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AXIOMATIC DESIGN OF CUSTOMIZABLE AUTOMOTIVE SUSPENSION

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ABSTRACT

The design of existing suspension systems typically involves a compromise solution for the conflicting requirements of comfort and handling. For instance, cars need a soft suspension for better comfort, whereas a stiff suspension leads to better handling. Cars need high ground clearance on rough terrain, whereas a low center of gravity (CG) height is desired for swift cornering and dynamic stability at high speeds. It is advantageous to have low damping for low force transmission to vehicle frame, whereas high damping is desired for fast decay of oscillations. To avoid these trade-offs, we have proposed a novel design for a customizable automotive suspension system with independent control of stiffness, damping and ride-height, which is capable of providing the desired performance depending on user preference, road conditions and maneuvering inputs. A suspension prototype has been built to demonstrate the concept. Axiomatic design theory was used for the development of the concept, design and fabrication of the prototype and design and implementation of the control system for the suspension system. The mechanical design of the proposed system is decoupled with respect to the functional requirements (FRs) of stiffness and ride-height; moreover ride-height is affected by the load on the vehicle (noise factor). A feedback control system for the customizable suspension was designed and implemented to uncouple the system and to make it robust to the noise factor. With this example, feedback control is proposed as a strategy for converting coupled or decoupled designs to uncoupled designs and for achieving robustness to noise factors.

Keywords: automotive, suspension, axiomatic design, customizable, adaptive, variable stiffness, variable ride-height

1 INTRODUCTION

Active vehicle suspensions have attracted a large number of researchers in the past few decades and comprehensive surveys on related research can be found in the papers by Elbeheiry et al [1995], Hedrick and Wormely [1975], Sharp and Crolla [1987], Karnopp [1995], and Hrovat [1997]. These review papers classify various suspension systems discussed in literature as passive, active, semi-active, slow-active, self-leveling and adaptive systems. In passive systems, the vehicle chassis is supported by only springs and dampers. Active systems (fully active or high frequency active) replace, in part or full, the springs and dampers of passive systems by actuators which act as force producers according to some control law, using feedback from the vehicle. The actuator control bandwidth extends beyond the wheel hop frequency, which is typically 8-10 Hz. Semi-active suspension systems are considered to be derived from active systems, with the actuator replaced by controllable damper (whose force-velocity relation can be modulated at relatively high frequencies) and a spring in parallel. These employ feedback control to track the force demand signal which is similar to a corresponding active system, except that in circumstances where the active system would supply work, the force demanded of the damper is zero. Slow-active systems (low frequency active) use actuator bandwidths in the range of body resonant frequencies in bounce, pitch and roll, and the frequency range of interest as far as responses to steering control are concerned, but lower than the wheel hop frequency. The applicability of fully active suspensions is restricted as the size, weight, power requirements and cost increase prohibitively with the bandwidth of the actuators. Semi-active suspensions have only dissipative elements and slow-active suspensions are band-limited; and hence are limited in their capabilities. This work will look at adaptive suspension systems, which are essentially passive systems in which the parameters of the system can be changed in response to some information. Karnopp and Margolis [1984] have discussed the effects of parameter variation on frequency response and proposed that suspensions with adaptive stiffness and damping coefficient have potential in improvement of ride comfort and handling. Damping control, typically achieved through orifice control, is an established technology in existing vehicles [Karnopp, 1983; Crosby and Karnopp, 1973]. Several road vehicles with pneumatic springs are capable of achieving self-leveling and variable ride-height [Cho and Hedrick, 1985]. Although advantages of variable stiffness have been illustrated in literature [Karnopp and Margolis, 1984], no system with independent control of stiffness has been proposed so far. In this paper, we propose a design for a novel customizable automotive suspension with independent control of stiffness, damping and ride-height, which is capable of providing the desired performance depending on user preference, road conditions and maneuvering inputs. This can be classified as an adaptive suspension with variable ride-height and the novelty of this work is that this is the only system with independent control of stiffness and ride-height.

Moreover this paper proposes the use of feedback control to uncouple a decoupled system and to achieve robustness to noise factors. In earlier research, system-wide rearrangement of leaf-level FR/DP elements as a collective set, to decouple a design that is coupled at a higher level to achieve a non-iterative design process, has been investigated [Melvin and Suh, 2002]. Use of mathematical transforms to achieve uncoupling and robustness has also been proposed [Deo and Suh, 2004]. Suh [2001] has discussed reduction of the sensitivity to noise factors, and Melvin and Deo [2002] have discussed introduction of robustness FRs as strategies for achieving robustness in the axiomatic design framework.

2 INTRODUCTION TO AXIOMATIC DESIGN

Axiomatic Design is a structured design method created to improve design activities by establishing criteria on which potential designs may be evaluated and by developing tools for implementing these criteria. Axiomatic design discusses the existence of four domains in the design world- customer, functional, physical and process domains. Customer attributes {CAs}, functional requirements {FRs}, design parameters {DPs}, and process variables {PVs} are the characteristic vectors of these domains. Design of products involves mapping from the functional domain to the physical domain and design of processes involves mapping from the physical domain to the process domain.

The axiomatic design process is centered on the satisfaction of FRs, which are defined as the minimum set of independent requirements that completely characterize the functional need of the product. Given a minimum set of independent FRs, the designer conceives a physical embodiment or a design containing a set of DPs, which are key physical variables in the physical domain that characterize the design that satisfies the specified FRs. The design and the choice of DPs are guided by the two design axioms.

- Axiom 1: Independence Axiom- Maintain the independence of all functional requirements.
- Axiom 2: Information Axiom- Minimize the information content of the design.

The design matrix (DM) is used to note the effect of DPs on FRs as follows:

$$\begin{Bmatrix} FR1 \\ FR2 \end{Bmatrix} = \begin{bmatrix} A_{11} & O \\ A_{21} & A_{22} \end{bmatrix} \begin{Bmatrix} DP1 \\ DP2 \end{Bmatrix} \tag{1}$$

where A_{11} denotes the effect of DP 1 on FR 1, A_{21} denotes the effect of DP 1 on FR 2, etc. To satisfy the Independence Axiom, the DM must be must be either diagonal or triangular. In an uncoupled design, the DM is diagonal and each of the FRs can be satisfied independently by adjusting one DP. In a decoupled design, the matrix is triangular and the independence of FRs can be guaranteed only if the DPs are determined in a proper sequence. In the case shown, we need to set the DPs in the order: DP 1 followed by DP 2. A full design matrix leads to a coupled design and the satisfaction of FRs becomes difficult.

The Information Axiom guides the designer to maximize the probability of satisfaction of the FRs. It becomes increasingly difficult to satisfy FRs when FRs are coupled by the chosen DPs.

3 MOTIVATION FOR CUSTOMIZABLE SUSPENSION

3.1 NEED FOR VARIABLE STIFFNESS: EFFECT OF STIFFNESS ON SUSPENSION PERFORMANCE

To understand the trade-off between comfort and handling caused by stiffness, we define high-frequency road noise isolation and low-frequency wheel alignment parameter changes as parametric measures of comfort and handling respectively and study the effect of stiffness on these parametric measures.

The orientation of the wheel and wheel axis with respect to the vehicle and the road are described by the wheel alignment parameters such as camber, caster, toe, steering axis inclination etc. Suspension travel causes the wheel alignment parameters to change. This creates lateral forces on the vehicle and may cause directional instability [Gillespie, 1992]. The nature of the tire-road interactions is such that the vehicle is relatively insensitive to the high-frequency wheel alignment parameter changes. Hence low-frequency wheel alignment parameter change is a good parametric measure for directional stability or handling. A stiff suspension reduces suspension travel and the ensuing wheel alignment parameter changes, thereby reducing the destabilizing lateral forces and directional instability.

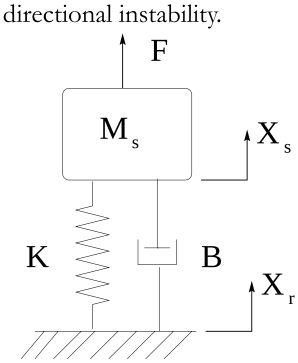

Figure 1: Quarter-car single DOF model

A simple quarter-car single degree of freedom (SDOF) model, as shown in Figure 1, is used to study the response of the sprung mass x_s to the road disturbance x_r and the force F acting on the sprung mass. The governing equation of motion is:

$$M\ddot{x}_s = B(\dot{x}_r - \dot{x}_s) + K(x_r - x_s) + F \tag{2}$$

where M, B and K are the suspension parameters mass, damping coefficient and stiffness respectively. The vertical force F, included in this formulation, could be the weight of the passengers or cargo (to study the static deflection of the vehicle) or the inertial forces acting on the vehicle caused by acceleration, braking or cornering. (For instance F could be the force transferred from the right wheel to left when the car is turning right). Laplace transform of equation 2 leads to the following two transfer functions of interest relating road disturbance x_r, and force F to the chassis displacement x_s. We will compare these two transfer functions for soft and stiff suspensions to understand how soft suspensions provide better comfort, while stiff suspensions provide better handling.

$$X_s = \left(\frac{Bs+K}{Ms^2+Bs+K}\right)\cdot X_r + \left(\frac{1}{Ms^2+Bs+K}\right)\cdot F \tag{3}$$

The first transfer function given by equation 4 shows the effect of road disturbance x_r on the chassis displacement x_s and hence is indicative of road noise isolation or comfort. The Bode plot (Figure 2) for this transfer function for two different values of stiffness shows the same performance at low frequency, but a soft suspension provides better comfort due to better high-frequency road noise isolation. (Note that the damping coefficient B is also changed to maintain the same damping ratio ς in the two cases).

$$\frac{X_s}{X_r} = \frac{Bs+K}{Ms^2+Bs+K} \tag{4}$$

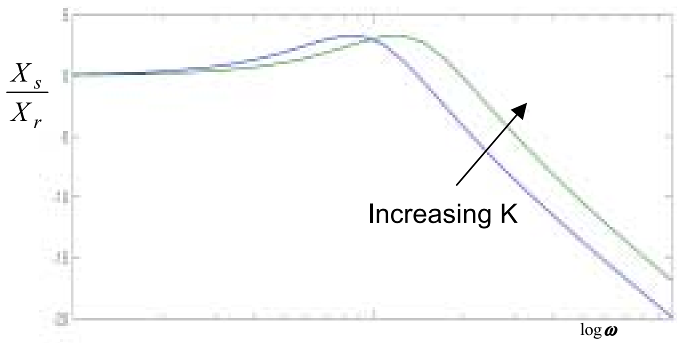

Figure 2: Bode plot showing road noise isolation (comfort) for soft and stiff suspensions

The second transfer function, given by equation 5, shows the effect of the force F on the chassis displacement x_s and hence is indicative of wheel alignment parameter changes or handling. The Bode plot (Figure 3) for this transfer function for two different values of stiffness shows the same response at high frequencies, but a stiff suspension provides better handling as it reduces the low-frequency wheel alignment parameter changes caused by force F (i.e. due to overload, cornering, acceleration or braking). Note that the same damping ratio ς is maintained in the two cases.

$$\frac{X_s}{F} = \frac{1}{Ms^2+Bs+K} \tag{5}$$

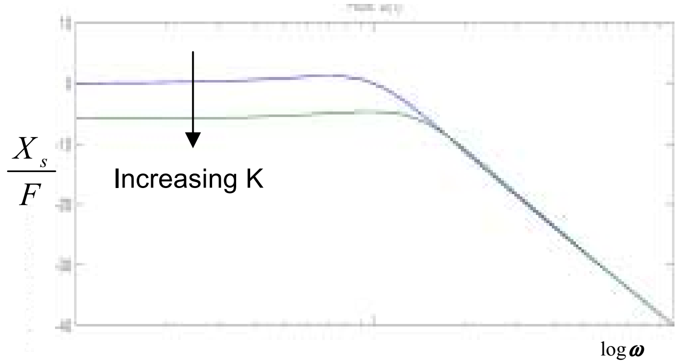

Figure 3: Bode plot showing suspension response to forces for soft and stiff suspensions (handling)

The analysis above shows that a stiff suspension leads to a better handling because of smaller low-frequency wheel alignment parameter changes and a soft suspension leads to better comfort due to better high-frequency road noise isolation. Hence, with a customizable suspension with user-control of stiffness, the user can get the desired performance (say comfort mode or sporty mode) by changing the stiffness.

Road noise is characterized by a certain power spectral density in terms of spatial frequency υ. If the vehicle is driven at constant speed V, the temporal excitation frequency ω is related to the spatial frequency υ by $\omega=2\pi V\upsilon$. The power spectral density in terms of temporal frequency keeps changing with the speed of the vehicle and hence the optimum suspension parameters keep changing with speed. Zuo and Nayfeh [2004] have presented optimum suspension parameters as a function of vehicle speed to minimize the cost function, that includes the requirements of ride comfort, road handling, vehicle attitude and suspension workspace. A suspension with adaptive suspension parameters (damping and stiffness) provides the capability to a get an optimum ride over the entire speed range by changing the suspension parameters as a function of speed according to a suitable algorithm. Adaptive suspension parameters can also be changed based on maneuvering inputs such as steering, braking, or throttle changes.

3.2 NEED FOR VARIABLE RIDE-HEIGHT

Cars need high ground clearance on rough terrain and greater height for better vision; whereas a low center of gravity (CG) height is desired for swift cornering, dynamic stability at high speeds. A suspension capable of ride-height control can avoid this trade-off as the ride-height can be changed on the fly based on user input or automatically based on vehicle speed and maneuvering inputs.

Soft suspension is necessary for good high-frequency road-noise isolation (comfort). We cannot use an excessively soft suspension, because of the disadvantages of excessive unfavorable suspension travel redistribution between jounce and rebound under overload, excessive wheel attitude changes (leading to directional instability) and excessive vehicle attitude changes (leading to passenger discomfort and excess headlight beam swaying). Ride-height control can take care of handling requirements such as low-frequency body and wheel attitude control, and also fix the unfavorable suspension travel redistribution. This allows the use of lower stiffness (as compared to a passive suspension) for better comfort without compromising on handling. This is an example of uncoupling the conflicting requirements of comfort and handling.

3.3 PRIOR ART ON VARIABLE STIFFNESS AND RIDE-HEIGHT

Variable stiffness can be achieved through the use of a non-linear spring such as an air spring, the stiffness of which depends on the equilibrium pressure and volume of the working fluid. We can use the amount of air in the air spring as a DP to achieve variable stiffness. But this DP affects ride height as well, as shown in equation 6. This leads to coupling as the number of FRs exceeds the number of DPs [Suh, 2001] and we cannot satisfy the two FRs of stiffness and ride-height independently. In practice, the DP: Amount of air is varied to get the desired ride-height, and the user has to live with the stiffness that results from it. To overcome this drawback, we have proposed a novel design for a customizable automotive suspension system with independent control of stiffness and ride-height, which is described below.

$$\begin{Bmatrix} \text{FR1: Control Ride Height} \\ \text{FR2: Control Stiffness} \end{Bmatrix} = \begin{bmatrix} X \\ X \end{bmatrix} \{\text{DP: Amount of air}\} \quad (6)$$

4 PROPOSED DESIGN FOR CUSTOMIZABLE SUSPENSION

4.1 CONCEPT DEVELOPMENT

The highest level functional requirements and constraints for a customizable automotive suspension are stated as follows:
Parent FR: Provide a customizable automotive suspension
FR1: Control stiffness
FR2: Control ride-height
FR3: Control damping
Constraint 1: Proposed customizable suspension design must be compatible with existing suspension kinematics.
Constraint 2: Since the size, weight, power requirements and cost of an actuator increase prohibitively with the bandwidth, the bandwidth of the actuators has to be lower than the wheel-hop frequency (8-10 Hz).

Suspension kinematics are designed for desired vehicle dynamics performance, which is often characterized by performance indices such as camber curve, caster curve, anti-pitch, anti-dive, understeer gradient etc. The vehicle dynamics performance is very sensitive to changes in suspension kinematics. Hence it is accepted as a highest level constraint that the proposed modifications, that introduce customization, should not require any change in the existing suspension kinematics. In this paper, we will design the customizable suspension for the SLA (short long arm) suspension, which is the most widely used architecture for front-wheel independent suspensions. SLA suspension system can be kinematically represented as shown in Figure 4.

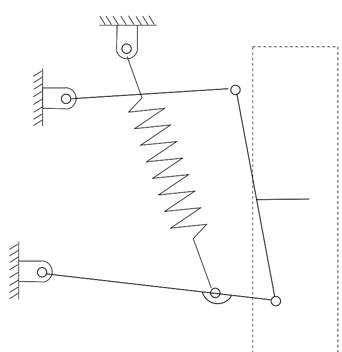

Figure 4: Kinematic representation of existing independent SLA suspension

FR1: Variable stiffness can be achieved by making the lower spring pivot movable along the LCA (lower control arm). Change in the lower spring pivot position alters the effective stiffness seen at the wheel K_w by changing the relation between the wheel travel and spring deflection. The lower spring pivot can be driven by a linear stage, consisting of a stepper motor, a lead screw and a linear bearing as shown in Figure 5. The effective stiffness seen at the wheel K_w is related to the spring stiffness K_s and the lower spring pivot position DP1: x as given by equation 7.

$$K_w = K_s \left(\frac{x}{L} \right)^2 \quad (7)$$

FR2: Variable ride-height can be achieved by making the upper spring pivot movable with respect to the vehicle frame in the vertical direction. There are several ways in which this can be done; a hydraulic actuator or a servo-motor is the most likely choice for actuating the motion of the upper pivot. Figure 5 shows one possible mechanism in which ride-height can be changed by moving the upper spring pivot by a motor driven cam. Movement of the upper pivot (lift of the cam) is used as DP2: U as depicted in the design matrix in equation 8.

FR3: Control damping is achieved by DP3: Orifice control, which is an established technology in existing vehicles. The damper (not shown in the figure for clarity) is connected between the vehicle frame and LCA, in parallel with the spring. Since FR3 is not affected by any other DP and DP3 doesn't affect any other FR, we will neglect this FR/DP pair in the subsequent analysis for simplicity.

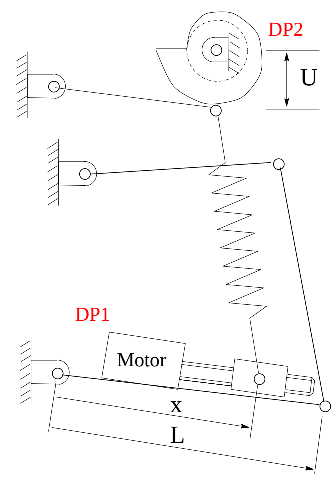

Figure 5: Proposed modifications to achieve independent control of stiffness and ride-height

The DM in equation 8 shows that the design is decoupled with respect to the FRs of stiffness and handling.

$$\begin{Bmatrix} \text{FR1: Stiffness} \\ \text{FR2: Ride-height} \\ \text{FR3: Damping} \end{Bmatrix} = \begin{bmatrix} X & O & O \\ X & X & O \\ O & O & X \end{bmatrix} \begin{Bmatrix} \text{DP1: Pivot position x} \\ \text{DP2: Cam position U} \\ \text{DP3: Orifice control} \end{Bmatrix} \quad (8)$$

4.2 PROTOTYPE DESIGN AND DEVELOPMENT

A 1:1 scale half-car 2-DOF model prototype was designed and built to demonstrate the capabilities of variable stiffness, variable ride-height, and wheel and vehicle attitude control. The suspension kinematics, with SLA architecture, were adopted from an existing vehicle and as per constraint 1, the modifications in the prototype did not change the kinematics.

Several arrangements of linear/rotary bearings and various choices of actuators were considered for the prototype and will be reported elsewhere. For the prototype, kinematics depicted in Figure 5 were used. For changing stiffness, the prototype employs a linear stage on the LCA (lower control arm). The lower spring pivot is pivoted to a carriage supported by a linear bearing, stepper motor, and an ACME screw to avoid back-drivability. The upper pivot is moved by a cam, driven by a stepper motor through a planetary gearhead and a bearing. The upper spring pivot is constrained by the top-arm to follow an arc with the length of the top-arm as the radius. Since the length of the top-arm is significantly greater than the length of travel of the upper pivot, the motion of the upper pivot is very close to a straight line. This arrangement achieves an almost linear motion of the upper pivot without the use of expensive and bulky linear bearings. (A servo motor or a hydraulic actuator is the most likely to be used for ride-height change in an actual system instead of the stepper motor which was used in the prototype for low cost). A roller cam-follower is used to reduce friction and the required torque. The upper and lower spring seats in existing conventional suspension systems are fixed to the chassis and the LCA respectively. The spring seats for the proposed customizable suspension need to have an additional degree to allow for the lower spring pivot motion. The lower spring seat has to be pivoted to the carriage on the linear drive, and the upper spring seat has to be pivoted to the top arm as shown in Figure 5. A picture of the actual prototype is shown in Figures 6.

Controls execution was done using the NI-Motion module in National Instruments™-LabVIEW software. NI PCI-7344 4-axis controller board was used for controlling the four stepper motors for control of stiffness and ride-height for the right and left suspensions of the half-car prototype.

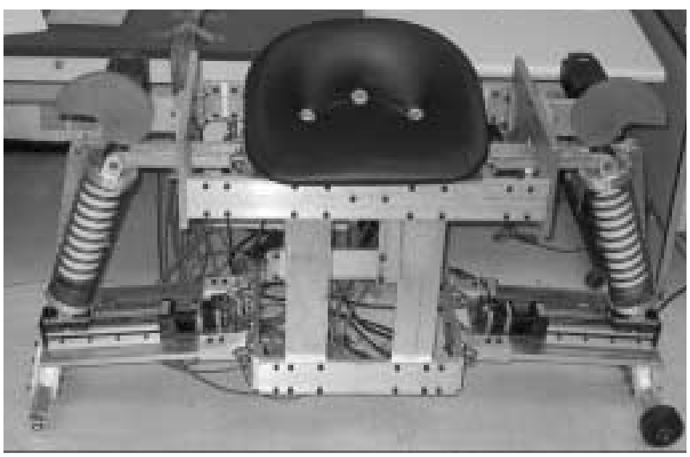

Figure 6 Photograph of customizable suspension prototype (front view)

5 CONTROL SYSTEM DESIGN

5.1 STIFFNESS CONTROL

FR1: Stiffness is not affected by any noise factor; hence we can use open loop control of stiffness. Equation 7 is used to calculate the required position of the lower pivot (DP1: x) from the desired value of FR1: Stiffness K_w, and the controller directs the stepper motor to position the lower spring pivot as required. The desired value of FR1: Stiffness could be input by the user depending on the desired ride (say comfort mode or sporty mode). Alternatively, depending on speed of the vehicle, the road conditions and maneuvering inputs, the stiffness could be automatically set to the optimum value according to a suitable algorithm.

5.2 RIDE-HEIGHT CONTROL

FR2: Ride-height depends not only on the cam position U (DP2), but also on stiffness (hence on DP1), and load on the vehicle (noise factor, which will be denoted subsequently as DP_{nf}). Note that the noise factor DP_{nf} is not a normal design parameter that the user can set to satisfy the FR. It is introduced in the design equation to indicate effect of the noise factor (DP_{nf}) on the FRs [Melvin, 2003]. To study the effect of the DPs on FR2: Ride-height, the system is modeled as a quarter-car single degree of freedom model as shown in Figure 7. The actuator (motor driven cam in this case) is modeled as a low bandwidth displacement provider. The actuator provides displacement U (DP2) in series with the spring. The response of the sprung mass x_s to the road disturbance x_r, the actuator input U, and the force F acting on the sprung mass is given by the equation of motion:

$$M\ddot{x}_s + B\dot{x}_s + Kx_s = B\dot{x}_r + Kx_r + K(U) + F \qquad (9)$$

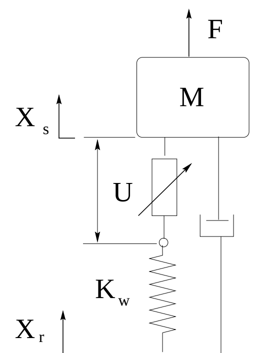

Figure 7: Modeling of the system as a quarter car single DOF model for feedback control of ride-height

Laplace transform of equation 9 gives the following three transfer functions of interest as shown in equation 10, relating road disturbance x_r, actuator input U and force F to the chassis displacement x_s. These are used to construct the block diagram in Figure 8, which will be used as part of the plant to be controlled.

$$X_s = \left(\frac{Bs+K}{Ms^2+Bs+K}\right)\cdot X_r + \left(\frac{K}{Ms^2+Bs+K}\right)\cdot U + \left(\frac{1}{Ms^2+Bs+K}\right)\cdot F \qquad (10)$$

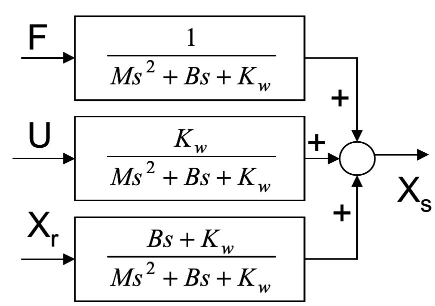

Figure 8: Block diagram representation of the proposed system

$$\begin{Bmatrix} \text{FR1:Stiffness} \\ \text{FR2:Ride-height} \end{Bmatrix} = \begin{bmatrix} O & A & O \\ D & C & B \end{bmatrix} \begin{Bmatrix} DP_{nf}\text{: Load F} \\ \text{DP1: Pivot position x} \\ \text{DP2: Cam position U} \end{Bmatrix} \quad (11)$$

The design matrix in equation 11 shows that this is a decoupled system (due to off-diagonal term C) and FR2: Ride-height is affected by noise factor DP_{nf}: Load on the vehicle (as shown by term D). As a result, any change in DP1: stiffness setting or DP_{nf} : load necessitates a change in DP2 by the user to maintain ride-height at the desired value. To achieve insensitivity to stiffness change and load change, a feedback control system for ride-height was designed as depicted in Figure 9. Since the system is decoupled and we set the DPs in an appropriate order (set stiffness followed by ride-height), we can treat ride-height control as a single-input single-output (SISO) system. This enables the use of classical control techniques, where DP2: U is treated as an input to the plant and K_w and F are treated as noise factors.

The feedback control system consists of a minor loop and a major loop. The minor loop is a motor position control loop, which comprises of the actuator dynamics (modeled as a servo-motor and cam in this case) and the PID controller of the motor, with unity feedback. The minor loop (actuator) accepts the desired value for DP2: U_{des} as input and provides a displacement U in series with the spring. The major loop is the ride-height control loop, which comprises of the plant, the minor loop (actuator) and the controller block. In the major loop, the actual ride-height $(X_s\text{-}X_r)_{actual}$ is measured and compared with the desired ride-height $(X_s\text{-}X_r)_{desired}$. An encoder connected to the suspension UCA (upper control arm) gives a measurement for $(X_s\text{-}X_r)_{actual}$. The controller determines the desired value for DP2: U_{des} according to a control law based on the difference between the actual and desired ride-height values. The controller in the customizable suspension prototype is a PI controller in series with a low pass filter, which are described below.

The plant and the actuator are type-0 systems. A PI controller is used to make it a type-1 system and ensure zero steady state error for a step input.

Suspension motion has two components; the first component is a low frequency component caused by the static load or other low frequency inertial forces acting on the vehicle, and the second component is a high frequency component caused by high frequency road noise. According to the control strategy, we intend to isolate the high frequency road noise passively and use the actuator and control loop to counter the suspension deflection due to low frequency load changes and inertial forces. To filter out the high frequency component of the actual ride-height change: $(X_s\text{-}X_r)_{actual}$ caused due to road-noise, we use a 2nd order low-pass filter with cut-off frequency a. The cut-off frequency a is designed such that the actuator signal U_{des} is not affected by the high-frequency road noise.

After the introduction of the feedback control loop, the resultant system accepts ride-height command $(X_s\text{-}X_r)_{desired}$ as an input from the user and sets FR2: ride-height to that value. Hence we use the ride-height command $(X_s\text{-}X_r)_{desired}$ as DP_{user} 2. The design matrix of the resultant system, given by equation 12, shows that introduction of the feedback control system converts the decoupled system to an uncoupled system. Note that we have used different DPs in equations 11 and 12. In equation 11, cam position U is used as DP2, whereas in equation 12, ride-height command $(X_s\text{-}X_r)_{desired}$ is used as $DP_{user}2$. $DP_{user}2$ is an operational design parameter that the user sets to satisfy the FR and in this example, DP2 is used as an intermediate DP [Deo and Suh, 2004]. The advantage of this formulation is that, with minimal hardware change, the physically decoupled system (equation 11) has been converted to a system (equation 12) that the user sees as uncoupled during the operation of the system! Also equation 12 shows that ride-height is independent of the load on the vehicle and hence this transform has also achieved robustness to a noise factor. Also the imaginary complexity is eliminated as the system appears uncoupled to the user during operation.

$$\begin{Bmatrix} \text{FR1:Stiffness} \\ \text{FR2:Ride-height} \end{Bmatrix} = \begin{bmatrix} O & A & O \\ O & O & 1 \end{bmatrix} \begin{Bmatrix} DP_{nf}\text{: Load F} \\ DP_{user}\text{1: Pivot position x} \\ DP_{user}\text{2: RH } (X_s\text{-}X_r)_{desired} \end{Bmatrix} \quad (12)$$

This example illustrates the utility of axiomatic design theory as a tool for innovation. The existing suspensions with air springs have less number of DPs than FRs as illustrated in equation 6, and hence it is impossible to satisfy the two FRs independently. By following axiomatic design principles, we were able to achieve a novel decoupled design as illustrated in equation 11. This design requires compensation for ride-height whenever stiffness or load on the vehicle changes. By designing a feedback control system, we have converted the decoupled design to an uncoupled design and also achieved robustness to noise factor load as shown in equation 12.

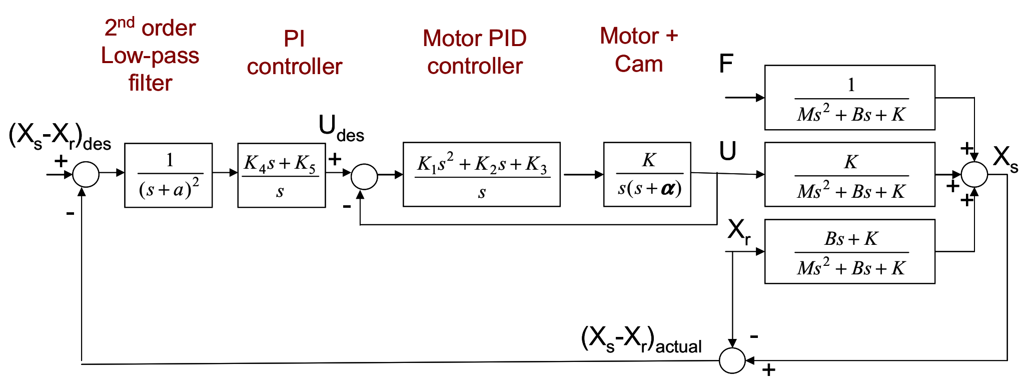

Figure 9: Ride-height feedback control system

6 CONCLUSIONS

This paper makes the following two major contributions:

1. Advantages of a customizable suspension have long been known, but no system with independent control of stiffness and ride-height has so far been proposed. The first major contribution of this work is the proposal of a novel suspension system with independent control of stiffness, damping and ride-height. The use of axiomatic design theory in the concept development and design and fabrication of the prototype is discussed. This example illustrates the use of axiomatic design theory as a tool for innovation.
2. The second major contribution of this work is the proposal of a method to uncouple a decoupled or coupled design and also to achieve robustness to noise factors, by superimposing it with a feedback control system. The mechanical design of the customizable suspension prototype is decoupled with respect to the FRs of ride-height and stiffness; moreover ride-height is affected by load (noise factor). Through the design and implementation of a feedback control system, insensitivity to stiffness change and load change is demonstrated.

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