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Axiomatic Design of Automobile Suspension and Steering Systems: Proposal for a novel six-bar suspension

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ABSTRACT

The existing vehicle designs exhibit a high level of coupling. For instance the coupling in the suspension and steering systems manifests itself through the change in wheel alignment parameters (WAP) due to suspension travel. This change in the WAP causes directional instability and tire-wear. The approach of the industry to solve this problem has been twofold. The first approach has been optimization of suspension link lengths to reduce the change in WAP to zero. Since this is not possible with the existing architecture, the solution used is the optimization of the spring stiffness K to get a compromise solution for comfort (which requires significant suspension travel and hence a soft spring) and directional stability (which demands least possible change in wheel alignment parameters and hence a stiff spring).

This paper presents an axiomatic design solution to this problem and an attempt to remove the coupling in the steering and suspension systems by making the WAP independent of suspension travel. The four-bar linkages used in the existing independent suspension systems are incapable of satisfying their FRs and cause coupling at a higher level. The proposed solution uses a six-bar Watt-I linkage suspension, which removes the coupling. It offers other advantages like the *hardening characteristics* for the suspension. A new steering system conformal to the new suspension system has been proposed.

FR/DP decomposition of the vehicle systems is presented. This indicates other couplings and DP redundancies in the vehicle system and also provides the framework for design of novel vehicles.

INTRODUCTION

INTRODUCTION TO AXIOMATIC DESIGN [1, 2]

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:

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

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. This is because the allowable tolerance for DPs decreases with the increase in the number of FRs and the number of off-diagonal elements in the design matrix.

WHEEL ALIGNEMENT PARAMETERS [3, 5, 8]

Orientation of the wheels and steering axes with respect to the vehicle frame and with respect to the terrain changes due to suspension travel. Figure 1 shows the wheel alignment parameters which describe the orientation of the wheel and the wheel axis. Excess camber causes tire wear and camber spread causes directional instability. Caster spread causes directional instability. Toe change due to suspension travel causes *Bump Steer* and excess toe causes tire-wear. Because of these factors vehicles exhibit tire-wear and directional instability due to suspension travel under conditions of overload, offset load or road undulations.

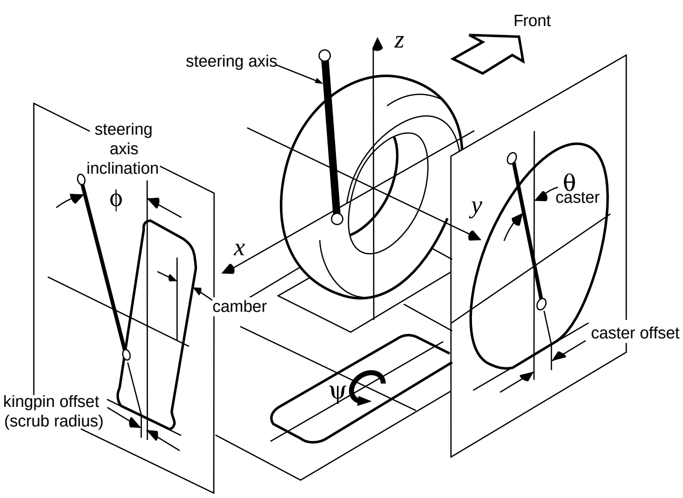

Figure 1 Wheel Alignment Parameters

FRDP DECOMPOSITION OF VEHICLE SYSTEM

Table 1 shows the top level FR/DP decomposition for the vehicle system and the design matrix (DM) is shown in Equation 2. The DM indicates two sets of couplings- coupling between FR 13 and FR 15 and coupling between FR 14 and FR 15. The effect of DP 15 on FR 13 is small (as indicated by x in the DM in Equation 2) and the design works in spite of this coupling due to the presence of a feedback control system- the driver. Identification and removal of the coupling between FR 14 and FR 15 is the subject matter of this paper. These two FRs have been decomposed further to understand this coupling better.

Table 2 shows the decomposition of FR 14: Hold passengers. As indicated in the decomposition, the passenger compartment must provide safety, comfort and pleasure to the passenger. FR/DP decomposition for FR 142: Ensure comfortable ride is shown in Table 3. The corresponding DM is given in Equation 3.

Table 1: Top level FR/DP decomposition of vehicle system

	Functional Requirements	Design Parameters
Parent	**Need for transportation**	**Vehicle system**
11	Allow low resistance motion	Rolling motion (Wheels)
12	Hold cargo	Cargo space
13	Control speed	Wheel rotation speed
14	Hold passengers	Passenger space
15	Control direction	Turning torque
16	Attractive appearance	Exterior Bodywork

$$\begin{Bmatrix} FR11 \\ FR12 \\ FR13 \\ FR14 \\ FR15 \\ FR16 \end{Bmatrix} = \begin{bmatrix} X & O & O & O & O & O \\ X & X & O & O & O & O \\ X & O & X & O & x & O \\ X & O & X & X & X & O \\ X & O & X & X & X & O \\ X & X & O & X & X & X \end{bmatrix} \begin{Bmatrix} DP11 \\ DP12 \\ DP13 \\ DP14 \\ DP15 \\ DP16 \end{Bmatrix} \ldots (2)$$

Table 2 : FR/DP decomposition of FR14 (Hold passengers)

	Functional Requirements	Design Parameters
Parent	**Hold passenger**	**Passenger compartment**
141	Provide crash protection	Impact strength
142	Ensure comfortable ride	Suspension dynamics
143	Provide pleasing environment	Interior design

Table 3 : FR/DP decomposition of FR142

	Functional Requirements	Design Parameters
Parent	**Ensure comfortable ride**	**Suspension dynamics**
1421	Limit maximum relative motion	Spring rate
1422	Dissipate energy	Damping coefficient
1423	Set equilibrium position	Spring initial length

$$\begin{Bmatrix} FR1421 \\ FR1422 \\ FR1423 \end{Bmatrix} = \begin{bmatrix} X & O & O \\ X & X & O \\ X & O & X \end{bmatrix} \begin{Bmatrix} DP1421 \\ DP1422 \\ DP1423 \end{Bmatrix} \ldots (3)$$

Table 4 shows the decomposition of FR 15 (Control direction) and the corresponding design matrix is given in Equation 4.

Table 4 : FR/DP decomposition of FR15

	Functional Requirements	Design Parameters
Parent	Control direction	Turning torque
151	Maintain wheel alignment	Suspension kinematics
152	Maintain tire-road contact	Suspension travel
153	Adjust desired torque	Wheel angle

$$\begin{Bmatrix} FR151 \\ FR152 \\ FR153 \end{Bmatrix} = \begin{bmatrix} X & O & O \\ X & X & O \\ X & O & X \end{bmatrix} \begin{Bmatrix} DP151 \\ DP152 \\ DP153 \end{Bmatrix} \ldots (4)$$

In several complex systems, a coupled DM at the highest level may be decoupled by system-wide rearrangement of the DPs and FRs [4]. It was observed that it is difficult to come up with a design which would remove the coupling between FR 14 and FR 15 at the highest level. But it is possible to have a decoupled system after decomposition. To illustrate this, FR 142 and FR 15 are decomposed together and the corresponding DM presented in Equation 5.

$$\begin{Bmatrix} FR151 \\ FR152 \\ FR1421 \\ FR1422 \\ FR1423 \\ FR153 \end{Bmatrix} = \begin{bmatrix} X & X_1 & X_3 & O & O & O \\ X & X & O & O & O & O \\ X & O & X & O & O & O \\ X & O & X & X & O & O \\ X & O & X & O & X & O \\ X & X_2 & X_4 & O & O & X \end{bmatrix} \begin{Bmatrix} DP151 \\ DP152 \\ DP1421 \\ DP1422 \\ DP1423 \\ DP153 \end{Bmatrix} \ldots (5)$$

EXISTING DESIGNS: IDENTIFICATION OF COUPLING

The elements X_1, X_2, and X_3 in the DM in Equation 5, indicate the coupling. X_1 indicates that suspension travel causes the WAP to change and this causes unwanted turning torque changes, as indicated by X_2. The extent to which the WAP change and hence the magnitude of the unwanted turning torque change depends on the spring stiffness as indicated by the elements X_3 and X_4. Note that X_2 being non-zero does not make the DM coupled. But DP 152 (Suspension travel) is a dynamic DP and it affects FR 153 (Adjust desired torque). Hence, to satisfy FR 153, we would require real-time adjustment of DP 153 (Wheel angle). To avoid this, we need a design that is uncoupled with respect to the dynamic design parameter DP 152 (X_1=0, X_2=0).

These elements can be made zero and coupling can be removed by making the WAP independent of suspension travel. This section examines the change in WAP due to suspension travel in the existing designs, lists the problems due to this coupling and explores the possibility of removing this coupling by making the WAP independent of suspension travel.

EXISTING SUSPENSION SYSTEMS: FOUR-BAR LINKAGES

All existing front-wheel independent suspension systems are variations of the four-bar mechanism. For instance, the parallel arm suspension, the short long arm (SLA) suspension and the McPherson strut suspension can be kinematically represented as shown in Figure 2.

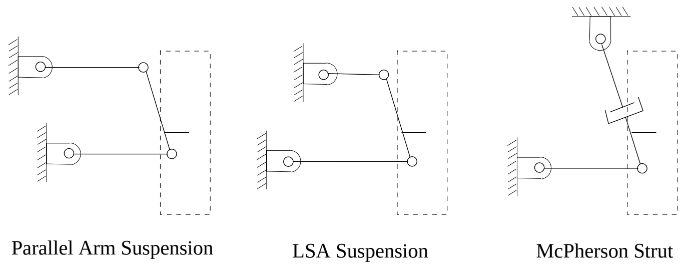

Figure 2: Kinematic representation of independent suspensions

The FRs that DP 151: Suspension Kinematics (Four-bar linkage) is expected to satisfy are given in the form of FR/DP decomposition in Table 5. We will only look into the following three important FRs for simplifying the analysis in this paper: Provide relative Z-motion, Avoid track changes (Δy=0) and Avoid camber and caster changes (Δø=0). Here Δy indicates tire scrub and Δø indicates camber change. The other FRs are uncoupled and can be easily satisfied.

Table 5 : FR/DP decomposition of FR151 (Maintain wheel alignment)

	Functional Requirements	Design Parameters
Parent	Maintain wheel alignment	Suspension kinematics
1511	Permit relative Z-motion	Single degree of freedom system
1512	Avoid track changes (Δy=0)	Effective swing axle radius
1513	Avoid camber and caster changes (Δø=0)	Equal motion of steering axis joints

Analysis of the parallel arm suspension shows that it is capable of providing relative Z motion by change of angle θ as shown in Figure 4 and can maintain Δø=0 as both joints of the steering axis have equal vertical motion during suspension travel. But the parallel-arm suspension is incapable of satisfying FR 1512: Δy=0 during suspension travel. This causes tire-scrub due to suspension travel as illustrated in Figure 4.

In the SLA suspension, we can achieve Δy=0 (no tire scrub) through assignment of appropriate values to the link lengths, but this doesn't allow ∆ø=0 during suspension travel. This causes camber change and caster change due to suspension travel [5]. A compromise solution for ∆y and ∆ø can be obtained through optimization of the link lengths and joint positions, but we cannot satisfy all three FRs simultaneously using a four bar linkage. Both ∆y and ∆ø can be reduced by increasing the link lengths, but this is limited by the constraints of cost, packaging and unsprung weight of the vehicle

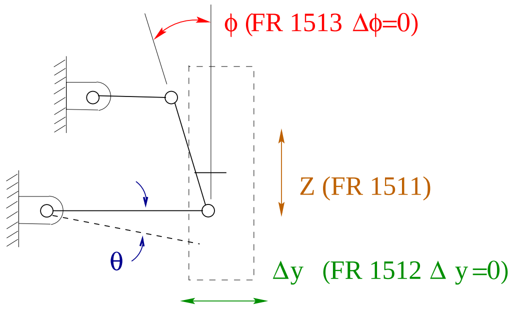

Figure 3: Representation of the FRs of the suspension system

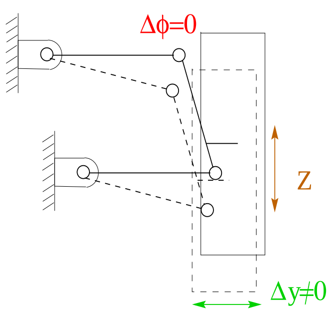

Figure 4: Kinematic representation of parallel-arm suspension showing tire-scrub

The McPherson strut suspension, also a four-bar linkage with one prismatic joint, is incapable of satisfying the three FRs simultaneously as well. It exhibits tire-scrub as well as WAP changes due to suspension travel.

This implies that in the existing designs, suspension travel affects the WAP. This makes the system coupled as indicated by the elements X_1, X_2 and X_3 in Equation 5. This coupling leads to several problems. The changes in camber angle and toe due to excess suspension travel under overload causes unnecessary tire-wear. This could be a serious issue in trucks as the WAP could change significantly from unloaded to fully loaded condition. Under offset load, different suspension travel for the wheels could cause camber spread, caster spread or toe spread leading to directional instability or *Drift/pull* of the vehicle. Toe change due to suspension travel causes *Bump Steer* due to road undulations. Toe change due to suspension travel is also a possible source for the *Nibble* problem, in which the high frequency road noises are transmitted back to the steering wheel.

Manifestation of coupling in existing systems

Very often in coupled designs, when one DP affects two or more FRs, these FRs require the DP to have different values. This leads to a trade-off between the conflicting FRs and the designer has to resort to optimization of the DPs to achieve the best compromise solution. The coupling in the automobile suspension and steering system is manifested by the following trade-offs to achieve compromise solutions:

1. Compromise between ∆ø=0 and ∆y=0 through optimization of link lengths.
2. Compromise between control and comfort through optimization of spring stiffness K. Control demands a stiff suspension, whereas comfortable ride demands a soft suspension.

Since the existing designs cannot make the WAP independent of suspension travel, optimization of the spring stiffness has been the approach of the industry to get a compromise solution for FRs of comfort and control. The axiomatic design approach points out the coupling between the FRs and indicates the need for developing a new uncoupled solution so that we do not have to live with the compromise solutions indicated above. Axiomatic design theory suggests the need for a new design that can satisfy both ∆ø=0 and ∆y=0 simultaneously. Such a design would make the WAP independent of suspension travel and spring stiffness. Hence the control of the vehicle will improve. Spring stiffness can be designed only from comfort considerations and this will also improve passenger comfort.

PROPOSED NEW DESIGN

This section discusses the new suspension and steering system proposed to remove the identified coupling.

PROPOSED DESIGN OF A SUSPENSION SYSTEM

Analysis in the previous section indicated that the four-bar linkage is incapable of satisfying all three FRs (provide suspension travel, maintain ∆ø=0 and maintain ∆y=0) simultaneously. This leads to coupling at a higher level. The DM indicates that the coupling can be removed by a change of DP 151: Suspension kinematics (Four-bar mechanism). A decision was made to change to DP 151: Suspension kinematics (Single degree of freedom system). Stating this as the DP presents several single degree of freedom systems as options for the hardware of the suspension kinematics, namely- a single revolute or prismatic joint, six-bar linkage and so on, apart from the four-bar linkage.

A prismatic joint is used in two wheeler suspensions, but there are issues involved in incorporating it in an automobile suspension. Revolute joint is used in the swing-axle suspension and it is capable of meeting only one FR out of the three FRs simultaneously. In a swing-axle suspension, both camber and track changes during suspension travel (neither $\Delta y=0$, nor $\Delta ø=0$).

The five possible inversions of a six-bar linkage using all revolute joints are shown in Figure 5. Investigation revealed that the Stephenson chains and the Watt-II chain are incapable of satisfying the three FRs simultaneously. Hence we proceed with the dimensional synthesis of the Watt-I linkage to see if it can satisfy the three FRs simultaneously.

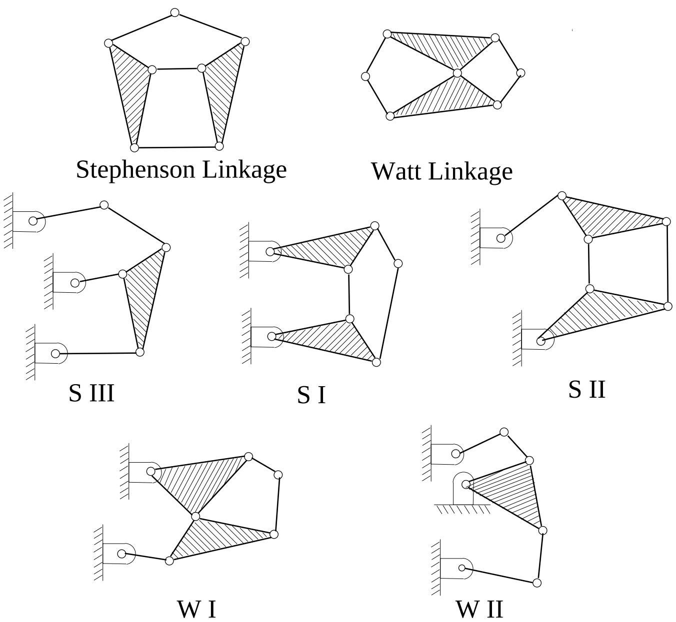

Figure 5: Different possible six-bar mechanisms

DIMENSIONAL SYNTHESIS OF WATT-I LINKAGE FOR THE SUSPENSION

Complex number representation is very convenient to model linkage members and their motions, and is very commonly used in kinematic analysis and synthesis [6]. A great majority of planar linkages can be thought of as combinations of vector pairs called *dyads*. For instance the four-bar linkage in Figure 8 can be perceived of as two dyads: the left side of the linkage represented as a vector pair **W** and **Z**, and the right side represented by the dyad **W*** and **Z***. The path point P of the coupler moves along a path from position P_1 to P_j defined in an arbitrary complex coordinate system by $\mathbf{R}_1$ and $\mathbf{R}_j$.

Suppose we specify two positions for an unknown dyad by prescribing the values of $\mathbf{R}_1$, $\mathbf{R}_j$ α_j and β_j. To find the unknown starting position vectors of the dyad **W** and **Z**, a loop-closure equation may be derived by summing the vectors clockwise around the loop containing $\mathbf{W}\exp\{i\alpha_j\}$, $\mathbf{Z}\exp\{i\beta_j\}$, **W** and **Z**:

$$\mathbf{W}\,(e^{i\boldsymbol{\alpha}_j}-1)+\mathbf{Z}(e^{i\boldsymbol{\beta}_j}-1)=\boldsymbol{\delta}_j \cdots (6)$$

where $\boldsymbol{\delta}_j = \mathbf{R}_j - \mathbf{R}_1$ s the displacement vector along the prescribed trajectory from P_1 to P_j. This equation is called the *standard-form* equation and is simply the vector sum around the loop containing the first and the j[th] positions of the dyad forming the left side of the four-bar linkage. This synthesis technique is well developed for synthesis of four-bar mechanisms for different tasks: Path generation, motion generation and function generation [6]. It involves expressing the desired motions and/or angular displacements of the input, output or coupler links (depending on the application) in the *standard-form* equation and solving these equations to get the desired link lengths. This enables us to get an analytical solution to the synthesis problem. The next section discusses the application of this technique for the synthesis of the suspension system from the Watt-I linkage

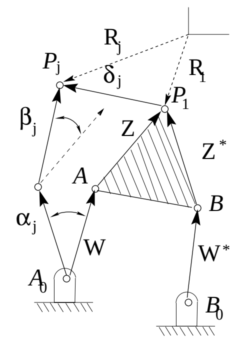

Figure 6 : Notation for the dyad

APPLICATION TO SUSPENSION DESIGN

Table 6 shows the FRs and the constraints that DP 151: Suspension kinematics is supposed to satisfy. Figure 7 shows the application of the dyad technique for the synthesis of the suspension system using the six-bar Watt-I linkage. Here $\boldsymbol{\delta}_j$ indicates the displacement of the steering axis joints A and B. Since $\boldsymbol{\delta}_j$ are same for A and B, $\boldsymbol{\delta}_j$ is also the displacement of the steering axis as a whole. $\mathbf{Z}_1$ through $\mathbf{Z}_6$ are vectors which characterize the link lengths and the existing state of the mechanism. The angular rotations of the links 1 to 4 are given by α_i, β_i, γ_i and θ_i respectively as shown in Figure 7.

Table 6 : FR/DP decomposition of FR 151 (Maintain wheel alignment)

	Functional Requirements	Design Parameters
Parent	Maintain wheel alignment	Suspension kinematics
1511	Permit relative Z-motion	Single degree of freedom system
1512	Avoid track changes ($\Delta y=0$)	$Im(\boldsymbol{\delta}_j)=0$
1513	Avoid camber and caster changes ($\Delta ø=0$)	Equal motion of steering axis joints
1514	Hardening characteristics	α_i

$$\begin{Bmatrix} FR1511 \\ FR1512 \\ FR1513 \\ FR1514 \end{Bmatrix} = \begin{bmatrix} X & O & O & O \\ O & X & O & O \\ O & O & X & O \\ X & O & O & X \end{bmatrix} \begin{Bmatrix} DP1511 \\ DP1512 \\ DP1513 \\ DP1514 \end{Bmatrix} \ldots (7)$$

The conditions imposed to meet the FRs are incorporated in the synthesis technique. The six-bar Watt-I linkage is a single degree of freedom system and we can attain relative Z-motion (FR 1511). To avoid camber and caster changes (FR 1513), we specify that the motion of the two steering axis joints A and B be the same (All $\boldsymbol{\delta}_j$ are the same for both joints as shown in Figure 7). To avoid tire-wear (FR 1512), we specify the imaginary parts of all $\boldsymbol{\delta}_j$ to be zero. This ensures the wheel moves straight up and down, and not sideways, ensuring $\Delta y=0$. FR 1514 can be easily met by specifying α_i.

Figure 7 shows four prescribed positions of the two joints of the steering axis. We formulate the *standard-form* equations for the three dyads for the three desired displacements $\boldsymbol{\delta}_1$, $\boldsymbol{\delta}_2$ and $\boldsymbol{\delta}_3$. This gives us the 9 equations given below. Each of these 9 complex equations actually consists of two equations- one equating the real parts and the other equating the imaginary parts of the complex equation. Note that, since $\mathbf{Z}_2$ and $\mathbf{Z}_4$ are a part of the same rigid body, they undergo the same angular displacements β_i. Same is the case for $\mathbf{Z}_3$ and $\mathbf{Z}_5$, which have equal angular displacements γ_i

For the first dyad ($\mathbf{Z}_1$ and $\mathbf{Z}_2$)

$$\begin{aligned} \boldsymbol{\delta}_1 &= \mathbf{Z}_1(e^{i\alpha_1}-1)+\mathbf{Z}_2(e^{i\beta_1}-1) \\ \boldsymbol{\delta}_2 &= \mathbf{Z}_1(e^{i\alpha_2}-1)+\mathbf{Z}_2(e^{i\beta_2}-1) \\ \boldsymbol{\delta}_3 &= \mathbf{Z}_1(e^{i\alpha_3}-1)+\mathbf{Z}_2(e^{i\beta_3}-1) \end{aligned} \ldots (8)$$

For the second dyad ($\mathbf{Z}_3$ and $\mathbf{Z}_4$)

$$\begin{aligned} \boldsymbol{\delta}_1 &= \mathbf{Z}_3(e^{i\gamma_1}-1)+\mathbf{Z}_4(e^{i\beta_1}-1) \\ \boldsymbol{\delta}_2 &= \mathbf{Z}_3(e^{i\gamma_2}-1)+\mathbf{Z}_4(e^{i\beta_2}-1) \\ \boldsymbol{\delta}_3 &= \mathbf{Z}_3(e^{i\gamma_3}-1)+\mathbf{Z}_4(e^{i\beta_3}-1) \end{aligned} \ldots (9)$$

For the third dyad ($\mathbf{Z}_5$ and $\mathbf{Z}_6$)

$$\begin{aligned} \boldsymbol{\delta}_1 &= \mathbf{Z}_5(e^{i\gamma_1}-1)+\mathbf{Z}_6(e^{i\theta_1}-1) \\ \boldsymbol{\delta}_2 &= \mathbf{Z}_5(e^{i\gamma_2}-1)+\mathbf{Z}_6(e^{i\theta_2}-1) \\ \boldsymbol{\delta}_3 &= \mathbf{Z}_5(e^{i\gamma_3}-1)+\mathbf{Z}_6(e^{i\theta_3}-1) \end{aligned} \ldots (10)$$

The values of $\boldsymbol{\delta}_1$, $\boldsymbol{\delta}_2$ and $\boldsymbol{\delta}_3$ depend on the desired suspension travel (FR 1511). We can specify α_1, α_2, and α_3 based on the desired hardening characteristics. $\boldsymbol{\delta}_3/\alpha_3 > \boldsymbol{\delta}_2/\alpha_2$ will give a hardening suspension (FR 1514).

The first set of equations has 6 equations and 7 unknowns, $\mathbf{Z}_1$, $\mathbf{Z}_2$, β_1, β_2 and β_3. $\mathbf{Z}_1$ and $\mathbf{Z}_2$ are planar vectors and hence constitute four unknowns. We can fix one unknown and solve for the other six. This gives us $\mathbf{Z}_1$ and $\mathbf{Z}_2$.

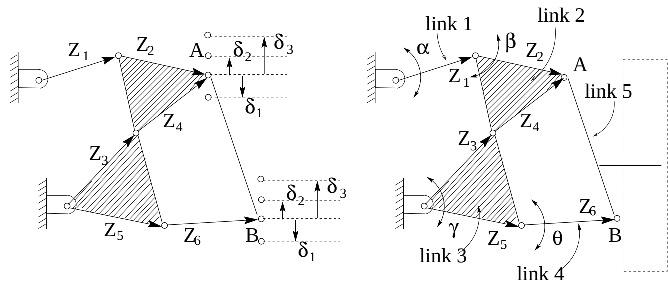

Figure 7 : Dimensional synthesis for the six-bar Watt-I linkage

The second set of equations has β_1, β_2 and β_3 specified from the previous equation. This again is a set of 6 equations with 7 unknowns. We can fix one unknown and solve for the other six to get $\mathbf{Z}_3$ and $\mathbf{Z}_4$. Similarly, we can solve for $\mathbf{Z}_5$ and $\mathbf{Z}_6$ from the third set of equations. The other link lengths can be obtained from vector additions of the known vectors $\mathbf{Z}_1$ through $\mathbf{Z}_6$.

PROPOSED DESIGN OF STEERING SYSTEM

Existing steering system

Figure 8 shows the top view schematic of the existing steering systems. Rotation of the steering wheel causes rotation of the Pitman arm through the steering column and the steering gear. This motion of the Pitman arm is transmitted through the tie rod to the steering knuckle which is rigidly connected to the vehicle wheel. This linkage transmits the motion of the steering wheel to the vehicle wheel. The two joints of the tie-rod are ball and socket joints to allow the suspension travel (in this case in and out of the plane of the paper).

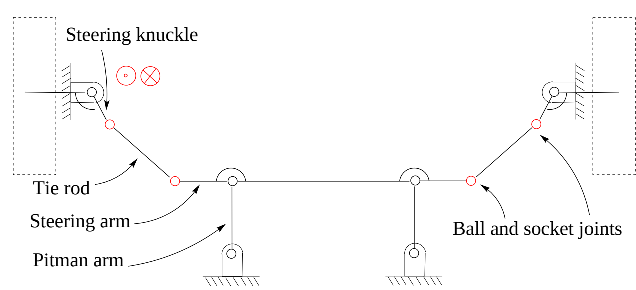

Figure 8: Top view schematic of existing steering system

Need for a new steering system

As discussed in a previous section, toe-changes due to suspension travel in existing suspension systems could cause tire-wear, directional instability, bump steer and the nibble problem. Also suspension travel changes the effective tie-rod length according to the relation

$L=L_1 cos\theta$. The steering mechanism link lengths are designed for, amongst other features, toe-out on turns. Thus the *toe-out on turns* characteristics of the linkage will change under overload load causing unnecessary tire-wear

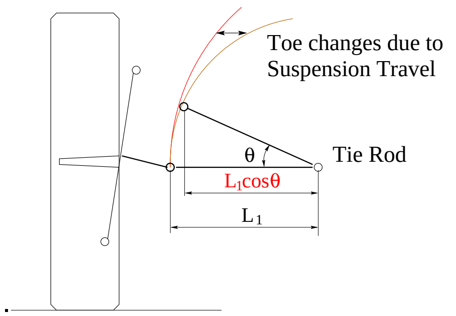

Figure 9 : Suspension travel causes toe-change and change in turning characteristics

Also the existing steering system is incompatible with the new six-bar suspension system as it shows excessive toe-sensitivity as shown in Figure 10 due to the vertical suspension travel.

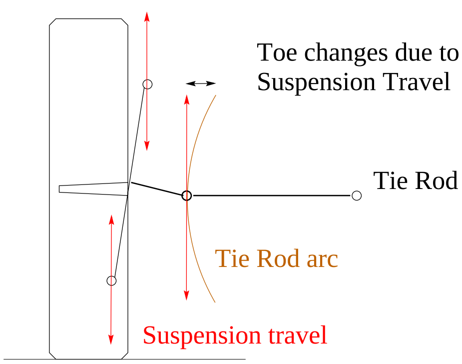

Figure 10 : Existing steering system is incompatible with the proposed suspension systems due to excessive toe-sensitivity

Design of a new steering system

Figure 11 shows the top view schematic of the proposed steering system. The proposed modification has a tie rod with a revolute joint at the inner tie rod end and a cylindrical joint at the outer tie rod end, instead of the ball and socket joints at both tie rod ends. In the new six-bar suspension system, the steering knuckle moves exactly vertically (in the Z-direction). It does not have any horizontal motion or any angle changes. This allows the use a cylindrical joint in place of ball and socket joints.

This system has a drawback that it does not allow the camber to change. We desire a camber roll on turns for suitable turning characteristics [3]. Hence we need to modify this steering system to allow camber roll on turns.

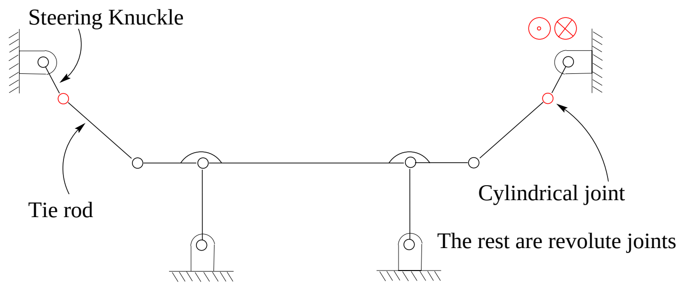

Figure 11 : Top view schematic of the proposed steering system

Modifications to allow camber change

To allow camber changes, a universal joint is introduced between the outer tie-rod end and the cylindrical joint as shown in Figure 12. Thus the steering system does not restrict camber changes and hence allows for camber roll on turns possible. Note that although the steering system does not restrict the camber changes, the suspension system does.

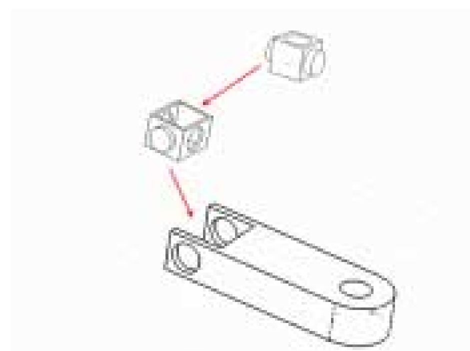

Figure 12 : Modification for camber roll on turns

EVALUATION OF THE PROPOSED DESIGN

ADVANTAGES

Wheel Alignment Parameters: Independent of suspension travel

Equations 5 and 11 show the design matrices for the existing and proposed suspension and steering systems. The elements X_1, X_2 and X_3 in equation 5 indicate the coupling in the existing design. The DM for the proposed design is a lower triangular matrix and hence satisfies the Independence axiom. Moreover, the design is uncoupled with respect to the dynamic DPs, namely DP 152 (Suspension travel) and DP 153 (Wheel angle), i.e. these DPs do not affect any other FR.

A careful look at the DM for the existing systems indicates that we must fix DP 152 before DP 1422 and DP 1423. But this is a contradiction since DP 1422 is static in most cases and DP 1423 is static in all cases, whereas DP 152 is dynamic.

$$\begin{Bmatrix} FR151 \\ FR1421 \\ FR1422 \\ FR1423 \\ FR152 \\ FR153 \end{Bmatrix} = \begin{bmatrix} X & O & O & O & O & O \\ X & X & O & O & O & O \\ X & X & X & O & O & O \\ X & X & O & X & O & O \\ X & O & O & O & X & O \\ X & O & O & O & O & X \end{bmatrix} \begin{Bmatrix} DP151 \\ DP1421 \\ DP1422 \\ DP1423 \\ DP152 \\ DP153 \end{Bmatrix} \dots (11)$$

Both the DMs show that DP 1422 (Damping coefficient) does not affect any FR other that FR 1422. Axiomatic design theory indicates that DP 1422 can be used as a dynamic DP. It is indeed used in variable damping suspension systems and slow-active suspension systems. This is another example of application of axiomatic design theory to facilitate the rapid identification of such novel ideas through the FR/DP decomposition and the DM.

In the proposed system, WAP are independent of suspension travel and hence there are no unwanted turning torque changes (X_1=0, X_2=0). This leads to better control. Since the WAP are independent of suspension, the problems of directional instability and tire-wear due to overload, offset load or road undulations are also eliminated. Since X_3=0 and X_4=0 in the proposed design, spring stiffness does not affect WAP or turning torque. Hence spring stiffness can be designed based only on comfort considerations and independent of control consideration. This will lead to better comfort.

Hardening characteristics

We want suspension to be responsive to the bumps for small displacements, so that we get a smooth ride. But we also want to limit the relative displacements between the wheel and the frame to some reasonable value. This can be achieved if we have a suspension that hardens with displacement.

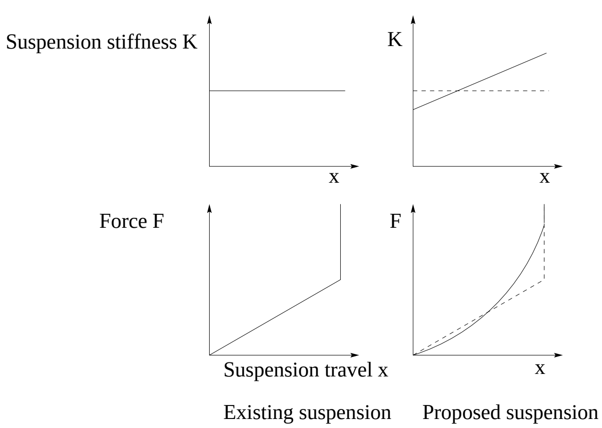

Figure 13 : Hardening characteristics in proposed suspension

In existing suspension systems, the spring compression or extension is directly proportional to suspension travel, giving a linear relation between force and suspension travel as shown in Figure 13. The force increases suddenly when the control arm hits the jounce bumper.

In the proposed six-bar Watt-I linkage, suitable design can achieve a non-linear relation between the suspension travel and spring compression giving rise to a progressive effective spring rate and a hardening suspension as shown in Figure 13.

LIMITATIONS

The six-bar Watt-I linkage has the disadvantages of more number of links and joints, higher cost and higher unsprung weight.

Unfavorable camber changes due to body roll and possible solutions

The proposed suspension maintains $\Delta ø$=0, but ø is measured with respect to the vehicle frame, whereas camber is measured with respect to ground. If the vehicle frame tilts with respect to the ground (exhibits roll), we will have an equal and opposite camber on both wheels, with magnitude equal to the vehicle roll. This leads to a positive camber on the outer wheels, which is unfavorable on turns.

One solution to avoid this problem could be to eliminate body-roll. This can be done by *active roll-stabilization* (ARS). Several existing vehicles have this ARS feature. Incorporating ARS will further improve handling and comfort levels.

CONCLUSION

This paper presents an axiomatic design approach to remove the coupling in the vehicle suspension and steering systems. FR/DP decomposition of the existing suspension and steering system is presented and coupling is identified. A new suspension system has been proposed which removes the coupling by making the wheel alignment parameters independent of suspension travel and hence delivers a better performance in terms of comfort, control and tire-wear. An analytical technique for the kinematic synthesis of the suspension system using a six-bar Watt-I mechanism is presented. A new steering system conformal to the new suspension system has been proposed. FR/DP decomposition of the vehicle system is presented. This indicates other couplings and DP redundancies in the vehicle system and also provides a framework for design of novel vehicles.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

WAP: Wheel Alignment Parameters

FR: Functional Requirement

DP: Design Parameter

DM: Design Matrix