.

$$
K = R^{-1}B^TP \quad (3-12)
$$

where the matrix $P$ is for following Ricatti equation.

$$PA + A^TP - PBR^{-1}B^TP + Q = 0 \quad (3-13)$$

Next, the parametric study for the weighting functions in the performance index is introduced for obtaining the root loci of the regulator. The values in this model are supposed as follows in concerning to the model apparatus of this system, which is used in experimental study as shown in the following chapter.

$m_1 = 45.9\text{kg}, \quad m_2 = 224\text{kg}, \quad k_1 = 186\text{N/mm}, \quad k_2 = 16.2\text{N/mm}, \quad T = 0.005\text{sec}$
Using these values, the control parameters of the optimal system and eigenvalues are obtained by varying the weighting parameter \( \alpha_1, \alpha_2 \) and \( R \) in the performance index. They are chosen from the three categories as follows.

1. Case LQ1: In the cases of \( \alpha_1 = 1, \alpha_2 = 0 \) and various \( R \) as shown in Table 3-1.
2. Case LQ3: In the cases of \( \alpha_1 = 1 \), constant \( R \) and various \( \alpha_2 \).
3. Case LQ2: In the cases of \( \alpha_1 = 0, \alpha_2 = 1 \) and various \( R \).

Case LQ1 is intended to make the acceleration of the sprung mass smaller. Case LQ2 is also intended to make the displacement of the sprung mass smaller. These are concerning to a ride comfort and a stable attitude of the sprung mass, respectively. And case LQ3 is intended to make both components smaller at the same time.

Figure 3-2 shows the root loci for these three conditions. As for the loci of unsprung mass vibrations as shown in the upper right hand in the figure, it is almost no change according to the parameters. On the other hand, the loci of sprung mass vibrations are varied remarkably according to the value of parameter. In the cases of (1), the pole assignment moves to zero according to the weighting value \( R \). It means the decrease of the resonance frequency and increase of the damping factor. In the cases of (2) and (3), the pole assignments move away from zero as the weighting value \( \alpha_2 \) increases and the weighting value \( R \) decreases. This tendency indicates the increase of the resonant frequency.

Noticing the movements of these poles, the control gains are obtained for three typical cases (a), (b) and (c) as shown in Figure 3-2 and Table 3-1. As mentioned before, case (a) means a condition for the best ride comfort because of the minimum acceleration. Case (b) means a condition for the best stable attitude. Case (c) is a condition for both components. These three conditions are selected for deciding the target control forces in the coming chapter.
Table 3-1 Setting conditions, control gains and eigenvalues

<table>
<thead>
<tr>
<th>Case</th>
<th>$\alpha_1$</th>
<th>$\alpha_2$</th>
<th>$R$</th>
<th>$K_1$</th>
<th>$K_2$</th>
<th>$K_3$</th>
<th>$K_4$</th>
<th>$K_5$</th>
<th>Eigenvalues</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>64.0</td>
<td>-1.76</td>
<td>46.3</td>
<td>19.5</td>
<td>0.017</td>
<td>-0.46±j66.67, -0.22±j67.6, -201.</td>
</tr>
<tr>
<td>LQ 1</td>
<td>1</td>
<td>0</td>
<td>$10^{-1}$</td>
<td>183.9</td>
<td>-3.71</td>
<td>116.3</td>
<td>43.2</td>
<td>0.063</td>
<td>-0.97±j66.5, -0.47±j67.5, -209.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>0</td>
<td>$2\times10^{-2}$</td>
<td>294.3</td>
<td>-4.90</td>
<td>171.6</td>
<td>60.6</td>
<td>0.114</td>
<td>-1.31±j66.3, -0.62±j67.4, -218.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>0</td>
<td>$10^{-2}$</td>
<td>1324.</td>
<td>-8.83</td>
<td>490.</td>
<td>170.</td>
<td>0.761</td>
<td>-2.35±j45.5, -0.82±j66.6, -346.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>0</td>
<td>$5\times10^{-4}$</td>
<td>1985.</td>
<td>-9.13</td>
<td>583.</td>
<td>221.</td>
<td>1.264</td>
<td>-2.35±j37.2, -0.69±j66.3, -447.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>0</td>
<td>$10^{-4}$</td>
<td>4731.</td>
<td>-3.58</td>
<td>432.</td>
<td>289.</td>
<td>3.598</td>
<td>-1.48±j66.3, -0.22±j66.0, -916.</td>
</tr>
<tr>
<td>LQ 2</td>
<td>0</td>
<td>1</td>
<td>$10^{-8}$</td>
<td>4.37</td>
<td>1.42</td>
<td>376.</td>
<td>140.</td>
<td>0.030</td>
<td>-2.95±j7.34, -0.003±j67.7, -200.</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>1</td>
<td>$10^{-9}$</td>
<td>245.2</td>
<td>9.11</td>
<td>30573.</td>
<td>1335.</td>
<td>0.259</td>
<td>-25.3±j27.1, -0.468±j66.6, -200.</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>1</td>
<td>$10^{-10}$</td>
<td>555.4</td>
<td>13.0</td>
<td>99037.</td>
<td>2573.</td>
<td>0.462</td>
<td>-44.7±j49.1, -0.859±j66.9, -201.</td>
</tr>
<tr>
<td>LQ 3</td>
<td>1</td>
<td>1</td>
<td>$10^{-4}$</td>
<td>4730.</td>
<td>-3.58</td>
<td>442.</td>
<td>292.</td>
<td>3.598</td>
<td>-1.50±j11.66, -0.22±j66.0, -916.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>$10^{2}$</td>
<td>$10^{-4}$</td>
<td>4725.</td>
<td>-3.55</td>
<td>1041.</td>
<td>455.</td>
<td>3.606</td>
<td>-2.30±j12.41, -0.23±j66.0, -916.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>$10^{4}$</td>
<td>$10^{-4}$</td>
<td>4692.</td>
<td>-3.30</td>
<td>9356.</td>
<td>1413.</td>
<td>3.654</td>
<td>-7.04±j7.07, -0.22±j66.0, -916.</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>$10^{6}$</td>
<td>$10^{-4}$</td>
<td>4601.</td>
<td>-1.10</td>
<td>99596.</td>
<td>4593.</td>
<td>3.806</td>
<td>-22.3±j22.3, -0.22±j66.0, -916.</td>
</tr>
</tbody>
</table>

Fig. 3-2 Root loci for the optimum regulator
3.2.2. Vibration Control using Frequency-Shaped Methods

As to the vibration sensitivity of the passenger on the vehicle, it is known that human sensitivity is depending on the frequency range which is remarkable in the vehicle behaviors. There are many attempts to introduce the human sensitivity to the performance indexes of the controls\textsuperscript{27-30}). In these methods, the optimization of control system is evaluated by not only the squared mean value of the variables, but also the frequency weighted value.

In designing the control systems, there is a strict trade-off between human sensitivity and vehicle movements. The passenger is preferred to be comfortable for comfort sensitive frequency range and the vehicle must be controlled in the stable attitude during running.

In this section, the application for the frequency shaped methods to the ride comfort and vehicle attitude control systems are discussed.

The control system is designed from the point of view of how to manage the trade-off of the two kinds of frequency domains, namely ride comfort and attitude as illustrated in previous chapter.

The construction of the control system by frequency shaped method is explained as follows. As mentioned in the previous section, the ordinary optimum regulator is used for the performance index of the quadratic form of time domain as the squared values of target variables. There is no frequency dependence on the values of the weighting parameters. On the contrary, the frequency shaped optimum regulator is converted for the performance index to frequency area from time area by using the Perseval equations. Therefore the performance index for the control system is shown as follows.

\[ J = \frac{1}{2}\pi \int \left( W^T(-j\omega)X(j\omega) + V^T(-j\omega)V(j\omega) \right) d\omega \quad (3-14) \]

where,

\[ W(j\omega) = L(j\omega)X(j\omega), \quad V(j\omega) = M(j\omega)U(j\omega) \]

\( L(j\omega), M(j\omega) \): Weighting Function

\( X(j\omega), U(j\omega) \): Fourier Transform of \( X(t) \), \( U(t) \)
Next, the design of the filter for frequency weighting function will be presented. To establish the frequency weighting function improving ride comfort and vehicle attitude at the same time, the characteristics of the response on the sprung mass of vehicle are measured. Figure 3-3 shows the results of the experiments in the following chapter. In the figure, a solid line shows the acceleration of the sprung mass and a broken line shows the displacement of the sprung mass. As shown in Figure 3-3, the acceleration is dominant in the frequency range of 2-8 Hz which is most closely related to human sensitivity in ride comfort, and the displacement is dominant in the frequency range below 1 Hz, which is related to the vehicle motion in the maneuverability. Fig.3-3 Frequency response of sprung mass

As the frequency weighting functions, the following two kinds of filters are designed. Each filter \( L_1 \) and \( L_2 \) are for the acceleration and the displacement of the sprung mass, respectively. Here, symbol \( s \) is Laplacian.

The peak frequency at the maximum value of each weighting function is 3.25 Hz on \( L_1 \) and 0.7 Hz on \( L_2 \), respectively.

\[
W_1(s)=L_1(s)x_1(s) \quad \text{(3-15)}
\]
\[
W_2(s)=L_2(s)x_2(s) \quad \text{(3-16)}
\]

where,

\[
L_1(s)=q_1 \frac{s^2+93.9s+417}{s^2+29.696s+417} \quad \text{(3-17)}
\]
\[
L_2(s)=q_2 \frac{s^2+20.0s+19.3}{s^2+6.22s+19.3} \quad \text{(3-18)}
\]

where, \( q_1 \) and \( q_2 \) are the parameters for the acceleration and the displacement of the sprung mass in the cases of no frequency shaped controls.
From the equations (3-17) and (3-18), the state space expressions are as follows. Here, the acceleration of sprung mass as input is used from the relative displacement and the acting force.

\[
\begin{bmatrix}
\dot{z}_1 \\
\dot{z}_2
\end{bmatrix} = \begin{bmatrix} a_{p_1} & a_{p_2} \\ 1 & 0 \end{bmatrix} \begin{bmatrix} z_1 \\
 0 & 0 \end{bmatrix} + \begin{bmatrix} a_4 & a_s \\ 0 & 0 \end{bmatrix} \begin{bmatrix} y \\
 f \end{bmatrix} \tag{3-19}
\]

\[
\ddot{x}_2 = q_2 \begin{bmatrix} c_{p_1} & 0 \end{bmatrix} \begin{bmatrix} z_1 \\
 z_2 \end{bmatrix} + q_1 \begin{bmatrix} a_4 & a_s \end{bmatrix} \begin{bmatrix} y \\
 f \end{bmatrix} \tag{3-20}
\]

\[
\begin{bmatrix}
\dot{z}_3 \\
\dot{z}_4
\end{bmatrix} = \begin{bmatrix} a_{p_1} & a_{p_4} \\ 1 & 0 \end{bmatrix} \begin{bmatrix} z_3 \\
 0 & 0 \end{bmatrix} + \begin{bmatrix} 1 \\
 0 \end{bmatrix} x_2 \tag{3-21}
\]

\[
x_2 = q_2 \begin{bmatrix} c_{p_1} & 0 \end{bmatrix} \begin{bmatrix} z_3 \\
 z_4 \end{bmatrix} + q_1 \cdot x_2 \tag{3-22}
\]

where,

\[
\begin{align*}
a_{p_1} &= -29.7, \\
a_{p_2} &= -417, \\
c_{p_1} &= 64.2, \\
a_{p_4} &= -6.22, \\
a_{s} &= -19.3, \\
c_{p_2} &= 13.8
\end{align*}
\]

Figure 3-4 shows the characteristics for two kinds of the weighting filter \(L_1(s)\) and \(L_2(s)\). Here, the gain of the weighting function becomes small except in the objective frequency range.

![Fig. 3-4 Frequency characteristics](image-url)
As for the performance indexes for the frequency shaped control, two kinds of index are offered as LQ1 for weighted in the acceleration of the sprung mass and LQ3 for weighted in the displacement as well as the acceleration of the sprung mass. The performance indexes are shown as follows. In these equations, weighting coefficient $r$ is disposed on the input $U$. Figure 3-5 shows the two kinds of frequency weighting function $Q_1$ and $Q_2$ in the frequency area concerning two kinds of parameter.

$$J_1 = \frac{1}{2\pi} \int_{-\infty}^{\infty} [W_1(-j\omega)W_2(j\omega)]^2 U(-j\omega)U(j\omega) d\omega \quad (3-23)$$

$$J_2 = \frac{1}{2\pi} \int_{-\infty}^{\infty} [W_3(-j\omega)W_4(j\omega)]^2 + rU(-j\omega)U(j\omega) d\omega \quad (3-24)$$

![Graph showing frequency shaped functions](image)

Fig.3-5 Frequency shaped functions

The state equations on the extended system including weighting functions are shown as follows. In the case of the extended system of adding the acceleration filter $L_1(s)$ of the sprung mass is the followings.

$$\begin{bmatrix}
\dot{z}_1 \\
\dot{z}_2 \\
\dot{y} \\
\dot{x}_1 \\
\dot{x}_2 \\
\dot{f}
\end{bmatrix} =
\begin{bmatrix}
a_1 & a_2 & a_3 & 0 & 0 & 0 & a_s \\
1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & a_1 & 0 & a_2 & 0 & a_s \\
0 & 0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & a_s \\
\end{bmatrix}
\begin{bmatrix}
z_1 \\
z_2 \\
y \\
x_1 \\
x_2 \\
f
\end{bmatrix} +
\begin{bmatrix}
0 \\
0 \\
0 \\
0 \\
0 \\
b_1
\end{bmatrix} u \quad (3-25)$$

Hence the control force value $u$ is given from the equation.

$$u = -[KR_3, KR_1] \cdot [Z, X]^T \quad (3-26)$$

where,

$$Z = [z_1, z_2]^T.$$
The each matrix $R_{xx}, R_{uu}$ is shown as follows.

**LQ1 type:**

$$R_{xx} = \begin{bmatrix} r_1 & 0 & r_2 & 0 & 0 & 0 & r_3 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ r_2 & 0 & r_4 & 0 & 0 & 0 & r_5 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & r_7 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ r_3 & 0 & r_5 & 0 & 0 & 0 & r_6 \\ \end{bmatrix}, \quad R_{uu} = r^2 \quad (3-27)$$

Similarly in the case of LQ3, the matrix $R_{xx}, R_{uu}$ is shown as follows.

**LQ3 type:**

$$R_{xx} = \begin{bmatrix} r_1 & 0 & 0 & 0 & r_2 & 0 & 0 & 0 & r_3 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & r_4 & 0 & 0 & 0 & r_5 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & r_6 & 0 & 0 & 0 & r_7 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ r_3 & 0 & r_5 & 0 & 0 & 0 & r_6 & 0 & 0 \\ \end{bmatrix}, \quad R_{uu} = r^2 \quad (3-28)$$

where,

\[
P_1 = q_1^2, P_2 = q_2^2, r_1 = P_1 \cdot c_{p_1}, r_3 = P_1 \cdot a_{4} \cdot c_{p_1}.
\]

\[
r_2 = P_1 \cdot a_{4} \cdot c_{p_1}, r_4 = P_1 \cdot a_{4} \cdot a_{5}, r_5 = P_1 \cdot a_{4}^2, r_6 = P_1 \cdot a_{5}^2.
\]

\[
r_1 = P_2 (LQ1type: r_7 = 0), r_3 = P_2 \cdot c_{p_1}, r_6 = P_2 \cdot c_{p_1}.
\]

To obtain the optimal control condition, many numerical calculations are carried out. As the result, value $q_1$, $q_2$ and $r$ are supposed as follows.

In the case of LQ1: $q_1 = 1.1$, \quad $r = 0.01$

In the case of LQ3: $q_1 = 1.0$, \quad $q_2 = 150.0$, \quad $r = 0.01$
Next, the robustness for the control system is discussed from the circle conditions for the regulators. If the Nyquist trajectory of the control condition is located out of the eigen circle, the control condition has the robustness. The comparisons of trajectories around the circle are carried out in the case of with and without control for each LQ1 and LQ3 type. The circle conditions for each control are examined in the range of frequency 0.1-1000 Hz.

Figure 3-6 shows the comparisons of Nyquist diagrams. In the case of ordinary LQ control, $KR_1(sI-A)^{-1}B$ is calculated and in the case of frequency shaped LQ control, $(KR_2, KR_1)(sI-A')^{-1}B'$ is also calculated. Here, $A'$ and $B'$ are $A$ and $B$ matrix in the extended system in equation (3-25). As shown in the figure, each Nyquist trajectory does not enter to the inside of the eigen circle and the circle condition is satisfied. Therefore the control system is stable and control robustness is also confirmed.

![Fig. 3-6 Comparison of Nyquist diagrams](image-url)
3.3. Vehicle Vibration Control for Half and Whole Car Model

3.3.1. Vibration Control for Half Car Model

The active controls to the front and rear suspensions are applied for the vibration control for half car model. The half car model is shown in Figure 3-7. It is composed of four degrees of freedom, namely vertical motion of two unsprung masses and vertical and rotational motions of the sprung mass. The equations of motion are described as follows.

\[
m_r \ddot{x}_i = k_i (x_0 - x_i) - k_2 (x_i - x_0) - c_r (\dot{x}_i - \dot{x}_0) - f_r \quad (3-29)
\]

\[
m_r \ddot{z}_i = k_i (x_0 - x_i) - k_2 (x_i - x_0) - c_r (\dot{z}_i - \dot{z}_0) - f_r \quad (3-30)
\]

\[
M \ddot{X} = k_2 (x_i - x_2) + c_r (\dot{X}_i - \dot{X}_2) + k_2 (x_r - x_2) + c_r (\dot{X}_r - \dot{X}_2) + f_r + f_r \quad (3-31)
\]

\[
I \ddot{\theta} = -l_1 k_2 (x_i - x_2) - l_1 c_r (\dot{\theta}_i - \dot{\theta}_2) - l_1 f_r + l_1 k_2 (x_r - x_2) + l_1 c_r (\dot{\theta}_r - \dot{\theta}_2) + l_1 f_r \quad (3-32)
\]

\[
\dot{f}_r = -f_r / T_f + k_r u_r / T_r \quad (3-33)
\]

\[
\dot{\theta}_r = -f_r / T_r + k_r u_r / T_r \quad (3-34)
\]

Where, \(x_s = X - 1, \theta, x_s = X + 1, \theta, m_j (j=f, r), M\) denote each of masses, \(k_{ij}, k_{2j} (j=f, r)\) denote each spring constant, \(x_{0j} (j=f, r)\) denote input displacement, \(x_{1j}, x_{2j} (j=f, r)\) denote the absolute displacement of each masses. The notation \(f_j (j=f, r)\) is generated force and supposed with the first order time lag (time constant \(T_j\)) for the optimal generating force \(U_j (j=f, r)\) of the vibration system.

Hereunder for the state variables of the vibration control, choosing the relative displacement \(y_j = (x_{1j} - x_{2j})\), relative velocity \(\dot{y}_j\), the sprung displacement \(x_{2j}\), the sprung velocity \(\dot{x}_{2j}\) and generated force \(f_j\), the state space
Expressions are as follows.

\[
\begin{align*}
\mathbf{x} &= A \cdot \mathbf{x} + B \cdot \mathbf{u} + D \cdot \mathbf{x} \quad \text{(3-36)} \\
\mathbf{y} &= C \cdot \mathbf{x} \\
\end{align*}
\]

where,

\[
\mathbf{X} = [\mathbf{y}_t, \mathbf{y}_r, \mathbf{y}_t, \mathbf{y}_r, \mathbf{x}_{2t}, \mathbf{x}_{2r}, \mathbf{x}_{2t}, \mathbf{x}_{2r}, \mathbf{f}_t, \mathbf{f}_r]^T \quad \text{(3-38)}
\]

\[
\begin{bmatrix}
\dot{\mathbf{y}}_t \\
\dot{\mathbf{y}}_r \\
\ddot{\mathbf{y}}_t \\
\ddot{\mathbf{y}}_r \\
\dot{\mathbf{x}}_{2t} \\
\dot{\mathbf{x}}_{2r} \\
\ddot{\mathbf{x}}_{2t} \\
\ddot{\mathbf{x}}_{2r} \\
\dot{\mathbf{f}}_t \\
\dot{\mathbf{f}}_r
\end{bmatrix} =
\begin{bmatrix}
0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
a_1 & a_2 & a_3 & a_4 & a_5 & 0 & 0 & 0 & a_4 & a_7 \\
a_8 & a_9 & a_{10} & a_{11} & 0 & a_{12} & 0 & a_3 & a_4 & a_7 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & a_7 & 0 & a_2 \\
a_1 & a_2 & a_3 & a_4 & a_5 & 0 & 0 & 0 & a_3 & a_4 \\
a_1 & a_2 & a_3 & a_4 & a_5 & 0 & 0 & 0 & a_3 & a_4 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & a_2 & 0 & a_2 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & a_2
\end{bmatrix}
\begin{bmatrix}
\mathbf{y}_t \\
\mathbf{y}_r \\
\dot{\mathbf{y}}_t \\
\dot{\mathbf{y}}_r \\
\mathbf{x}_{2t} \\
\mathbf{x}_{2r} \\
\dot{\mathbf{x}}_{2t} \\
\dot{\mathbf{x}}_{2r} \\
\mathbf{f}_t \\
\mathbf{f}_r
\end{bmatrix} +
\begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
x_{2t} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
x_{2r} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\mathbf{x}_{2t} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\mathbf{x}_{2r} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\mathbf{f}_t & b_1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\mathbf{f}_r & 0 & b_2 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}_t \\
\mathbf{u}_r
\end{bmatrix}
\quad \text{(3-39)}
\]

\[
\begin{align*}
a_1 &= -\left(k_{11}/m + k_{22}/m + k_{2r}/M + l^2 k_{22}/l\right) \\
a_2 &= -\left(c_r/m + c_r/M + l^2 c_r/l\right) \\
a_3 &= -(k_{22}/M - l^2, k_{2r}/l) \\
a_4 &= -(c_r/M - l^2, c_r/l) \\
a_5 &= -k_{11}/m \\
a_6 &= -(1/m + l^2, l^2)/l \\
a_7 &= -(c_r/M - l^2, c_r/l) \\
a_8 &= -(k_{22}/M - l^2, k_{2r}/l) \\
a_9 &= -(c_r/m + c_r/M + l^2 c_r/l) \\
a_{10} &= -(k_{11}/m + k_{22}/m + k_{2r}/M + l^2 k_{22}/l) \\
a_{11} &= -k_{11}/m \\
a_{12} &= -(1/m + l^2, l^2)/l \\
a_{13} &= -(1/m + l^2, l^2)/l \\
a_{14} &= -(1/m + l^2, l^2)/l \\
a_{15} &= c_r/M + l^2 c_r/l \\
a_{16} &= c_r/M + l^2 c_r/l \\
a_{17} &= c_r/M + l^2 c_r/l \\
a_{18} &= c_r/M + l^2 c_r/l \\
a_{19} &= c_r/M + l^2 c_r/l \\
a_{20} &= 1/M - l^2, l^2/l \\
a_{21} &= c_r/M - l^2, c_r/l \\
a_{22} &= c_r/M - l^2, c_r/l \\
a_{23} &= c_r/M - l^2, c_r/l \\
a_{24} &= c_r/M - l^2, c_r/l \\
a_{25} &= c_r/M - l^2, c_r/l \\
a_{26} &= 1/M + l^2, l^2/l \\
a_{27} &= 1/M + l^2, l^2/l \\
a_{28} &= 1/M + l^2, l^2/l \\
a_{29} &= 1/M + l^2, l^2/l \\
\beta_1 &= 1/T_r \\
\beta_2 &= 1/T_r \\
\end{align*}
\]

b_1 = 1/T_r \\
b_2 = 1/T_r \\
d_1 = k_{11}/m \\
d_2 = k_{11}/m
The following performance index is assumed for the optimal state feedback control.

\[ J = \int \left( \begin{bmatrix} \dddot{x} \cr \dddot{z} \end{bmatrix} Q_1 \begin{bmatrix} \dddot{x} \cr \dddot{z} \end{bmatrix} + \begin{bmatrix} x \cr z \end{bmatrix} Q_2 \begin{bmatrix} x \cr z \end{bmatrix} + R \begin{bmatrix} u \cr u \end{bmatrix} \right) \, dt \]  

(3-40)

In the index, function \( Q_1 \) and \( Q_2 \) are the weighting for the acceleration and the displacement of the sprung mass. Function \( R \) is the weighing for control forces. For the ride comfort control, the vertical acceleration and the pitching angular acceleration of the sprung mass should have to be minimized. According to the performance index, the optimal control forces are expressed as follows.

\[ u_f = -K_1x \\ u_r = -K_2x \]  

(3-41)

\[ K_j = [K_{j1}, K_{j2}, K_{j3}, K_{j4}, K_{j5}, K_{j6}, K_{j7}, K_{j8}, K_{j9}, K_{j10}, K_{j11}, K_{j12}] \quad (j=f,r) \]  

(3-42)

Next, the pole assignments of the regulators are obtained by increasing weighting function \( R \). In this design the dimensions of the half car model are supposed as follows in the reference of the experimental vehicle in the previous chapters.

\[
\begin{align*}
K_{f1} &= 22400 \text{ kgf/m} \\
m_f &= 10.24 \text{ kgf's/m} \\
k_{f2} &= 2400 \text{ kgf/m} \\
c_f &= 170 \text{ kgf's/m} \\
l_f &= 1.2695 \text{ m} \\
T_f &= 0.005 \text{ sec} \\
M &= 97.75 \text{ kgf's/m} \\
K_{r1} &= 22400 \text{ kgf/m} \\
m_r &= 10.24 \text{ kgf's/m} \\
k_{r2} &= 2400 \text{ kgf/m} \\
c_r &= 170 \text{ kgf's/m} \\
l_r &= 1.4005 \text{ m} \\
T_r &= 0.005 \text{ sec} \\
I &= 228.8 \text{ kgf's/m} \end{align*}
\]

Figure 3-8 shows the root loci for the obtained pole assignments. As for the unsprung mass vibration, the trajectories of the root loci are changed remarkably according to the values of parameter.

In the case of (1) for the weightings of the acceleration terms, the root locus moves along half circle as the weighting value of \( R \) increases. Namely, both of heave and pitch motion are damped and the both of the frequencies are decreasing. On the contrary, in the case of (2) for the weightings of the acceleration and the displacement of the sprung mass, the root locus is moving...
toward left upper side from the curved circle mentioned above as increasing the weighting value of R. Finally in the case (3) of weightings for the vehicle attitude, the root locus is drawing directly to left upper side. In these conditions, the heave and pitch frequencies are moved to higher frequencies and the damping factors are also increased.

The given examples of the gains for case (I) are shown in Table 3-2. In the table, notation $K_f$ is for front wheel and $K_r$ is for rear wheel, respectively. In this case function $Q_1$ and $R$ are supposed as follows:

$$Q_1 = \begin{pmatrix} 1.21 & 0 \\ 0 & 1.21 \end{pmatrix}, \quad R = \begin{pmatrix} 2.5e-04 & 0 \\ 0 & 2.5e-04 \end{pmatrix}$$

The pole locations of this example are the nearest signs to the right bottom of Fig. 3-8. From the location, the both effects of lower resonance frequency and damping effect of heave and pitch motions are expected by this control.

![Root loci for half car model](image)

**Fig. 3-8 Root loci for half car model**

**Table 3-2** Example of the control gain for case (1)

<table>
<thead>
<tr>
<th></th>
<th>$y_f$</th>
<th>$y_r$</th>
<th>$\dot{y}_f$</th>
<th>$\dot{y}_r$</th>
<th>$x_{2f}$</th>
<th>$x_{2r}$</th>
<th>$\dot{x}_{2f}$</th>
<th>$\dot{x}_{2r}$</th>
<th>$f_f$</th>
<th>$f_r$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_f$</td>
<td>518.16</td>
<td>-22.423</td>
<td>87.006</td>
<td>0.22617</td>
<td>-612.68</td>
<td>-203.54</td>
<td>190.04</td>
<td>-36.55</td>
<td>0.19333</td>
<td>1.743e-3</td>
</tr>
<tr>
<td>$K_r$</td>
<td>-26.646</td>
<td>-63.289</td>
<td>0.43326</td>
<td>67.205</td>
<td>-218.79</td>
<td>-917.31</td>
<td>-21.319</td>
<td>198.45</td>
<td>2.8569e-3</td>
<td>0.2930</td>
</tr>
</tbody>
</table>
3.3.2. Control Considering Sprung Mass Vibration Mode

The active controls to the four each wheel are applied to four wheels model considering the modes of vehicle body. The four wheels model is shown in Figure 3-9. It is composed of three degrees of freedom, namely the vertical, rolling and pitching motion of the vehicle body, which is related with front and rear as well as right and left four wheels. The equations of motion are described as follows.

\[ M \ddot{z} = \sum_{i=1}^{S_1} S_i - \Sigma u_i \]  
(3-43)

\[ \dot{\phi} = T_i ((S_1 - S_2) - (u_1 - u_3))/2 + T_i ((S_3 - S_4) - (u_4 - u_4))/2 \]  
(3-44)

\[ \dot{z}_i = -a_i ((S_1 - S_2) - (u_1 - u_3)) + a_i ((S_3 - S_4) - (u_4 - u_4)) \]  
(3-45)

\[ \ddot{f}_i = -f_i/T_i + ku_i/T_i \]  
(3-46)

\[ S_i = -k_i z_i - c_i \dot{z}_i \]  
(3-47)

\[ z_i = z_1 + (-1)^{i-1} a_i \theta - (-1)^i T_i \phi/2 \]  
(3-48)

Here \( z_{S_i} \) denotes the relative displacement for each wheel. \( S_1 \) denotes side force, \( f_i \) denotes the active generated force and \( u_i \) denotes the active optimal force. The active generated force has the first order time lag (time constant \( T_i \)) for the optimal force.

Here under for the state variables of the control, choosing the relative displacement \( z \) of the body, the relative velocity \( \dot{z} \), the body rolling angle \( \phi \), the rolling angular velocity \( \dot{\phi} \), the body pitching angle \( \theta \), the pitching angular velocity \( \dot{\theta} \) and the control force \( f_i \) for each wheel, the state space expressions are as follows.

Fig. 3-9 Four wheels model
\[ \dot{X} = A \cdot X + B \cdot u \]  
\[ Y = C \cdot X \]  
\[ X = [z, \dot{z}, \phi, \dot{\phi}, \theta, \dot{\theta}, f_1, f_2, f_3, f_4]^T \]

where,
\[
A = \begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-2(k_x + k_r)/M & 0 & a_{23} & a_{24} & a_{25} & a_{26} & a_{27} & a_{28} & a_{29} & a_{210} \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & a_{41} & a_{42} & 0 & a_{43} & a_{44} & a_{45} & a_{46} & a_{47} & a_{48} \\
0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
a_{11} & a_{12} & 0 & a_{13} & a_{14} & a_{15} & a_{16} & a_{17} & a_{18} & a_{19} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & a_{39} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & a_{40} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & a_{10} \\
\end{bmatrix}
\]

\[
B = \begin{bmatrix}
0 \\
0 \\
0 \\
k/T \\
0 \\
0 \\
0 \\
k/T \\
0 \\
k/T \\
\end{bmatrix}
\]

where,
\[
a_{23} = 2(k_x + k_r)/M \\
a_{24} = 2(a_x k_x - a_y k_r)/M \\
a_{25} = 2(a_x a_y - a_z）/M \\
a_{26} = -(T_x k_x + T_y k_r)/21 \\
a_{27} = -T_x/21 \\
a_{28} = -T_y/21 \\
a_{29} = -2(k_x + k_r)/1 \\
a_{39} = -2(a_x k_x + a_y k_r)/1 \\
a_{40} = -2(a_x a_y - a_z)/1 \\
a_{10} = -1/T_i \\
a_{41} = 2(a_x k_x - a_y k_r)/i \\
a_{42} = 2(a_x a_y - a_z)/i \\
a_{43} = -2(a_x k_x + a_y k_r)/i \\
a_{44} = -2(a_x a_y - a_z)/i \\
a_{45} = a_{46} = a_{47} = a_{48} = 0 \\
a_{11} = a_{12} = a_{13} = a_{14} = 0 \\
a_{15} = a_{16} = a_{17} = a_{18} = 0 \\
a_{19} = a_{20} = a_{21} = a_{22} = 0 \\
a_{23} = a_{24} = a_{25} = a_{26} = a_{27} = a_{28} = a_{29} = a_{30} = 0 \\
a_{41} = a_{42} = a_{43} = a_{44} = a_{45} = a_{46} = a_{47} = a_{48} = 0 \\
a_{51} = a_{52} = a_{53} = a_{54} = a_{55} = a_{56} = a_{57} = a_{58} = a_{59} = a_{60} = a_{61} = a_{62} = a_{63} = a_{64} = a_{65} = a_{66} = a_{67} = a_{68} = a_{69} = a_{70} = a_{71} = a_{72} = a_{73} = a_{74} = a_{75} = a_{76} = a_{77} = a_{78} = a_{79} = a_{80} = 0 \\
a_{81} = a_{82} = a_{83} = a_{84} = a_{85} = a_{86} = a_{87} = a_{88} = a_{89} = a_{90} = 0 \\
a_{91} = a_{92} = a_{93} = a_{94} = a_{95} = a_{96} = a_{97} = a_{98} = a_{99} = a_{100} = 0 \\
a_{101} = 0 \\
a_{102} = 0 \\
a_{103} = 0 \\
a_{104} = 0 \\
a_{105} = 0 \\
a_{106} = 0 \\
a_{107} = 0 \\
a_{108} = 0 \\
a_{109} = 0 \\
a_{110} = 0 \\
a_{111} = 0 \\
a_{112} = 0 \\
a_{113} = 0 \\
a_{114} = 0 \\
a_{115} = 0 \\
a_{116} = 0 \\
a_{117} = 0 \\
a_{118} = 0 \\
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a_{121} = 0 \\
a_{122} = 0 \\
a_{123} = 0 \\
a_{124} = 0 \\
a_{125} = 0 \\
a_{126} = 0 \\
a_{127} = 0 \\
a_{128} = 0 \\
a_{129} = 0 \\
a_{130} = 0 \\
a_{131} = 0 \\
a_{132} = 0 \\
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a_{134} = 0 \\
a_{135} = 0 \\
a_{136} = 0 \\
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a_{148} = 0 \\
a_{149} = 0 \\
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a_{151} = 0 \\
a_{152} = 0 \\
a_{153} = 0 \\
a_{154} = 0 \\
a_{155} = 0 \\
a_{156} = 0 \\
a_{157} = 0 \\
a_{158} = 0 \\
a_{159} = 0 \\
a_{160} = 0 \\
a_{161} = 0 \\
a_{162} = 0 \\
a_{163} = 0 \\
a_{164} = 0 \\
a_{165} = 0 \\
a_{166} = 0 \\
a_{167} = 0 \\
a_{168} = 0 \\
a_{169} = 0 \\
a_{170} = 0 \\
a_{171} = 0 \\
a_{172} = 0 \\
a_{173} = 0 \\
a_{174} = 0 \\
a_{175} = 0 \\
a_{176} = 0 \\
a_{177} = 0 \\
a_{178} = 0 \\
a_{179} = 0 \\
a_{180} = 0 \\
a_{181} = 0 \\
a_{182} = 0 \\
\end{bmatrix}
\]

The following performance index is assumed for the optimal state feedback control.
\[
J = \int_0^\infty \left\{ p_{11} z^2 + p_{12} \dot{z}^2 + p_{13} \ddot{z}^2 + p_{21} \phi^2 + p_{22} \dot{\phi}^2 + p_{23} \ddot{\phi}^2 + p_{31} \theta^2 + p_{32} \dot{\theta}^2 + p_{33} \ddot{\theta}^2 + p_{41} u_1^2 + p_{42} u_2^2 + q_{11} \dot{u}_1^2 + q_{12} u_1 u_2 + q_{21} u_2^2 + q_{22} u_2 u_4\right\} dt
\]

Namely for the ride comfort control, the quantities of the vertical acceleration, velocity and displacement of the sprung mass and the pitching and rolling angular accelerations, velocities and displacements of the sprung mass are induced for the target variables. Therefore, the optimal control forces are expressed as follows.
\[
u_i = -K_{ji} X \]
\[
K_{ji} = [K_{j1}, K_{j2}, K_{j3}, K_{j4}, K_{j5}, K_{j6}, K_{j7}, K_{j8}, K_{j9}, K_{j10}] \quad (j = f, r)
\]
Next, the parametric study for the weighting functions is carried out for obtaining the pole assignments of the regulators. In this design the dimensions of the car model are supposed as the same with the half car model in the previous section.

Figure 3-10 shows the root loci for the obtained pole assignments. As for the unsprung mass vibration, the trajectories of the root loci are changed remarkably according to the values of parameter.

In the case of (1) for the weightings of the acceleration of the sprung mass, the root locus is drawing the one half of the circle toward left hand side according to the change of the weighted value of $R$.

On the contrary, in the case of (2) for the weightings of the acceleration and the displacement of the sprung mass, the root locus is moving toward left upper side from the curved circle mentioned above.

Finally in the case (3) of weightings for the vehicle attitude, the root locus is drawing directly to left upper side.

Fig.3-10 Root loci for three vibration moods
From the result of the figure, the tendency of pole assignments is nearly same with previous section and it is shown that body modes are also controlled by this model. In the case (1), heave, pitch and roll frequencies are lowered and the damping factors are increased. On the contrary, heavy weightings for the acceleration terms mean the increase of the deviation of displacement. From the point of view of ride comfort, the flat ride component would be excessive by this condition. Therefore in this control method, case (2) is rather superior to case (1) and (3). Among these three modes, the weighting values are adjusted for getting a suitable combination.

Many numerical calculations are carried out for best control condition. The given example of the control gains for the case (2) is shown in Table 3-3. In this case, the weighting parameter is supposed for the acceleration and displacement of the sprung mass and the weighting for the roll angle is bigger than the others as shown in the table. As the result, the roll motion due to steering is fairly controlled in this condition as well as other heave and pitch motion. These considerations for the control methods are intended to introduce to the practical control systems.

<table>
<thead>
<tr>
<th>Performance Index</th>
<th>Control gain</th>
<th>Control Force</th>
<th>U1</th>
<th>U2</th>
<th>U3</th>
<th>U4</th>
</tr>
</thead>
<tbody>
<tr>
<td>P11 2x10^7</td>
<td>k11</td>
<td>-1.413649E+03</td>
<td>-1.413649E+03</td>
<td>-1.164660E+03</td>
<td>-1.164660E+03</td>
<td></td>
</tr>
<tr>
<td>P12 0</td>
<td>k12</td>
<td>-5.605706E+02</td>
<td>-5.605706E+02</td>
<td>-6.521051E+02</td>
<td>-6.521051E+02</td>
<td></td>
</tr>
<tr>
<td>P13 1x10^4</td>
<td>k13</td>
<td>-7.008839E+03</td>
<td>7.008839E+03</td>
<td>-6.541583E+03</td>
<td>6.541583E+03</td>
<td></td>
</tr>
<tr>
<td>P21 5x10^7</td>
<td>k14</td>
<td>-8.561318E+02</td>
<td>8.561318E+02</td>
<td>-7.990563E+02</td>
<td>7.990563E+02</td>
<td></td>
</tr>
<tr>
<td>P22 0</td>
<td>k15</td>
<td>5.135129E+01</td>
<td>5.135129E+01</td>
<td>1.511506E+02</td>
<td>1.511506E+02</td>
<td></td>
</tr>
<tr>
<td>P23 1x10^2</td>
<td>k16</td>
<td>2.430572E+02</td>
<td>2.430572E+02</td>
<td>-5.390766E+02</td>
<td>-5.390766E+02</td>
<td></td>
</tr>
<tr>
<td>P31 1x10^7</td>
<td>k17</td>
<td>2.135157E+00</td>
<td>4.299251E-01</td>
<td>1.082867E+00</td>
<td>-5.086825E-01</td>
<td></td>
</tr>
<tr>
<td>P32 0</td>
<td>k18</td>
<td>4.299251E-01</td>
<td>2.135157E+00</td>
<td>-5.086825E-01</td>
<td>1.082867E+00</td>
<td></td>
</tr>
<tr>
<td>P33 1x10^4</td>
<td>k19</td>
<td>1.160215E+00</td>
<td>-5.450169E-01</td>
<td>2.190757E+00</td>
<td>5.992066E-01</td>
<td></td>
</tr>
<tr>
<td>Q11 12 0.14</td>
<td>k20</td>
<td>-5.450169E-01</td>
<td>1.160215E+00</td>
<td>5.992066E-01</td>
<td>2.190757E+00</td>
<td></td>
</tr>
<tr>
<td>Q21 22 0.15</td>
<td>k21</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3-3 Example of control gain for case (2)
3.4. Vehicle Motion Control due to Steering Maneuver

3.4.1. Vehicle Motion Control due to Steering

The model for vehicle dynamics is shown in Figure 3-11. The equations of motion of the vehicle are expressed by the Euler's equations of motion. The Euler's equations of motion can be written in the form as follows.

\[
\begin{align*}
M(\dot{u}+wq-vr) &= F_x \\
M(\dot{v}+ur-wp) &= F_y \\
M(\dot{w}+vp-uq) &= F_z \\
\dot{h}x+qhz-rhy &= N_x \\
\dot{h}y+rhx-phi &= N_y \\
\dot{h}z+ph-y-qhx &= N_z
\end{align*}
\]  

(3-56)  

(3-57)

Fig. 3-11 Model of six degrees of freedom

where \((u, v, w), (p, q, r), (hx, hy, hz), (Fx, Fy, Fz)\) and \((Nx, Ny, Nz)\) are the scalar components of the linear velocity, the angular velocity, the angular momentum, the external forces and the external moments, along and about the vehicle body axes, respectively.

From the relation of the symmetric model, following two products of inertia are zero.

\[ l_{x,y} = l_{y,z} = 0 \]  

(3-58)

Then the Euler's equations for angular motions simplifying as follows.

\[
\begin{align*}
1_1\dot{r} - l_1\dot{p} + (l_1-l_2)pq-l_1pr &= N_x \\
1_2\dot{q} + (l_1-l_2)pr + l_2(q^2-r^2) &= N_y \\
1_3\dot{r} - l_3\dot{p} + (l_3-l_4)pq+l_4qr &= N_z
\end{align*}
\]  

(3-59)  

(3-60)  

(3-61)

where \(l_1, l_2, l_3\) are the moments of inertia about body axes, and \(l_{1z}\) is the product of inertia with respect to the \(x\) and \(z\) axes.
Treating the motion due to the steering behaviours in the steady running state of the vehicle, the motion model is supposed for sprung mass which is composed with lateral, vertical, yaw and roll motions, as shown in Figure 3-12.

The referential equations of motion are as follows.

(lateral motion)

\[
M \ddot{y} = \sum F_{yi} \quad \text{i=1, 2, 3, 4} (3-62)
\]

\[
\ddot{y} = \dot{v} + u \tau - g \phi \quad (3-63)
\]

(vertical motion)

\[
M_s \ddot{z} = M_s \ddot{g} + \sum S_i \quad \text{i=1, 2, 3, 4} (3-64)
\]

(yaw motion)

\[
I_y \ddot{\phi} = N_y \quad (3-65)
\]

\[
N_y = A_{r} F_{,r} + A_{r} F_{,r} \quad (3-66)
\]

\[
F_{,r} = F_{,r} + F_{,r} \quad (3-67)
\]

\[
F_{,r} = F_{,r} + F_{,r} \quad (3-68)
\]

(roll motion)

\[
I_z \ddot{\phi} = 2(M_s Z_s + M_s Z_s) \quad (3-69)
\]

\[
N_x = (z_s + H) F_{,s} + (z_s + H) F_{,s} + T_r(S_s - S_s) / 2 + T_r(S_s - S_s) / 2 \quad (3-70)
\]

(relative motion between sprung and unsprung mass)

\[
\dot{z}_s = \dot{w} - (-1)^i T, \phi / 2 \quad \text{i=1, 2, 3, 4} \quad \text{j=f, r} \quad (3-71)
\]

(generated force in the suspension)

\[
S_i = -k_i z_i - c_i \dot{z}_i - (-1)^i d_{ij} + S_{ij} \quad \text{i=1, 2, 3, 4} \quad \text{j=f, r} \quad (3-72)
\]

(reaction force by stabilizer)

\[
d_{s,i} = R_i (z_s - z_s) / T_r^2 \quad d_{s,r} = R_r (z_s - z_s) / T_r^2 \quad (3-73)
\]

(bearing load for each wheel)

\[
P_{r,1} = S_i - M_s g \quad \text{i=1, 2} \quad (3-74)
\]

\[
P_{s,1} = S_i - M_s g \quad \text{i=3, 4} \quad (3-75)
\]

(camber angle due to rolling)

\[
\phi_{1} = (-1)^i \phi_{r,0} + k \phi \quad \text{i=1, 2} \quad (3-76)
\]

\[
\phi_{1} = (-1)^i \phi_{r,0} + k \phi \quad \text{i=3, 4} \quad (3-77)
\]

(steering angle)

\[
\delta = \delta_{\*} (1 - e^{-\omega t}) \quad (3-78)
\]

60
(relation between side force and steering torque)

Figure 3-13 shows the steering model and the relations are shown as following equations.

gear torque:
\[ T_s = k_s (\delta_s - N_s \delta_s) \]  (3-79)

transfer torque:
\[ N_s T_s = k_{s1} (\delta_s - N_1 (\delta_{sw1} + \delta_{sw2}) / 2) \]  (3-80)

rotational angle for pinion:
\[ \delta_p = (N_s k_{s1} \delta_s + N_1 k_{s1} (\delta_{sw1} + \delta_{sw2}) / 2) / (N_s^2 k_{s1} + k_{s1}) \]  (3-81)

balance for the moment of front left wheel:
\[ M_1 = F_{s2} \]
\[ = -N_1 k_{s1} (\delta_s - N_1 \delta_{sw1}) / 2 \]  (3-82)

Fig. 3-13 Steering model

balance for the moment of front right wheel:
\[ M_2 = F_{s2} \]
\[ = -N_1 k_{s1} (\delta_s - N_1 \delta_{sw2}) / 2 \]  (3-83)

From the equation (3-81), (3-82) and (3-83), the following equation is obtained.

(balance for front-left steering torque and lateral force)
\[ N_1^2 (N_s^2 k_{s1} + k_{s1} / 2) \delta_{sw1} - N_1^2 k_{s1} / 2 \cdot \delta_{sw1} - 2 \delta (1 + N_s^2 k_{s1} / k_{s1}) \cdot F_{s1} - N_1 N_s k_{s1} \delta_{sw} = 0 \]  (3-84)

(balance for front-right steering torque and lateral force)
\[ -N_1^2 k_{s1} / 2 \cdot \delta_{sw1} + N_1^2 (N_s^2 k_{s1} + k_{s1} / 2) \delta_{sw2} - 2 \delta (1 + N_s^2 k_{s1} / k_{s1}) \cdot F_{s1} - N_1 N_s k_{s1} \delta_{sw} = 0 \]  (3-85)

(relation of real steering angle)
\[ \beta_{ri} = \delta_{swi} + \varepsilon_i \]
\[ \beta_{ri} = + \varepsilon_i \]
\[ i = 1, 2 \]  (3-86)
\[ i = 3, 4 \]  (3-87)
(lateral force for front and rear wheels)

\[ F_{r1} = -C_{r1}F_{r1}(\tan \frac{-v+A_{r1}r+(z+H)}{u} \phi - \beta_{r1}) - C_{e1}F_{r1} \phi, \quad (3-88) \]

\[ F_{r2} = -C_{r2}F_{r2}(\tan \frac{-v-A_{r2}r+(z+H)}{u} \phi - \beta_{r2}) - C_{e2}F_{r2} \phi. \quad (3-89) \]

From the equations (3-76), (3-86) and (3-88), the relation for front steering torque and lateral force is obtained.

(front left wheel)

\[ F_{r1}C_{r1} \delta_{r1} - F_{r1} \] \(-C_{r1}F_{r1}(\tan \frac{-v+A_{r1}r+(z+H)}{u} \phi + \varepsilon_{r1}) - C_{e1}F_{r1} \phi = 0 \quad (3-90) \]

(front right wheel)

\[ F_{r2}C_{r2} \delta_{r2} - F_{r2} \] \(-C_{r2}F_{r2}(\tan \frac{-v-A_{r2}r+(z+H)}{u} \phi - \varepsilon_{r2}) - C_{e2}F_{r2} \phi = 0 \quad (3-91) \]

(cornering coefficient)

\[ C_{r1} = \frac{C_{m1}}{\sqrt{300/F_{r1}}} \quad (3-92) \]

From the equation (3-84), (3-85) and (3-90), the equations of balance are obtained as follows.

\[
\begin{bmatrix}
N_{r1}^2(N_s^2k_{sr}^e+k_{sr}/2) - N_{r1}^2k_{sr}/2 - 2(1+N_s^2k_{sr}^e/k_{sr}) & 0 & \delta_{r1} \\
-N_{r1}^2k_{sr}/2 + N_{r1}^2(N_s^2k_{sr}^e+k_{sr}/2) & 0 & -2(1+N_s^2k_{sr}^e/k_{sr}) & \delta_{r2} \\
F_{r1}C_{r1} & 0 & -1 & 0 & F_{r1} \\
0 & F_{r2}C_{r2} & 0 & -1 & F_{r2} \\
\end{bmatrix}
\]

\[
\begin{bmatrix}
N_sN_{r1}k^e_{sr} \delta_{r1} \\
N_sN_{r1}k^e_{sr} \delta_{r2} \\
C_{r1}F_{r1}(\tan \frac{-v+A_{r1}r+(z+H)}{u} \phi + \varepsilon_{r1}) - C_{e1}F_{r1} \phi, \\
C_{r1}F_{r2}(\tan \frac{-v-A_{r2}r+(z+H)}{u} \phi - \varepsilon_{r2}) - C_{e1}F_{r2} \phi. \quad (3-93) \\
\end{bmatrix}
\]
Compensating the delay of control actuator, the feedforward control system corresponding to the amount of steering is introduced. The estimating model expected for steady state of rolling is supossed for instantaneous steering behaviour.

For the equations of motion in steady state for lateral, yaw and roll directions, the following equations are induced from (3-62), (3-65) and (3-69).

\[ M(-ur+g\phi) = \Sigma F_{z,i} \quad i=1,2,3,4 \quad (3-94) \]

\[ A_i(F_{z,i}-F_{z,i}) = 0 \quad (3-95) \]

\[ (z_r+H)(F_{z,i}+F_{z,i})+(z_r+H)(F_{z,i}+F_{z,i})+T_r(S_1-S_2)/2+T_r(S_3-S_4)/2=0 \quad (3-96) \]

Here the lateral force equations (3-88) and (3-89) are simplified with neglecting the transient amount as follows.

\[ F_{z,i} = -C_{r,Fi} \frac{v+A_{r,i}}{u_0} + C_{r,Fi} \phi_{sw} \quad (3-97) \]

\[ F_{z,i} = -C_{r,Fi} \frac{v-A_{r,i}}{u_0} \quad (3-98) \]

In the feedforward controlled stage, the estimated signals corresponding to a steady state of the vehicle according to the actual vehicle speed \( u_0 \) and steering angle \( \delta_{sw} \) are obtained from these equations. For the steady state case, the state estimated simultaneous equations are shown as follows.

\[ M\dot{u}_0+2C_{z,i} \frac{v+A_{r,i}}{u_0} + 2C_{z,i} \frac{v-A_{r,i}}{u_0} = 2C_{z,i} \phi \quad (3-99) \]

\[ 2A_{Cr,i} \frac{v+A_{r,i}}{u_0} + 2A_{Cr,i} \frac{v-A_{r,i}}{u_0} = 2A_{Cr,i} \phi \quad (3-100) \]

\[ 2Z_{Cr,i} \frac{v+A_{r,i}}{u_0} + 2Z_{Cr,i} \frac{v-A_{r,i}}{u_0} + K \phi \phi_{\infty} = 2Z_{Cr,i} \phi \quad (3-101) \]

where,

\[ C_{r,Fi} = C_{r,Fi} \]

\[ K = \frac{T_r}{2} \left( k_r T_r + 2 \frac{R_r}{T_r} \right) + \frac{T_r}{2} \left( k_r T_r + 2 \frac{R_r}{T_r} \right) \quad (3-102) \]

\[ \delta_{\omega} = \delta_{\omega} / N_{\omega} \quad (3-103) \]
From these equations the stable roll angle $\phi_\infty$ is given and the feedforward signal is also given in the functions of the vehicle speed and the steering angle.

$$
\begin{bmatrix}
  a_{11} & a_{12} & 0 \\
  a_{21} & a_{22} & 0 \\
  a_{31} & a_{32} & a_{33}
\end{bmatrix}
\begin{bmatrix}
  v \\
  r \\
  \phi_\infty
\end{bmatrix}
= 
\begin{bmatrix}
  b_1 \\
  b_2 \\
  b_3
\end{bmatrix}
$$

(3-104)

$$
\phi_\infty = \frac{(a_{11}a_{22}b_3 + a_{12}b_3 - a_{21}a_{32}b_1 - a_{22}a_{33}b_1) - b_1a_{31}a_{32} - b_3a_{31}a_{32}}{a_{11}a_{22}a_{33} - a_{12}a_{31}a_{32}}
$$

(3-105)

$$
\dot{\phi}_\infty = G_s(s) \phi_\infty
$$

(3-106)

$$
G_s(s) = \frac{kT_s}{1 + Ts}
$$

(3-107)

![Roll estimating model](image)

Fig. 3-14 Active attitude control system

The signal flow chart of active attitude control system is shown in Figure 3-14. The map of the predicted rolling angle is obtained as shown in the figure. The feedforward control signal is modified by the filter $G_s(s)$ to improve the dynamic response characteristics at transient state.

In the feedback control, the roll angle calculated from each relative displacement of the wheel is used here. This control law is called control 1.
Next, compensating the improvement of control response in the high speed running, another control law is offered. In such case, the emerged lateral acceleration is continued to some extent by the effect of the balance of tire side forces due to vehicle roll motion. Then, from the equations (3-69) and (3-70) for roll motion, the equation of balance of roll moment in the stationary state is obtained as follows:

\[ 0 = 2(M_r, Z_r + M_z, Z_r)(\ddot{Y}) + T, (S_i - S_i)/2 + T, (S_i - S_i)/2 + h, M(\ddot{Y}) \]  

where, \( h = Z_r + H, Z_r + H \)  

(3-108)

(3-109)

Considering the balance for both right and left wheels, the relation between lateral motion and tire side force is expressed.

\[ T, (S_i - S_i)/2 + T, (S_i - S_i)/2 = -(2M_r, Z_r + h, M_r)(\ddot{Y}) - (2M_z, Z_r + h, M_r)(\ddot{Y}) \]  

(3-110)

According to above equation,

\[ (S_i - S_i) = -2(2M_r, Z_r + h, M_r)(\ddot{Y})/T, \]  

(3-111)

\[ (S_i - S_i) = -2(2M_z, Z_r + h, M_r)(\ddot{Y})/T, \]  

(3-112)

Here, supposing the target additional pressure \( \Delta p_{cr} \) as deviate pressure against cylinder pressure \( p_{si} \), the next equation is induced for every wheel.

\[ \Delta p_{cr} = (S_i - S_i)/(2A_r) - p_{si}, \]  

(3-113)

\[ = k_{x} \ddot{Y} - p_{si}, \]

Accordingly, newly compensating feedback loop is added for the second control law in the case of higher speed conditions. In other feedback control parts, the feedback law uses the roll rate and roll angle calculated from each relative displacement of the wheel.

Finally the two kinds of control methods for active force generator are expressed as follows. Here the objective roll angle is zero.

**Control 1:** \( \dot{\phi}_\infty = \phi_r - k_x \phi_\infty - k_{x}\phi \)  

(3-114)

**Control 2:** \( \dot{\phi}_\infty = \phi_r - k_{x1}(k_{x2} \ddot{Y} - p_{si}) - k_{x3} \phi - k_{x4} \phi \)  

(3-115)

where, \( \phi = 0 \)  

(3-116)
3.4.2. Vehicle Motion Control using Driver Maneuver Model

The purpose of the active control for the steering system is to reduce the unstable motion by a side wind or to improve the handling response of the vehicle. The vehicle and driver model used in this section is shown in Figure 3-15. The vehicle model is two degrees of freedom as follows.

(lateral movement)

\[ M \ddot{y} = F_x + F_y \]  

(yaw motion)

\[ I \dot{\phi} = A \dot{\theta} - A \dot{\phi}, \]  

(cornering force)

\[ F_x = -C_r \frac{y - u_s \phi + A_s \dot{\phi}}{u_s} - \delta - u_\delta \]  

\[ F_y = -C_r \frac{y - u_s \phi - A_s \dot{\phi}}{u_s} \]  

The driver's steering operation model in the following expression is assumed to reduce the deviation of the course from the front view point. And the maneuver model consists of dead time of operation, first order time lag and first order lead elements.

The dimensions and supposed values are shown as follows. In the driver's model, parameters are obtained from the experimental results using driving simulator.26)
\[
\delta_i = K_x \frac{T_h s + 1}{T_h s + 1} \exp(-\tau s)(y + L_i \phi) \quad (3-121)
\]

\[
K_x = -0.00493 \text{[rad/m]}, T_h = 0.5 \text{[s]}, T_h = 0.1 \text{[s]}, \tau = 0.6 \text{[s]}
\]

The dead time element may be replaced using Pade's first order approximation methods as follows.

\[
\exp(-\tau s) = \frac{1-(\tau/2)s}{1+(\tau/2)s} \quad (3-122)
\]

Using these equations, the models for the vehicle and the driver are expressed in a state space expression and the optimal control is obtained as follows.

\[
\dot{x} = Ax + Bu \delta \quad (3-123)
\]

where,

\[
x = [y, \dot{y}, \phi, \dot{\phi}, \delta_i, \dot{\delta}, \dot{\delta}_i]^T \quad (3-124)
\]

\[
A = \begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 \\
0 & a_1 & a_2 & a_3 & a_4 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 \\
0 & a_5 & a_6 & a_7 & a_8 & 0 \\
a_9 & a_{10} & a_{11} & a_{12} & a_{13} & a_{14}
\end{bmatrix}
\]

\[
B = \begin{bmatrix}
b_1 \\
b_2 \\
b_3
\end{bmatrix}
\]

\[
a_1 = -2(c_i + c_r)/(M \omega_0) \quad a_2 = 2(c_i + c_r)/M \\
a_3 = -2(A_i c_i - A_r c_r)/(M \omega_0) \quad a_4 = 2c_i/M \\
a_5 = -2(A_i c_i - A_r c_r)/(l_x \omega_0) \quad a_6 = 2(A_i c_i - A_r c_r)/l_x \\
a_7 = -2(A_i c_i + A_r c_r)/(l_x \omega_0) \quad a_8 = 2A_i c_i/l_x \\
a_9 = -K_x/(T_h \tau / 2) \\
a_{10} = -K_x(K_h - \tau / 2) + 2K_x K_h (\tau / 2) ((c_i + c_r)/(M \omega_0)) \\
\quad + (A_i c_i - A_r c_r) l_x/(l_x \omega_0)))/(T_h \tau / 2) \\
a_{11} = -K_x L_x - 2K_x K_h (\tau / 2) ((c_i + c_r)/(M \omega_0 + (A_i c_i - A_r c_r) l_x/l_x)))/(T_h \tau / 2) \\
a_{12} = -K_x(K_h - \tau / 2) l_x + 2K_x K_h (\tau / 2) ((A_i c_i - A_r c_r)/(M \omega_0)) \\
\quad + (A_i c_i + A_r c_r) l_x/(l_x \omega_0)))/(T_h \tau / 2) \\
a_{13} = -(1 - 2K_x K_h (\tau / 2)(c_i/(M + A_i c_i) l_x/l_x)))/(T_h \tau / 2) \\
a_{14} = -(T_h + \tau / 2)/(T_h \tau / 2) \\
b_1 = 2c_i/M \quad b_2 = 2A_i c_i/M \\
b_3 = 2K_x(K_h(c_i/(M + A_i c_i) l_x/l_x))/T_h
\]

\[
u_0 = f_k x \quad (3-125)
\]

where,

\[
f_k = [f_1, f_2, f_3, f_4, f_5, f_6] \quad (3-126)
\]
Here for the verification of the vehicle model, the comparison with real running behaviours and simulated results are illustrated as following.

Figure 3-16 shows the comparison of simulation results and experimental results when the driver changes the lane at the speed of 100 km/h. Time histories of yaw rate and lateral acceleration are much similar to each other. From the figure the model is considered to be almost similar to the experimental results.

Fig. 3-16 Comparison in lane change
(\(\omega = 100\text{km/h}\), (——) for simulation and (-----) for experiment)

To calculate the control gains using the optimum regulator methods, the quadratic form performance index is composed for satisfying the aforementioned objects. At first stage of the designing, the performance index function is determined as follows.

\[
J = \int_{0}^{\infty} 
\left( g_{1}(y+L_{x}\dot{\psi})^2 + g_{2}(\dot{\phi})^2 + g_{3}(k_{x}\delta - \dot{\psi})^2 + g_{4}u_{\delta}^2 \right) \, dt \quad (3-127)
\]

In this function, the sum of four terms in right hand side should be minimized. Each term related with the deviation of the course from the driver's front view point, the deviation of the vehicle yaw rate, the phase lag of the yaw rate against the steering wheel angle and the active amount of controlled
steering angle, respectively.

The control gains are calculated by changing the weighting coefficients of these terms and coefficient $k_C$. The control effects of the active steering system were simulated by computer using the supposed set of the gains. These computer simulations show that the stability of the vehicle in the straight running increases with the growth of weighting parameter $g_1$ and $g_2$. On the contrary, as the weighting for the stability is heavy, the vehicle response in lane changing is not enough.

Furthermore, for achieving the response of vehicle motion, the weighting parameter $g_3$ and the coefficient $k_c$ should be taken into accounts. In the meaning of the third term, the response of vehicle is $k_C$ times of yaw rate in the case of steering and is controlled within minimal yaw rate in the case of straight running. To obtain the optimal condition, several combinations are selected for simultaneous improvement in stability and responsibility by numerical calculations.

After all, the following performance index function is adopted for the purposes of the control.

$$ J = \int_0^\infty \left( g_3 (k_e \delta - \dot{\phi})^2 + g_4 u \dot{\delta} \right) \, dt \quad (3-128) $$

The given examples of the control gains are shown in Table 3-4(a) according to the coefficients of the performance index. In this case the condition of vehicle speed is 100 km/h and the assumption of road surface friction is ordinary road. Under the condition that the control input is limited within 1 degree, the suitable condition is selected as case (b) in the table.

Next, in the case of (b), the control gains are calculated for the various running conditions. The given examples of the control gains are shown in Table 3-4(b). There are two cases for vehicle speed under the condition of high and low friction road. As shown in the table, the difference of tire cornering power between high and low friction road shows the different control conditions.
Table 3-4(a) Example of the control gain (100km/h, μ=0.8)

<table>
<thead>
<tr>
<th>Coefficient</th>
<th>f₁(y)</th>
<th>f₂(y)</th>
<th>Control gain</th>
<th>f₃(φ)</th>
<th>f₄(φ)</th>
<th>f₅(δ₁)</th>
<th>f₆(δ₁)</th>
</tr>
</thead>
<tbody>
<tr>
<td>g₁</td>
<td>kᵣ</td>
<td>g₄</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(a)</td>
<td>1.0</td>
<td>5.0</td>
<td>20.0</td>
<td>-0.0003</td>
<td>-0.0005</td>
<td>0.1261</td>
<td>-0.1329</td>
</tr>
<tr>
<td>(b)</td>
<td>1.0</td>
<td>8.0</td>
<td>20.0</td>
<td>-0.0005</td>
<td>-0.0049</td>
<td>0.1159</td>
<td>-0.1317</td>
</tr>
<tr>
<td>(c)</td>
<td>1.0</td>
<td>12.0</td>
<td>20.0</td>
<td>-0.0019</td>
<td>-0.0057</td>
<td>0.0779</td>
<td>-0.1943</td>
</tr>
<tr>
<td>(d)</td>
<td>1.0</td>
<td>8.0</td>
<td>4.0</td>
<td>-0.0005</td>
<td>-0.0085</td>
<td>0.2158</td>
<td>-0.3944</td>
</tr>
<tr>
<td>(e)</td>
<td>1.0</td>
<td>8.0</td>
<td>80.0</td>
<td>-0.0003</td>
<td>-0.0021</td>
<td>0.0465</td>
<td>-0.0423</td>
</tr>
</tbody>
</table>

Table 3-4(b) Example of the control gain (in the case (b))

<table>
<thead>
<tr>
<th>Condition Speed</th>
<th>Road</th>
<th>μ</th>
<th>f₁(y)</th>
<th>f₂(φ)</th>
<th>Control gain</th>
<th>f₃(φ)</th>
<th>f₄(φ)</th>
<th>f₅(δ₁)</th>
<th>f₆(δ₁)</th>
</tr>
</thead>
<tbody>
<tr>
<td>60 km/h</td>
<td>0.8</td>
<td>-0.0008</td>
<td>-0.0051</td>
<td>0.0520</td>
<td>-0.1045</td>
<td>0.8779</td>
<td>0.0292</td>
<td></td>
<td></td>
</tr>
<tr>
<td>100 km/h</td>
<td>0.8</td>
<td>-0.0005</td>
<td>-0.0049</td>
<td>0.1159</td>
<td>-0.1317</td>
<td>0.9417</td>
<td>0.0338</td>
<td></td>
<td></td>
</tr>
<tr>
<td>60 km/h</td>
<td>0.2</td>
<td>-0.0024</td>
<td>-0.0073</td>
<td>0.0176</td>
<td>-0.1110</td>
<td>0.5661</td>
<td>0.0329</td>
<td></td>
<td></td>
</tr>
<tr>
<td>100 km/h</td>
<td>0.2</td>
<td>-0.0024</td>
<td>-0.0053</td>
<td>0.0440</td>
<td>-0.1388</td>
<td>0.5644</td>
<td>0.0342</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The predictive effects of the control in the case (b) are shown as comparisons with non-controlled vehicle in Table 3-5 and Figure 3-17(a) and (b).

The table shows the eigenvalues of both controlled and non-controlled vehicle as the results of the calculations. There are two poles and each of them has meaning. The first pole assignment would be indicated as the stability of the control and the second pole would be as responsibility of vehicle motion.

According to the vehicle velocity and surface conditions, the pole assignments are changed as shown in the both figures by the symbol in the table. The designed control shows that both stability and responsibility are fairly improved in the both friction roads. They are all towards left upper side of these figures. In the case of 60 km/h, the responsibility of the vehicle should be mainly improved. In the high speed, the responsibility should be improved as well as the stability and controllability.

From these considerations, the control scheme in the real running situation would be adjusted for obtaining the suitable conditions for optimal control.
Table 3-5 Example of eigenvalues

<table>
<thead>
<tr>
<th>Speed</th>
<th>50km/h</th>
<th></th>
<th>100km/h</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Road μ</td>
<td>0.8</td>
<td>0.2</td>
<td>0.8</td>
<td>0.2</td>
</tr>
<tr>
<td>Control</td>
<td>control</td>
<td>No</td>
<td>control</td>
<td>No</td>
</tr>
<tr>
<td>Symbol</td>
<td>●</td>
<td>○</td>
<td>△</td>
<td>△</td>
</tr>
<tr>
<td>Root λ1</td>
<td>-0.39</td>
<td>-0.57</td>
<td>-0.17</td>
<td>-0.33</td>
</tr>
<tr>
<td></td>
<td>±j0.59</td>
<td>±j0.58</td>
<td>±j0.48</td>
<td>±j0.56</td>
</tr>
<tr>
<td>Root λ2</td>
<td>-3.41</td>
<td>-3.32</td>
<td>-1.53</td>
<td>-1.90</td>
</tr>
<tr>
<td></td>
<td>±j3.24</td>
<td>±j3.83</td>
<td>±j2.76</td>
<td>±j3.04</td>
</tr>
<tr>
<td>Root λ4</td>
<td>-7.26</td>
<td>-7.56</td>
<td>-2.79</td>
<td>-2.55</td>
</tr>
</tbody>
</table>

Fig.3-17 Root loci for vehicle motion control
3.5. Summary

In this chapter, the theoretical approaches for designing the active control are described for realizing the desired characteristics of vehicle vibration.

First in section 3.2., as for the quarter car model, the linear control theory is applied to vibration control for two degrees of freedom model. With considering the response of the used actuator, the feedback control system is constructed for single suspension. As the results of the study, the linear quadratic regulator was used for the performance index consisted of sprung mass acceleration and displacement, and the control effects of active control for suspension were confirmed. Moreover noticing the frequency characteristics for vehicle vibration, the frequency shaped control was designed for better ride comfort as well as stable vehicle attitude. These control effects were also verified by the simulation results. Moreover the robustness of the control methods was shown from the circular conditions for the control just as the same as the ordinary optimum regulator.

Next in section 3.3., the vibration control of vehicle body in the low frequency regions was designed for the half car model. First, the feedback controls of the front and rear suspensions are constructed for the pitching motion in the case of crossing across the road irregularities. In further considerations, as for other motions of the vehicle like rolling and diagonal movements, the vibration controls for four wheels model were designed. As the results of the studies, the effectiveness for the active control was confirmed by the simulations for the making the minimum change in the vehicle attitude as the low frequency vibration of the vehicle body.

Finally in section 3.4., the motion control due to steering maneuvers was designed. Considering the response of the control devices, the feedforward control was adopted for the ordinary feedback control systems and the effect of active feedforward control was introduced to the attitude control from the beginning of rolling phenomena at the vehicle steering. Moreover the actively
steering control was also introduced for the motion control. It was constructed from the point of view of driver-vehicle closed loop. According to the consideration including the driver maneuver model, the designed control system was verified to be stable in the steering performance of vehicle.

As the future subjects for the fields of the control, the way of setting the optimal performance index for human sensitivity of vibration and motion must be take into accounts, while more effective and practical methods for integrated control are also expected.

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CHAPTER 4 STUDY OF SEMI-ACTIVE VIBRATION CONTROL

4.1. Introduction

In the methods of vibration control for vehicle suspension, aiming improvement for riding comfort, there are two ways such as passive and active control: the former is made from ordinary spring and damper combined systems\(^1\)-\(^5\) and the latter is the way of supplying energy positively from external sources.

Taking the practical use for these methods into considerations, a simple system with few control powers would be expected and the methods by semi-active type elements have been proposed\(^6\)-\(^11\) for few energy dissipation. These are pursuing to realize the variable characteristics of elements attached to ordinary passive elements due to the change of state variables\(^12\)-\(^25\). There is, for example, vibration adjustment apparatus with variable devises for spring coefficients\(^26\)-\(^27\).

By the developments of micro-electronics, many useful software servo technologies have been introduced to these fields, by which the control methods are easily changed according to the change of structural dynamics. There are many applications of control methods based on the modern control theories\(^28\)-\(^36\).

In this chapter, the systems of semi-active control with low energy dissipation, which is simply constructed and excellent in the cost performance, are developed and applied to the control system of this study. Hereunder the technique to make suspension force follow the optimal force by using the active vibration control theory described in section 3.2. is examined corresponding to various test conditions.

In this study, the linear quadratic optimal regulator is not used to control the system actively but to determine the target function for semi-active
control of the system. Moreover, the vibration control system is constructed by continuously control of suspension damping force with successive control of damping valve opening corresponding to the behavior of suspension system. It is different from the conventional damping absorber, which is controlled by on-off valve opening. Using this system it is expected that the control energy is much saved for suspension control.

In section 4.2., the system configuration of the semi-active control system and the newly developed damping force control valve are described.

In section 4.3., the target functions to the system with the semi-active vibration control are calculated and the characteristics of the force control valve are examined, and then the effect of the continuously damping control is predicted. From these considerations, the optimizations of valve opening characteristics and control system are analyzed.

Furthermore in section 4.4., the experimental analysis concerning the optimal vibration control is carried out with the model apparatus of two degrees of freedom, which mounts the system designed in above sections. The performance and the effect of control system are verified from these experimental results.

4.2. System Configuration of Semi-active Vibration Control

4.2.1. Continuously Controlled Damper System

Figure 4-1 shows the construction of damping force control system in this study. In this system, control equipments in four wheels are controlled independently. As shown between sprung and unsprung masses in the right upper side of the figure, the continuously controlled damping valve is located between cylinder actuator and high pressurized gas spring. Among the controller in the figure, the state variables of the vibration system are instantaneously calculated from the signals of acceleration and stroke sensors. The target control force is also calculated at every moment here. The control valve is driven to
generate damping force to follow this target control force. That is, the damping force control is adjusted by valve opening according to the state changes.

Next, the following mechanism of damping control to the target control force $u$ is shown in the block diagram in Fig.4-1. Because the damping force is generated as the product of oil flow rate changes and oil flow resistance by continuously controlling of the valve opening, the valve opening is controlled to make the oil flow squeezing to follow the optimal force. However, the valve opening is adjusted according to the direction of the difference between

![Diagram](image)

**Fig.4-1 Semi-active vibration control system**

<table>
<thead>
<tr>
<th>Table 4-1 Logic for valve control</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{y}$</td>
</tr>
<tr>
<td>$u$</td>
</tr>
<tr>
<td>$\varepsilon = u - f$</td>
</tr>
<tr>
<td>$\varepsilon \dot{y}$</td>
</tr>
<tr>
<td>valve opening</td>
</tr>
</tbody>
</table>

(a) compressed (b) stretched

![Diagram](image)

**Fig.4-2 Oilflow and valve openings**
the generated damping force and the target control force as well as the direction of oil flow, that is the relative velocity $\dot{y}$. The following signal processings are carried out along the block diagram shown in the figure. To decrease the difference $\varepsilon$, which is that control force $u$ minus the damping force $f_c$, valve opening is controlled. The signal $\varepsilon \dot{y}$ is used as a signal by which the sign of difference $\varepsilon$ and oil flow directions are judged.

Table 4-1 shows the relation between the sign of these signals and valve opening. Here, as shown by part (a) and (b) of the table 4-1, when the sign of relative velocity $\dot{y}$ and target control force $u$ is different, the damping force should be decreased immediately, and valve opening is controlled to open the orifice completely for reducing the resistance of oil flow as shown in the schematic graphs of Figure 4-2(a) and (b).

Next, the composition of the continuous damping force control valve used in this study is shown in Figure 4-3(a). The valve which generates the damping force consists of orifice which is made by the sleeve and the spool as shown in the figure. The position of the spool is detected with non-contact displacement detector installed at the end of the spool and using the detected signal the spool is driven by direct moving actuator. Figure 4-3(b) shows the opening of orifice in the developed section diagram of the sleeve. The sectional area change of the squeezing part to the valve opening is divided into two parts, one is linear part A and another is nonlinear part B.

![Diagam of Drive actuator, Sleeve, Displacement detector, and Spool with A and B sections marked.](image)
The necessary valve opening characteristics is obtained by using such shaped parts. To control the damping force, a high response and a broadband performance are required in the control. Figure 4-4 shows the relation between relative velocity and the damping force changes in the case of constant valve opening. A nonlinear characteristic can be seen for oil flow rate according to the various flow coefficients due to the orifice part shape. The gain of the controlled force due to the velocity is high in the small valve opening. The valve opening characteristics is one of the important items of designing, which governs the controllability of damper. It is also predicted that the high gain performance in this region is one of the causes of unstable valve motion and vibration.

Next, the frequency characteristic of valve response is shown in Figure 4-5. The ratio of real valve opening due to the command signal is shown in vertical axis and the frequency in the horizontal axis. The response of this system is much influence on this performance concerning to the effect of controller's sampling period of control as well as depending on the valve form and the driving actuator characteristics. From the figure the response of this valve is almost flat below 10Hz and shows first order delay characteristics which cutoff frequency is about 30 Hz. As the control performance of this system is expected in the frequency regions below unsprung mass resonance, it is enough response for this valve to control damping of vibration.
4.2.2. Analysis of Controlled Damper System

Assuming one wheel suspension system of vehicle, the continuous damping force control system is modeled by two degrees of freedom vibration system. The operation analysis of the control valve is carried out as follows. The model is constructed by piston, cylinder, control valve parts which consist of sleeve and spool driven by actuator and gas spring which maintains the back-pressure. Moreover, there is controller parts where valve opening of control valve is controlled according to the change of the state variable.

Equations of each part are shown as follows by using the notations in Figure 4-1 and 4-3.

(oil flow in cylinder)
\[ Q_1 = A_2 (x_1 - x_2) + V_x P / K_a \]  \hspace{1cm} (4-1)

(control valve part)
\[ m \ddot{x}_a = -c \dot{x}_a - k x_a + f_s \]  \hspace{1cm} (4-2)
\[ f_s = -p_L Q_1 - p Q_1 v \cos \theta \]  \hspace{1cm} (4-3)

where,
\[ Q_1 = c_s B_s x_s \sqrt{2 p / \rho} \]  \hspace{1cm} (4-4)
\[ v = Q_1 / (c_s B_s x_s) \]  \hspace{1cm} (4-5)

(gas spring part)
\[ V = -Q_1 \]  \hspace{1cm} (4-6)
\[ P_s = -\kappa P_0 V_s / V_0 \]  \hspace{1cm} (4-7)

(controller part)
\[ u = F(y, \dot{y}, x_2, \dot{x}_2, f_s) \]  \hspace{1cm} (4-8)

Where,
\[ y = x_1 - x_2 \]  \hspace{1cm} (4-9)
\[ \dot{x}_s = -x_s / T + k' x_s / T \]  \hspace{1cm} (4-10)
In the control valve part, the self induced vibration may be generated; which is caused by the fluid power fluctuation on the spool, the driving stiffness of spool actuator and the pressure changes of high pressure side. The frequency of this excitation movement is governed by the next equation.

\[ f_o = \frac{1}{2 \pi} \sqrt{\frac{k_s}{m_s}} \]  \hspace{1cm} (4-11)

where \( k_s \) is the equivalent stiffness for spool support and \( m_s \) is the mass of the driven parts.

The self induced vibration is depended on the equivalent viscosity coefficient of the squeezing part of the oil, and the vibration characteristics are closely related to the valve opening characteristic.

On the other hand, the damping force generated in this vibration system is given by the next expression from the relations between the cross sectional area and the pressure of the flow because of conservation of flow in squeezing.

\[ f_o = A_o \cdot P_o = A_o \cdot P \]  \hspace{1cm} (4-12)

where \( A_o = B \cdot x \).  \hspace{1cm} (4-13)

As the pressure change is caused by flow change due to vibration turbulence, and the damping force is generated by squeezing part. After all, the generated damping force is shown by the next equation with an equivalent sectional area of orifice \( A_c \) and the turbulence velocity \( \dot{y} \).

\[ f_o = \frac{A_c}{A_o} \rho \cdot 2 \cdot (A_c / A_o)^2 \cdot (\dot{y} / A_o)^2 \]  \hspace{1cm} (4-14)

The damping force, therefore, would be more unstable when the turbulence velocity is larger and the equivalent sectional area is smaller, that is, the gain of controlled valve opening is higher. In the practical control system, other factors should be accounted for the response; these are linearity of the system, time-lag due to the length of oil tube and nonlinearity of gas spring.
4.3. Analysis of Semi-Active Control

4.3.1. Prediction of Control Effect

Hereunder the method of full-active vibration control is discussed for obtaining the target force of semi-active damping control system. The effect of full-active control is also predicted. The simulation is carried out to obtain the time histories by Runge-Kutta methods based on the two degrees of freedom model expressed by (3-1) and (3-2).

The input displacement was assumed for a normal road and to be included colored noise due to road roughness $A_z$, vehicle speed $v$ and the spatial frequency $f_{g0}$. That is, the following road disturbance equation was assumed.

$$x_i = \left( A_i / (1 + T_i \cdot s) \right) \cdot w$$

(4-15)

where

$$T_i = 1 / (2 \pi f_{g0} \cdot v)$$

$w$: white noise (normal random number)

The input waveform and frequency distribution of road in the calculation are shown in Figure 4-6(a) and (b).

![Fig. 4-6 Road input for predictive calculation](image_url)
The effects of the active control in random input are examined by the numerical calculations. In Figure 4-7(a), (b) and (c), the control effects of each optimal regulator are described. The results are obtained using the control gains shown in the paragraph 3.2.1.

Acceleration \( \ddot{x} \)
of sprung mass
Non-controlled case
Displacement \( x \)
of sprung mass

\[
\begin{align*}
\text{Acceleration} & \quad \ddot{x} \\
of {\text{sprung mass}} & \\
\text{Non-controlled case} & \\
\text{Displacement} & \quad x \\
of {\text{sprung mass}} & \\
\end{align*}
\]

Fig. 4-7 Predictive effects of LQ control
In each figure, the example of the time histories for the most effective condition is shown as comparison with the non-controlled case. The response characteristics of the acceleration of sprung mass due to input displacement are shown in the bottom of the figures. In these figures, the effects of the weighting parameter are described with the notice of arrow. The effects of the control are verified with the predicted results mentioned in Fig 3-2.

Next, the effects of the frequency weighted optimal regulator are also described in the Figure 4-8(a) and (b). The frequency response characteristics of the acceleration of sprung mass due to input displacement and the displacement of sprung mass due to active force are shown respectively.

In these figures, broken line (LQ) indicates without frequency weighting, chained line (LQ1) with frequency weighting for sprung mass acceleration and rigid line (LQ3) with frequency weighting for both sprung mass acceleration and displacement.

From these response characteristics, the control performance in the case of LQ3 is the most remarkable effect in the vibration control and is also understood to improve in the frequency region at which the effect of vibration control is obtained using the weighting filter. After obtaining these ideal control forces, the damping force control system is constructed to follow as shown in the next section.

Fig. 4-8 Predictive effects of weighted LQ control
4.3.2. Optimization of Damper Valve Characteristics

The control system shown in the previous section was designed by the simulation of the operating characteristics of the control valve. To analyze the valve opening characteristic of the continuous damping force control, various valve shapes were assumed and the effect of opening characteristics and the suitable sampling period for control were examined by the simulation. The examples are shown in Figure 4-9 as valve A for linear characteristics of the sectional area changes due to valve movement and valve B for non-linear one. The effect of variable sectional area on the operating characteristics of control valve was examined.

Figure 4-10(a) and (b) are the simulation results for the valve opening control and damping force waveform at that time in the cases of assuming valve characteristic A and B.

An unstable vibration is occurred in the case of valve A. On the other hand, there is no such vibration in valve B, where the sectional area change of the orifice is non-linear as shown in Figure 4-9.
4.3.3. Analysis for the Vibration Control

The simulation study was carried out for damping force control with pursuing the target force given by the active control. Hereunder the results of predictive analysis are shown for applying the control condition (a), (b) and (c). These conditions are shown in Figure 3-2 of section 3.2.1. In the case of (a) (LQ1), the weighting of performance index is on the sprung acceleration concerned with ride comfort. In the case of (b) (LQ3), the weighting of performance index is added on the sprung mass displacement as well as the acceleration. In the case of (c) (LQ2), the weighting of the sprung displacement is remarkable concerned with vehicle motion. The simulation model is half-car model with 4 degrees of freedom for sprung mass bounce and pitch motions and front or rear unsprung masses. The control gain for each condition is the same value as described in Table 3-1. Figure 4-11 shows the vibration transfer function of sprung mass from the road input. Compared with the case of control (a) and (b).

![Vibration transfer ratio](image)

**Fig. 4-11 Simulated results of control**

![Damping force change during control](image)

(a) Damping force change during control

![Critical damping ratio](image)

(b) Critical damping ratio

**Fig. 4-12 Controlled damping for case (b)**
the vibration level of case b in the resonance frequency of sprung mass about 1 Hz up to several Hz is less than that of case a. The damping force characteristics in the control b and the frequency characteristics of the equivalent damping coefficient are shown in Figure 4-12(a) and (b). As for the change of damping force at every moment, the trajectory of damping force is moved along the sector shape as shown in the figure(a) where the suspension relative velocity is in horizontal axis. Moreover the frequency characteristics of
the damping coefficient are given over a wide range which covers over 0.1 to 10 Hz as shown in figure (b).

The example of the effect of the control for supposed normal road is also shown in Figure 4-13. Comparing with the cases with and without control, both of the vibration acceleration and displacement of sprung mass with control are reduced by half of that of without control.

Next, the effect of damping control by frequency weighted methods is examined. The predictive results for the effect of the control were shown in Figure 4-14(a) and (b), when the weighting function of the damping force control was set already shown in former Figure 4-8.

In these figures the passive control cases (NO LQ) were also shown for comparisons. The effect of the control was not so much in comparison with the active control (Fig. 4-8) because of the little power in the damping control. However comparing with ordinary control (LQ), the vibration suppression performance of frequency weighted control (LQ3) is fairly improved in the frequency region at which the weighted filter is aimed to decrease the vibration level.

Fig. 4-14 Simulation results for semi-active control
4.4. Experimental Analysis of Optimal Vibration Control

4.4.1. Experimental Apparatus and Control System

Figure 4-15 shows the appearance of the experimental apparatus for the verifications of this control system. The equipment is a model apparatus for two degrees of freedom, consists of the sprung and unsprung masses. The both of masses slide vertically along with two guiding bars.

![Fig. 4-15 Model apparatus of system](image)

The suspension unit is installed between these masses. The unit is composed of the oil actuator which consists of piston and cylinder, the high-pressure gas accumulator and the attenuation force control valve which is located between them. Two pressure gauges are also set up to obtain the damping force from the difference pressure through the orifice in the damping control valve. Absolute displacement detectors are attached to individual mass and base. The accelerometer for performance evaluation is also located on each mass. The excitation input displacement equivalent to turbulence is given by an electro-hydraulic shaker according to the signal stored in a data recorder. Each numerical value of the actual experimental device is as the following.

\[
m_1=45.9\text{kg}, \quad m_2=224\text{kg}, \quad k_1=186\text{N/mm}, \quad k_2=16.2\text{N/mm}
\]

In the above-mentioned vibration system, to make the control force follow the target force which is preliminary set by the optimal state feedback control, the damping force is controlled.
Figure 4-16 shows the block diagram of the control system. The personal computer as the attenuation controller used PC-98x12 (32 bit) and analog and digital conversion of both the state variable input and control output is used for 12 bit. The processing language used C, the processing time from state variable input to the control output is about 1 msec for non-frequency weighted control and about 1.2 msec for frequency weighted control. The control period is assumed to be 5 msec. From the measurement results at the processing time, as a load of CPU, there is no remarkable difference between an usual control and the frequency weighted control. Figure 4-17 shows the flowchart of the processing program in the case of the frequency weighted control.

Fig. 4-17 Flow chart of the control
4.4.2. Optimization of the Controlled Damper Valve

To optimize the squeezing part shape of the damping force control valve in the actual experiment, two kinds of opening characteristics A and B were set as well as the above-mentioned simulation and the stability of the control system was examined.

Figure 4-18 (a) is shown the generated damping force of two kinds of valve in 0.3 m/s of the suspension velocity at each valve opening. In the case of linear valve A, the generated damping force is rapidly increased in small valve opening. Figure (b) shows the damping force control gains from this figure. That is, the control gain changes rapidly in valve A because of the nonlinear characteristics in generated force. On the contrary, the change rate of the control gain of valve B is rather small.

The examples of experimental waveforms are shown in Figure 4-19 (a) and (b). These are the comparisons with valve A and B for the same vibration condition of sine wave input. In these figures both of the acceleration of sprung mass and the damping force are shown for each valve. From these figures the system with valve A is unstable, on the contrary, that with the valve B is stable. It is clear that the oil flow in damping control valve vibrated in the linear cross-sectional area because the damping control gain is extremely increased.
in small valve opening. On the other hand, good stabilizing is given for valve B with nonlinear characteristic for cross-sectional area of orifice. The generated vibration with valve A is mostly corresponding to the resonant frequency of the excitation movement from the equivalent spring constant of mass and the drive part of the spool due to the pressure change of the orifice chamber.

![Graphs of acceleration and damping force for valves A and B.](image)

(a) In the case of valve A  
(b) In the case of valve B  

Fig. 4-19 Experimental results for effects and controls

Next, the effect of the control period on the valve response characteristics was examined. In this experiment the sampling period of control is chosen for three kinds of 0, 5 and 10 msec, and the controlled waveforms are compared. When the control period is longer, the spiky and impulsive vibration is occurred according to the delay of control under the random excitation condition.

Figure 4-20(a) and (b) show the situation of the vibration generation. These figures show the waveforms of damping force, relative velocity between sprung and unsprung mass and valve opening signal for (a) sampling period 10 msec and (b) 0 msec respectively. In the sampling period 10msec as shown in the figure, the damping force is emerged with the relative velocity and the spiky vibra-
tion is generated partially. The reason of this vibration may be thought to be the response delay in the oil pressure, which is caused by the delay only for the control period between real control signal (solid line in valve opening chart) and output waveform of controller (dotted line). By this response delay the spiky vibration is generated when a relative velocity is large.

Figure 4-20 Spiky vibration by control delay

Figure 4-21 shows the spiky vibration toward sampling period.

From this figure it is clear that sampling period should be lowered less than 5msec at least.

By the above-mentioned examinations, an unstable vibration of the controlled damping force could be prevented by valve shape modification and sampling period optimization. The most of vibration is fairly damped and is able to solve the problem of disturbance.

Figure 4-21 Spiky vibrations
4.4.3. Results and Discussions of Control Effect

Experiments were carried out for each case of the parameter sets a, b, c in Table 3-1, and two kinds of passive conditions for case e as hard absorber and case d as soft absorber. The vertical input \( x_0 \) was made to simulate actual road roughness. The electrical hydraulic shaker vibrated the wheel spring by the table displacement \( x_0 \).

The results of the experiments are shown in Figure 4-22 and Figure 4-23. The time histories for each case are shown in Figure 4-22(a)-(e). In each figure there are six time histories namely the acceleration \( \ddot{x}_2 \) and displacement \( x_2 \) of sprung mass, control objective force \( u \), the relative velocity \( \dot{y} \), the actual damping force \( f \) and the actual input displacement \( x_0 \). Noticing the variations of each wave form, the response is changed by the gain parameters in spite of the same amount of excitation. In case a, there is low level in the acceleration \( \ddot{x}_2 \) compared with case c. On the other hand, the displacement \( x_2 \) is the lowest in case b, which is fairly consistent with parameter choice in the performance index.

The feasibility of the control strategy of this system is examined by the comparison with the desired force variation \( u \) and actual generating force \( f \). As shown in each figure, the damping force is generated according to the desired control force \( u \). In the suitable occasion that the directions of relative velocity \( \dot{y} \) and the objective force \( u \) are the same, the damping force \( f \) is actually able to follow the desired one. From these figures in each parameter, there are some coincidence with the direction of generated force and desired one. The envelope of the damping force change in each figure is similar to the desired force, respectively.

Figure 4-23(a)-(e) show the generated damping force changes during control according to the relative velocity \( \dot{y} \) between the unsprung and sprung masses. These figures are described by the Lissajous form of the velocity-force. In these figures the right upper sides indicate the region of bump and the left that of rebound. From the comparison with each case a, b and c, the generat-
Fig. 4-22 Comparison of measured time histories

Fig. 4-23 Generated damping forces during the control
ed damping force are not described in one line like 4 and 5, a passive shock absorber characteristics shown in Fig. 4-23(d) and (e). The damping forces are fairly scattered in both directions for bump and rebound according to the control gains. In cases a and 5, the levels of the generated forces are rather lower than in case 6. In case 6 the generated level of the damping force is higher in the low velocity region and the effect of the control is revealed as the small variation of the displacement of the sprung mass as shown in Fig. 4-22(c).

The frequency transfer function for each case were given by a fast fourier transform analyser. The transfer ratio was \( \frac{x_2}{x_0} \) and the frequency range was 0.5-20 Hz. Figure 4-24 shows various distributions in shapes of the transfer functions. It is obvious that the distribution is changed by the control parameters. Case 6 is the lowest transfer ratio below 2 Hz, and case 5 is the highest. Case 4, 5 and 6 are lower transfer ratio within 2-8 Hz compared with case 4. From the view point of the control effect, case 5 shows the most desirable characteristics in the all cases.

Fig. 4-24 Comparison of frequency response
The effects of this control method should be discussed in terms of how well the measured characteristics match the target ones. This is carried out by comparison of the eigenvalues listed in Table 3-1 with the shapes of transfer function shown in Fig. 4-24. The results of the transfer function indicate the changes in the shape according to the supposed control gains. The peak at the left hand side of the figure due to the resonance of the sprung mass m2 is moved to the right hand side according to the movement of the pole assignment estimated by the eigenvalue as shown in Fig. 3-2. From these relationships between the control parameter in active control design and the experimental results using the semi-active control methods, the effect of the control is obvious and this method is shown to be useful for adjusting the vibration responses in the vehicle suspensions.

Next, in the case of frequency weighted control for the target force, the experimental results are shown in Figure 4-25(a) and (b). In each figure the comparison of the cases with and without frequency weighted control is shown as well as no damping force control. No LQ denotes the status of the fixed valve opening in neutral location. Note LQ means usual control condition and LQ3 means frequency weighted control as shown in section 3.2.2.

![Graphs showing experimental results for frequency weighted control](image-url)
The frequency shaped control shows the effect of decreasing of acceleration level and increasing of the stability of displacement of sprung mass. As shown in figure (a), the effect of frequency weighted control is recognized to be improved in near 3 Hz region compared with usual control, and the stability is also improved in the low frequency region about 0.5 Hz as shown in figure (b). This control method results in the excellent attitude control of the sprung mass. From these experimental results and discussions, when the frequency weighted control is applied to suspension damping force control system, the vibration would be significantly suppressed in comparison with the usual control methods.

Thus the vibration characteristics of such a control system mentioned above are shown to be adjustable by using the control equipment proposed here.

4.5. Summary

The continuous damping force control system which is used for semi-active control of vehicle suspension is constructed and the effect of the control is verified by the calculation and the experiment.

Aiming at optimization of device characteristics and controller, the designing conditions to the valve opening characteristics and control system are also examined. Furthermore the frequency shaped optimal regulator is applied to the control system. The control performance of the system for two kinds of frequency region concerning the trade-off with ride comfort and attitude is studied by the simulation and experiment.

The results obtained in this study are shown as follows:

(1) The effect of the proposed damping force control system was confirmed by the examination with a model experimental apparatus. By applying the optimum regulator to the linearized vibration system, the damping force was obtained by coping with the optimal target force.
(2) From the examination for valve opening characteristics with the experimental apparatus, it was verified that the valve opening property is corresponding to stabilize the control system in the wide range of the operation. The cross-sectional area change of the valve orifice due to spool displacement should be non-linear to design the valve characteristics. The area change of the squeezing part to the valve opening is divided into two parts, one is linear part and another is nonlinear part.

(3) Examining the influence of sampling period in controller on the valve response characteristics, it was verified that the sampling period for this control system should be lower than 5 msec at least. When the period is longer the spiky vibration would be generated due to the delay of the control.

(4) From the experimental results with the apparatus, the preferred methods for the control system were discussed. As for the control logic by the frequency shaped linear quadratic regulator, a higher effect of control was obtained because of the controllability for both of frequency regions concerning the trade-off with the ride comfort and the attitude control of vehicle.

(5) The processing time of the control was compared with the frequency weighted control and the usual control by using 32 bit personal computer. It was clarified that the frequency weighted control needed 1.2 msec and 20% increase in the control period. In comparison with the usual control, the load of CPU is comparatively small.

It is still more necessary for improvement of the control that the more detailed considerations should be taken into accounts for practical applications including the valve opening control logic. Moreover it would be a problem in the future such as improving the safety of the control against the system failure and development of conservative sensor system for practical use.
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CHAPTER 5 STUDY OF ACTIVE VIBRATION CONTROL

5.1. Introduction

In vehicle suspension system, various control methods are examined over both performances of vibration and motion to improve vehicle comfort\(^1\)-\(^{14}\). These control methods include an active type and a semi-active type described in the preceding chapter. Active controlling by oil pressure energy is popular in suspension control. The control method of vibration is frequently applied the optimum regulator with state variable feedback\(^{15}\)-\(^{19}\). There are also many attempts to use recently developed control theories\(^{20}\)-\(^{29}\).

In the design of these control systems, the problems include not only stability of vehicle attitude during running but also improvement of vibration of vehicle due to road disturbances. It is necessary to adapt efficiently various control technologies with fine tuning according to various road surfaces and vehicle maneuvers. On the contrary, the necessities for pursuing the energy dissipation are also expected for improving both energy efficiency and the cost of the elements. Therefore the riding comfort as well as the control of stability and controllability of vehicle is still needed in the application of these technologies\(^{30}\)-\(^{32}\).

In this chapter, the oil pressure control is used to suppress the vibration. The active control of vehicle motion described in section 3.4.1. is examined experimentally using real car.

First, in section 5.2., the control element characteristics used for this vibration control are described. The control actuator and the control elements are shown and the results of their properties are also examined by experiment.

Next, in section 5.3., the effect of the control is analyzed by simulation.

At the end, in section 5.4., the experimental analysis is carried out for confirming the effect of the control by using real car. The test procedure of the vehicle is also shown. The applicability of this control method is also examined in this field test.
5.2. Active Vibration Control System

5.2.1. Construction of Controlling Actuator

Figure 5-1 shows a detailed composition of suspension control system which is used to examine the active type vibration control. Table 5-1 shows the principal dimensions of this experimental system.

The control device is a so-called "integrated suspension unit" that is accumulated with a pump to execute an active oil pressure control and a control valve for semi-active damping force attenuation. The unit is consist of two parts: one is the attitude control by a servo-motor and a gearpump and an electromagnetic valve and another is the ride comfort control by a damping force control valve and an accumulator for gas spring. The damping force control valve is installed between the accumulator and the cylinder. The control valve is a rotary type and is driven by a pulse-motor for continuously controlling a damping force with every suspension movement. The variable orifice of multi-hole type is adapted for decreasing the flow force due to oil jet stream in squeezing part.

<table>
<thead>
<tr>
<th>Table 5-1 Dimensions for unit</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter of cylinder</td>
<td>30mm</td>
</tr>
<tr>
<td>Frictional force between piston and cylinder</td>
<td>0.1kN</td>
</tr>
<tr>
<td>Capacity of accumulator</td>
<td>350cm³</td>
</tr>
<tr>
<td>Precharged gas pressure in accumulator</td>
<td>5.0MPa</td>
</tr>
<tr>
<td>Displacement of gear pump</td>
<td>0.25cm³/rev.</td>
</tr>
<tr>
<td>Max. power of servo motor</td>
<td>0.550kW</td>
</tr>
<tr>
<td>Max. speed of servo motor</td>
<td>10,000rpm</td>
</tr>
<tr>
<td>Max. pressure in cylinder</td>
<td>17.0MPa</td>
</tr>
<tr>
<td>Damping force range of DCV</td>
<td>60N—(∞)</td>
</tr>
<tr>
<td>Weight of suspension unit</td>
<td>10kg</td>
</tr>
</tbody>
</table>

Fig. 5-1 Integrated suspension unit
For the main element of attitude control part, a gearpump has a pair of external gears, and the gear shaft is directly driven with the servo-motor. The adjustment of active force by the gearpump is done by supplying working oil to the actuated cylinder according to the direction of suspension movement.

5.2.2. Characteristics of Controlling Elements

A basic characteristic of the control element was examined by the individual experiment for each element used in this study. The characteristic of the experimental unit was measured by the excitation test of the cylinder. In the tests, the rod of the cylinder was controlled at constant piston speed based on the triangular wave signal from oscillator and the constant vibration input was given to the unit. The results are shown below.

(1) Gas spring characteristics

Figure 5-2 shows the gas spring property of the suspension unit. Gas spring constant in vertical axis shows the amount of increase and decreases of generated force for unit amount of unsprung mass displacement. Three cases of charged pressure were examined.

It is understood that the gas spring constant is varied according to the piston speed and is approximately proportional to the charged pressure.

![Graph showing gas spring characteristics](image)

Fig. 5-2 Stiffness of gas spring
(2) Pressure characteristics

In the case of the fixed unsprung mass, the pressure characteristics were measured. When the command signal as operational input is varied stepwise to 2500, 5000 rpm gearpump revolution from 0 rpm, the pressure $P_C$ of the cylinder is rapidly changed. Figure 5-3 shows the relations between the required time of up and down pressure of $P_C$ with 0.5 Mpa and the valve opening angle $\theta_v$. As the result, the required time for the pressure is in inverse-proportional to the revolution and there is no remarkable difference between (a) and (b) in the region of $\theta_v$ above 10 degree. At $\theta_v = 5$ deg., the effect of shortening of required time is more remarkable in higher revolution.

![Graph of pressure characteristics](image)

(a) Case of increasing pressure (b) Case of decreasing pressure

Fig. 5-3 Time response for increasing/decreasing cylinder pressure

(3) Actuator friction force

A static frictional force that acted on the sliding area of the piston was calculated from the measurement value of $P_C$ and $F_C$ and the expansion side is smaller than the shrinking side in the whole. There is the tendency that in the expansion side it is nearly 80 N irrespective to the value of $P_C$, and on the other hand, it becomes larger in proportional to $P_C$ in the shrinking side. And it reaches about 100 N in the case at $P_C=10$ Mpa.
(4) Damping force characteristics

Figure 5-4 shows the damping force characteristics in the case at the piston velocity $v = 0.3$ m/sec. There are three kinds of valves of cross sectional area changes in the experiment.

As shown in this figure, in the case of valve A, whose orifice area $A_d$ is approximately proportional to valve rotating angle $\theta_v$, the damping force change is extremely high in $F_d = 2$ kN neighborhood operating region. Valve B and C have slow and gradual change of the orifice area and are confirmed in the experiment. From these considerations, the valve C is selected to the suspension unit for the experimental vehicle. The response of the damping control valve is governed by the property of driven pulsemotor itself namely 1500 pulse/sec, and in a whole response the maximum response frequency is 25 Hz at the condition of the pressure state $P_c = 10$ Mpa, valve opening within 10 degrees.

![Graph showing damping force characteristics](image)

**(a) Bound side**

**(b) Rebound side**

Fig. 5-4 Characteristics of damping force

5.2.3. Controller circuit

There are two kinds of control for vehicle suspension system: one is attitude control of the vehicle and other is vibration control for each wheel. The controller is designed to combine both controls in a whole system.

Figure 5-5 shows the construction of the controller that divides 1 CPU for
the attitude control and 4 CPUs for every wheel vibration control because of the need for high speed processing. The 68B09E type was used in the CPU for this suspension control system. The number of A/D converter was 32 channel because of many sensors. The resolution of the converter is 12 bit for the actuator and 8 bit for the monitor during calculation. Moreover one-tip microcomputer was used for the pulse converter. Here the control period was 5 msec for the damping control and 10 msec for the attitude control, and then assembler was used in the program and carried out in integer type. As shown in the figure, the setting parameter was also used by the LT terminal.

![Diagram of suspension controller]

**Fig. 5-5 Construction of suspension controller**

### 5.3. Analysis of Active Control Effect

#### 5.3.1. Adaptation of Control Methods

In the active type vibration control, the control performance should be carefully considered for the vehicle movement and the delay of the control. Therefore, to compensate the delay, the feed forward control is needed in addition to the usual feedback control. Hereunder two types of the feed forward control are examined as follows. In the both cases the effects of the control are simulated.

**Control 1:** the combined method for feed forward control based on the steering angle for the first maneuver and state feedback due to vehicle roll angle.
Control 2: method of being added the vehicle lateral acceleration during attitude control to the usual feedback control with roll angle/rate.

5.3.2. Analysis of Vibration Control

For the preparation of applying the each control to the vehicle, the predictive considerations are carried out by simulations. The calculation model is four degrees of freedom as for the vehicle motions besides two degrees of freedom as for the vibration of the sprung and unsprung masses.

Figure 5-6(a) and (b) show the examples of active roll attitude control when stepwise steering and slalom steering (control 1). In each figure, the time histories of steering angle, roll angle, pump revolution and slip angle are described. From figure (a), the roll angle of the vehicle body is decreased to 50 % or less compared with the case without the control. The pump revolution is rapidly changed to the high rotation by the feedforward control and the initial roll inclination is suppressed at the same time.

Next, in figure (b), the effects of the control are also shown for slalom test. As shown in the figure the side slip angle is decreased by the attitude control as well as the roll angle.

![Graphs showing control results](image)

(a) In the case of stepwise steering  
(b) In the case of slalom test

Fig. 5-6 Simulation results for active attitude controls  
(\(u_0=40\text{km/h}, \text{with (---) and without (-----) control})
5.4. Analysis of Vibration Control Using an Experimental Vehicle

5.4.1. Experimental Vehicle and Procedure

Table 5-2 shows evaluation methods for vehicle stability and controllability with the experimental vehicle. The test items are as follows.

1) motion characteristics in the steady state.
2) response in the cornering motion.
3) subjective judgment of closed loop response including driver-vehicle.

As for above 1), the constant speed full circular running test of radius 30 m was carried out. For 2), the quarter cornering test with stepwise steering was carried out in constant speed 40 Km/h. For 3), the lane change test in the speed 70 Km/h on 30 m length/3 m width lane was carried out. In these tests, the vehicle behaviors and state variables were measured with steering wheel angle, pump revolution, yaw rate, lateral acceleration and so on.

<table>
<thead>
<tr>
<th>Item</th>
<th>Test</th>
<th>Measurements</th>
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<tr>
<td>Body roll</td>
<td>Steady state</td>
<td>Roll angle</td>
</tr>
<tr>
<td>characteristics</td>
<td>cornering test</td>
<td>30m</td>
</tr>
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<tr>
<td>Subjective</td>
<td>Cornering test</td>
<td></td>
</tr>
<tr>
<td>judgment</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Lane change test</td>
<td></td>
</tr>
</tbody>
</table>
5.4.2. Results and Discussions of Control Effect

The experimental results of the motion control of the vehicle are described here. At the beginning the comparison of roll movements is shown concerning to the active controlled vehicle and non-controlled vehicle. In the case of control the control 1 and 2 were also examined.

The roll characteristics of the controlled and non-controlled car are shown in Figure 5-7, which are obtained from circular turning tests.

Comparing the steady roll angle at 0.5 G of centripetal acceleration for controlled and non-controlled, the roll angle was 65% for control 1, 70 % for control 2 compared with non-control case.

Next, the result of the cornering tests was shown in Figure 5-8 as the roll behavior due to steering. The control method in this case was control 1. From the time histories of pump revolution in this figure, the both sides of the pressures of the suspension cylinder were rapidly increased at the initial moment by the feedforward control. As the result of the control, the initial roll angle of the control vehicle is suppressed to about 50 % to the non-controlled one.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig5-7.pdf}
\caption{Body roll characteristics}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig5-8.pdf}
\caption{Attitude control in cornering ($u_s=40km/h$, with (——) and without (-----) control)}
\end{figure}
Next, the result of the lane change test is shown in Figure 5-9(a) and (b). The figure shows the comparison with controlled and non-controlled in both control 1 and control 2 respectively. The peak value of the roll angle of the controlled car is decreased to non-controlled by 50% when changing the lane. Moreover as for the phase of the roll angle, it depended on control method, in the case of control 1 the rising of the initial roll was controlled by the feed forward control and the peak time was delayed compared with non-control. On the other hand, it was the same as non-control for control 2 and the peak time was advanced to the non-control.

Fig. 5-9 Attitude control in lane-change maneuver (70 km/h)
without control  suspension control

Photo 5-1 Continuous pictures during lane change test
In Photo 5-1, the continuous pictures during lane changing are shown in every 0.4 second. The left hand pictures are for the case of without control and right hand pictures are for the case of with control 1. As shown in these pictures, the roll angle due to steering is fairly decreased from the beginning of lane change by the active suspension control.

As for the subjective judgment of the driver at that time, all the drivers answered that the driving maneuver in the control 1 was easier because of the slow roll motion. Moreover, the return steering of handling maneuver had changed depending on the roll angle comparing with controlled and non-controlled vehicle. That is, in the small roll angle with controlled, the time lag in the steering angle was almost zero and the handling maneuver was smooth when return steering. On the contrary, in the large roll angle with non-control, the time lag was felt to be uncomfortable to these drivers. As the result of the control, the feeling of handling maneuver is fairly improved.

5.4.3. Discussion of Control Methods

The comparison of the effect of control is discussed for the individual control system. Figure 5-10(a), (b) and (c) showed the experimental data of impulsive steering test, the simulation result for the control 1 respectively and the simulation result for the control 2 individually. In every figure the vehicle speed is the same to 100 Km/h.

Comparing with figure (a) and (b), the experiment and the simulation are mostly corresponding. Here in the case of applying the control 1, according to the feedforward term due to the steering angle, the pump moves reversely in advance when returning handle maneuver. The tendency becomes more remarkable when the vehicle speed is faster and the steering is more impulsive.

On the other hand, in the case of applying the control 2 as shown in figure (c), the reverse-rotation of pump is not occurred and the peak value of the roll angle is decreased in compared with the control 1.

From these considerations as practical use for the control, it is necessary
to adopt the gain schedule methods according to the vehicle speed and the steering maneuver. On the other hand, when the attitude control was applied, the damping force control was changed to the direction of high damping for using the energy of active control efficiently. Including these controls it must be considered to judge and select the control method due to real running of the vehicle.

![Graphs showing control methods comparison](image)

Fig. 5-10 Comparison of control methods in attitude control

5.5. Summary

Applying the active type vibration control to the vehicle suspension, the attitude control characteristic by oil pressure control was examined.

The results in this study are concluded as follows:

(1) Concerning to the attitude control during handling maneuver, actively control system was applied for improvement of vehicle stability and controllability. Integrated with active controlled actuator, hydropneumatic spring and damping force control valve, effective control suspension units were composed.

(2) Each control response characteristic of the control element was examined and controller was composed for attitude control.
(3) The predictive examination for the effect of the control was carried out by model simulations. The effects of both the feedforward controls by steering and feedback controls preceding attitude change were clarified.

(4) As the results of experimental vehicle analysis by real running tests, the effect of the roll attitude control was confirmed. The roll control effect was shown to be stable maneuver in the handle steering.

Moreover, from the consideration for the control methods, the gain schedule methods are necessary to adopt the control methods according to the vehicle speed and the steering maneuver.

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6.1. Introduction

In the active control of vehicle steering systems there are two methods for the motion control, one is additional front steering and the other is four wheel steering. In general, as for the latter case, the degree of freedom in design is larger and the effect of the control ability is also better at the critical movement\(^1\). However if the purpose of the design is in high-speed running, the main subject is to secure the stability of a driver-vehicle system and the real steering angle is limited within several degrees. Therefore an active control for the front wheel steering was carried out here and the real steering angle was adopted by so-called drive-by-wire method to synthesize the active control signal from the vehicle movement given by the controller according to the driver’s steering.

The purposes of the design for the front active steering are:

1) To improve stability against disturbance like cross wind.
2) To improve response and controllability of steering at lane changing.\(^2\)\(^{-4}\).

To achieve these purposes, not only the control system of the vehicle but also the steering characteristic of the driver should be considered simultaneously. Driver’s operation model as mentioned in section 3.4.2. is considered for the control system of this study. The driver-vehicle systems were referred to many preceding studies\(^5\)\(^{-31}\). The control logic for active steering is made of the state feedback loop designed by the optimum regulator, and is also intended to improve the properties for both response and stability of vehicle movements.

In this chapter, the experimental vehicle for the control system is constructed, the predictive performances are simulated and the experiments by the test vehicle are carried out.
First in section 6.2, the control unit and controller used for vehicle motion control are shown.

Next in section 6.3, the control effects are analyzed by numerical simulations.

In section 6.4, at the end, the experimental analysis for vehicle handling and movement is carried out and the effects of the vehicle motion control are examined using the test vehicle.

6.2. Active Motion Control System

6.2.1. Actuator for Control

Figure 6-1 shows the steering actuator for the control system. The mechanism to add an assistant force was installed in the power steering system. In the actuator the rotary valve of oil flow is driven by the stepmotor making a usual power steering mechanism to make an additional steer. Namely the rotary valve is driven by both the driver operation and added input from controller.

Table 6-1 shows the dimensions for the system.

![Diagram of Active Motion Control System]

Fig. 6-1 Construction of control unit
Table 6-1 Dimensions for active steering

<table>
<thead>
<tr>
<th>Steering gear</th>
<th>Type</th>
<th>Rack and pinion</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Rack stroke</td>
<td>146mm</td>
</tr>
<tr>
<td></td>
<td>Lock to lock</td>
<td>3 rev.</td>
</tr>
<tr>
<td>Control valve</td>
<td>Type</td>
<td>Rotary directional control valve</td>
</tr>
<tr>
<td>Pulse motor and motor driver</td>
<td>Exciting method</td>
<td>4 - 5 phase exciting</td>
</tr>
<tr>
<td></td>
<td>Step angle</td>
<td>0.36deg</td>
</tr>
<tr>
<td></td>
<td>Maximum speed</td>
<td>2,000PPS</td>
</tr>
<tr>
<td></td>
<td>Rated current</td>
<td>1.4A/phase</td>
</tr>
<tr>
<td></td>
<td>Input power source</td>
<td>DC24V ± 10%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.5A</td>
</tr>
<tr>
<td>Steering angle sensor</td>
<td>Type</td>
<td>Potentiometer (Sakae, 46HDS-35ΩF No.1)</td>
</tr>
<tr>
<td></td>
<td>Linearity</td>
<td>±0.1% (±1deg)</td>
</tr>
<tr>
<td>Wheel angle sensor</td>
<td>Type</td>
<td>Potentiometer (Midori, LP-200F (5 kΩ))</td>
</tr>
<tr>
<td></td>
<td>Linearity</td>
<td>±0.5% (±1mm)</td>
</tr>
</tbody>
</table>

The stepmotor is activated by the control signal calculated in the controller based on the steering angle, the yaw rate, the lateral acceleration, the vehicle speed and the wheel angle. This motor moves the rotary valve to control the front wheel steering by controlling the pressure in the power cylinder. This steering system is still controllable in case of the unit failure, being equipped with a fail safe system. In the abnormal circumstances the fail recovering valve, by which the pressure difference of both left and right power cylinder is reduced, is set on the control system.

Next the block diagram of the steering actuator control system is shown in Figure 6-2. In the figure the solid lines indicate the mechanical operative signal flow and the broken lines indicate the electric signal flow.

When the derivative $x_y$ between steering wheel angle and real steering operational value exceed a preset value (it is about 16 degrees in nominal steering wheel angle), the mechanism (which is sign $C_{fs}$ in the figure) acts on the power piston for fail safe operations.

In the electric control shown by the broken lines, two kinds of control parts are combined: open proportional loop and electric feedback loop. The open loop is controlled by the signal of driver's steering and the active control steering angle.
The electric feedback loop is the output due to the difference signal between these sums and the real steering angle. This loop is installed for dealing with the case of the out of control in the stepmotor. The dead zone of the motor is set with 3 degrees, which is shown Kg in the figure.

In the control system, the steering wheel angle is detected electrically and the amount of active steering angle is calculated. According to these signals, the control command is introduced to the stepmotor and the pressure due to the deviation between rotational angle of the motor and real steering angle acts on the power cylinder to make an additional steering angle.

6.2.2. Characteristics of Controlling Elements

To obtain the control stability, the conditions and the response characteristics are examined by experiments.

The examples of the wheel angle responses in the cases of three kinds of steering angle are shown as time histories in Figure 6-3 (a), (b) and (c).
The control conditions are as follows: sampling period $T=35 \text{msec}$, calculation time $t_c=2 \text{msec}$ and the pulse interval $t_p=3 \text{msec}$. From the figure it is stable in case (a) and (b), but unstable in case (c) in more than 3 Hz.

![Graphs showing steering angle and gain response](image.png)

Fig. 6-3 Response of control unit  
Fig. 6-4 Response of installed actuator

The tuned result of the parameter and the response characteristic of wheel angle for steering angle was shown in Figure 6-4 by varying control periods. To stabilize the wheel angle, the pulse interval is changed by varying control periods. From the figure the gain would be decreased and the phase lag would be larger when the control period is longer. According to these considerations, the optimal conditions are selected as follows: $T=5 \text{msec}, t_c=1.2 \text{msec}$ for calculation time and $t_p=0.5 \text{msec}$ for the pulse interval.
6.2.3. Controller System

The steering controller is shown in Figure 6-5. Signal from each sensor such as real steering angle, steering wheel angle, lateral acceleration and yaw rate is input to the controller, on the basis of these signals the control signal is calculated for the output signal by this system. 16bit CPU 68000 is used for the calculation and 12 bit analog-digital converter is used to obtain the necessary resolution. As mentioned above, steering actuator is adopted for pulsemotor to drive the control valve, and pulse converter is used for converting the analog voltage output to pulsemotor input by analog-pulse conversion.

Next, when the breakdown is occurred in the control system or the drive system, the system becomes dangerous because of using the Drive-by-Wire methods. In the steering system, the detection of such a state is executed by the computer and sensor systems. In the computer WDT (Watch-dog-timer) is installed for detecting the abnormal computing of the CPU. Preventing a dangerous operation by wrong sensor information, the detection system of abnormal sensing is also installed.

To make the control period of 5msec, in consideration of the followability of real operational steering angle, the steering control program was written by assembler. Moreover the active steering operation program was written by C language and result in 40 msec period.

Fig. 6-5 Construction of steering controller
6.3. Analysis of Active Motion Control

6.3.1. Control System

Figure 6-6 shows steering control system. The control gain \( f_k : k=1, 2, \ldots, 6 \) in the figure is calculated by using the optimum regulator methods. The suitable controls for the various situations are realized by the gain schedule based on the vehicle velocity and the estimated value for the road friction coefficient. The friction value is required by comparing with the both of the peak values of the yaw rate measured on the vehicle and the calculated value in the vehicle model in the computer.

![Fig.6-6 Active steering control system](image)

Next, as for the control logic applications, the effect of the control in vehicle motion was predicted by simulations according to the design shown in the section 3.4.2. The running conditions for the simulation are supposed for two cases: one is the crosswind disturbance during straight running at the constant speed (100 Km/h) and the disturbance is also supposed to be an equivalent force of 980 N from the left-hand side on the front bumper for the period of 0.5 sec. Another case is the single lane change at 100 km/h with 3.0 m lane width.
The simulation conditions as for the control logic are as follows: The control gain is provided for the weight coefficients (g_3=1.0, g_4=20.0 and k_C=8.0) of the performance index in the equation (3-116).

6.3.2. Analysis of the Motion Control

Figure 6-7 shows the simulation results for the crosswind disturbance test. From the figure, it is clarified in the controlled case that the sudden displacement from the straight line caused by the disturbance became to be a half in comparison with non-controlled case, and either yaw rate change or lateral acceleration peak shows the same tendency. Moreover the followabilities are also improved and it is obvious that total stability of the vehicle motion is fairly improved by the control.

Figure 6-8 shows the results of the lane change simulations in comparison of the controlled and non-controlled cases. From the figure, the both of the response of the vehicle and the stability were also improved. From the predictive analysis, these controls for steering are expected for improvement of vehicle motion, especially for the stability and controllability.

![Figures showing simulation results](image-url)

**Fig.6-7** Simulation results in cross wind disturbance (u_w=100km/h, with (••••••) and without (----) control)
6.4. Experimental Analysis of Motion Control by Test Vehicle

6.4.1. Test Vehicle and Measuring Methods

Table 6-2 shows the test procedure for the experiments with the controlled vehicle. The two cases are evaluated by the tests:

1. To imitate the crosswind disturbance, the compressed water is injected during straight running.
2. The vehicle response test during impulsive steering.

In the case (1), the reaction force by the compressed water is applied transversely from the nozzle located in the front bumper of the vehicle during straight running at constant speed 70 Km/h. The driver was fixing his handling during the water injection and active steering was controlled by the state feedback in the control systems.

![Graphs showing simulation results in lane-change maneuver](image)

Fig. 6-8 Simulation results in lane-change maneuver
(\(u=100\)km/h, with \(-\) and without \(-\)- control)
In the case (2), the driver made an impulsive steering along with the lane after straight running at constant speed 70 Km/h. The impulse steering was made to 80-100 degrees in the period of about 0.3 seconds. In the lane-change test, 3 m width lane crossing was attempted within 30 m length. In these tests, the quantities of vehicle behavior as steering wheel angle, pump revolution, yaw rate and lateral acceleration were measured.

<table>
<thead>
<tr>
<th>Item</th>
<th>Test</th>
<th>Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cross wind stability</td>
<td>Wind disturbance simulating test</td>
<td>$\omega = 70km/h$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Lateral travel</td>
</tr>
<tr>
<td></td>
<td></td>
<td>water injection</td>
</tr>
<tr>
<td>Vehicle response</td>
<td>Impulse steering test</td>
<td>$\delta_{\omega}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80-100 deg</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.3 sec</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\omega = 70km/h$, 100km/h</td>
</tr>
<tr>
<td>Subjective judgment</td>
<td>Lane change test</td>
<td>$\omega = 70km/h$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3 m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>30 m</td>
</tr>
</tbody>
</table>

6.4.2. Results and Discussions of Control Effect

The performance of vehicle motion was evaluated from the both points of view as the open loop characteristics of the vehicle and the closed loop one of driver-vehicle system. The comparison of the performance for the controlled and the controlled case is shown.

Figure 6-9 shows the result of the crosswind disturbance test. When the external force was applied to the non-controlled car in straight running, the yaw rate was generated in proportion to the received force and after all the lateral displacement was about 2 m in 3 seconds. On the other hand, in the case of the active steering control, after the disturbance the real steering angle was controlled to suppress the yaw rate. As a result, the yaw rate peak value was decreased and the settling time of the yaw rate was very short. The
lateral displacement was about 1 m, for 50% decreased as compared with the non-controlled car.

![Diagram](image)

Fig. 6-9 Cross wind stability test
(\(u_w=70\) km/h, with (—) and without (••••) control)

Figure 6-10 shows the result of the lane change test. At the beginning of the lane change in the non-controlled car, the generation of yaw rate was delayed against the steering maneuver and the change of the yaw rate was relatively small. Moreover, in the after half of the lane change movement, the amount of the steering wheel angle was large and the lateral displacement of the car was overshot. Furthermore, the slip angle of the car body was changed largely in the after half of the lane change movement and the stability was not so good. On the contrary, according to the active steering, the yaw rate was given rise to by handling maneuver at the beginning of lane change and the yaw motion of the vehicle was fairly improved. The slip angle of the controlled car became larger than the non-controlled because of the rapid yaw
rate generation. By the effect of the control, the vehicle movement in the after half of the lane change was settled and stabilized by the rapid decreasing of both the slip angle and yaw rate. As for the amount of the steering, wheel correction maneuver during the lane change was small and the load of the driver was fairly decreased. Further the lateral displacement of the controlled vehicle was small and the trace of the vehicle was very smooth during the lane change.

These vehicle maneuvers are clarified by the continuous pictures during lane change as shown in Photo 6-1. There are comparisons of the steering control vehicle (right) with non-control (left) and suspension control one (middle). As shown in the third pictures from the top, the vehicle under the steering control is crossing the boundary pyroon, but the others are not yet crossing. As the result, the driver would be able to manage the after half of lane changing because of the quick turning of vehicle head by the control.

![Vehicle response in lane change test](image-url)

---

Fig. 6-10 Vehicle response in lane change test
(\(v_0=70\text{km/h}\), with --- for active steering control, and without — for control)
without control  suspension control  front steer control

Photo 6-1 Continuous pictures during lane change
As the result of these tests, it may be clarified that the effects of stability and controllability against the crosswind disturbance and unexpected turbulence as well as the rapid responsibility as the emergency handling were both improved by using the active control of the front steering.

6.5. Summary

Aiming at the improvement of motion performance of the vehicle, the test of the control methods of front active steering was carried out.

The following results are obtained.

(1) An active control device was made to steering over a usual power steering system. The controller was newly designed and developed for giving the further steering to that of driver's real steering. The characteristics were also examined for each control element.

(2) According to the model simulation for the vehicle performance, the control gains for the experimental vehicle were determined.

(3) By the experimental vehicle mounted these devices, the crosswind disturbance test and the lane change test were carried out and the effect of these controls of the system was examined.

(4) From the results of the experiments, as for the crosswind test, the lateral displacement of the vehicle was reduced by half of the uncontrolled one. As for lane change, the both of rapid yaw motion and stability of the vehicle were confirmed.

In the conclusion of the studies, additional active control is effective for vehicle handling and controllability. By using the active control of the front steering, the stability and controllability of the vehicle against the crosswind disturbance and unexpected disturbance as well as the rapid responsibility as the emergency handling were significantly improved.

According to the application of the active steering control, the long term driving would be able to realize with smallest physical stress.
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CHAPTER 7 EXPERIMENTAL STUDY OF ACTIVE CONTROL

7.1. Introduction

The vehicle is always controlled by driver in his sight information, bodily sensation and also his judgement. Driving is affected by a state of the traffic and an environmental condition like a crosswind, a road disturbance and so on. If promptly judging these situations and instantaneously applying the suitable control are possible, performances of the vehicle motion and riding comfort are sure to be improved. For these improvements of vehicle control, there are many studies for the integrated control system\textsuperscript{[1-11]} and applications of new control techniques\textsuperscript{[12-15]}. They are treating steering, suspension and traction systems as integrating elements.

From this point of view, in this chapter, the chassis control system is composed to obtain the vehicle safety and driving comfort. This system originates from the question: " How much improvement of the vehicle performance is expected for the active control of suspension and steering systems? "

Figure 7-1 shows the concept for this system. The system consists of following three subsystems of control.

1) Suspension control system: Four wheel independent distributed system, compactly integrated devices with high response and energy efficiency.

2) Steering control system: Active front wheel control considering a driver-vehicle system.

3) Multi-CPU control system: Decentralized control system for each suspension and total coordinated system.

Using these subsystems, active hierarchical control system is applied to the instantaneous state changes of vehicle and unexpected disturbance from environments. These subsystems are not connected with a mechanical link system but with a so-called Drive-by-Wire (DBW) system.
First in section 7.2., an overall idea for the system composition and design architecture are described. Next the composition of newly developed actuator, controller and the experimental vehicle are shown respectively as well as the combining technique of individual control systems.

Next in section 7.3., using the test vehicle that installs these elements, the experiments by the real driving are carried out for the vibration and the motion control of the vehicle.

![Diagram](image)

**Fig. 7-1 System concept for chassis control**

7.2. Hierarchical Active Control

7.2.1. Concept of Controlling System

In the actual driving, three steps of running situation are available as straight running, cornering and steering. For better driving, a control adaptation should be carried out according to the running state.
Figure 7-2 shows the control adaptation and system composition of the experimental vehicle in this study. It is a hierarchical structure of three steps when the operation system is divided into each situation.

a) At first, in the case of crosswind disturbance or driver’s handling, the small amount of steering is added to front wheel actively according to the vehicle motion and driver handling characteristics.

b) Next, in the case of vehicle attitude change due to cornering and so on, the body attitude control is available for each pump in each suspension by the feedforward and feedback signal of vehicle state variables.

c) In the case of straight running, vibration control due to road surface conditions is adapted instantaneously for each suspension by damping control according to feedback control of the state variables.

---

**Fig. 7-2 Chassis control vehicle**
According to the running state of vehicle, one of these control systems is selected for a suitable control mode. The highest superiority is given for (a) active steering control and secondarily for (b) active attitude control. In this system, (c) semi-active damping control is always adapted. This system is called partial active control with energy saving.

7.2.2. Construction of Experimental Vehicle

The experimental vehicle used in this study is so-called chassis controlled vehicle and installs an integrated actuator for suspension, a steering actuator, power supply units, sensors and amplifiers to detect the state variables and the controller.

Figure 7-3 shows the system configuration for the suspension control. The integrated suspension unit is composed by motor and pump that are oil pressure source, a gas spring and damping force control valve. This unit has two functions for an attitude control of whole body and a ride comfort control of individual wheel vibration. The pump is driven by the servomotor and the damping control valve is directly driven by the pulsemotor.
Moreover, the sensor and amplifier for detecting the state variables of vehicle and the controller are installed. The suspension actuator unit is operated by the optimal amount in the controller based on the signal from the sensor. The control systems are also divided into the two functions. The multiprocessor system of 5 CPU is used in this system. Four of them are for each wheel vibration control individually because of necessary high-speed operation and the last one for attitude control of whole body. The gain schedule control method is used in this system for the better responsibility to the various road conditions.

Figure 7-4 shows the system configuration for steering control. The actuator for active steering control is converted from usual power steering unit. Between the steering wheel and the pinion axis, the additional steer is applicable about 1 degree. Drive-by-Wire (DBW) system is realized by the steering system. This actuator is driven by the pulsemotor based on the signal from the controller and controls the real steering angle.

![System configuration for steering control](image)
Figure 7.5. and 7.6. show the system block diagrams for the power supply and control system respectively. The power supply to drive these actuators is installed in the trunk room, and drive circuits for servomotor and pulsemotor are also installed in both trunk room and front passenger's seat bottom. The AC supply uses two of methods of a portable dynamometer that is installed in the trunk room and DC-AC inverter by the battery. The former power supply is used for servomotor driving. For the drive circuits of the servomotor and the pulsemotor the latter power supply is selected because of low fluctuation of the source voltage. As the result the power is used efficiently in this system.

![Power supply system block diagram](image)

**Fig. 7-5 Power supply system block diagram**

![Active control system block diagram](image)

**Fig. 7-6 Active control system block diagram**

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Photographs of the experimental vehicle are shown in Photo 7-1(a), (b) and (c). They show the suspension unit in front left wheel, front steering mechanism from the view of the underbody and the controller units on the rear sheet location, respectively.

(a) Integrated suspension unit

(b) Active front steering actuator unit

(c) Controller unit
The controller specification and the parameters for each control as well as the sensor parameters are also summarized from Table 7-1 to 7-4.

Table 7-1 Specification for controller

<table>
<thead>
<tr>
<th>Item</th>
<th>Specification</th>
<th>Feature</th>
</tr>
</thead>
<tbody>
<tr>
<td>CPU</td>
<td>68B09E (2 MHz) ROM32kB, RAM64kB</td>
<td>High-performance 8bit microcomputer</td>
</tr>
<tr>
<td>Bus arbiter</td>
<td>Fixed-priority</td>
<td>Arbitration at 1 byte, no dead time</td>
</tr>
<tr>
<td>A/D converter</td>
<td>12bit x 32ch.</td>
<td>Automatic conversion and storing memory</td>
</tr>
<tr>
<td>D/A converter</td>
<td>12bit x 8ch., 8bit x 8ch.</td>
<td>Output data readable</td>
</tr>
<tr>
<td>Common memory</td>
<td>8 kB</td>
<td>Private writing area for CPUs</td>
</tr>
<tr>
<td>Timer</td>
<td>5 ms</td>
<td>Period variable</td>
</tr>
<tr>
<td>Communication interface</td>
<td>RS-232C, 9,600 baud</td>
<td>Baudrate variable</td>
</tr>
<tr>
<td>Voltage/pulse converter</td>
<td>up &amp; down x 4ch., 1kHz max</td>
<td>Single chip microcomputer</td>
</tr>
</tbody>
</table>

Table 7-2 Specification for vibration control

<table>
<thead>
<tr>
<th>Item</th>
<th>Executing CPU</th>
<th>Control period</th>
<th>Language</th>
<th>Arithmetic accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vibration control</td>
<td>CPU 1 - 4</td>
<td>5 ms</td>
<td>Assembler</td>
<td>16bit (or 24/32bit) Integer</td>
</tr>
<tr>
<td>Attitude control</td>
<td>CPU 5</td>
<td>10ms</td>
<td>Assembler</td>
<td>16bit (or 24/32bit) Integer</td>
</tr>
<tr>
<td>Parameter setup</td>
<td>CPU 5</td>
<td>......</td>
<td>Pascal</td>
<td>32bit Floating point</td>
</tr>
</tbody>
</table>

Table 7-3 Specification for steer control

<table>
<thead>
<tr>
<th>Item</th>
<th>Control period</th>
<th>Language</th>
<th>Arithmetic accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering control</td>
<td>5 ms</td>
<td>Assembler</td>
<td>16bit integer (partly 32bit)</td>
</tr>
<tr>
<td>Active steering</td>
<td>40ms (operation cycle)</td>
<td>C language</td>
<td>32bit real</td>
</tr>
<tr>
<td>Parameter setup</td>
<td>......</td>
<td>C language</td>
<td>32bit real</td>
</tr>
</tbody>
</table>

Table 7-4 Detective objects and the sensors

<table>
<thead>
<tr>
<th>Detective object</th>
<th>Sensor</th>
<th>Cut-off frequency</th>
<th>Use</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Name</td>
<td>Method</td>
<td>Range</td>
</tr>
<tr>
<td>Wheel angle</td>
<td>Linear potentiometer</td>
<td>Variable resistor</td>
<td>±33deg</td>
</tr>
<tr>
<td>Steering angle</td>
<td>Rotary potentiometer</td>
<td>t</td>
<td>±540deg</td>
</tr>
<tr>
<td>Steering effort</td>
<td>Force sensor</td>
<td>Strain gauge</td>
<td>9.8N/m</td>
</tr>
<tr>
<td>Damping force</td>
<td>Pressure sensor</td>
<td>Semiconductor</td>
<td>14.7MPa</td>
</tr>
<tr>
<td>Orifice</td>
<td>Rotary potentiometer</td>
<td>Variable resistor</td>
<td>64deg</td>
</tr>
<tr>
<td>Relative displacement</td>
<td>t</td>
<td>t</td>
<td>±0.1m</td>
</tr>
<tr>
<td>Yaw rate</td>
<td>Rate gyro</td>
<td>Precession</td>
<td>±80deg/s</td>
</tr>
<tr>
<td>Roll angle</td>
<td>Gyro</td>
<td>t</td>
<td>±8 deg</td>
</tr>
<tr>
<td>Vertical acceleration (strut)</td>
<td>Acceleration sensor</td>
<td>Strain gauge</td>
<td>19.6m/s²</td>
</tr>
<tr>
<td>Vertical acceleration (floor)</td>
<td>t</td>
<td>t</td>
<td>9.8m/s²</td>
</tr>
<tr>
<td>Lateral acceleration (floor)</td>
<td>t</td>
<td>t</td>
<td>10Hz</td>
</tr>
<tr>
<td>Front wheel speed</td>
<td>Speed sensor</td>
<td>Electromagnetic detector</td>
<td>27.8m/s</td>
</tr>
<tr>
<td>Rear wheel speed</td>
<td>t</td>
<td>t</td>
<td></td>
</tr>
</tbody>
</table>
7.3. Experimental Analysis of Hierarchical Control

7.3.1. Results and Discussions of Vibration Control Effect

The vehicle vibration and riding comfort performance are evaluated by experiments. The measurement of vibration characteristics is carried out in the ordinary road at vehicle speed 50 km/h and the rough road at 30 km/h as shown in Table 7-5.

<table>
<thead>
<tr>
<th>Item</th>
<th>Road</th>
<th>Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power spectrum</td>
<td>Ordinary road</td>
<td>$u = 50,\text{km/h}$</td>
</tr>
<tr>
<td>density distribution of vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>acceleration</td>
<td>Uneven road</td>
<td>$u = 30,\text{km/h}$</td>
</tr>
<tr>
<td>Damping force characteristics</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Subjective judgment</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The test results of vibration on ordinary road are compared with the controlled vehicle and non-control vehicle that has passive damping characteristics. In the non-control vehicle the damping property is set soft and hard, and the damping force is determined directly in proportion by the relative velocity between unsprung and sprung masses.

The characteristics of the acceleration on the center of the floor and the damping force of the non-control vehicle during running on ordinary road are shown in Figure 7-7(a) and (b), respectively. In the figures, the soft damping characteristics of the non-control car is shown by the solid line and the vibration level of 1 Hz that is the resonance frequency of sprung mass becomes to be higher. On the contrary, the hard damping one of the non-control car is shown by the broken line and the vibration level of 4-8 Hz that is most sensitive to human body resonance becomes also to be higher. Thus the both levels of these vibration areas could not lower at the same time by the passive damping characteristics. As shown in figure (b), the damping force changes
due to the relative velocity are not symmetrical for the bound and rebound direction of the suspension.

Next, the characteristics for controlled vehicle are shown in Figure 7-8 (a) and (b). For the comparison, in the figure (a), the results of the non-controlled car are also described. As shown in the figure, the both levels of sprung mass resonance and human sensitive regions are also decreased as compared with non-control car. According to the diagram in the figure (b), the damping force has changed instantaneously in the various regions due to the various movements of suspension. These damping force changes are coincided with the result of the simulation as shown in Figure 4-12(a). As the result that the both levels of the vibration response of 1 Hz and 4-8 Hz are lower than that of non-control car, the effect on ride comfort is remarkable and the subjective ride feeling of the passenger is fairly improved.

![Graph](image1.png)

(a) Floor vibration response  
(b) Damping force characteristics

Fig. 7-7 Damping and vibration characteristics (conventional)

![Graph](image2.png)

(a) Floor vibration response  
(b) Controlled damping force

Fig. 7-8 Performance and vibration characteristics
Next, the riding comfort characteristics are examined on the different road. As the comparison of the control effects for the two kinds of frequency regions mentioned above, the relations of both characteristics are chosen for the peak values of the acceleration power spectrum densities and are also shown in Figure 7-9 (a) and (b). Here, the figure (a) shows the case of running on the ordinary paved road and the figure (b) shows the one of running on the rough road. The controlled modes are two cases (A) and (B) for each road, and the non-controlled cases are soft and hard mode, as well as the open and closed states of the damping valve.

In each figure, the results for the cases of controlled, namely in the control mode (A) for ordinary road and (B) for rough road, are plotted together. In these figures the effect of the optimal control is directed to the lower region. As the results from the comparison, control (A) is much effective in the ordinary road and control (B) is much effective in the rough road. Clearly from these results, the gain scheduling methods have to be used for the optimal control and to give an adequate damping force characteristic for various road conditions.

![Diagram](image-url)
As the conclusion of the section shown in Figure 7-10, an overall evaluation of attitude and vibration control are shown from the viewpoints of the various control effects. In the figure, the broken line indicates for the case of the non-controlled car for 100% and the solid line for the controlled car.

In the attitude control, as mentioned in chapter 5, it is realized from the roll control that vehicle motion performances are improved and stable handling maneuvers are achieved as well as roll stability is managed at the beginning of steering. As the results of damping force control, it is clarified that the transmitted vibration level from road to the vehicle floor is able to control in the frequency range from 1 to several Hz and it is also achieved to be adaptable for various running road conditions.

Fig. 7-10 Improvement evaluation in vehicle performance (suspension control)
7.3.2. Results and Discussions of Motion Control Effect

The vehicle motion performances are evaluated by examination of the simultaneous control effect of both suspensions and steering control. As shown in Table 7-6, (1) impulsive steering and (2) lane change running tests are carried out.

Table 7-6 Evaluation method for vehicle stability and controllability

<table>
<thead>
<tr>
<th>Item</th>
<th>Test</th>
<th>Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle response</td>
<td>Impulse steering test</td>
<td><img src="image" alt="Diagram" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\delta_w = 80-100$ deg</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$u_w = 70\text{km/h}, 100\text{km/h}$</td>
</tr>
<tr>
<td>Subjective judgment</td>
<td>Lane change test</td>
<td><img src="image" alt="Diagram" /></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$u_w = 70\text{km/h}$</td>
</tr>
</tbody>
</table>

First, the vehicle responses of non-control and controlled vehicle are examined by the impulsive steering test. The responses of yaw rate, lateral acceleration and roll angle are shown in Figure 7-11 (a), (b) and (c), respectively. From the figures, the difference among four cases, namely non-control, the steering control, the suspension control and the simultaneous control, is examined for the motion response due to handling.

(a) **Yaw rate response**: Gain levels in the frequency region range 1-2 Hz are higher in both of steering and simultaneous control than that of non-control. The amounts of phase lag are smaller in both steering and simultaneous control than that of non-control.

(b) **Lateral acceleration response**: Gain levels in 0.8-1.8 Hz and the phase lag are almost same tendency with yaw rate responses.

(c) **Roll angle response**: Gain level in the frequency region below 0.9 Hz is lower in simultaneous and suspension control than that of non-control.

From the results of figure (a) and (b), amounts of the phase lag of yaw rate and lateral acceleration due to steering are compared as shown in Figure 7-12. In the figure, the response characteristic is chosen as the representative for
the phase lag in 1 Hz. From the figure, the amount of phase lag in active steering control is smallest and the effect of rapid response of vehicle motion is obvious.

\[
\begin{array}{c}
\text{(a) } \tau/\delta_w \\
\text{(b) } \ddot{y}/\delta_w \\
\text{(c) } \phi/\delta_w
\end{array}
\]

\((u_w=70\text{km/h, with } - - - \text{ for simultaneous, with } --- \text{ for active steering control, with } \cdots \cdots \text{ for active suspension control and without } \cdots \cdots \text{ control).)}

\text{Fig. 7-11 Vehicle response in impulsive steering tests}

\[
\begin{array}{c}
\text{Phase lag of yaw rate (deg)} \\
\text{Phase lag of lateral acceleration (deg)}
\end{array}
\]

\text{Fig. 7-12 Phase lag comparisons with yaw rate and lateral acceleration (1Hz)
The above-mentioned performance is the open loop characteristics of the vehicle. Next, the closed loop characteristics as man-vehicle systems are discussed from the result of the lane-change test as shown in Figure 7-13.

From the time histories shown in the figure, it can be seen in comparing the case of steering control with that of non-control or suspension control that the response of yaw rate and rapid increase of side slip angle are apparent at the beginning of the lane change. This tendency means the improvement of vehicle yaw motion and quick turning by the active front steering control.

At the after half of the lane changing, the driver of the steering controlled vehicle does not need to turn the steering wheel so much as shown in the time histories. The convergences of yaw rate and slip angle are greater than those of the uncontrolled vehicle. As the result, the overshoot of lateral motion does not occur in the steering controlled vehicle.

The behaviors of the vehicle in the test are also shown in Photo 7-2 as continuous 8 pictures of every 0.4 second.

The quick turning motion at the beginning can also be clarified from the continuous photograph. Comparing with the pictures of the second and third frame from the top, the vehicle under steering control as well as simultaneous controlled one is crossing the pyron that set in the boundary of two lanes. In the case of non-controlled, on the contrary, the vehicle is not yet crossing the pyron. It means the rapid yaw motion of the controlled car is better than that of non-controlled.

Next, the sixth and seventh pictures in Photo 7-2 clearly show the convergence of the vehicle motion. Namely, the simultaneous control car shows the small amount of yaw angle of the body and returns to the straight position. On the other hand, non-control car shows the larger amount of yaw angle as seen at left side of the body in the picture. Furthermore it is obvious that the attitude stabilization is achieved by the active roll control of suspension and the controlled car is stable at the end of the lane change.
Fig. 7-13 Vehicle response in lane change tests

($u_0=70$km/h, with (-----) for simultaneous, with (----) for active steering control, with (------) for active suspension control and without (-----) control)
without control

simultaneous control

Photo 7-2. Continuous pictures during lane change tests
Figure 7-14 shows another verification for the stability of the controls. It is comparison of yaw rate generations due to steering in Lissajous figure. The rapid increase of yaw rate is obvious and the followability of the vehicle is better in the lane changing. In the simultaneous control, the rapid yaw motion due to steering control is improved at the beginning of lane change and the stable returning motion due to suspension control is also improved at the after half of the lane change. This would be an easiness for handling to the driver.

As the result of both controls, the amount of the time to stabilize vehicle is improved to be one fourth of that of non-control, and the squared integral value of driver's handling operation decreases to be 65%. It is clarified that this control improves significantly the vehicle motion as the man-vehicle system.

\[(u_0=70km/h, \text{ with } \text{—— for simultaneous, with } \cdots \cdots \text{ for active suspension control}} \text{ with } \cdots \cdots \text{ for active steering control, and without } \cdots \cdots \text{ control})\]

**Fig. 7-14 Yaw rate change due to steering maneuver**

After these discussions, the total effect for both open and closed loop characteristics of the control is evaluated. The results are shown in the diagram of Figure 7-15. These performances are all from the above-mentioned impulsive steering test and lane change test and are compared between the
simultaneous control and non-control. Upper half of the figure is concerning to the former test and under half is to the latter test.

As the results from the figure:
1) Centrifugal and lateral motions: Yaw rate and lateral acceleration response are fairly improved and the followability due to steering is much increased.
2) Rolling motion: The response of roll angle for attitude control is improved and the roll occurring time is prolonged.

As the results of the experimental studies for real running of the vehicle, the easiness for handling to the driver is realized.

It can be said that the motion control system in this study is achieved the better performance of the vehicle by the total combined control strategies of integrated suspension and hierarchical systems according to running status.

![Diagram of vehicle stability and controllability](image-url)
7.4. Summary

The effect of an active steering and suspension control was examined for a usual state of running of the vehicle. Each control system was designed to be hierarchically constructed and practical experimental vehicle was made along with theoretical designs. The control system with an active hierarchy was applied to an active suspension system and active front steering system to realize the chassis control of the vehicle. The control performance was evaluated by real running examinations.

The following results are obtained.

(1) Constructing the control experimental vehicle and using it, the various experiments were carried out. The effect of control with the hierarchy was examined by mounting the active suspension system and the steer controls on the chassis control experimental vehicle.

(2) It was confirmed that the vibration control with small energy consumption was achieved and the transmitted vibration from road to the floor was decreased in the frequency range up to 1 to several Hz as compared with non-control car.

(3) It was also verified that the suitable combination of both active steering and suspension control was effective for the improvement of vehicle motion control. As the result of these controls, the stabilized time of the lateral motion was significantly improved to be one fourth of non-controlled case and driver's handling operation decreased to be 60%. Namely, the effect of vehicle stability against the crosswind and unexpected disturbance from the road, vehicle response due to driver's steering maneuvers as a sudden obstacle avoidance is equally improved by the control systems.

Thus the active control application was shown to be able to improve remarkably the vehicle stability and controllability as well as riding comfort. Several control methods in this system would be applied to practical vehicle and are utilized for the vehicle safety and comfort.
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CHAPTER 8 CONCLUSION

The purpose of this study has been to clarify the feasibility of various forms of active control for the vehicle chassis. In the basic stage of designing vehicle suspension and steering systems, it is important to take all the fundamental properties of vehicle motion, vibration and dynamic behavior into account. The semi-active control of conventional elements and the introduction of active control technology are expected to improve the comfort and safety of vehicles by stabilizing the body attitude and decreasing the vibration of the vehicle under various conditions during vehicle running. For instance, for superior motion performance of the vehicle, it is necessary to optimize the tire-ground contact, and for suitable handling it is necessary to achieve control and stability in the driver-vehicle closed loop systems.

From these considerations, for the fundamental performances of the vehicle such as straight running and cornering, it is necessary to control the vibration characteristics of the suspension that are due to simultaneous changes in road contacts, and to control the handling characteristics of the steering that are due to maneuvers of the driver's intention. Moreover, automobile manufacturers require that such equipment would be able to be constructed easily, efficiently, and with low energy consumption.

The study begins with precise observation and analysis of the phenomena of vehicle vibration and motion behavior in order to clarify the target level of the controls. Next, for the control of these individual targets, various applications of control methods are surveyed and located effectively in the vehicle. Finally, controlling systems for these various states of vehicle running conditions are proposed using simple and effective construction. The feasibility of these design methods is confirmed by the individual model simulations and discussions. The experimental verifications are also introduced to clarify the effect of these controls using practical experiments with
experimental vehicles.

In these design methods, the suspension units are constructed with independent hydropneumatic systems that integrate the control elements and hydraulic power sources. This system is aimed at the functional integration in each unit of active and semi-active controls. The system uses the combined methods of feedforward and feedback control. The steering system is also constructed with newly devised front active steering. Vehicle controllability and stability at high speeds can be improved by a slight control of front steering. A design procedure utilizing modern control theory and a sophisticated element technique using actuator and multi-microprocessors is also applied. The benefits of these methods include integration of the elements that induce good response of the control, low energy dissipation, ease of construction with high reliability and easy system maintenance. Moreover, such a system is expected to reduce both the noise generated from the pipe line and the energy spent to compress the oil.

The contributions described in each chapter of this overall approach are as follows:

In CHAPTER 2, "STUDY OF EVALUATION AND ANALYSIS FOR VEHICLE VIBRATION", first, evaluation of vehicle vibration is undertaken for the floor acceleration in ordinary commercial cars. Next, noticing the vibration inputs from the tire to the vehicle, the vibration occurring specifically because of the rolling tire is examined. As a result of the examination, the need for measuring tire uniformity leads to the development of a newly designed tire uniformity measuring apparatus. Finally, considering the principle input to vehicle, the measurement of the road surface profile during real running conditions is also achieved by developing another measuring apparatus. Finally, the target level for improving floor vibration of the vehicle is clarified and the target characteristics are recognized by these evaluations.
In CHAPTER 3, "BASIC THEORIES OF ACTIVE CONTROL FOR VEHICLE DYNAMICS", to achieve active control of the vehicle at the above-mentioned target level, the adaptable control theories are considered. First, vibration control for a linearized quarter car model is designed taking into consideration the characteristics of delay of controlling elements. The selected control methods involve an optimal regulator with feedback control of state variables.

Next, considering the tradeoff between human sensitivity and vehicle motions, vibration control using frequency-shaped cost functions is applied to obtain improvements in both ride comfort and vehicle attitude characteristics. Moreover, vehicle vibration control for half and whole car models and vehicle motion control accompanying steering maneuvers are also discussed. These ideal control theories are applied to the real controls in the following chapters.

In CHAPTER 4, "STUDY OF SEMI-ACTIVE VIBRATION CONTROL", in preparation for achieving the optimal control mentioned in chapter 3, a system configuration of semi-active vibration control using a newly designed continuously controlled damper is proposed. In this method the damping valve is controlled successively according to the simultaneous changes due to state variables, which differs from the conventional multi-stage damping control. Analysis of characteristics and optimization of the damper valve are discussed here and predictive analysis is also shown to be effective for vibration suppression of the vehicle. Moreover, the effect of the system is verified experimentally using an apparatus with two degrees of freedom.

In CHAPTER 5, "STUDY OF ACTIVE VIBRATION CONTROL", aiming at the low frequency vibration control of the vehicle that could not be realized by adopting the above-mentioned methods, the control of the vehicle attitude is designed using compensating methods for the delay of the active control elements. The system configuration for active attitude control and control system with combination of feedforward and feedback loops is introduced. After showing the
predicted analysis for the control effects, experimental verification of this system's practicality is proposed by real running tests using an experimental vehicle.

In CHAPTER 6, "STUDY OF ACTIVE MOTION CONTROL", the active control of vehicle handling is discussed in regard to the vehicle attitude control during handling as mentioned in chapter 5. The control for stability and controllability is often evaluated by the closed loop between driver and vehicle system. The control system is therefore designed to include the dynamic characteristics of the driver. Control methods are shown, and the predicted analysis is offered for the effectiveness for stability and steerability of the vehicle during handling maneuvers. The effect of the control is verified by the fact that the controlled experimental vehicle is preferred to the non-controlled vehicle and the system can be constructed by simply adding active control to the ordinary steering systems.

In CHAPTER 7, "EXPERIMENTAL STUDY OF ACTIVE CONTROL", the adaptive control system is proposed with a hierarchical active control system utilizing the three kinds of controls mentioned in chapters 4 to 6. The system combines active control of steering and suspension and semi-active control of suspension damper. It is constructed with effective energy dissipation and priorities for control according to vehicle running status. The system's superiority is confirmed experimentally by using an experimental vehicle, and the comfort and ease of steering are also derived from these results.

In CHAPTER 8 as conclusion, it has been shown that the remarkable combined effects on vehicle vibration and motion achieved by the total combined control strategies of integrated and hierarchical systems cannot be realized by the conventional individual control schemes.
Several control methods in this system have been applied to practical vehicles and are utilized for the vehicle safety and comfort. The methods proposed in this study for measuring tire uniformity and road profile could also be used in many other practical applications. These techniques are promoted and utilized for extending the existing database and in the quality evaluation of vehicle ride comfort.

The following areas need to be considered for future study:

In this study, various conditions of the vehicle environment and certain driving conditions could not be treated. The individual control system is based on linear control theories. Moreover, the optimization for the system elements should be considered from the point of view of energy consumption.

As to the design of the control systems, all control methods need to adapt to the non-linear characteristics of tire and other complicated, critical areas. More suitable performance indexes for the control, which are more representative for human sensitivity are needed.

For the practical application of this study, simplification of the methods is needed to decrease the number of sensors, which was partially tried by the attempt to make use of state observers. The development of failure detection and an isolation system for the sensors and control computer is also needed for practical use of these control methods. A review control for compensating the response of the actively controlled element is still needed for more advanced controls.

Finally, we need to understand the necessity of a new concept of design for the so-called man-machine system, namely how to compromise and optimize the relationship between the human operator as driver and the machine system as vehicle.