

ANALYSIS OF VEHICLE SUSPENSION SYSTEM SUBJECTED TO FORCED VIBRATION USING MAT LAB/SIMULINK

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ABSTRACT

A safe vehicle must be able to stop and maneuver over a wide range of road conditions. Good contact between the tires and the road will be able to stop and maneuver quickly and it is achieved by vehicle suspension system. Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. Shock absorber is an important part of automotive suspension system which has an effect on ride characteristics. Shock absorbers are also critical for tire to road contact which to reduce the tendency of a tire to lift off the road. This affects braking, steering, cornering and overall stability. The removal of the shock absorber from suspension can cause the vehicle bounce up and down. It is possible for the vehicle to be driven, but if the suspension drops from the driving over a severe bump, the rear spring can fall out. The main role of a suspension engineer is to tune the suspension in such a way as to maximize the road holding performance of the vehicle. One of the more difficult components in achieving this is the selection of the dampers which was done empirically in the past. This paper outlines the development of the equations of motion for some simple vehicle models and demonstrates how the increasing availability of numerical simulation software's MATLAB could be used to solve these equations to optimize the vehicle before it arrives at the road.

1. Introduction

Suspension systems have been widely applied to vehicles, from the horse-drawn carriage with flexible leaf springs fixed in the four corners, to the modern automobile with complex control algorithms. The suspension of a road vehicle is usually designed with two objectives; to isolate the vehicle body from road irregularities and to maintain contact of the



wheels with the roadway. Isolation is achieved by the use of springs and dampers and by rubber mountings at the connections of the individual suspension components. From a system design point of view, there are two main categories of disturbances on a vehicle, namely road and load disturbances. Road disturbances have the characteristics of large magnitude in low frequency (such as hills) and small magnitude in high frequency (such as road roughness). Load disturbances include the variation of loads induced by accelerating, braking and cornering. Therefore, a good suspension design is concerned with disturbance rejection from these disturbances to the outputs. Roughly speaking, a conventional suspension needs to be “soft” to insulate against road disturbances and “hard” to insulate against load disturbances. Therefore, suspension design is an art of compromise between these two goals (Wang2001).

Today, nearly all passenger cars and light trucks use independent front suspensions, because of the better resistance to vibrations. One of the commonly used independent front suspension system is referred as double wishbone suspension. Described a method for optimization of the motion characteristics of a double wishbone front suspension system by using a genetic algorithm. The analysis considered only the kinematics of the system (Esat 1999). T.Yamanaka, H.Hoshino, K. Motoyama developed prototype of optimization system for typical double wishbone suspension system based on genetic algorithms. In this system, the suspension system was analyzed and evaluated by mechanical system simulation software ADAMS (Yamanaka, Hoshino and Motoyama 2000). Hazem Ali Attia presented dynamic modeling of the double wishbone motorvehiclesuspension system using the point-joint coordinate's formulation. In his paper, the double wishbone suspension system is replaced by an equivalent constrained system of 10 particles. Then the laws of particle dynamics are used to derive the equations of motion of the system (Attia 2002). The aim of this study is to find the effects of damping constant on the vehicle vibrations and also to obtain the vibration displacements of the car body for different suspension models under typical sinusoidal base excitations. Dynamic analyses of these models are investigated by the analytic method. MATLAB software is used for numerical calculations.

2. Working Principle

The primary functions of a vehicle's suspension systems are to isolate the structure and the occupants from shocks and vibrations generated by the road surface. The suspension systems basically consist of all the elements that provide the connection between the tires and the vehicle body and are designed to meet the following requirements: (1) Ride comfort, (2) Road-holding, and (3) Handling. The first requirement mentioned above for the suspension system requires an elastic resistance to absorb the road shocks. This primary function is fulfilled by the suspension springs. Various different types of springs have been used in vehicle suspensions such as leaf springs, helical coil springs, torsion bar springs, air springs,

rubber springs. It is obvious that a suspension system must be able to withstand the loads acting on it. These forces may be in the longitudinal direction such as acceleration and braking forces, in the lateral direction such as cornering forces, and in the vertical direction.

2.1 Solid Axle Suspension Systems

In solid axle suspension systems, wheels are mounted at the ends of a rigid beam so that any movement of one wheel is transmitted to the opposite wheel causing them to steer and camber together. Solid drive axles are used on the rear of many cars and most trucks and on the front of many four-wheel-drive trucks. Solid beam (non-driven) axles are commonly used on the front of heavy trucks where high load-carrying capacity is required. Solid axles have the advantage that wheel camber is not affected by body roll. Thus there is little wheel camber in cornering, except for that which arises from slightly greater compression of the tires on the outside of the turn. In addition, wheel alignment is readily maintained, minimizing tire wear. The major disadvantage of solid steerable axles is their susceptibility to tramp-shimmy steering vibrations. The most common solid axles are Hotchkiss, Four link and DeDion.

2.2 Independent Suspension Systems

In contrast to solid axles, independent suspensions allow each wheel to move vertically without affecting the opposite wheel. Nearly all passenger cars and light trucks use independent front suspensions, because of the advantages in providing room for the engine and the better resistance to steering vibrations. The independent suspension also has the advantage that it provides inherently higher roll stiffness relative to the vertical spring rate. Further advantages include easy control of the roll centre by choice of the geometry of the control arms, larger suspension deflections, and greater roll stiffness for a given suspension vertical rate. Over the years, many types of independent front suspension have been tried such as Macpherson, Trailing arm, Swing axle, Multi link and Double wishbone suspension. Many of them have been discarded for a variety of reasons, with only two basic concepts, the double wishbone and the Macpherson strut, finding widespread success in many varied forms. The most common design for the front suspension of American car following World War II used two lateral control arms to hold the wheel. The upper and lower control arms are usually of unequal length from which the acronym SLA (short-long arm) gets its name. These are often called "A-arms" in the United States and "wishbones" in Britain. This layout sometimes appears with the upper A-arm replaced by a simple link, or the lower arm replaced by a lateral link, the suspensions are functionally similar. The SLA is well adapted to front-engine,

rear-wheel-drive cars because of the package space it provides for the engine oriented in the longitudinal direction.

Design of the geometry for a SLA requires careful refinement to give good performance. The camber geometry of an unequal-arm system can improve camber at the outside wheel by counteracting camber due to body roll, but usually carries with it less-favorable camber at the inside wheel (equal-length parallel arms eliminate the unfavorable condition on the inside wheel but at the loss of camber compensation on the outside wheel). At the same time, the geometry must be selected to minimize tread change to avoid excessive tire wear. The compact design of a coil spring makes it ideal for use in front suspension systems. Two types of coil spring mountings are used. In the first type the spring is positioned between the frame and the lower control arm as shown in Figure 1. This mounting is most often used on cars with a conventional frame or a partial front frame. The second type of mounting is shown in Figure 2. In this mounting, the coil spring is positioned between the upper control arm and a spring tower formed in the inner section of the fender (Remling 1983). The wishbones may or may not be equal or parallel. The wishbones are parallel and equal in length as shown in Figure 3.(a). The parallel and unequal length wishbone suspension system is shown in Figure 3.(b). A further refinement is the nonparallel, unequal length wishbone suspension system illustrated in Figure 3.

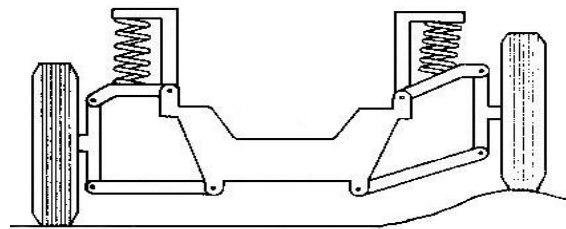


Fig. 1 A first type of independent front suspension system.

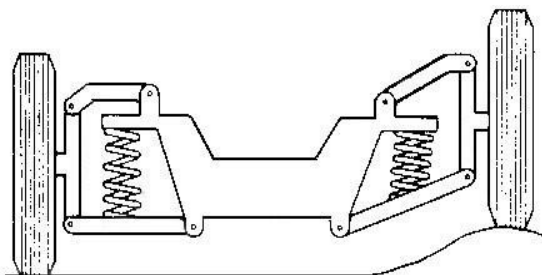


Fig. 2 A second type of independent front suspension system.

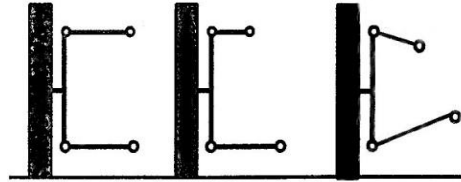


Fig 3 Double wish bone suspension designs(a)Parallel and equal (b) Parallel and unequal (c) N on parallel and un equal.

1. SYSTEM

1.1 Modeling Assumptions

Figure 4 shows a part of a chassis with a double wishbone suspension system. The mechanical system consists of a main chassis, a double wishbone suspension subsystem and a wheel. A suspension spring, lower and upper arms are included in the suspension sub-system. The lower and upper arms are modeled by simple links. The chassis is constrained to move vertically upward or downward. The wheel can be modeled as a linear translational spring. The motion of the wheel over the road provides a vertical input which excites the body of the vehicle. For analysis purpose, the model of the quarter car with the double wishbone suspension assumed to travel with constant velocity on a road surface characterized by a displacement $Z_1(t)$.

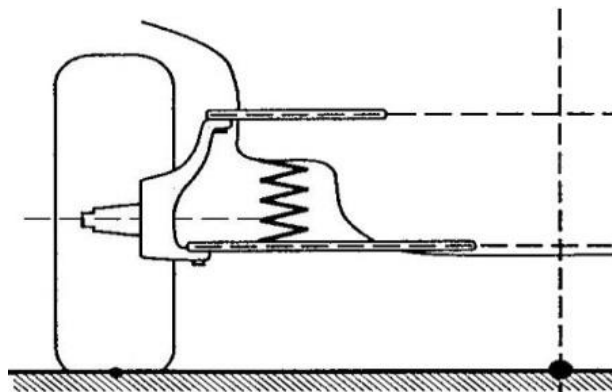


Fig 4.A quarter car with the double wishbone suspension.

1.2 Quarter Car Model

The advantage of the 2-DOF quarter car model is that while still a relatively simple system to analyze, it allows a good approximation of the motion of both the chassis and the wheels of a vehicle.

Fig.5 2-DOF quarter car model.

System Parameter	Value
Sprung Mass (m_s)	450 Kg
Unsprung Mass (m_u)	45 Kg
Suspension Stiffness(k_s)	22000 N/m
Passive Suspension Damping Coefficient(c_s)	2300 N-sec/m
Tire Stiffness (k_t)	176000 N/m
Tire Damping Coefficient (c_t).	230 N-sec/m

$$|H_u(\omega)| = \left| \frac{(-m_1\omega^2 + k_s + c_s j\omega)(k_t + j\omega c_t)}{(-m_u\omega^2 + k_t + c_t j\omega + c_s j\omega)(-m_1\omega^2 + k_s + c_s j\omega) - (k_s + c_s j\omega)^2} \right|$$

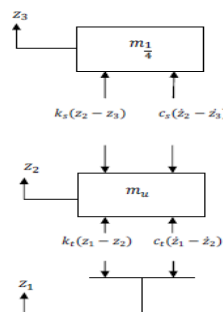


Fig.6 Free Body Diagram of 2-Dof Quarter Car Model

A free body diagram of the quarter car model is given in Figure 6. The degrees of freedom of this system are the vertical displacements z_2 and z_3 of masses m_u and $m_{1/4}$ respectively. The system is subjected to base excitation which is defined by the displacement z_1 . Displacements z_1 , z_2 and z_3 are measured relative to their static equilibrium positions. m_u represents the unsprung mass, which for this model is the mass of the wheel, tyre and a proportion of the suspension linkages, while $m_{1/4}$ refers to the sprung mass, in this case, the remaining mass of the suspension linkages, and a quarter of the mass of the chassis. K_s and K_t refer to the spring rate of the suspension elements and spring rate of the tyre respectively, while C_s and C_t refer to the damping coefficient of the suspension elements and tyre.

1.1 Suspension parameters of quarter car model

Table .1 Suspension Parameters of Quarter Car Model

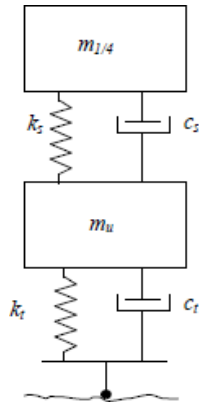
System Parameter	Value
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Using Newton's second law of motion, the equations of motion for this system become:

$$m_u \ddot{z}_2 = k_t(z_1 - z_2) + c_t(\dot{z}_1 - \dot{z}_2) - k_s(z_2 - z_3) - c_s(\dot{z}_2 - \dot{z}_3)$$

$$m_{1/4} \ddot{z}_3 = k_s(z_2 - z_3) + c_s(\dot{z}_2 - \dot{z}_3)$$

$$|H_{\frac{1}{4}}(\omega)| = \left| \frac{(k_t + c_s j \omega)(k_t + j \omega c_t)}{(-m_u \omega^2 + k_t + k_s + j \omega(c_t + c_s))(-m_{1/4} \omega^2 + k_s + c_s j \omega) - (k_s + c_s j \omega)^2} \right|$$



3.4 Frequency response functions

Using the assumed parameters, and the equations derived above it is now possible to draw the frequency response function curves for the various degrees of freedom of the vehicle. This is achieved by solving the equations of motion simultaneously at each value of input frequency. These frequency response functions we are created using MATLAB codes.

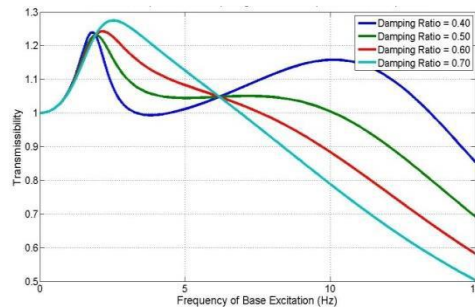


Fig.7 Ratio of amplitude of unsprung mass to amplitude of road profile

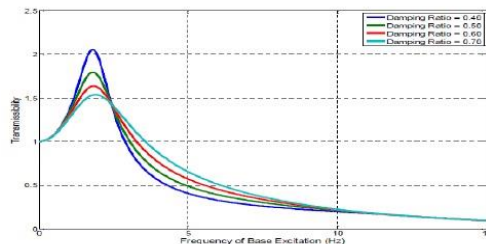


Fig .8 Ratio of Amplitude of Sprung Mass to Amplitude of Road Profile

4. Conclusion

Figure 7 is the frequency response function of the unsprung mass for the 2-DOF quarter car model. Several different damping ratios have been plotted on the same axis to illustrate how the behaviour of the car may change due to different damping ratios. It is clearly visible in this graph that even a small change in damping ratio, from 0.4 to 0.7, can have a very drastic effect on the motion of the unsprung mass. In the frequency range between 0 and approximately 2 Hz, the different damping ratios have very little effect. In the region approximately between 2 and 6 Hz, the lower damping ratio (of 0.4) will result in a lower amplitude of displacement of the unsprung mass. In contrast, in the region beyond 6 Hz, the higher damping ratio of 0.7 will result in less displacement of the unsprung mass. It can also be observed from this figure that with the higher damping ratios there is a single local maximum for the FRF curve. Lowering the damping ratio may add a second local maximum of approximately 11 Hz, and the height of this increases as the amount of damping in the system decreases. The frequency response function of the sprung mass of the quarter car model is shown in Figure 8. It can be seen that there is a single peak in the FRF for all of the damping ratios examined. This peak occurs at approximately the same frequency regardless of the amount of damping applied. One important factor to note is that the lower the damping ratio, the higher the peak amplitude of the FRF. This figure confirms what is already commonly known. That is, that if the vehicle is subjected to random broadband input, most of the motion of the sprung mass will occur at a low frequency and in order to control the movement of the sprung mass, a higher damping ratio is required

References

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