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- Formation Flying Reconfiguration
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World's Largest Passenger Airliner

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Relativistic Elasticity & the Universal Equation of Elasticity for Next Generation Aircrafts & Spacecrafts

By E.G. Ladopoulos
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Abstract - The theory of "Relativistic Elasticity" is proposed for the design of the new generation large aircrafts with turbojet engines and speeds in the range of 50,000 km/h. This theory shows that there is a considerable difference between the absolute stress tensor and the stress tensor of the moving frame even in the range of speeds of 50,000 km/h. For bigger speeds like c/3, c/2 or 3c/4 (c=speed of light), the difference between the two stress tensors is very much increased. Therefore, for the next generation spacecrafts with very high speeds, then the relative stress tensor will be very much different than the absolute stress tensor. Furthermore, for velocities near the speed of light, the values of the relative stress tensor are very much bigger than the corresponding values of the absolute stress tensor. The proposed theory of "Relativistic Elasticity" is a combination between the theories of "Classical Elasticity" and "Special Relativity" and results to the "Universal Equation of Elasticity". For the structural design of the new generation aircrafts and spacecrafts the stress tensor of the airframe will be used in combination to the singular integral equations method. Such a stress tensor is reduced to the solution of a multidimensional singular integral equation and for its numerical evaluation will be used the Singular Integral Operators Method (S.I.O.M.).

Keywords : Relativistic Elasticity, Aircrafts, Spacecrafts, Relative Stress Tensor, Absolute Stress Tensor, Stationary and Moving Frames, Energy-Momentum Tensor, Multidimensional Singular Integral Equations, Singular Integral Operators Method (S.I.O.M.), Universal Equation of Elasticity.

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RELATIVISTIC ELASTICITY THE UNIVERSAL EQUATION OFELASTICITY FOR NEXT GENERATION AIRCRAFTS SPACECRAFTS

Strictly as per the compliance and regulations of:

Relativistic Elasticity & the Universal Equation of Elasticity for Next Generation Aircrafts & Spacecrafts

E.G. Ladopoulos

Abstract - The theory of "Relativistic Elasticity" is proposed for the design of the new generation large aircrafts with turbojet engines and speeds in the range of 50,000 km/h. This theory shows that there is a considerable difference between the absolute stress tensor and the stress tensor of the moving frame even in the range of speeds of 50,000 km/h. For bigger speeds like c/3, c/2 or 3c/4 (c=speed of light), the difference between the two stress tensors is very much increased. Therefore, for the next generation spacecrafts with very high speeds, then the relative stress tensor will be very much different than the absolute stress tensor.

Furthermore, for velocities near the speed of light, the values of the relative stress tensor are very much bigger than the corresponding values of the absolute stress tensor. The proposed theory of "Relativistic Elasticity" is a combination between the theories of "Classical Elasticity" and "Special Relativity" and results to the "Universal Equation of Elasticity". For the structural design of the new generation aircrafts and spacecrafts the stress tensor of the airframe will be used in combination to the singular integral equations method. Such a stress tensor is reduced to the solution of a multidimensional singular integral equation and for its numerical evaluation will be used the Singular Integral Operators Method (S.I.O.M.).

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I. Future Applications of Aircrafts and Spacecrafts Design

The possibilities of turbomachines applied in aircrafts have been very much increased because of the big evolution of the jet engines and the high performance axial – flow compressor. The concern for very light weight in the aircraft propulsion application, and the desire to achieve the highest possible isentropic efficiency by minimizing parasitic losses, led inevitably speed operation. The increasing evolution of aeroelasticity in aircraft turbomachines to axial-flow compressors with cantilever airfoils of high aspect ratio. Also, the turbojet engines were found to experience severe vibration of the rotor blades at part Continues to be under active investigation, driven by the needs of aircraft powerplant and turbine designers.

The target of international Aeronautical Industries is therefore to achieve a competitive technological advantage in certain strategic areas of new and rapidly developing advanced technologies, by which in the longer terms, can be achieved increased market share. This considerably big market share includes the design of a new generation large aircraft with speeds even in the range of 50,000 km/h. The application of new generation turbojet engines makes possible the design of such type of large aircrafts and therefore there is a need of elastic stress analysis for the construction of the total parts of such type of new generation aircrafts.

Furthermore, the target of the International Space Agencies (ESA, NASA, etc.) is to achieve in the future, next generation spacecrafts moving with very high speeds, even approaching the speed of light. In such cases the relative stress tensor will be much different than the absolute stress tensor and so special material will be used for the construction of such spacecrafts. The type of the proper material for the construction of the next generation spacecrafts is under investigation and will be very much different than the usual composite materials.

In the present investigation it will be shown that there is a difference between the absolute stress tensor and the stress tensor of the airframe even in the range of speeds of 50,000 km/h. On the other hand, for bigger speeds the difference of the two stress tensors is very much increased. Thus, for bigger velocities like c/3, c/2 or 3c/4 (c=speed of light) the relative stress tensor is very much different than the absolute one, while for velocities near the speed of light the values of the relative stress tensor are much bigger than the corresponding values of the absolute stress tensor. The study of the connection between the stress tensors of the absolute frame and the airframe is included in the theory proposed by E.G.Ladopoulos [30] - [32] under the term *"Relativistic Elasticity"* and the final formula which results from the above theory is called the *"Universal Equation of Elasticity"*. Hence, in the present study the theory of *"Relativistic Elasticity"* will be applied for the elastic stress analysis design of the next generation aircrafts and spacecrafts.

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Beyond the above, E.G.Ladopoulos [1]-[16] and E.G.Ladopoulos et al. [17]-[22] proposed several linear singular integral equation methods applied to elasticity, plasticity and fracture mechanics applications. In the above studies the Singular Integral Operators Method (S.I.O.M.) is investigated for the numerical evaluation of the multidimensional singular integral equations in which is reduced the stress tensor analysis of the linear elastic or plastic theory. Also, the theory of linear singular integral equations was extended to non-linear singular integral equations, too. [23]-[29]. The theory of *"Relativistic Elasticity"* will be applied to the design of the elastic stress analysis for the airframes. *"Relativistic Elasticity"* is derived as a generalization of the classical theory of elastic stress analysis for stationary frames. For future aerospace applications the difference between the relative and the absolute stress tensors will be of increasing interest. Furthermore, the classical theory of elastic stress analysis began to be analyzed in the early nineteenth century and was further developed in the twentieth century. In the past were written several important monographs on the classical theory of elasticity. [33]- [52}.

On the other hand, during the past years special attention has been concentrated on the theoretical aspects of the special theory of relativity. Hence, some classical monographs were written, dealing with the theoretical foundations and investigations of the special and the general theory of relativity. [53]–[60].Furthermore, a very important point which will be shown in the present research is that the *"relative stress tensor is not symmetrical"*, while, as it is well known, the *"absolute stress tensor is symmetrical"*. This difference is very important for the design of the next generation aircrafts and spacecerafts of very high speeds. Thus, the foundations of the theory of *"Relativistic Elasticity"* for airstructures lead to a general theory, in which no restriction is made with regard to the relative motion. This general theory is further reduced to one class of relative motion, uniform in direction and velocity.

II. Relative Stress Tensor Formulation for Airframes

The state of stress at a point in the stationary frame S^0, is defined by the following symmetrical stress tensor: (Fig.1).

$$\sigma^0 = \begin{bmatrix} \sigma^0_{11} & \sigma^0_{12} & \sigma^0_{13} \\ \sigma^0_{21} & \sigma^0_{22} & \sigma^0_{23} \\ \sigma^0_{31} & \sigma^0_{32} & \sigma^0_{33} \end{bmatrix} \qquad (2.1)$$

Where: $$\sigma^0_{21} = \sigma^0_{12},\ \sigma^0_{31} = \sigma^0_{13},\ \sigma^0_{32} = \sigma^0_{23} \qquad (2.2)$$

Consider an infinitesimal face element df with a directed normal, defined by a unit vector n, at definite point p in the three-space of a Lorenz system. The matter on either side of this face element experiences a force which is proportional to df.

Thus, the force is valid as:

$$\mathrm{d}\,\boldsymbol{\sigma}(\mathbf{n}) = \boldsymbol{\sigma}(\mathbf{n})\,\mathrm{d}\,f \qquad (2.3)$$

The components σi(n) of σ(n) are linear functions of the components $\mathrm{n_k}$ of n:

$$\sigma_i(\mathbf{n}) = \sigma_{ik} n_k\,,\ i,k = 1,2,3 \qquad (2.4)$$

Where $\boldsymbol{\sigma}_{ik}$ is the elastic stress tensor, which can be also called the relative stress tensor, in contrast to the space part σ^0_{ik} of the total energy-momentum tensor T_{ik}, referred as the absolute stress tensor. [53], [54} (Fig. 2).

The connection between the absolute and relative stress tensors is:

$$\sigma^0_{ik} = \sigma_{ik} + g_i u_k\,,\ i,k = 1,2,3 \qquad (2.5)$$

where gi are the components of the momentum density g and u_k the components of the velocity u of the matter.

Furthermore, the connection between g and the energy flux s, is valid as:

$$\mathbf{g} = \mathbf{s}/c^2 \qquad (2.6)$$

in which c denotes the speed of light (= 300.000 km/sec).

The total work done per unit time by elastic forces on the matter inside the closed surface f is equal to:

$$W = \int_f (\boldsymbol{\sigma}(\mathbf{n})\cdot\mathbf{u})\mathrm{d}\,f = \int_f \sigma_{ik} n_k u_i\,\mathrm{d}\,f = -\int_\upsilon \frac{\vartheta(u_i\sigma_{ik})}{\vartheta x_k}\mathrm{d}\,\upsilon\,,\ i,k = 1,2,3 \qquad (2.7)$$

Where the integration in the last integral is extended over the interior υ of the surface f.

Hence, the work done on an infinitesimal piece of matter of volume $\delta\upsilon$ is valid as:

$$\delta W = -\frac{\vartheta(u_i\sigma_{ik})}{\vartheta x_k}\delta\upsilon \qquad (2.8)$$

Moreover, (2.8) must be equal to the increase per unit time of the energy inside $\delta\upsilon$:

$$\frac{\mathrm{d}}{\mathrm{d}\,t}(h\delta\upsilon) = \delta W \qquad (2.9)$$

where h is the total energy density, including the elastic energy and denotes the substantial time derivative.

Eq. (2.9) is valid as:

$$\frac{\mathrm{d}}{\mathrm{d}t}(h\delta\upsilon)=\left(\frac{\vartheta h}{\vartheta t}+\frac{\vartheta h}{\vartheta x_k}u_k\right)\delta\upsilon+h\delta\upsilon\frac{\vartheta u_k}{\vartheta x_k}=\left[\frac{\vartheta h}{\vartheta t}+\frac{\vartheta}{\vartheta x_k}(hu_k)\right]\delta\upsilon \qquad (2.10)$$

which leads to the relation:

$$\frac{\vartheta h}{\vartheta t}+\frac{\vartheta}{\vartheta x_k}(hu_k+u_i\sigma_{ik})=0 \qquad (2.11)$$

So, the total energy flow is valid as:

$$\mathbf{s}=\mathbf{hu}+(\mathbf{u}\cdot\boldsymbol{\sigma}) \qquad (2.12)$$

Where $(\mathbf{u}\cdot\boldsymbol{\sigma})$ is a space vector with components $(\mathbf{u}\cdot\boldsymbol{\sigma})_k=u_i\sigma_{ik}$.

Hence, the total momentum density can be written as:

$$\mathbf{g}=\frac{\mathbf{s}}{c^2}=\mu\mathbf{u}+\frac{(\mathbf{u}\cdot\boldsymbol{\sigma})}{c^2} \qquad (2.13)$$

Where $\mu=h/c^2$ is the total mass density, including the mass of the elastic energy.
From (2.5) and (2.13) one obtains:

$$\sigma_{ik}-\sigma_{ki}=-g_iu_k+g_ku_i=[-(\mathbf{u}\cdot\boldsymbol{\sigma})_iu_k+(\mathbf{u}\cdot\boldsymbol{\sigma})_ku_i]/c^2\neq 0 \qquad (2.14)$$

which shows that the relative stress tensor is not symmetrical, in contrast to the absolute stress tensor (2.1) which is symmetrical.

In the stationary frame S^0 the velocity $u^0=0$ and hence, from (2.5), (2.12) and (2.13) one obtains the following expressions:

$$\sigma^0_{ik}=\sigma_{ik}=\sigma_{ki}=\sigma^0_{ki}\ (i,k=1,2,3) \qquad (2.15)$$

Beyond the above, the mechanical energy-momentum tensor satisfies the following relation:

$$T_{ik}U_k=-h^0U_i \qquad (2.16)$$

where U_i is the four-velocity of the matter, in the Lorentz system and $U^0_i=(0,0,0,ic)$.

Thus, the following scalar can be formed:

$$U_iT_{ik}U_k/c^2=U^0_iT^0_{ik}U^0_k/c^2=-T^0_{44}=h^0(x_1) \qquad (2.17)$$

With $h^0(x_1)$ the invariant rest energy density considered as a scalar function of the coordinates (x_i) $(i=1,2,3)$ in S. (Fig. 2)

By applying further the tensor:

$$\Delta_{ik}=\delta_{ik}+U_iU_k/c^2 \qquad (2.18)$$

which satisfies the relations:

$$U_i\Delta_{ik}=\Delta_{ik}U_k=0 \qquad (2.19)$$

then, we can form the following symmetrical tensor:

$$S_{ik}=\Delta_{i1}T_{1m}\Delta_{mk}=S_{ki} \qquad (2.20)$$

which is orthogonal to U_i:

$$U_iS_{ik}=S_{ik}U_k=0 \qquad (2.21)$$

By combining eqs. (2.16), (2.17) and (2.20) we obtain:

$$S_{ik}=T_{ik}-h^0U_iU_k/c^2 \qquad (2.22)$$

Furthermore, in the stationary system S_0 one has:

$$S^0_{ik}=\sigma^0_{ik}=\sigma_{ik},\ S^0_{i4}=S^0_{4i}=0 \qquad (2.23)$$

Eq. (2.22) may also be written as:

$$T_{ik}=\xi_{ik}+S_{ik} \qquad (2.24)$$

where:

$$\xi_{ik}=h^0U_iU_k/c^2=\mu^0U_iU_k \qquad (2.25)$$

is the kinetic energy-momentum tensor for an elastic body and:

$$\mu^0=h^0/c^2 \qquad (2.26)$$

is the proper mass density.

Also, let us introduce in every system S the quantity:

$$\sigma_{ik}=S_{ik}-S_{i4}U_k/U_4 \qquad (2.27)$$

which, on account of (2.24) and (2.25) is valid as:

$$\sigma_{ik}=T_{ik}-T_{i4}U_k/U_4 \qquad (2.28)$$

From (2.1) and (2.2) the three-tensor:

$$S^0_{ik}=\sigma^0_{ik}=\sigma_{ik}$$

in the stationary system is a real symmetrical matrix. The corresponding normalized eigenvectors $\mathbf{h}^{0(j)}$ satisfy the orthonormality relations:

$$\mathbf{h}^{(j)0} \cdot \mathbf{h}^{(\rho)0} = \delta^{je} \qquad (2.29a)$$

and:

$$h_i^{(j)0} h_k^{(j)0} = \delta_{ik} \quad (j, \rho = 1,2,3) \qquad (2.29b)$$

The eigenvalues $p^0_{(j)}$, the principal stresses, are the three roots of the following algebraic equation, where λ is the unknown:

$$\left|S^0_{ik} - \lambda\delta_{ik}\right| = \left|\sigma^0_{ik} - \lambda\delta_{ik}\right| = 0 \qquad (2.30)$$

The matrix S^0_{ik} may also be written in terms of the eigenvalues and eigenvectors as:

$$S^0_{ik} = \sigma^0_{ik} = p^0_{(j)} h_i^{(j)0} h_k^{(j)0} \qquad (2.31)$$

From eqs. (2.23) and (2.31) one obtains the following form of the stress four-tensor in S^o:

$$S^0_{ik} = p^0_{(j)} h_i^{(j)0} h_k^{(j)0} \qquad (2.32)$$

Hence, in any system S we have

$$S_{ik} = p^0_{(j)} h_i^{(j)} h_k^{(j)} \qquad (2.33)$$

From (2.24), (2.25), (2.27) and (2.33) we obtain the following expressions

$$T_{ik} = \mu^0 U_i U_k + p^0_{(j)} h_i^{(j)} h_k^{(j)} \qquad (2.34)$$

$$\sigma_{ik} = S_{ik} - S_{i4} U_k / U_4 = p^0_{(j)} h_k^{(j)} \left(h_k^{(j)} + i h_4^{(j)} u_k / c\right) \qquad (2.35)$$

By putting:

$$h_i^{(j)} = (\mathbf{h}^{(j)}, h_4^{(j)}) \qquad (2.36)$$

and introducing the notation $\mathbf{a} \bullet \mathbf{b}$ for the direct product of the vectors a and b, we may write (2.35) for the relative stress tensor σ as:

$$\boldsymbol{\sigma} = p^0_{(j)} \left[\mathbf{h}^{(j)} \bullet \mathbf{h}^{(j)} + \frac{i}{c} h_4^{(j)} (\mathbf{h}^{(j)} \bullet \mathbf{u})\right], \; j = 1,2,3 \qquad (2.37)$$

Beyond the above, the triad vectors $h_i^{(j)}$ satisfy the tensor relations:

$$h_i^{(j)} h_i^{(\rho)} = \delta^{j\rho} \qquad (2.38)$$

$$h_i^{(j)} h_k^{(j)} = \Delta_{ik} \qquad (2.39)$$

with Δ_{ik} given by (2.18).

If the stationary system S^0 for every event point is chosen in such a way that the spatial axes in S^0 and in S have the same orientation, one obtains:

$$\mathbf{h}^{(j)} = \mathbf{h}^{(j)0} + \left\{\mathbf{u}(\mathbf{u} \cdot \mathbf{h}^{(j)0})(\gamma - 1)\right\} / u^2$$

$$h_4^{(j)} = i \mathbf{u} \cdot \mathbf{h}^{(j)0} \gamma / c$$

with:

$$\gamma = 1 / (1 - u^2/c^2)^{1/2} \qquad (2.41)$$

From (2.34) and (2.40) with $i = k = 4$ we obtain:

$$h = -T_{44} = -\mu^0 U_4^2 - p^0_{(j)} (\mathbf{u} \cdot \mathbf{h}^{(j)0})^2 \cdot \gamma^2 / c^2 \qquad (2.42)$$

In the stationary system, (2.37) reduces to:

$$\boldsymbol{\sigma}^0 = p^0_{(j)} \left(\mathbf{h}^{(j)0} \bullet \mathbf{h}^{(j)0}\right) \qquad (2.43)$$

Thus, from (2.42) we obtain the following transformation law for the energy density:

$$h = \frac{h^0 + \mathbf{u} \cdot \boldsymbol{\sigma}^0 \cdot \mathbf{u} / c^2}{1 - u^2/c^2}$$

$$\mathbf{u} \cdot \boldsymbol{\sigma}^0 \cdot \mathbf{u} = u_i \sigma^0_{ik} u_k \qquad (2.44)$$

and the mass density:

$$\mu = \frac{\mu^0 + \mathbf{u} \cdot \boldsymbol{\sigma}^0 \cdot \mathbf{u} / c^4}{1 - u^2/c^2} \qquad (2.45)$$

From (2.40) and (2.34) with k = 4, one obtains the momentum density $\mathbf{g}$ with the components $g_i = T_{i4} / ic$:

$$\mathbf{g} = \mathbf{u}\left[h^0 + \mathbf{u} \cdot \boldsymbol{\sigma}^0 \cdot \mathbf{u}(1 - \gamma^{-1}) / u^2\right] \gamma^2 / c^2 + (\boldsymbol{\sigma}^0 \cdot \mathbf{u}) \gamma / c^2 \qquad (2.46)$$

Also, from (2.40) and (2.35) we obtain the relative stress tensor:

$$\boldsymbol{\sigma} = \boldsymbol{\sigma}^0 + \mathbf{u} \bullet (\boldsymbol{\sigma}^0 \cdot \mathbf{u})(\gamma - 1)/u^2 - (\boldsymbol{\sigma}^0 \cdot \mathbf{u}) \bullet \mathbf{u}(\gamma - 1)/\gamma u^2$$
$$- (\mathbf{u} \bullet \mathbf{u})(\mathbf{u} \cdot \boldsymbol{\sigma}^0 \cdot \mathbf{u})(\gamma - 1)^2 / \gamma u^4 \qquad (2.47)$$

In the special case u = (u,0,0), where the notation of the matter at the point considered is parallel to the x1-axis (see Figs.1 and 2), the transformation equations (2.44), (2.46) and (2.47) reduce to:

$$h = \left(h^0 + \frac{u^2}{c^2} \sigma^0_{11}\right) \gamma^2$$

$$g_{x_1} = \gamma^2 \left(\mu^0 + \frac{\sigma^0_{11}}{c^2}\right) u \qquad (2.48)$$

$$g_{x_2} = \frac{\gamma \sigma_{21}^{0}}{c^2} u$$

$$g_{x_3} = \frac{\gamma \sigma_{31}^{0}}{c^2} u$$

and the relative stress tensor:

$$\boldsymbol{\sigma} = \begin{bmatrix} \sigma_{11} & \sigma_{12} & \sigma_{13} \\ \sigma_{21} & \sigma_{22} & \sigma_{23} \\ \sigma_{31} & \sigma_{32} & \sigma_{33} \end{bmatrix} = \begin{bmatrix} \sigma_{11}^{0} & \gamma\sigma_{12}^{0} & \gamma\sigma_{13}^{0} \\ \frac{1}{\gamma}\sigma_{21}^{0} & \sigma_{22}^{0} & \sigma_{23}^{0} \\ \frac{1}{\gamma}\sigma_{31}^{0} & \sigma_{32}^{0} & \sigma_{33}^{0} \end{bmatrix} \quad (2.49)$$

where γ is given by (2.41). Finally, as it could be easily seen the relative stress tensor is not symmetrical, in contrast to the absolute stress tensor which is symmetrical.

III. Elastic Stress Analysis for Stationary Frames and Airframes

Let us consider the stationary frame of Fig. 1 with Γ_1 the portion of the boundary of the body on which displacements are presented, Γ_2 the surface of the body on which the force tractions are employed and Γ the total surface of the body equal to $\Gamma_1+\Gamma_2$.

For the principal of virtual displacements, for linear elastic problems then the following formula is valid:

$$\int_{\Omega} (\sigma_{jk,j}^{0} + b_k) u_k \,\mathrm{d}\Omega = \int_{\Gamma_2} (p_k - \overline{p}_k) u_k \,\mathrm{d}\Gamma \qquad (3.1)$$

Where u_k are the virtual displacements, which satisfy the homogeneous boundary conditions $\overline{u}_k \equiv 0$ on Γ_1, b_k the body forces (Fig. 1) and p_k the surface tractions at the point k of the body. (Fig. 3)

Beyond the above, (3.1) takes the following form if u_k do not satisfy the previous conditions on Γ_1:

$$\int_{\Omega} (\sigma_{jk,j}^{0} + b_k) u_k \,\mathrm{d}\Omega = \int_{\Gamma_2} (p_k - \overline{p}_k) u_k \,\mathrm{d}\Gamma + \int_{\Gamma_1} (\overline{u}_k - u_k) p_k \,\mathrm{d}\Gamma \qquad (3.2)$$

where $p_k = n_j \sigma_{jk}^{0}$ are the surface tractions corresponding to the u_k system. By integrating (3.2) follows:

$$\int_{\Omega} b_k u_k \,\mathrm{d}\Omega - \int_{\Omega} \sigma_{jk}^{0} \varepsilon_{jk} \,\mathrm{d}\Omega = -\int_{\Gamma_2} \overline{p}_k u_k \,\mathrm{d}\Gamma - \int_{\Gamma_1} p_k u_k \,\mathrm{d}\Gamma + \int_{\Gamma_1} (\overline{u}_k - u_k) p_k \,\mathrm{d}\Gamma \qquad (3.3)$$

in which ε_{jk} are the strains.

By a second integration (3.3) reduces to:

$$\int_{\Omega} b_k u_k \,\mathrm{d}\Omega + \int_{\Omega} \sigma_{jk,j}^{0} u_k \,\mathrm{d}\Omega = - \int_{\Gamma_2} \overline{p}_k u_k \,\mathrm{d}\Gamma - \int_{\Gamma_1} p_k u_k \,\mathrm{d}\Gamma + \int_{\Gamma_1} \overline{u}_k p_k \,\mathrm{d}\Gamma + \int_{\Gamma_2} u_k p_k \,\mathrm{d}\Gamma \qquad (3.4)$$

Furthermore, a fundamental solution should be found, satisfying the equilibrium equations, of the following type:

$$\sigma_{jk,j}^{0} + \Delta_l^i = 0 \qquad (3.5)$$

Where Δ_l^i is the Dirac delta function which represents a unit load at i in the l direction.

The fundamental solution for a three-dimensional isotropic body is: [31]

$$u_{lk}^{*} = \frac{1}{16\pi G(1-v)r}\left[(3-4v)\Delta_{lk} + \frac{\vartheta r}{\vartheta x_l}\frac{\vartheta r}{\vartheta x_k}\right]$$

$$p_{lk}^{*} = -\frac{1}{8\pi(1-v)r^2}\left[\frac{\vartheta r}{\vartheta n}\left[(1-2v)\Delta_{lk} + 3\frac{\vartheta r}{\vartheta x_l}\frac{\vartheta r}{\vartheta x_k}\right] - (1-2v)\left[\frac{\vartheta r}{\vartheta x_l} n_k - \frac{\vartheta r}{\vartheta x_k} n_l\right]\right] \qquad (3.6)$$

where G is the shear modulus, v Poisson's ratio, n the normal to the surface of the body, Δ_{lk} Kronecker's delta, r the distance from the point of application of the load to the point under consideration and n_j the direction cosines (Fig.3).

The displacements at a point are given by the formula:

$$u^i = \int_\Gamma up \,\mathrm{d}\,\Gamma - \int_\Gamma pu \,\mathrm{d}\,\Gamma + \int_\Omega bu \,\mathrm{d}\,\Omega \tag{3.7}$$

Hence, (3.7) takes the following form for the "l" component:

$$u_l^i = \int_\Gamma u_{lk}\, p_k \,\mathrm{d}\,\Gamma - \int_\Gamma p_{lk} u_k \,\mathrm{d}\,\Gamma + \int_\Omega b_k u_{lk} \,\mathrm{d}\,\Omega \tag{3.8}$$

By differentiating u at the internal points, one obtains the stress-tensor for an isotropic medium:

$$\sigma_{ij}^0 = \frac{2Gv}{1-2v}\Delta_{ij}\frac{\vartheta u_l}{\vartheta x_l} + G\left(\frac{\vartheta u_i}{\vartheta x_j} + \frac{\vartheta u_j}{\vartheta x_i}\right) \tag{3.9}$$

Also, after carrying out the differentiation we have:

$$\sigma_{ij}^0 = \int_\Gamma \left[\frac{2Gv}{1-2v}\Delta_{ij}\frac{\vartheta u_{lk}}{\vartheta x_l} + G\left(\frac{\vartheta u_{ik}}{\vartheta x_j} + \frac{\vartheta u_{jk}}{\vartheta x_i}\right)\right] p_k \,\mathrm{d}\,\Gamma +$$

$$+ \int_\Omega \left[\frac{2Gv}{1-2v}\Delta_{ij}\frac{\vartheta u_{lk}}{\vartheta x_l} + G\left(\frac{\vartheta u_{ik}}{\vartheta x_j} + \frac{\vartheta u_{jk}}{\vartheta x_i}\right)\right] b_k \,\mathrm{d}\,\Omega - \tag{3.10}$$

$$- \int_\Gamma \left[\frac{2Gv}{1-2v}\Delta_{ij}\frac{\vartheta p_{lk}}{\vartheta x_l} + G\left(\frac{\vartheta p_{ik}}{\vartheta x_j} + \frac{\vartheta p_{jk}}{\vartheta x_i}\right)\right] u_k \,\mathrm{d}\,\Gamma$$

Eq. (3.10) can be further written as following:

$$\sigma_{ij}^0 = \int_\Gamma D_{kij}\, p_k \,\mathrm{d}\,\Gamma - \int_\Gamma S_{kij} u_k \,\mathrm{d}\,\Gamma + \int_\Omega D_{kij} b_k \,\mathrm{d}\,\Omega \tag{3.11}$$

Where the third order tensor components D_{kij} and S_{kij} are:

$$D_{kij} = \frac{1}{8\pi(1-v)r^2}\left[(1-2v)\left[\Delta_{ki} r_{,j} + \Delta_{kj} r_{,i} - \Delta_{ij} r_{,k}\right] + 3 r_{,i} r_{,j} r_{,k}\right] \tag{3.12}$$

$$S_{kij} = \frac{G}{4\pi(1-v)r^3}\left[3\frac{\vartheta r}{\vartheta n}\left[(1-2v)\Delta_{ij} r_{,k} + v(\Delta_{ik} r_{,j} + \Delta_{jk} r_{,i}) - 5 r_{,i} r_{,j} r_{,k}\right]\right. \tag{3.13}$$

$$\left. + 3v(n_i r_{,j} r_{,k} + n_j r_{,i} r_{,k}) + (1-2v)(3 n_k r_{,i} r_{,j} + n_j \Delta_{ik} + n_i \Delta_{jk}) - (1-4v) n_k \Delta_{ij}\right]$$

whith: $r_{,i} = \dfrac{\vartheta r}{\vartheta x_i}$

Finally, because of eqs (2.49) and (3.11) by considering the moving system S of Fig. 2, then the stress-tensor reduces to the following form:

$$\begin{aligned}
\sigma_{11} &= \sigma_{11}^0 \\
\sigma_{12} &= \gamma\sigma_{12}^0 \\
\sigma_{13} &= \gamma\sigma_{13}^0 \\
\sigma_{21} &= \frac{1}{\gamma}\sigma_{21}^0 \\
\sigma_{22} &= \sigma_{22}^0 \\
\sigma_{23} &= \sigma_{23}^0 \\
\sigma_{31} &= \frac{1}{\gamma}\sigma_{31}^0 \\
\sigma_{32} &= \sigma_{32}^0 \\
\sigma_{33} &= \sigma_{33}^0
\end{aligned} \tag{3.14}$$

Where σ_{ij}^0 are given by. (3.11) to (3.13).

The following Table 1 shows the values of γ as given by (2.41) for some arbitrary values of the velocity u of the moving aerospace structure:

Table 1

Velocity u	$\gamma = 1/\sqrt{1-u^2/c^2}$	Velocity u	$\gamma = 1/\sqrt{1-u^2/c^2}$
50,000 *km/h*	1.000000001	0.800c	1.666666667
100,000 *km/h*	1.000000004	0.900c	2.294157339
200,000 *km/h*	1.000000017	0.950c	3.202563076
500,000 *km/h*	1.000000107	0.990c	7.088812050
10E+06 *km/h*	1.000000429	0.999c	22.36627204
10E+07 *km/h*	1.000042870	0.9999c	70.71244596
10E+08 *km/h*	1.004314456	0.99999c	223.6073568
2x10E+8 *km/h*	1.017600788	0.999999c	707.1067812
c/3	1.060660172	0.9999999c	2236.067978
c/2	1.154700538	0.99999999c	7071.067812
2c/3	1.341640786	0.999999999c	22360.67978
3c/4	1.511857892	C	∞

From the above Table follows that for small velocities 50,000 km/h to 200,000 km/h, the absolute and the relative stress tensor are nearly the same. On the other hand, for bigger velocities like c/3, c/2 or 3c/4 (c = speed of light), the variable γ takes values more than the unit and thus, relative stress tensor is very different from the absolute one. Finally, for values of the velocity of the moving structure near the speed of light, the variable γ takes bigger values, while when the velocity is equal to the speed of light, then γ tends to the infinity.

The Singular Integral Operators Method (S.I.O.M.) as was proposed by E.G.Ladopoulos [4], [8], [9], [11], [12], [13], [15] and E.G.Ladopoulos et all [22] will be used for the numerical evaluation of the stress tensor (3.11), for every specific case.

IV. Conclusions

In the present investigation in the area of aeronautics technologies the theory of "Relativistic Elasticity" has been introduced and applied for the design of a new generation large aircraft with turbojet engines and speeds in the range of 50,000 km/h. Such a design and construction of the new generation aircraft will be applied to an increased market share of International Aeronautical Industries. Furthermore, the theory of "Relativistic Elasticity" has been applied for the design of the next generation spacecrafts moving with very high speeds, even approaching the speed of light, as the target of the International Space Agencies (ESA, NASA, etc.) is to achieve such spacecrafts in the future. The future investigation concerns to the determination of the proper composite materials for the construction of the next generation spacecfracts, as usual composite solids are not proper for such a construction.

The theory of "Relativistic Elasticity" and the "Universal Equation of Elasticity" show that there is a considerable difference between the absolute stress tensor of the airframe even in the range of speeds of 50,000 km/h. For bigger speeds the difference between the two stress tensors is very much increased. "Relativistic Mechanics" is a combination of the theories of "Classical Elasticity" and "Special Relativity".

For the structural design of the next generation aircrafts and spacecrafts will be used the stress tensor of the airframe in combination to the singular integral equations. Such a stress tensor is reduced to the solution of a multidimensional singular integral equation and for its numerical evaluation will be used the Singular Integral Operators Method (S.I.O.M.).

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Figure Captions

Figure 1 : The state of stress σ_{ik}^{0} in the stationary system.

Figure 2 : The state of stress σ_{ik}^{0} in the stationary system S^{o} and σ_{ik} in the airframe system S, with velocity u parallel to the x_1 - axis.

Figure 3 : The stationary system S^{o}.

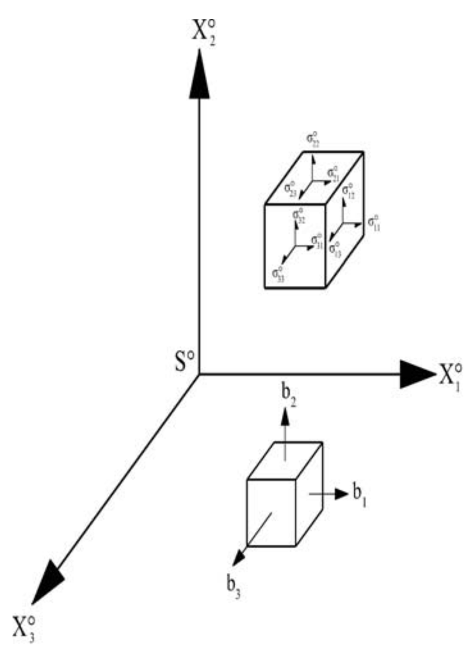

Figure 1

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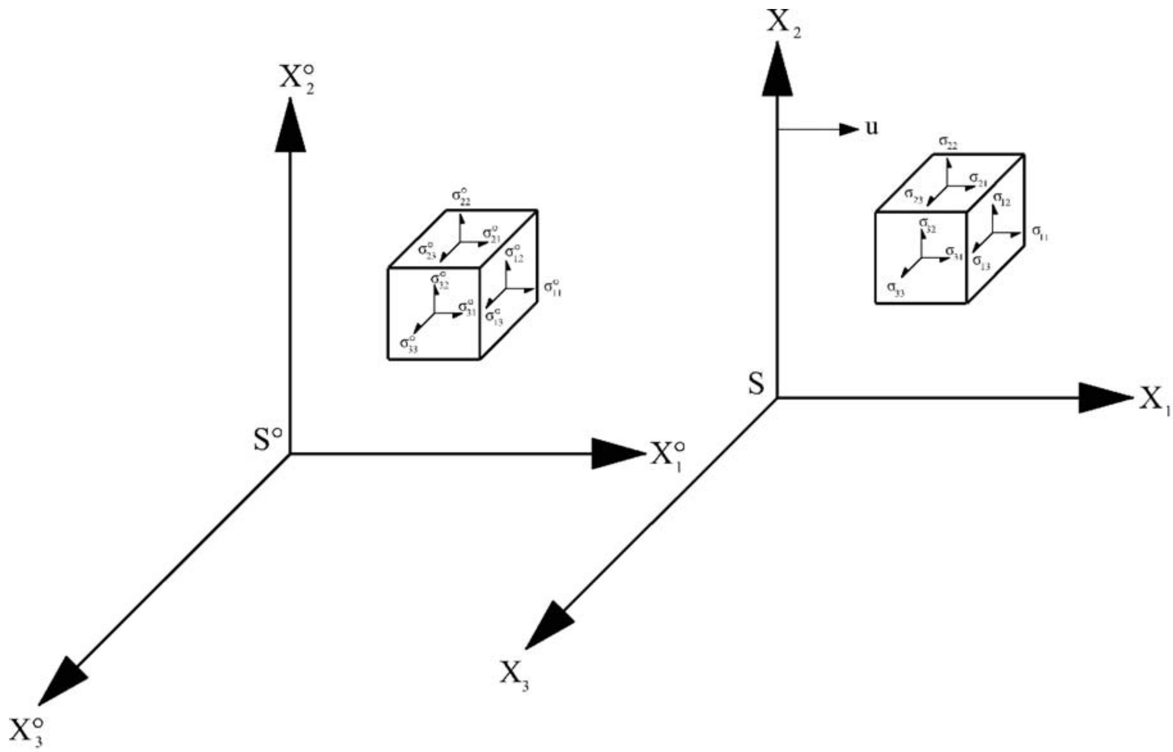

Figure 2

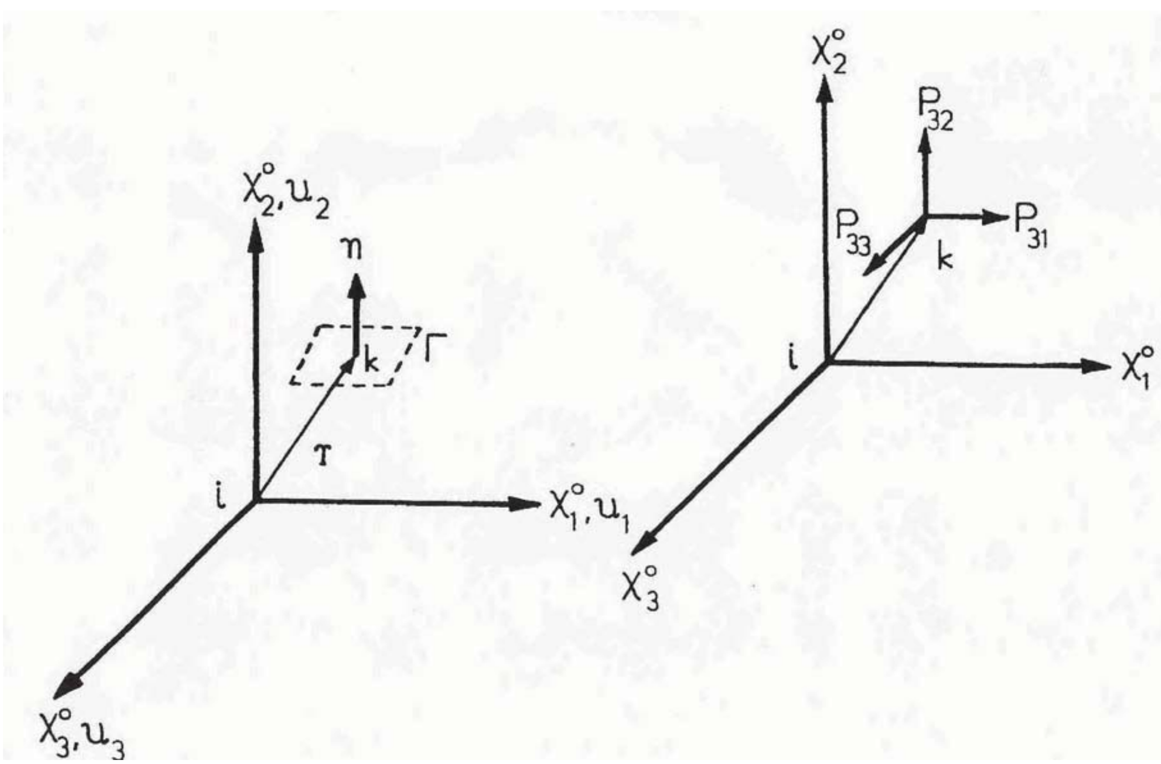

Figure 3

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Robust Algorithms for Formation Flying Reconfiguration

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Abstract - Over the last 20 years spacecraft formation flying has been the subject of numerous research activities due to the advantages offered when compared with large, complex, single purpose satellites. With the obvious advantages of increased functionality and enhanced reliability, come however, also substantial challenges in the maintenance and reconfiguration of the spacecraft formation. The present paper addresses these problems by proposing two approaches that can be mathematically validated thus making it attractive for safety critical applications such as proximity operations. The first approach hinges on the implementation of pursuit algorithms first studied by French scientist Pierre Bouguer in the 18th century. The proposed approach separates the control law into two distinct stages: planar movement control and orthogonal displacement suppression. The second approach relies on the use of motion camouflage which is a hunting technique widely used in the natural world that allows a predator to approach a prey while appearing to remain stationary. A number of different scenarios are presented and the two approaches implemented within them. Numerical results shows that both methods are robust to dynamical uncertainties and do ensure the correct reconfiguration manoeuvres.

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ROBUST ALGORITHMS FOR FORMATION FLYING RECONFIGURATION

Strictly as per the compliance and regulations of:

Robust Algorithms for Formation Flying Reconfiguration

Gianmarco Radice[α], Tao Yang [σ], Weihua Zhang [ρ]

Abstract - Over the last 20 years spacecraft formation flying has been the subject of numerous research activities due to the advantages offered when compared with large, complex, single purpose satellites. With the obvious advantages of increased functionality and enhanced reliability, come however, also substantial challenges in the maintenance and reconfiguration of the spacecraft formation. The present paper addresses these problems by proposing two approaches that can be mathematically validated thus making it attractive for safety critical applications such as proximity operations. The first approach hinges on the implementation of pursuit algorithms first studied by French scientist Pierre Bouguer in the 18th century. The proposed approach separates the control law into two distinct stages: planar movement control and orthogonal displacement suppression. The second approach relies on the use of motion camouflage which is a hunting technique widely used in the natural world that allows a predator to approach a prey while appearing to remain stationary. A number of different scenarios are presented and the two approaches implemented within them. Numerical results shows that both methods are robust to dynamical uncertainties and do ensure the correct reconfiguration manoeuvres.

I. Introduction

In recent years, the idea of distributing the functionality of large satellites among smaller has become increasingly popular as a traditional, large single spacecraft may not be sufficient to meet mission requirements [1]. Several scenarios entailing cooperative satellites have been considered for numerous space missions. To this end spacecraft formation flying has become a promising means of reducing operational costs and increase mission flexibility and functionality [2-6]. Due to the often precise navigation and positioning requirements of these missions, the spacecraft station keeping and orbit and increase mission flexibility and functionality [2-6]. Due to the often precise navigation and positioning requirements of these missions, the spacecraft station keeping and orbit control become crucial for mission success. Different approaches exist and have been proposed in literature to tackle these challenging problems [7-12]. The main drawback of these approaches is that they generally require costly computational resources making them thus unsuitable for on-board scheduling. The development of autonomy technologies is the key to three vastly important strategic technical challenges facing future spacecraft missions. The reduction of mission operation costs, the continuing return of quality science products through increasingly limited communications bandwidth and the launching of a new era of solar system exploration, characterised by sustained presence and in depth scientific studies. Spacecraft autonomy will bring significant advantages by improving resource management, increasing fault tolerance and simplifying payload operations. Also, when considering the communication delays in deep space missions, the requirement for autonomy becomes clear. Ground stations and controllers will not be able to communicate and control distant spacecraft in real-time to guarantee pointing precision and safety. As the number of satellites within the formation and the distance of the operational orbit from the Earth increase, conventional methods show their limits and become less practical. New control methods are therefore required; approaches that enhance the automation of the system, enabling the formation to perform deployment, maintenance and re-configuration manoeuvres autonomously.

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a) Pursuit Algorithms

An interesting line of research, inspired by pursuit algorithms, was first studied by French scientist Pierre Bouguer in the 18th century. Simply put, if a point *A* in space moves along a known curve, then another point *P* describes a pursuit curve if its motion is always directed towards *A* and the two points move with equal speeds. More than a century later, scholars found that if three agents, initially placed at the vertices of an equilateral triangle, were to run one after the other, then their pursuit curves would be a logarithmic spiral and they would eventually meet at a common point, known now as the Brocard point of a triangle as shown in Figure1[13].

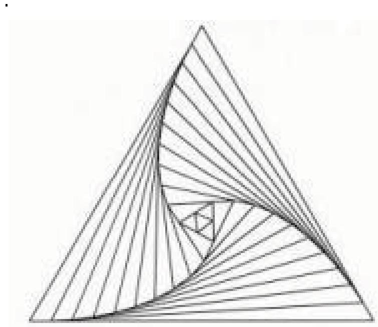

Figure 1: Pursuit curve pattern for an equilateral triangle.

b) Motion Camouflage

Motion camouflage is a stealth technique that allows a predator to approach a moving target (e.g. the prey) whilst appearing to remain stationary. To achieve this, the predator follows a path such that it always lies on the line connecting the predator and a fixed point (knowas the camouflage background) as shown in Figure 2. Biologists have used stereo cameras to reconstruct the movements in three dimensions of dragonflies, and verify that these insects successfully use motion camouflage to disguise themselves as stationary during aerial maneuvers. A more elaborate behavior is performed by the male dragonflies that periodically appear to switch fixed point locations, sometimes to nearby points, sometimes to points at infinity [14].

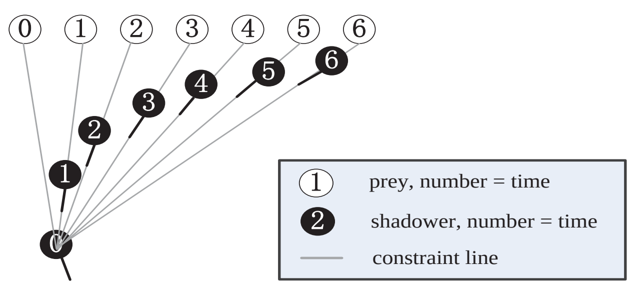

Figure 2 : A predator motion camouflage trajectory

The only visual cue to the predator's approach is its graduallooming. In a psychophysical experiment based on a 3D computergame, humans became prey, defending themselves against attacks from motion camouflaged missiles. Alternative missile approach Strategies included a homing approach and direct interception approach. The experimental results demonstrated that motioncamouflaged missiles were in general able to get closer to the object before being shot than the alternative strategies.

II. Control Algorithms

We assume that the formation is orbiting the Earth at an altitude which is much larger than the relative distance between the satellites. We can therefore define the equations of motion of a chaser satellite about a target satellite through the Clohessy- Wiltshire equations:

$$\begin{aligned} \ddot{x} - 2n\dot{y} - 3n^2 x &= a_x \\ \ddot{y} + 2n\dot{x} &= a_y \\ \ddot{z} + n^2 z &= a_z \end{aligned} \qquad (1)$$

Motion in the *z* direction and along the orbital plane is decoupled; hence if necessary the control law can be designed in two stages: planar and orthogonal control.

a) Pursuit Algorithms Control

If the satellite lies on the reference centre, then under cyclic pursuit it will remain stationary. Generally the initial position of the agentis however not superposed to the reference centre, thus it is necessary to combine this with beacon's guidance to achieve reorientation. Suppose the reference centre to be a virtual beacon, together with angular rotation control of $\dot{\theta}_i' = \omega_i$, another control denoted as $\dot{\theta}_i'' = u_i$ would be required to maintain the relative distance maintenance with respect to the beacon. This linear control is expressed as:

$$u(t) = \begin{cases} k_b g_b(\rho(t))\alpha_b(\gamma(t)) & if\ \rho(t) > 0 \\ 0 & if\ \rho(t) = 0 \end{cases} \qquad (2a)$$

with

$$g_b(\rho) = \ln\left(\frac{(c_b - 1)\cdot\rho + \rho_e}{c_b \cdot \rho_e}\right) \qquad (2b)$$

$$\alpha_b(\gamma) = \begin{cases} \gamma & if\ 0 \le \gamma \le 3\pi/2 \\ \gamma - 2\pi & if\ 3\pi/2 < \gamma < 2\pi \end{cases} \qquad (2c)$$

$$\alpha_b(\gamma) = \begin{cases} \gamma & if\ 0 < \gamma < \pi/2 \\ \gamma - 2\pi & if\ \pi/2 \le \gamma \le 2\pi \end{cases} \qquad (2d)$$

Where, is the distance between the vehicle and the beacon$\gamma \in [0, 2\pi)$represents the angular distance between the heading of the vehicle and the position vector of the beacon Note that Eq. 2c valid in the case of counterclock wise equilibrium and Eq. 2d valid in the case of clockwise equilibrium. A combined control law for multi-agent motion would then be:

$$\dot{\theta}_i = \dot{\theta}_i' + \dot{\theta}_i'' = \omega_i + u_i(t) = \begin{cases} k_\alpha \alpha_i + k_b g_b(\rho_i)\alpha_b(\gamma) & \text{for} \quad \rho_i > 0 \\ k_\alpha \alpha_i \quad \text{for} \quad \rho_i = 0 \end{cases} \qquad (3)$$

In the orthogonal direction, a linear feedback control is designed.

To suppress possible oscillations, the velocity value is taken into account. Here the parameters are adjusted to be:

$$k_z = 0.0002,\ k_v = 0.0002\,.$$

$$u_z = -k_z z - k_v \dot{z} \qquad (4)$$

This provides the control in the out of plane direction.

b) Motion Camouflage Control

The ideal motion camouflage equations are built on the assumption that the position of the target is given in advance. Let us assume that the position of the target is $\vec{z}(t)$ and that of the predator is $\vec{r}(t)$, both of which lie either in a plane or three-dimensional Euclidean space.

If the predator uses motion camouflage, then lies $\vec{r}(t)$ on the line connecting the target and some fixed reference point r_0. This means that:

$$\vec{r}(t)=\vec{r}(0)+u(t)(\vec{z}(t)-\vec{r}_0) \tag{5}$$

Where $u(t)=[0,1]$ is the position ratio of $\overrightarrow{r_0 r}$ to $\overrightarrow{r_0 z}$ To perform the formation control we assume impulsive manoeuvres such that the velocity vector changes instantaneously. The chaser transfers from state of $(\rho_1,\dot{\rho}_1^-)$ at t_1 to $(\rho_2,\dot{\rho}_2^+)$ at t_2. Superscripts of " - " and " + " refer to the state of before and after an impulse respectively. Defining $\Delta t=t_2-t_1, \psi=n\Delta t, s=\sin\psi, c=\cos\psi$, The state transition matrix becomes:

$$\Phi(t_1,t_2)=\Phi(\Delta t)=\begin{bmatrix}\Phi_{\rho\rho} & \Phi_{\rho\dot{\rho}}\\ \Phi_{\dot{\rho}\rho} & \Phi_{\dot{\rho}\dot{\rho}}\end{bmatrix}$$

$$=\begin{bmatrix}4-3c & 0 & 0 & s/n & 2(1-c)/n & 0\\ 6(s-\psi) & 1 & 0 & 2(1-c)/n & (4s-3\psi)/n & 0\\ 0 & 0 & c & 0 & 0 & s/n\\ 3ns & 0 & 0 & c & 2s & 0\\ 6n(c-1) & 0 & 0 & -2s & 4c-3 & 0\\ 0 & 0 & -ns & 0 & 0 & c\end{bmatrix} \tag{6}$$

with

$$\begin{cases}\dot{\rho}_1^+=\Phi_{\rho\dot{\rho}}^{-1}(\rho_2-\Phi_{\rho\rho}\rho_1)\\ \dot{\rho}_2^-=\Phi_{\dot{\rho}\rho}\rho_1+\Phi_{\dot{\rho}\dot{\rho}}\dot{\rho}_1^+)\end{cases} \tag{7}$$

and impulse vectors of

$$\begin{cases}\Delta v_1=\dot{\rho}_1^+-\dot{\rho}_1^-\\ \Delta v_2=\dot{\rho}_2^+-\dot{\rho}_2^-\end{cases} \tag{8}$$

Integrating all the velocity changes provides the fuel mass required for the manoeuvre through the rocket equation.

III. Numerical Results – Pursuit Algorithms

To simplify the stability analysis, a formation of only two satellites is investigated at first. We consider two reconfiguration manoeuvres: separation increase and phase angle adjustment. In this task for the reason of initial symmetric states, cyclic pursuit is sufficient to achieve radius enlargement. Applying cyclic pursuit control to this scenario requires the linear velocity to be constant. Whereas to keep the periodicity invariant to the reference centre under orbital dynamics, the velocity value relative to the reference centre should change as well. Then the cyclic pursuit in the orbit direction and the feedback control in the orthogonal direction are applied. Setting $k_\alpha=2\omega_e/\pi,\ \omega_i=k_\alpha\alpha_i-\omega_e$ as control input,

Where ω_e is the expected angular rotation rate, $k_z=0.0002,\ k_v=0.0002$ Assuming the satellites has the same mass $m=367kg$ and electric thrusters with $I_{sp}=1640s,\ Thrust=7.22e-5kN$ Is used in the first phase of about 20 minutes after which it is decreased to 3.67e-5kN. These correspond to the values of the centripetal forces in initial and target positions respectively. The results are shown in Fig 3-4.

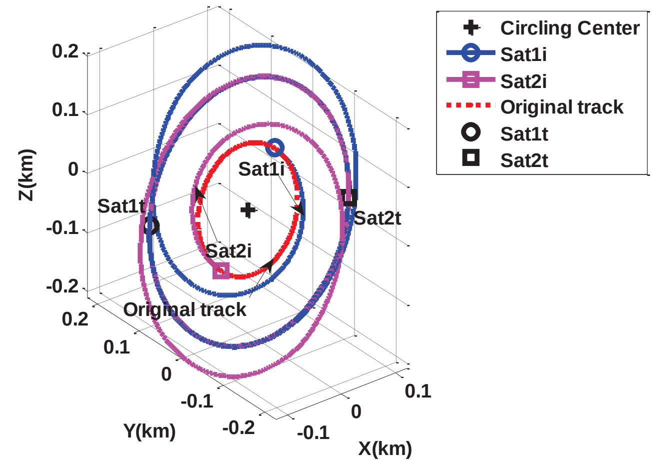

Figure 3 : Propagation of radius enlargement.

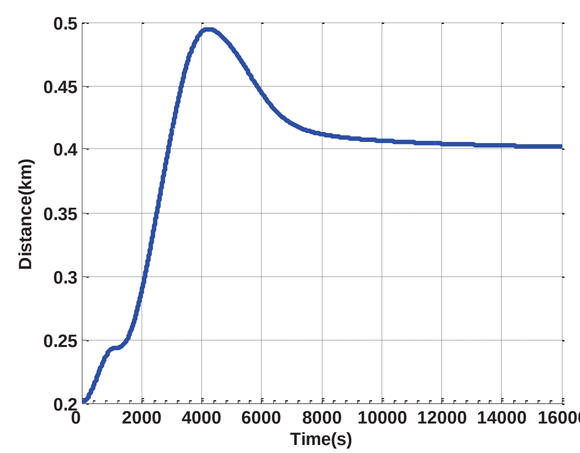

Figure 4 : Spacecraft relative distance

The eigenvalues of this system are $\pm0.9918ie-3$, - 0.6314, $-0.3157\pm0.9402ie-3$, 0. Eliminating the fake eigenvalues of $\pm0.9918ie-3$ and 0 through coordinate constraints, leaves the remaining with negative real parts. Hence the planar movement is stable. Figure 4 shows the radius of this formation increases while maintaining a constant phase angle. The radius increase following the velocity increase is rapid, but it still takes a relatively long time to finally reach the desired orbital configuration. In the first phase, a higher thrust is required to increase the relative

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distance. In the second phase, the thrust should be reduced to avoid overshooting the desired relative distance. If the thrust is maintained to the initial level throughout the manoeuvre then, the convergence rate is very slow.

In the second scenario want to modify the relative phase angle between the satellites. Applying control law to planar movement with parameters $\rho_e = 0.1$km, $c_b = 2$, $k_b = 0.02$ and $k\alpha = 2\omega_e/\pi$. Initial spacecraft mass are the same as before while the propulsion system employs SMART-1 Hall Effect Thrusters with $I_{sp} = 1640s$.

Figure 5 shows that the two satellites gradually evolve to the new required angular phase distance.

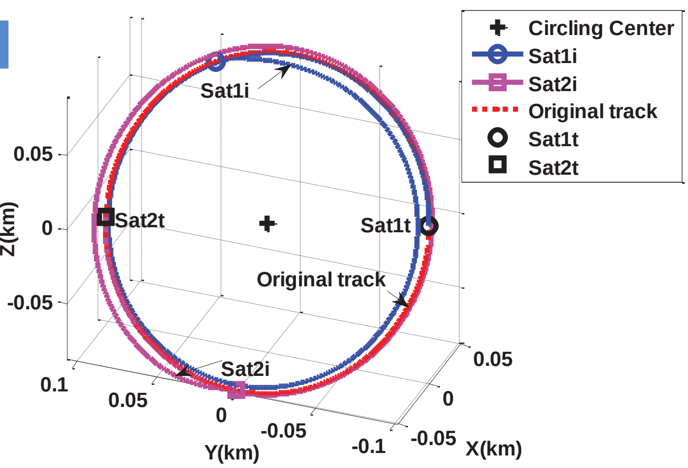

Figure 5 : Propagation of the phase angle adjustment.

If an impulsive propulsion system is used, the relative distance between the satellites would oscillate slightly before reaching the required phase angle separation as shown in Figure 6. It can be seen that the manoeuvre takes more than twice the time than using a low thrust propulsion system, as shown in Figure 7.

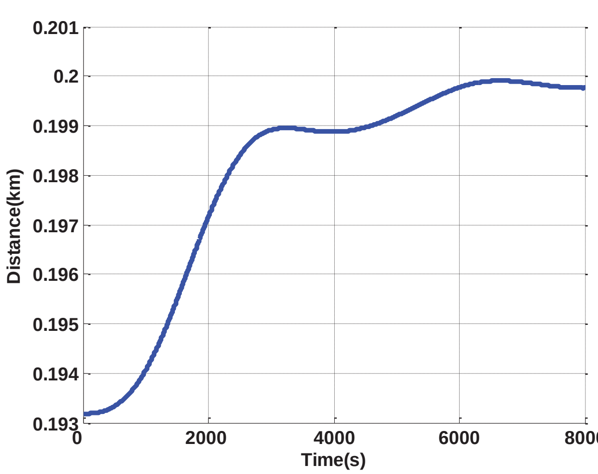

Figure 6 : Spacecraft relative distance.

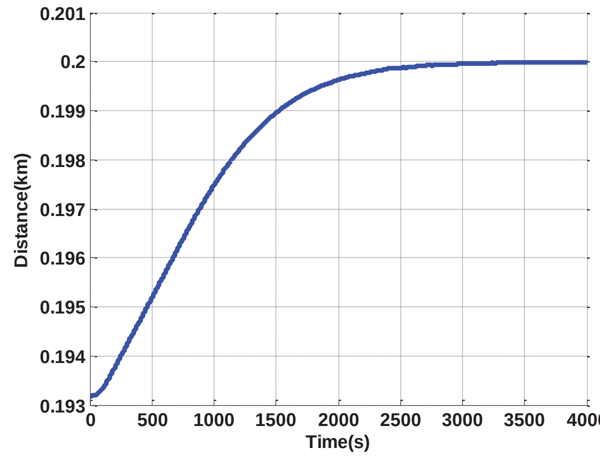

Figure 7 : Spacecraft relative distance.

IV. Numerical Results –Motion Camouflage

To simplify the preliminary analysis, let us assume the target is circling around a spacecraft with parameters of:

$$A_t = 2000m, B_t = 2000\sqrt{3}m/s,\ \phi_t = 0, \psi_t = \pi$$

$n \approx 9.92e-4\ rad/s$ The target's motion can be expressed as

$$\begin{aligned} x_t &= -A_t \cos(nt + \phi_t) \\ y_t &= 2A_t \sin(nt + \phi_t) \\ z_t &= B_t \cos(nt + \phi_t + \psi_t) \end{aligned} \tag{9}$$

We assume the chaser initiates its trajectory takes from the centre of the reference frame. When the parameters, u_c, N_c and φ_c are defined the trajectory shown in Figure 8 with velocity consumption of $\Delta V = 2.717m/s$ is followed.

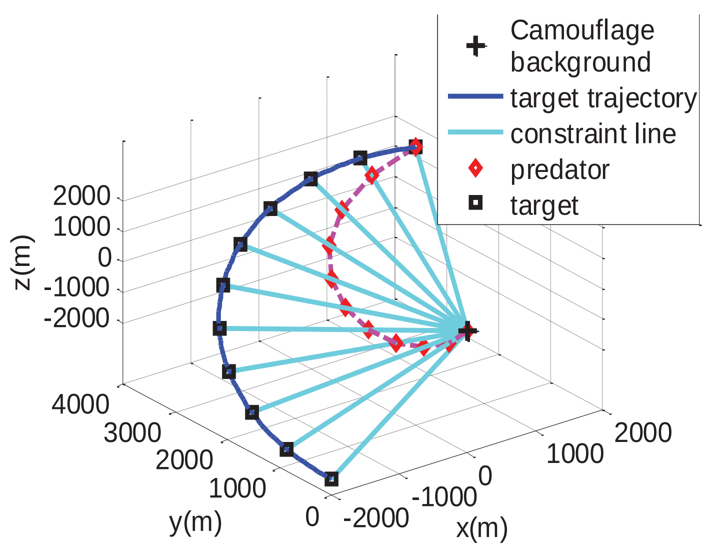

Figure 8 : Motion camouflage trajectory

In between impulsive intervals, the chaser will not be precisely located on the constraint lines all the

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time. This phenomenon would probably result in the failure of a possible stealthy approach. To address this failing more frequent impulses need to be applied as shown in Figure 9. This however comes at the expense of a more costly manoeuvre with a $\Delta v = 12.33$ m/s

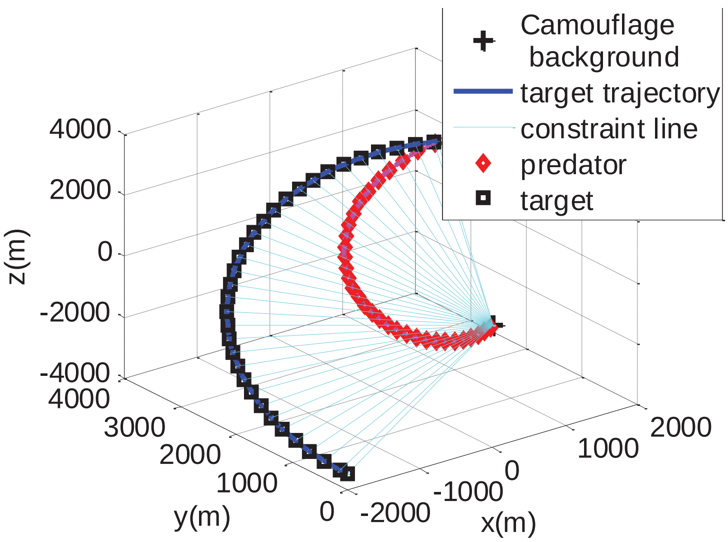

Figure 9 : Motion camouflage trajectory

V. Conclusions

This paper presented two different methodologies for group coordination and cooperative control of *n* satellites to achieve formation reconfiguration and phase angle adjustment. The first approach is based on pursuit algorithms while the second takes inspiration from motion camouflage. To validate the methodologies different scenarios are presented: a formation reconfiguration, an angular phase shift and a rendezvous manoeuvre. In summary, it has been shown that the control schemes proposed in this paper may have some potential for implementation in space missions, particularly since these approaches can be validated analytically. Future work would be the application of these control schemes in various scenarios while optimizing fuel consumption.

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Stability analysis of a landing gear mechanism with torsional degree of freedom

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Abstract - In this study, stability of a landing gear mechanism with torsional degree of freedom is analyzed. Derivation of the equations of motion of the model with torsional degree of freedom and the von Schlippe tire model are presented. Nonlinear model is linearized and Routh-Hurwitz criterion is applied. Stability analysis is conducted in the e-v plane for different values of the torsional spring rate c and in the k-v plane for different values of the relaxation length σ and vertical force Fz . Percentages of the stable regions are computed. Effects of the variation of the caster length e, half contact length a and their ratio on stable regions are analyzed. Results and conclusions about the variation of stability are presented and constructive recommendations are given.

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STABILITY ANALYSIS OF A LANDING GEAR MECHANISM WITH TORSIONAL DEGREE OF FREEDOM

Strictly as per the compliance and regulations of:

Stability analysis of a landing gear mechanism with torsional degree of freedom

Elmas Atabay [α], Ibrahim Ozkol [α]

Abstract - In this study, stability of a landing gear mechanism with torsional degree of freedom is analyzed. Derivation of the equations of motion of the model with torsional degree of freedom and the von Schlippe tire model are presented. Nonlinear model is linearized and Routh-Hurwitz criterion is applied. Stability analysis is conducted in the e–v plane for different values of the torsional spring rate c and in the k–v plane for different values of the relaxation length σ and vertical force F_Z. Percentages of the stable regions are computed. Effects of the variation of the caster length e, half contact length a and their ratio on stable regions are analyzed. Results and conclusions about the variation of stability are presented and constructive recommendations are given.

I. Introduction

Vibration of aircraft steering systems has been a problem of great concern since the production of first airplanes. Shimmy is an oscillatory motion of the landing gear in lateral and torsional directions, caused by the interaction between the dynamics of the tire and the landing gear, with a frequency range of 10-30 Hz. Though it can occur in both nose and main landing gear, the first one is more common. Shimmy is a dangerous condition of self - excited oscillations driven by the interaction between the tires and the ground that can occur in any wheeled vehicle. Problem of shimmy occurs in ground vehicle dynamics and aircraft during taxiing and landing. In other words, shimmy takes places either during landing, take-off or taxi and is driven by the kinetic energy of the forward motion of the aircraft. It is a combined motion of the wheel in lateral, torsional and longitudinal directions.

II. Shimmy

Shimmy can occur in steerable wheels of cars, trucks and motorcycles, as well as trailers and tea carts. Invehicle dynamics, shimmy is the unwanted oscillation of a rolling wheel about a vertical axis. It can occur in taxiing aircraft, as well. In the case of a shopping cart wheel, it is caused by the coupling between transverse and pivot degrees of freedom of the wheel. In the case of landing gear, shimmy is the result of the coupling between tire forces and landing gear bending and torsion. In other words, basic cause of shimmy is energy transfer from tireground contact force and vibration modes of the landing gear system. Shimmy is an unstable phenomenon and it occurring with a certain combination of parameters such as mass, elastic quantities, damping, geometrical quantities, speed, excitation forces and nonlinearities such as friction and freeplay. It is difficult to determine shimmy analytically since it is a very complex phenomenon, due to factors such as wear and ground conditions that are hard to model. Small differences in physical conditions can lead to extremely different results. For example, it is reported in [1] that a new small fighter aircraft whose name is withheld, has displayed to vibrations during low and high speed taxi tests and first several landings and take - offs, but shimmy vibrations with frequencies in the range 22-26 Hz were experienced during next several landings and take-offs at certain speeds, especially during landing. This demonstrates the effect of wear on landing gear shimmy. In the reported case, it was seen that tightening the rack too tight against the pinion prevented the wheel from turning, while tightening it less tight caused the vibration to disappear but reappear in the following flights.

Ground control of aircraft is extremely important since severe shimmy can result in loss of control or fatigue failure of landing gear components. Vibration of aircraft steering systems deserves and has gained attention since shimmy is one of the most important problems in landing gear design. Shimmy is reported to be due to the forces produced by runway surface irregularities and nonuniformities of the wheels [2-5]. Modeling of aircraft tires presents similar challenges to those involved in modeling automotive tires in ground vehicle dynamics, on a much larger scale in terms size and loads on the tire [6]. Shimmy is a complex phenomenon influenced by many parameters. Causes of shimmy can be listed as follows [2,7-10].

- Insufficient overall torsional stiffness of the gear about the swivel axis
- Inadequate trail, since positive trail reduces shimmy
- Improper wheel mass balancing about the swivel axis
- Excessive torsional freeplay
- Low torsional stiffness of the strut
- Flexibilities in the design of the suspension
- Surface irregularities
- Nonuniformities of the wheels
- Worn parts

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III. Detection and Suppression of Shimmy

Shimmy is a great concern in aircraft landing gear design and maintenance. Prediction of nose landing gear shimmy is an essential step in landing gear design because shimmy oscillations are often detected during the taxi or runway tests of an aircraft, when it is no longer feasible to make changes on the geometry or stiffness of the landing gear. Although shimmy was observed in earlier aircraft as well, there were no extra shimmy damping equipments installed. Historically, France and Germany tended to deal with shimmy in the design phase, while in United States, the trend was to solve the problem after its occurrence. Currently, the general methodology is to employ a shimmy damper and structural damping. A shimmy damper, acting like a shock absorber in a rotary manner, is often installed in the steering degree of freedom to damp shimmy. It is a hydraulic damper with stroke limited to a few degrees of yaw. A shimmy damper restrains the movement of the nose wheel, allowing the wheel to be steered by moving it slowly, but not allowing it to move back and forth rapidly. It consists of a tube filled with hydraulic fluid causing velocity dependent viscous damping forces to form when a shaft and piston are moved through the fluid. Oleo-pneumatic shock absorbers are the most common shock absorber system in medium to large aircraft, since they provide the best shock absorption ability and effective damping. Such an absorber has two components: a chamber filled with compressed gas, acting as a spring and absorbing the vertical shock and hydraulic fluid forced through a small orifice, forming friction, slowing the oil and causing damping. Another common cure is to replace the tires even though they may not be worn out [10-12].

Shimmy started being investigated in 1920's both theoretically and experimentally and soon it became clear that it is caused not by a single parameter but by the relationships between parameters. Effects of acceleration and deceleration on shimmy have been reported to be examined, and the accelerating system is found to be slightly less stable [13]. Number of publications available in literature on landing gear shimmy is limited because many developments are proprietary and are not published in literature.

IV. Literature Survey

Many papers have been published addressing shimmy as a vehicle dynamics problem. In that perspective, tire is the most important item, and tire models have been investigated. [13] examines the wheel shimmy problem and its relationship with longitudinal tire forces, vehicle motions and normal load oscillations. [8] compares different dynamic tire models for the analysis of shimmy instability. [3] is an investigation of tire parameter variations in wheel shimmy, by considering the shimmy resulting from the elasticity of a pneumatic tire, particularly in taxiing aircraft. [14] is on the application of perturbation methods to investigate the limit cycle amplitude and stability of the wheel shimmy problem. [7] deals with the shimmy stability of twin-wheeled cantilevered aircraft main landing gear. The objective in [15] is to develop software on assessing shimmy stability of a general class of landing gear designs using linear and nonlinear landing gear shimmy models. [16] studies the periodic shimmy vibrations and chaotic vibrations of a simplified wheel model using bifurcation theory. [17] is on tire dynamics and is a development to deal with large camber angles and inflation pressure changes. [18] is another study on tire dynamics, where stability charts show the behavior of the system in terms of certain parameters such as speed, caster length, damping coefficient and relaxation length. [19] is an experimental study on wheel shimmy where system parameters are identified, stability boundaries and vibration frequencies are obtained on a test rig for an elastic tire. Dependence of shimmy oscillations in the nose landing gear of an aircraft on tire inflation pressure are investigated in [20]. The model derived in [21] is used and it is concluded that landing gear is less susceptible to shimmy oscillations at inflation pressures higher than the nominal.

Transverse vibrations of landing gear struts with respect to a hull of infinite mass have been studied theoretically in [22]. Similarly, [23] presents a nonlinear model describing the dynamics of the main gear wheels relative to the fuselage.

Lateral dynamics of nose landing gear shimmy models has gained some attention. Lateral response of a nose landing gear has been investigated in [10] where nonlinearities arise due to torsional freeplay. In [24], lateral response to ground-induced excitations due to runway roughness is taken into consideration as well. Lateral stability of a nose landing gear with a closed loop hydraulic shimmy damper is presented in [12]. Closed form analytical expressions for shimmy velocity and shimmy frequency are derived in regard to the lateral dynamics of a nose landing gear in [25].

A dynamic model of an aircraft nosegear is developed in [9] and effects of design parameters such as energy absorption coefficient of the shimmy damper, the location of the center of gravity of the landing gear, shock strut elasticity, tire compliance, friction between the tire and the runway surface and the forward speed on shimmy are investigated. It is shown in [26] that dry friction is one of the principal causes of shimmy. Bifurcation analysis of a nosegear with torsional and lateral degrees of freedom is performed in [21]. Similarly, bifurcation analysis of a nosegear with torsional, lateral and longitudinal modes is performed in [27].

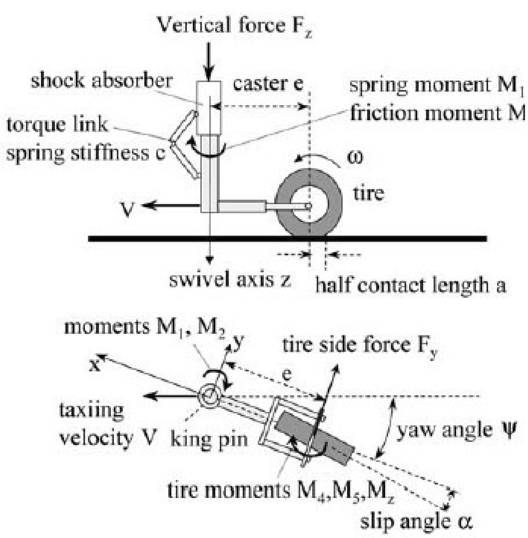
In a more mathematical study, incremental harmonic balance method is applied to an aircraft wheel shimmy system with Coulomb and quadratic damping [28] and amplitudes of limit cycles are predicted.

Theoretical research on shimmy has a long history, with the initial focus on tire dynamic behavior because tires play an important role in causing shimmy instability. Theories on tire models can be divided into stretched string models and point contact models. In the stretched string model proposed by von Schlippe, the tire centerline is represented as a string in tension, the tire sidewalls are represented by a distributed spring where the string rests and the wheel is represented by a rigid foundation for the spring. Pacejka has proposed replacing the string by a beam. The point contact method assumes the effects of the ground on the tire act at a single contact point and is much easier to implement in an analytical model.

V. Mathematical Model

a) Landing gear model

In this study, stability of a landing gear model with torsional degree of freedom is analyzed. The nonlinear mathematical shimmy model presented in [11], [29] and [30] describes the torsional dynamics of the lower parts of a landing gear mechanism and stretched string tire model. Figures 1 a and b show the physical and mathematical nose landing gear models. Dynamics of the lower part of the landing gear is described by a second order ordinary differential equation for the yaw angle ψ about the vertical axis z, while the dynamics of the tire modeled with respect to the stretched string tire model is described by a first order ordinary differential equation for the lateral tire deflection y.



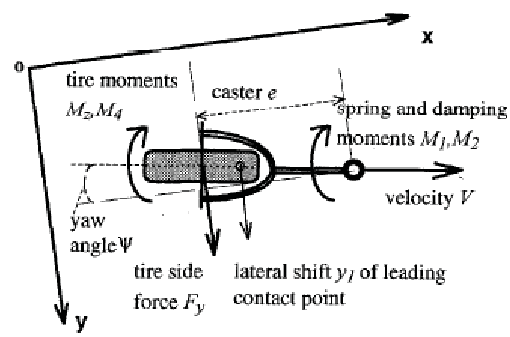


Figure 1 : a. Nose landing gear model [30], b. shimmy dynamics model [29].

$$I_z\ddot{\psi} = M_1 + M_2 + M_3 + M_4 \qquad (1)$$

Where I_z is the moment of inertia about the z axis,
M_1 is the linear spring moment between the turning tube and the torque link,
M_2 is the combined damping moment from viscous friction in the bearings of the oleo-pneumatic $_2$ shock absorber and from the shimmy damper,
M_3 is the tire moment about the z axis and
M_4 is the tire damping moment due to tire tread width..
M_1 and M_4 are external moments.
M_3 and M_4 are caused by lateral tire deformations due to side slip.
M_3 is composed of M_z, tire aligning moment about the tire center, and tire cornering moment eF_y. F_y is the wheel cornering force or the sideslip force acting with caster e as lever arm.

$$M_1 = k\dot{\psi} \qquad (2)$$

$$M_2 = c\psi \qquad (3)$$

$$M_3 = M_z - eF_y \qquad (4)$$

$$M_4 = \frac{\kappa}{v}\dot{\psi} \qquad (5)$$

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Where k is the torsional spring rate, c is the torsional damping constant, v is the taxiing velocity and κ is the tread width moment constant defined as [29]

$$\kappa = -0.15\, a^2 c_{F\alpha}\, F_z \tag{6}$$

F_y and M_z depend on the vertical force F_z and slip angle α . Tire sideslip characteristics are nonlinear. Cornering force F_y and vertical force F_z are related as

$$F_y / F_z = c_{F\alpha}\alpha, \qquad \text{for } |\alpha| \le \delta \tag{7}$$

$$F_y / F_z = c_{F\alpha}\delta\, sign(\alpha), \qquad \text{for } |\alpha| > \delta \tag{8}$$

Where δ is the limiting slip angle or the limit angle of tire force and **sign** (α) is the sign function defined as

$$sign(\alpha) = \begin{cases} 1, \text{if } \alpha > \delta \\ -1, \text{if } \alpha \le \delta \end{cases} \tag{9}$$

Slip angle may be caused by either pure yaw or pure sideslip. Pure yaw occurs when the yaw angle ψ is allowed to vary while the lateral deflection y is held at zero. Pure sideslip, on the other hand, occurs when the lateral deflection y is allowed to vary as the yaw angle ψ is held at zero [11].

An expression is given for the nonlinear sideslip characteristic in the widely used Magic Formula [7, 11, 17] as the following

$$F_y = D\sin\left[C\arctan\{B\alpha - E(B\alpha - \arctan(B\alpha))\}\right] \tag{10}$$

Where B,C, D and E are functions of the wheel load, slip angle, slip ratio and camber. B and E are related to vertical force F_z , C is the shape factor and D is the peak value of the curve.

Plots of F_y / F_z versus α will not be presented here due to lack of space, but they have similar characteristics when obtained using either (7) and (8) or the Magic Formula, thus the simple approximations given by (7) and (8) are used instead of the complicated Magic Formula. Only force and moment derivatives are needed as parameters for (7) and (8).

Aligning moment M_z is defined using a half-period sine. M_z / F_z is approximated by a sinusoidal function and the constant zero given by (11) and (12).

$$M_z / F_z = c_{M\alpha}\frac{\alpha_g}{180}\sin\left(\frac{180}{\alpha_g}\alpha\right), \quad \text{for } |\alpha| \le \alpha_g \tag{11}$$

$$M_z / F_z = 0, \qquad \text{for } |\alpha| > \alpha_g \tag{12}$$

Where α_g the limiting angle of tire moment.

b) Tire model

Tire is modeled using the elastic string theory. Lateral deflection of the tire is described as [11, 29]

$$\dot{y} + \frac{v}{\sigma} y = v\psi + (e-a)\dot{\psi} \tag{13}$$

Ground forces are transmitted to the wheel through the tire, and these forces acting on the tire footprint deflect the tire. Elastic string theory states that lateral deflection y of the leading contact point of the tire with respect to tire plane can be described as a first order differential equation given by (13). This equation is derived as follows.

Tire sideslip velocity V_t is expressed as

$$V_t = \dot{y} + \frac{y}{\tau} \tag{14}$$

Where $\tau = \dfrac{\sigma}{V}$ is the time constant, σ is the relaxation length, which is the ratio of the slip stiffness to longitudinal force stiffness.

The tire also undergoes yaw motion, leading to a yaw velocity V_r which is approximated as

$$V_r = v\psi + (e-a)\,\dot{\psi} \tag{15}$$

As the wheel rolls on the ground,

$$V_t = V_r \tag{16}$$

Substituting (14) and (15) into (16) yields (13).

An equivalent side slip angle caused by lateral deflection is used to compute cornering force F_y and aligning moment M_z and is approximated as

$$\alpha \approx \arctan\alpha = \frac{y}{\sigma} \tag{17}$$

Equations (1), (13) and (17) constitute the governing equations of the torsional motion of the landing gear and include nonlinear tire force and moment. Parameters of a light aircraft used in the computations are given in table 1.

Table 1 : Parameters used in the torsional dynamics.

Parameter	Description	Value	Unit
v	velocity	0…80	m/s
a	half contact length	0.1	m
e	caster length	0.1	m
I_z	moment of inertia	1	kg m^2
F_z	vertical force	9000	N
c	torsional spring rate	-100000	Nm/rad
$c_{F\alpha}$	side force derivative	20	1/rad
$c_{M\alpha}$	moment derivative	-2	m/rad
k	torsional damping constant	0…-50	Nm/rad/s
κ	tread width moment constant	-270	Nm^2/rad
$\sigma = 3a$	relaxation length	0.3	m
α_g	limit angle of tire moment	10	deg
δ	limit angle of tire force	5	deg

c) Linearization

In order to use linear analysis tools, nonlinear landing gear model has to be linearized. Following this, linear stability analysis will be performed.

Within a small range of the side slip angle α, cornering force F_y and the ratio M_z / F_z can be approximated proportional to the side slip angle. Based on this assumption, substituting equations (7), (8), (11) and (12) into (4) yields (20), the complete expression for the tire moment M_3

$$F_y = \begin{cases} c_{F\alpha}\alpha F_z, & |\alpha| \le \delta \\ c_{F\alpha}\delta\, sign(\alpha), & |\alpha| > \delta \end{cases} \tag{18}$$

$$M_z = \begin{cases} c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z, & |\alpha| \le \alpha_g \\ 0, & |\alpha| \ge \alpha_g \end{cases} \tag{19}$$

$$M_3 = \begin{cases} -ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \alpha \le -\alpha_g \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & -\alpha_g < \alpha < -\delta \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, F_z, & -\delta < \alpha < \delta \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \delta < \alpha < \alpha_g \\ ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \alpha > \alpha_g \end{cases} \tag{20}$$

Substituting (17) into (20) and expressing M_3 in the neighborhood of $\alpha = 0$ or $y = 0$ yields

$$M_3 = c_{M\alpha}\frac{\alpha_g}{180}\sin\left(\frac{180}{\alpha_g}\frac{y}{\sigma}\right)F_z - ec_{F\alpha}\frac{y}{\sigma}F_z \tag{21}$$

M_3 can be linearized using the Taylor series expansion as

$$\left.\frac{\partial M_3}{\partial y}\right|_{y=0} = c_{M\alpha}\frac{\alpha_g}{180}\cos\left(\frac{180}{\alpha_g}\frac{y}{\sigma}\right)F_z\frac{180}{\alpha_g\sigma} - ec_{F\alpha}\frac{1}{\sigma}F_z\Bigg|_{y=0} = \frac{F_z}{\sigma}\left(c_{M\alpha} - ec_{F\alpha}\right) \tag{22}$$

Defining the state variables as $(\psi, \dot{\psi}, y)$ gives the linearized model as three ordinary differential equations of first order as

$$\begin{bmatrix} \dot{\psi} \\ \ddot{\psi} \\ \dot{y} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ c_1 & c_2 & c_3 \\ v & c_4 & c_5 \end{bmatrix} \begin{bmatrix} \psi \\ \dot{\psi} \\ y \end{bmatrix} \tag{23}$$

Where

$$c_1 = \frac{c}{I_z} \tag{24}$$

$$c_2 = \frac{k}{I_z} + \frac{\kappa}{vI_z} \tag{25}$$

$$c_3 = \frac{(c_{M\alpha} - ec_{F\alpha})F_z}{I_z \sigma} \tag{26}$$

$$c_4 = e - a \tag{27}$$

$$c_5 = \frac{-v}{\sigma} \tag{28}$$

Characteristic equation is obtained as

$$\lambda^3 - (c_2 + c_5)\lambda^2 + (c_2 c_5 - c_1 - c_3 c_4)\lambda + (c_1 c_5 - vc_3) = 0 \tag{29}$$

Routh - Hurwitz criterion is applied to determine stability boundaries of the linear model. This criterion states that for a third order polynomial $a_3 s^3 + a_2 s^2 + a_1 s + a_0 = 0$ to be stable, $a_n > 0$ and $a_2 a_1 > a_3 a_0$. By inspecting the characteristic equation (29)

$$a_3 = 1$$
$$a_2 = -(c_2 + c_5)$$
$$a_1 = c_2 c_5 - c_1 - c_3 c_4$$
$$a_0 = c_1 c_5 - vc_3$$

Thus, for the landing gear model to be stable

$$-(c_2 + c_5) > 0 \tag{30}$$

$$c_2 c_5 - c_1 - c_3 c_4 > 0 \tag{31}$$

$$c_1 c_5 - vc_3 > 0 \tag{32}$$

$$-(c_2 + c_5)(c_2 c_5 - c_1 - c_3 c_4) > c_1 c_5 - vc_3 \tag{33}$$

VI. STABILITY REGIONS

Stability plots will not be presented in order to save space, but numerical values regarding the stable percentages of the parameter space will be presented.

1. *Stability boundaries in the* $\mathrm{e} - \mathrm{v}$ *plane for different values of* c

Stability regions are analyzed in the $\mathrm{e} - \mathrm{v}$ plane for different values of the torsional spring rate c. Torsional damping constant k is taken as -50 Nm/rad/s. Caster length evaries between -0.1 and 0.3 m, while the velocity v varies between 0 and 200 m/s. ctakes the values 0, -50000 and -100000 Nm/rad. Table 2 shows the percentages of the area of the stable region in the $\mathrm{e} - \mathrm{v}$ plane for the values of c considered.

Table 2 : Percentage of stable region in the $\mathrm{e} - \mathrm{v}$ plane for different values of c.

	Percentage of stable region
c =-100000 Nm/rad	97.3 % stable
c =-50000 Nm/rad	68.7 % stable
c =0 Nm/rad	33.6 % stable

2. *Stability boundaries in the* $\mathrm{k} - \mathrm{v}$ plane for different values of σ and F_z

Stability regions are analyzed in the $k - v$ plane for different values of the relaxation length σ and vertical force F_z. Velocity v varies between 0 and 100 m/s while torsional damping constant k *varies* between -120 and 20 Nm/rad/s when analyzing stability for different values of σ and between -100 and 50 Nm/rad/s when analyzing stability for different values of F_z. It is seen that for $\sigma <$ 0.1, there is more instability at small velocities and more stability at large velocities, while for $\sigma > 0.1$, there is more stability at small velocities and more instability at large velocities. Larger values of F_z and v require larger values of $-k$ for stability and there is no instability for negative damping coefficients below 16 m/s. Generally speaking, shimmy occurs under a certain damping value, depending on the velocity. There is stability for all values of the damping constant k for small velocities for velocities below 16 m/s. Tables 3 and 4 show the percentages of the area of the stable region in the $k - v$ plane for the values of σ and F_z considered, respectively.

Table 3 : Percentage of stable region in the $\mathrm{k} - \mathrm{v}$ plane for different values of σ .

	Percentage of stable region
σ =0.02 m	78.3 % stable
σ =0.07 m	65.4 % stable
σ =0.12 m	61.3 % stable
σ =0.17 m	60.7 % stable
σ =0.22 m	61.4 % stable
σ =0.27 m	62.9 % stable
σ =0.32 m	64.4 % stable

Table 4 : Percentage of stable region in the $\mathrm{k-v}$ plane for different values of F_z

	Percentage of stable region
$F_z = 0$ N	72.5 % stable
$F_z = 5000$ N	58.5 % stable
$F_z = 10000$ N	45.1 % stable
$F_z = 15000$ N	32.5 % stable

3. *Effects of the caster length e and half contact length a on stability boundaries*

Effects of the variation of the caster length e, half contact length a and their ratio on stability boundaries are analyzed below. Increments and decrements in the stable portion of the $\mathrm{e-v}$ and $\mathrm{k-v}$ planes are presented quantitatively in tabular form.

i. *Effects of the half contact length a on stability boundaries in the $\mathrm{e-v}$ plane for different values of c*

This part of the stability analysis of the linear model is conducted in the $\mathrm{e-v}$ plane, thus the effect of the caster length e is already contained within the calculations. For this reason, effect of the half contact length a on stability of the model will be analyzed. Effects of 5 % and 10 % increase and decrease of the half contact length a are analyzed in this section.

- A 5 % increase in the half contact length a from 0.1 m to 0.105 m leads to an increase in the unstable region in the $\mathrm{e-v}$ plane, as can be seen by inspecting table 9. It is observed that there is a greater increase in the unstable region for large values of the torsional spring rate c .
- A 10 % increase in the half contact length a from 0.1 m to 0.11 m leads to a further increase in the unstable region in the $\mathrm{e-v}$ plane, as can be seen by inspecting table 9. As was the case for a half contact length of 0.105 m, there is a greater increase in the unstable region for large values of the torsional spring rate c.
- A 5 % decrease in the half contact length a from 0.1 m to 0.095 m leads to a increase in the stable region in the $\mathrm{e-v}$ plane, as seen by inspecting table 9. There are almost no instabilities in the $\mathrm{e-v}$ plane for a high torsional spring rate c. It observed that there is a greater increase in the stable region for large values of the torsional spring rate c.
- A 10 % decrease in the half contact length a from 0.1 m to 0.09 m leads to a further increase in the stable region in the $\mathrm{e-v}$ plane, as seen by inspecting table 9. As was the case for a half contact length of 0.095 m, there are almost no instabilities in the $\mathrm{e-v}$ plane for a high torsional spring rate c and there is a greater increase in the stable region for large values of the torsional spring rate c .

The following table quantifies the amounts of increments and decrements in the stability of the $\mathrm{e-v}$ plane for variations of the half contact length a. Values given for a half contact length of 0.1 m show how much of the analyzed region in the $\mathrm{e-v}$ plane is stable. Values given in the following lines for half contact lengths of 0.105 m, 0.11 m, 0.095 m and 0.09 m show how much of the analyzed region are stable and how much increment or decrement exists with respect to the stability of the system having a half contact length of 0.1 m.

Table 5 : Effect of variation of the half contact length on stability in the $\mathrm{e-v}$ plane.

		→ direction of decreasing torsional spring rate		
		c=-100000 Nm/rad	c=-50000 Nm/rad	c=0 Nm/rad
	a=0.1 m	97.3 % stable	68.8 % stable	33.6 % stable
increment in the half contact length	a=0.105 m	92.8 % stable	62.2 % stable	26.8 % stable
		4.6 % decrement	9.5 % decrement	20 % decrement
	a=0.11 m	87.6 % stable	55.4 % stable	22.6 % stable
		10 % decrement	19.3 % decrement	32.7 % decrement
decrement in the half contact length	a=0.095 m	99 % stable	74.6 % stable	41.6 % stable
		1.7 % increment	8.5 % increment	23.9 % increment
	a=0.09 m	99 % stable	79.7 % stable	48.9 % stable
		1.7 % increment	15.9 % increment	45.7 % increment

ii. *Effects of the caster length e and half contact length a on stability boundaries in the $\mathrm{k-v}$ plane*

This part of the stability analysis of the linear model is conducted in the $\mathrm{k-v}$ plane. Effects of the caster length e, half contact length a and their ratio on stability of the model will be analyzed. Effects of 5 % and 10 % increase and decrease of e and a and variation of their ratio are also analyzed.

Effects of the caster length e *on stability boundaries in the* $k\text{–}v$ *plane for different values of* σ

This part of the stability analysis is conducted for different values of the relaxation length σ, thus the effect of the half contact length ais already contained since $\sigma = 3a$. For this reason, effect of the caster lengtheon stability of the model will be analyzed. Effects of 5 % and 10 % increase and decrease of e are analyzed in this section.

- A 5 % increase in the caster length e from 0.1 m to 0.105 m leads to an increase in the stable region in The $k\text{–}v$ plane, as can be seen by inspecting table 10. It is observed that there is a smaller increase in the stable region for large values of the relaxation length σ. Increase in the stable region is almost unnoticeable for relaxation lengths above 0.12 m.
- A 10 % increase in the caster length e from 0.1 m to 0.11 m leads to a further increase in the stable region in the $k\text{–}v$ plane, as can be seen by inspecting table 10. As was the case for a caster length of 0.095 m, there is a smaller increase in the stable region for large values of relaxation length σ and the increase in the stable region is almost unnoticeable for relaxation lengths above 0.12 m.
- A 5 % decrease in the caster length e from 0.1 m to 0.095 m leads to an increase in the unstable region in the $k\text{–}v$ plane, especially for low velocities, as seen from table 10. It is observed that there is a smaller increase in the unstable region for large values of the relaxation length σ. Increase in the stable region is almost unnoticeable for relaxation lengths above 0.12 m.
- A 10 % decrease in the caster length e from 0.1 m to 0.09 m leads to a further increase in the unstable region in the $k\text{–}v$ plane, especially for low velocities, as seen from table 10. As was the case for a caster length of 0.095 m, there is a smaller increase in the unstable region for large values of relaxation length σ and the increase in the unstable region is almost unnoticeable for relaxation lengths above 0.12 m.
- Table 6 quantifies the amount of increments and decrements in the stability of the $k\text{–}v$ plane for variations of the caster length e. Values given for a caster length of 0.1 m show how much of the analyzed region in the $k\text{–}v$ plane is stable. Values given in the following lines for half caster lengths of 0.105 m, 0.11 m, 0.095 m and 0.09 m show how much of the analyzed region are stable and how much increment or decrement exists with respect to the stability of the system having a caster length of 0.1 m.

Table 6 : Effect of variation of the caster length on stability in the $k\text{–}v$ plane.

		→ direction of increasing relaxation length						
		σ=0.02 m	σ=0.07 m	σ=0.12 m	σ=0.17 m	σ=0.22 m	σ=0.27 m	σ=0.32 m
	e=0.1 m	78.3 % stable	65.4 % stable	61.3 % stable	60.7 % stable	61.4 % stable	62.9 % stable	64.4 % stable
increment in the caster length	e=0.105 m	81.3 % stable	66.7 % stable	61.9 % stable	60.9 % stable	61.5 % stable	62.9 % stable	64.3 % stable
		3.7 % increment	3 % increment	2 % increment	0.9 % increment	0.1 % increment	0.07 % decrement	0.2 % decrement
	e=0.11 m	84.3 % stable	68.1 % stable	62.7 % stable	61.2 % stable	61.6 % stable	62.9 % stable	64.2 % stable
		7.7 % increment	4.1 % increment	2.2 % increment	0.9 % increment	0.3 % increment	0.04 % decrement	0.4 % decrement
decrement in the caster length	e=0.095 m	75.3 % stable	64.1 % stable	60.7 % stable	60.4 % stable	61.4 % stable	63 % stable	64.6 % stable
		3.8 % decrement	2 % decrement	1.1 % decrement	0.5 % decrement	0.1 % decrement	0.02 % increment	0.2 % increment
	e=0.09 m	72.7 % stable	63 % stable	60.2 % stable	60.2 % stable	61.3 % stable	63 % stable	64.8 % stable
		7.2 % decrement	4 % decrement	1.8 % decrement	0.9 % decrement	0.1 % decrement	0.2 % increment	0.5 % increment

iii. *Effects of the caster lengtheand half contact length* a *on stability boundaries in the* $k\!-\!v$ *plane for different values of* F_z

Effect of the variation of the ratio e/a on the stability of the model is analyzed. Effects of 5 % and 10 % increase and decrease of e/a are presented in table 7.

Table 7 : Effect of variation of the ratio e/a on stability in the $k\!-\!v$ plane.

	e/a =1				
	e =0.08 m a =0.08 m	e =0.09 m a =0.09 m	e =0.1 m a =0.1 m	e =0.11 m a =0.11 m	e =0.12 m a =0.12 m
F_z =0 N	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable
F_z =5000 N	58.5 % stable	58.3 % stable	58.7 % stable	59 % stable	58.2 % stable
F_z =10000 N	45.2 % stable	44.9 % stable	45.4 % stable	45.9 % stable	44.8 % stable
F_z =15000 N	32.5 % stable	32.2 % stable	33 % stable	33.7 % stable	32.1 % stable

	e/a =1.05				
	e =0.11 m a =0.105 m	e =0.105 m a =0.1 m	e =0.1 m a =0.095 m	e =0.095 m a =0.09 m	e =0.09 m a =0.085 m
F_z =0 N	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable
F_z =5000 N	58.3 % stable	58.4 % stable	58.4 % stable	58.6 % stable	58.6 % stable
F_z =10000 N	44.9 % stable	45.1 % stable	45.3 % stable	45.3 % stable	45.6 % stable
F_z =15000 N	32.4 % stable	32.5 % stable	32.7 % stable	33 % stable	33.3 % stable

	e/a =1.1				
	e =0.12 m a =0.11 m	e =0.11 m a =0.1 m	e =0.1 m a =0.09 m	e =0.09 m a =0.082 m	e =0.082 m a =0.075 m
F_z =0 N	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable
F_z =5000 N	58.1 % stable	58.4 % stable	58.5 % stable	58.6 % stable	58.9 % stable
F_z =10000 N	44.7 % stable	45.1 % stable	45.3 % stable	45.9 % stable	46.3 % stable
F_z =15000 N	32.1 % stable	32.5 % stable	33.1 % stable	33.7 % stable	34.2 % stable
	e/a =0.95				
	e =0.09 m a =0.095 m	e =0.095 m a =0.1 m	e =0.1 m a =0.105 m	e =0.105 m a =0.11 m	e =0.11 m a =0.116 m
F_z =0 N	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable
F_z =5000 N	58.6 % stable	58.5 % stable	58.5 % stable	58.4 % stable	58.4 % stable
F_z =10000 N	45.3 % stable	45.2 % stable	45.1 % stable	45 % stable	44.9 % stable
F_z =15000 N	32.8 % stable	32.6 % stable	32.5 % stable	32.3 % stable	32.2 % stable

	e/a =0.9				
	e =0.081 m a =0.09 m	e =0.09 m a =0.1 m	e =0.1 m a =0.11 m	e =0.11 m a =0.122 m	e =0.122 m a =0.135 m
F_z =0 N	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable	72.5 % stable
F_z =5000 N	58.8 % stable	58.6 % stable	58.5 % stable	58.5 % stable	58.5 % stable
F_z =10000 N	45.7 % stable	45.4 % stable	45.2 % stable	45.2 % stable	45.4 % stable
F_z =15000 N	33.2 % stable	32.7 % stable	32.5 % stable	32.3 % stable	32.3 % stable

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VII. Results and Conclusions

1. Results and conclusions about the variation of stability in the $e-v$ *plane and recommendations*

- A 5 % increase in the half contact length a leads to an increase in the unstable region in the $e-v$ plane.
- A 10 % increase in the half contact length a leads to a further increase in the unstable region in the $e-v$ plane.
- A 5 % decrease in the half contact length a leads to an increase in the stable region in the $e-v$ plane. For the parameters considered, there were no instabilities in the $e-v$ plane for a high torsional spring Rate c.
- A 10 % decrease in the half contact length a leads to a further increase in the stable region in the $e-v$ plane. For the parameters considered, there were no instabilities in the $e-v$ plane for a high torsional spring rate c.
- The increments in the stable and unstable regions are greater for large values of the torsional spring rate c.
- Increments in the half contact length lead to increments in the unstable region in the $e-v$ plane. In other words, increasing the half contact length decreases stability.
- Decrements in the half contact length lead to increments in the stable region in the $e-v$ plane. In other words, decreasing the half contact length increases stability.

2. Results and conclusions about the variation of stability in the $k-v$ *plane and recommendations*

- A 5 % increase in the caster length e leads to an increase in the stable region in the $k-v$ plane. There is a smaller increase in the stable region for large values of the relaxation length σ such that the increase in the stable region is almost negligible for relaxation lengths above 0.12 m.
- A 10 % increase in the caster length e leads to a further increase in the stable region in the $k-v$ plane. There is a smaller increase in the stable region for large values of relaxation length σ and the increase in the stable region is almost negligible for relaxation lengths above 0.12 m.
- A 5 % decrease in the caster length e leads to an increase in the unstable region in the $k-v$ plane, especially for low velocities. There is a smaller increase in the unstable region for large values of the relaxation length σ such that the increase in the stable region is almost negligible for relaxation lengths above 0.12 m.
- A 10 % decrease in the caster length e from leads to a further increase in the unstable region in the $k-v$ plane, especially for low velocities. There is a smaller increase in the unstable region for large values of relaxation length σ and the increase in the unstable region is almost negligible for relaxation lengths above 0.12 m.
- Increments in the stable and unstable regions are smaller for large values of the relaxation length σ.
- Increments in the caster length lead to increments in the stable region in the $k-v$ plane. In other words, increasing the caster length increases stability.
- Decrements in the half contact length lead to increments in the unstable region in the $k-v$ plane. In other words, decreasing the caster length decreases stability.

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LQ Previewed Tracking For Biproper Systems

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Abstract - In linear quadratic previewed control, strictly proper systems are used for tracking performance by the feedforward control proportional to the measurable exogenous input. However, state space models that employ sensors to measure exogenous inputs are sometimes biproper. A classical example for biproper system is a small aircraft regulation in cruise condition where the gust inputs are measured but the ride quality is deteriorated. For such systems, the previewed control with a biproper system is required. In this paper, the procedure for strictly proper system is extended and a modified Riccati matrix differential equation for biproper system is presented.

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LQ PREVIEWED TRACKING FOR BIPROPER SYSTEMS

Strictly as per the compliance and regulations of:

LQ Previewed Tracking For Biproper Systems

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Abstract - In linear quadratic previewed control, strictly proper systems are used for tracking performance by the feedforward control proportional to the measurable exogenous input. However, state space models that employ sensors to measure exogenous inputs are sometimes biproper. A classical example for biproper system is a small aircraft regulation in cruise condition where the gust inputs are measured but the ride quality is deteriorated. For such systems, the previewed control with a biproper system is required. In this paper, the procedure for strictly proper system is extended and a modified Riccati matrix differential equation for biproper system is presented.

I. Introduction

In linear quadratic previewed control, tracking performance by the feedforward control proportional to an exogenous input is well known [1-5]. The state space model in these problems incorporates a strictly proper system. However, models that employ sensors to measure exogenous inputs are sometimes biproper. A classical example is a small aircraft regulation in cruise condition wherein the normal acceleration is regulated for a smooth ride quality in the presence of gust inputs. For such systems, previewed control for biproper system is required. In this paper, the procedure for strictly proper system in Ref. 1 is extended and a modified Riccati matrix differential equation for biproper system is studied further.

There is substantial progress in gust alleviation [6,7] and in structural control problems with accelerometers [8] that are biproper systems. Yet, especially in gust alleviation, investments for forward-looking sensor have been made to measure the presence of gust ahead of a flight path [9]. We are required to use the previewed measurements and restore the performance in the time windows of gust using a feedforward control law. Therefore, linear quadratic previewed (LQP) control for biproper systems is considered. In normal acceleration regulation, the inner loop controller is assumed fixed. Thus, the feedforward actions linear to the measurements of exogenous inputs are considered in simulation.

It is possible to convert a biproper system into a strictly proper system and develop a LQP control within the framework of strictly proper system. To this end, consider a scalar differential equation with respect to time,

$$\dot{n}(t) = an(t) + bu(t)$$
$$y(t) = n(t) + du(t)$$

The non-zero constant 'd' defines a biproper system. With an actuator model,

$$\dot{u}(t) = \tau u(t) + g\, u_c(t),$$

the augmented system without the time variable in arguments becomes,

$$\begin{bmatrix} \dot{n} \\ \dot{u} \end{bmatrix} = \begin{bmatrix} a & b \\ 0 & \tau \end{bmatrix}\begin{bmatrix} n \\ u \end{bmatrix} + \begin{bmatrix} 0 \\ g \end{bmatrix} u_c(t)$$
$$y = \begin{bmatrix} 1 & d \end{bmatrix}\begin{bmatrix} n \\ u \end{bmatrix}.$$

By defining a command input $u_c(t)$, clearly the problem converts itself into a strictly proper system. However, the state feedback control problem simultaneously modifies itself into an output feedback problem. Thus a solution matrix to the Riccati differential equation (RDE) is not always direct as in the case of a state feedback system. In fact, a steady state solution using the algebraic Riccati equation itself calls for parameter optimization [10,11].

In Section 2, modified RDE and its symmetric matrices are presented. Section 3 provides stability and optimality conditions to solve the RDE. In Section 4, a scalar example is used to compare the tracking performance of biproper and strictly proper systems. Conclusions are presented in Section 5.

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II. Main Results

In deriving an optimal control law $u^*(t)$, consider the following problem statement.

$$\underbrace{\text{Minimize}}_{u} \; J(x,u) \tag{1}$$

subject to the following constraints,

$$\dot{x}(t) = Ax(t) + Bu(t) + Ew(t) \tag{2}$$

$$y(t) = Cx(t) + Du(t) + Fw(t) \tag{3}$$

The state, input and output vectors are represented by $x \in R^n$, $u \in R^m$ and $y \in R^r$, respectively. The disturbance input vector is given by $w \in R^p$. The compatible matrices A, B, C, D and F are assumed to be time invariant. Define the cost function J,

$$J =< e(T), \bar{Q}e(T) > + \int_{t_0}^{T} \{< e(t), Qe(t) > + < u(t), Ru(t) >\} dt \tag{4}$$

Where, $< v_1, v_2 >$ is the inner product for the compatible vectors v_1 and v_2. The error vector is $e(t) = z(t) - y(t)$ and $z(t)$ is the reference inputs. The Hamiltonian with costate vector $p(t)$ is,

$$\begin{aligned} H =& \tfrac{1}{2} < e(t), Qe(t) > + \tfrac{1}{2} < u(t), Ru(t) > \\ &+ < Ax(t), p(t) > + < Bu(t), p(t) > + < Ew(t), p(t) > \end{aligned} \tag{5}$$

Following the necessary conditions for optimality,

$$\frac{\partial H}{\partial u(t)} = 0 \quad \& \quad \frac{\partial H}{\partial x(t)} = -\dot{p}(t)$$

We have the control law as a function of the costate vector,

$$u = -(R + D'QD)^{-1}[B'p + D'QCx - D'Q(z - Fw)] \tag{6}$$

$$\dot{p} = -A'p - CQDu - C'QCx + C'Q(z - Fw) \tag{7}$$

Here $(.)'$ refers the transpose of the vector or matrix $(.)$. For brevity, the time variable in the arguments is suppressed. Since $Q \geq 0$ (positive semidefinite) and $R > 0$ (positive definite), the sufficient condition,

$$\frac{\partial^2 H}{\partial u^2} = \hat{R} = R + D'QD > 0,$$

for a minimum $u(t)$ is met. Rewriting Eqn.(7)

$$\dot{p} = -\bar{A}'p + (W\hat{R}^{-1}W' - C'QC)x + (C'Q - W\hat{R}^{-1}D'Q)[z - Fw], \tag{8}$$

the matrices $W = C'QD$ and $\bar{A} = A - B\hat{R}^{-1}W'$ are defined. To derive RDE, consider the costate vector $p(t)$

$$p = Kx - g \tag{9}$$

such that the control law in Eqn.(6) modifies to,

$$u^*(t) = -\hat{R}^{-1}\{\bar{K}x - B'g - D'Q[z - Fw]\}, \quad \forall t \in [t_0, T]. \tag{10}$$

The state feedback gain and the closed loop system matrix are defined as below,

$$\bar{K} = B'K + W' \tag{11}$$

$$\dot{x} = A_c x + B\hat{R}^{-1}B'g + Ew + B\hat{R}^{-1}D'Q[z - Fw]. \tag{12}$$

Note that in the stability matrix $A_c = (\bar{A} - B\hat{R}^{-1}B'K)$, $\bar{A}$ serves as an open loop matrix. It is important to guarantee that the matrix A_c is stable. Consider the time derivative of $p(t)$ in Eqn. (9),

$$\begin{aligned}\dot{p} &= [\dot{K} + K\bar{A} - KB\hat{R}^{-1}B'K]x + KB\hat{R}^{-1}B'g - \dot{g} \\ &\quad + KEw + KB\hat{R}^{-1}D'Q[z - Fw]\end{aligned} \tag{13}$$

Equating the coefficients of like terms in Eqn.(7) and (13), the RDE and $\dot{g}$ -equation for tracking performance are,

$$\dot{K} = -K\bar{A} - \bar{A}'K + KB\hat{R}^{-1}B'K + W\hat{R}^{-1}W' - C'QC \tag{14}$$

$$\dot{g} = -A_c{}'g + [(KB + W)\hat{R}^{-1}D' - C')]Q[z - Fw] + KEw \tag{15}$$

The boundary conditions for the forward integration are known to be $g(T) = 0$ and $K(t_0) = K_0$. For finite duration optimal control problem in time $[t_0, T]$, the transversality conditions [1], lead to the following end conditions,

$$K(T) = S^{-1}[C'\bar{Q}C - \bar{W}\hat{R}^{-1}W'] \tag{16}$$

$$g(T) = S^{-1}[C'\bar{Q} - \bar{W}\hat{R}^{-1}D'Q][z(T) - Fw(T)] \tag{17}$$

$$S = I + \bar{W}\hat{R}^{-1}B' \text{ and } \bar{W} = C'\bar{Q}D$$

Note that when $Fw(t) = z(t)$, the reference signal $z(t)$ is previewed. The optimal control law in Eqn.(10) minimizing J can be stated as follows:

Control Law: Given the linear time invariant system

$$\dot{x}(t) = Ax(t) + Bu(t) + Ew(t)$$

$$y(t) = Cx(t) + Du(t) + Fw(t)$$

and the desired output $z(t)$ with error $e(t) = z(t) - y(t)$. Given the cost functional J

$$J = < e(T), \bar{Q}e(T) > + \int_{t_0}^{T} \{< e(t), Qe(t) > + < u(t), Ru(t) >\}dt$$

Where $u(t)$ is unconstrained, T is specified, R is positive definite, and $\bar{Q}$ and Q are positive semidefinite. The optimal control exists, is unique, and is given by

$$u^*(t) = -\hat{R}^{-1}\{\bar{K}x - B'g - D'Q[z - Fw]\}, \quad \forall t \in [t_0, T].$$

The n by n real, symmetric and positive definite matrix K in $\bar{K} = B'K + W'$ is the solution of the Riccati type matrix differential equation in Eqn. (14) with boundary condition in Eqn. (16). The vector $g(t)$ (with n components) is the solution to the linear vector differential equation in Eqn. (15) with the boundary condition in Eqn. (17). The optimal trajectory is the solution of the linear differential equation in Eqn. (12).

III. Stability and Optimality Conditions

Consider matrix $\hat{H}$ and the sufficient condition $\hat{H} \geq 0$ for local optimality, where

$$\hat{H} = \begin{bmatrix} \dfrac{\partial^2 H}{\partial x^2} & \dfrac{\partial^2 H}{\partial x \partial u} \\ \dfrac{\partial^2 H}{\partial u \partial x} & \dfrac{\partial^2 H}{\partial u^2} \end{bmatrix} = \begin{bmatrix} Q & W \\ W' & \hat{R} \end{bmatrix}.$$

In cases where $D = 0$, a positive semidefinite $\hat{H}$ is guaranteed by the virtue $Q \geq 0$ and $R > 0$. In biproper systems, however, it is necessary to select quadratic weights $Q \geq 0$ and $R > 0$ such that $\hat{H}$ is positive semidefinite for a given non-zero W. To derive stability, consider the algebraic Riccati equation,

$$0 = -K\bar{A} - \bar{A}'K + KB\hat{R}^{-1}B'K + W\hat{R}^{-1}W' - C'QC$$

and its counterpart, the Lyapunov matrix equation,

$$K\bar{A}_c + \bar{A}_c{}'K = -(KB\hat{R}^{-1}B'K - W\hat{R}^{-1}W' + C'QC) = -\hat{Q}.$$

Clearly, stability is guaranteed if $\hat{Q} \geq 0$. Therefore, given $\hat{R} > 0$ and $Q \geq 0$, it is required to show that $\hat{Q} \geq 0$. Consider the feedback part of the control law for stability,

$$-u = \hat{R}^{-1}[B'K + W']x \ \text{ or } \ \hat{R}^{-1}B'Kx = -u - W'x. \qquad (18)$$

To prove $\hat{Q} \geq 0$, let

$$x'\hat{Q}x = x'(KB\hat{R}^{-1}B'K - W\hat{R}^{-1}W' + C'QC)x$$

$$= x'KB(-u - \hat{R}^{-1}W'x) - x'[W\hat{R}^{-1}W' - C'QC]x$$

$$= x'KBu - x'(KB + W)\hat{R}^{-1}W'x + x'C'QCx$$

$$= -x'KBu - u'W'x + x'C'QCx$$

$$= -u'(B'K + W')x + x'C'QCx$$

$$= u'\hat{R}^{-1}u + x'C'QCx \geq 0 \quad \forall \ \hat{R} > 0 \text{ and } Q \geq 0 \qquad \text{Q.E.D}$$

Thus the new symmetric matrices in the algebraic and Lyapunov equations preserve stability and optimality conditions.

IV. Example

To illustrate the optimal control of biproper systems, a scalar example is considered.

$$\dot{x} = -x + u + ew$$
$$y = x + du + fw$$

Let $z(t) = 1(t)$ and consider the boundary value problem $x(t_0 = 0) = 0$, $T = 1$, $w = \sin(60t)$ and $k(T) = g(T) = 0$. Eqn.(14) and (15) for $k(t)$ and $g(t)$ with $Q = 1$ and $R = r$ are,

$$\dot{k} = -2k\bar{a} + \frac{k^2}{\hat{r}} + \frac{d^2}{\hat{r}} - 1$$
$$\dot{g} = a_c g(t) + [\frac{(k+d)d}{\hat{r}} - 1][1 - fw] + ekw$$
$$\hat{r} = r + d^2, \quad \bar{a} = -(1 + \frac{d}{\hat{r}}), \quad a_c = \bar{a} - \frac{k}{\hat{r}}$$

The optimal control law and the closed loop system are,

$$u = -\frac{1}{\hat{r}}[(k+d)x - g - d(1 - fw)]$$
$$\dot{x} = a_c x + \frac{1}{\hat{r}} g + \frac{d}{\hat{r}}(1 - fw) + ew$$

In Figure 1, optimal trajectories for biproper (solid lines) and strictly proper (dotted lines) are compared. The presence of control input at the output node with a non-zero value for d introduces a steady state error in biproper systems. Further the rise time and settling time for strictly proper system is much faster than the biproper system. The control input and the solution to the Riccati differential equation are also plotted in Figure 1.

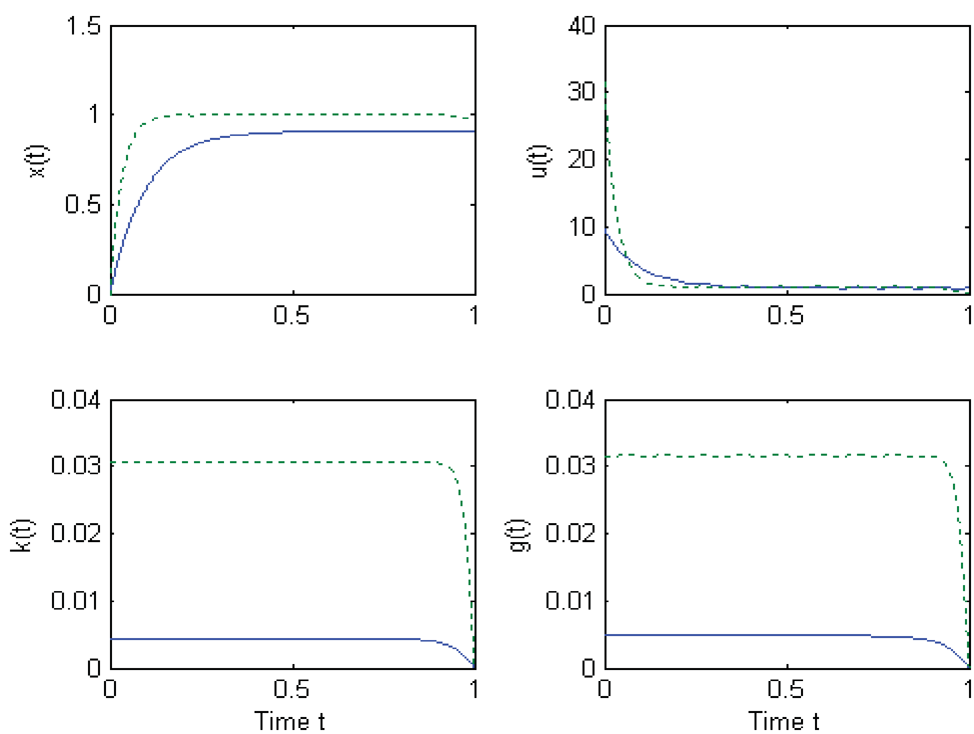

Figure 1 : Tracking Performance of Typical Biproper System

V. Concluding Remarks

In this paper, linear quadratic previewed control for strictly proper system is extended to biproper systems. Modified Riccati differential equation is presented. For normal acceleration regulation in a small aircraft at time windows of a gust input, the results of this paper is extendible to a control configuration where the inner loop is fixed and outer loop is used for regulation. This aspect of the paper is under investigation for medium size aircraft.

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On Dynamics of a Landing Gear Mechanism With Torsional Freeplay

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Abstract - In this study, dynamics of a landing gear mechanism with torsional degree of freedom and torsional freeplay is analyzed. Derivation of the equations of motion of the model with torsional degree of freedom and the von Schlippe tire model are presented. Freeplay is introduced into the model and effects of freeplay angles of 0°, 0.5°, 1° and 1.5° are observed by obtaining time histories of the torsion angle and lateral tire deformation and limit cycles of the torsionangle. Amplitudes and frequencies of oscillations of the time histories of the torsion angle and lateral tire deformation are presented.

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ON DYNAMICS OF A LANDING GEAR MECHANISMWITH TORSIONAL FREEPLAY

On Dynamics of a Landing Gear Mechanism With Torsional Freeplay

Elmas Atabay [α], Ibrahim Ozkol [σ]

Abstract - In this study, dynamics of a landing gear mechanism with torsional degree of freedom and torsional freeplay is analyzed. Derivation of the equations of motion of the model with torsional degree of freedom and the von Schlippe tire model are presented. Freeplay is introduced into the model and effects of freeplay angles of 0 °, 0.5°, 1° and 1.5° are observed by obtaining time histories of the torsion angle and lateral tire deformation and limit cycles of the torsion angle. Amplitudes and frequencies of oscillations of the time histories of the torsion angle and lateral tire deformation are presented.

I. Introduction

Vibration of aircraft steering systems has been a problem of great concern since the production of first airplanes. Shimmy is an oscillatory motion of the landing gear in lateral and torsional directions, caused by the interaction between the dynamics of the tire and the landing gear, with a frequency range of 10–30 Hz. Though it can occur in both nose and main landing gear, the first one is more common. Shimmy is a dangerous condition of selfexcited oscillations driven by the interaction between the tires and the ground that can occur in any wheeled vehicle. Problem of shimmy occurs in ground vehicle dynamics and aircraft during taxiing and landing. In other words, shimmy takes places either during landing, take–off or taxi and is driven by the kinetic energy of the forward motion of the aircraft. It is a combined motion of the wheel in lateral, torsional and longitudinal directions.

II. Shimmy

Shimmy can occur in steerable wheels of cars, trucks and motorcycles, as well as trailers and tea carts. In vehicle dynamics, shimmy is the unwanted oscillation of a rolling wheel about a vertical axis. It can occur in taxiing aircraft, as well. In the case of a shopping cart wheel, it is caused by the coupling between transverse and pivot degrees of freedom of the wheel. In the case of landing gear, shimmy is the result of the coupling between tire forces and landing gear bending and torsion. In other words, basic cause of shimmy is energy transfer from tireground contact force and vibration modes of the landing gear system.

Shimmy is an unstable phenomenon and it occurring with a certain combination of parameters such as mass, elastic quantities, damping, geometrical quantities, speed, excitation forces and nonlinearities such as friction and freeplay. It is difficult to determine shimmy analytically since it is a very complex phenomenon, due to factors such as wear and ground conditions that are hard to model. Small differences in physical conditions can lead to extremely different results. For example, it is reported in [1] that a new small fighter aircraft whose name is withheld, has displayed to vibrations during low and high speed taxi tests and first several landings and takeoffs, but shimmy vibrations with frequencies in the range 22–26 Hz were experienced during next several landings and take–offs at certain speeds, especially during landing. This demonstrates the effect of wear on landing gear shimmy. In the reported case, it was seen that tightening the rack too tight against the pinion prevented the wheel from turning, while tightening it less tight caused the vibration to disappear but reappear in the following flights.

Ground control of aircraft is extremely important since severe shimmy can result in loss of control or fatigue failure of landing gear components. Vibration of aircraft steering systems deserves and has gained attention since shimmy is one of the most important problems in landing gear design. Shimmy is reported to be due to the forces produced by runway surface irregularities and nonuniformities of the wheels [2–5]. Modeling of aircraft tires presents similar challenges to those involved in modeling automotive tires in ground vehicle dynamics, on a much larger scale in terms size and loads on the tire [6].

Shimmy is a complex phenomenon influenced by many parameters. Causes of shimmy can be listed as follows [2,7–10].

- Insufficient overall torsional stiffness of the gear about the swivel axis
- Inadequate trail, since positive trail reduces shimmy
- Improper wheel mass balancing about the swivel axis
- Excessive torsional freeplay
- Low torsional stiffness of the strut
- Flexibilities in the design of the suspension
- Surface irregularities
- Nonuniformities of the wheels
- Worn parts

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III. Detection and Suppression of Shimmy

Shimmy is a great concern in aircraft landing gear design and maintenance. Prediction of nose landing gear shimmy is an essential step in landing gear design because shimmy oscillations are often detected during the taxi or runway tests of an aircraft, when it is no longer feasible to make changes on the geometry or stiffness of the landing gear. Although shimmy was observed in earlier aircraft as well, there were no extra shimmy damping equipments installed. Historically, France and Germany tended to deal with shimmy in the design phase, while in United States, the trend was to solve the problem after its occurrence. Currently, the general methodology is to employ a shimmy damper and structural damping. A shimmy damper, acting like a shock absorber in a rotary manner, is often installed in the steering degree of freedom to damp shimmy. It is a hydraulic damper with stroke limited to a few degrees of yaw. A shimmy damper restrains the movement of the nose wheel, allowing the wheel to be steered by moving it slowly, but not allowing it to move back and forth rapidly. It consists of a tube filled with hydraulic fluid causing velocity dependent viscous damping forces to form when a shaft and piston are moved through the fluid. Oleo–pneumatic shock absorbers are the most common shock absorber system in medium to large aircraft, since they provide the best shock absorption ability and effective damping. Such an absorber has two components: a chamber filled with compressed gas, acting as a spring and absorbing the vertical shock and hydraulic fluid forced through a small orifice, forming friction, slowing the oil and causing damping. Another common cure is to replace the tires even though they may not be worn out [10–12].

Shimmy started being investigated in 1920's both theoretically and experimentally and soon it became clear that it is caused not by a single parameter but by the relationships between parameters. Effects of acceleration and deceleration on shimmy have been reported to be examined, and the accelerating system is found to be slightly less stable [13]. Number of publications available in literature on landing gear shimmy is limited because many developments are proprietary and are not published in literature.

IV. Literature Survey

Many papers have been published addressing shimmy as a vehicle dynamics problem. In that perspective, tire is the most important item, and tire models have been investigated. [13] examines the wheel shimmy problem and its relationship with longitudinal tire forces, vehicle motions and normal load oscillations. [8] compares different dynamic tire models for the analysis of shimmy instability. [3] is an investigation of tire parameter variations in wheel shimmy, by considering the shimmy resulting from the elasticity of a pneumatic tire, particularly in taxiing aircraft. [14] is on the application of perturbation methods to investigate the limit cycle amplitude and stability of the wheel shimmy problem. [7] deals with the shimmy stability of twin–wheeled cantilevered aircraft main landing gear. The objective in [15] is to develop software on assessing shimmy stability of a general class of landing gear designs using linear and nonlinear landing gear shimmy models. [16] studies the periodic shimmy vibrations and chaotic vibrations of a simplified wheel model using bifurcation theory. [17] is on tire dynamics and is a development to deal with large camber angles and inflation pressure changes. [18] is another study on tire dynamics, where stability charts show the behavior of the system in terms of certain parameters such as speed, caster length, damping coefficient and relaxation length. [19] is an experimental study on wheel shimmy where system parameters are identified, stability boundaries and vibration frequencies are obtained on a test rig for an elastic tire. Dependence of shimmy oscillations in the nose landing gear of an aircraft on tire inflation pressure are investigated in [20]. The model derived in [21] is used and it is concluded that landing gear is less susceptible to shimmy oscillations at inflation pressures higher than the nominal.

Transverse vibrations of landing gear struts with respect to a hull of infinite mass have been studied theoretically in [22]. Similarly, [23] presents a nonlinear model describing the dynamics of the main gear wheels relative to the fuselage.

Lateral dynamics of nose landing gear shimmy models has gained some attention. Lateral response of a nose landing gear has been investigated in [10] where nonlinearities arise due to torsional freeplay. In [24], lateral response to ground–induced excitations due to runway roughness is taken into consideration as well. Lateral stability of a nose landing gear with a closed loop hydraulic shimmy damper is presented in [12]. Closed form analytical expressions for shimmy velocity and shimmy frequency are derived in regard to the lateral dynamics of a nose landing gear in [25].

A dynamic model of an aircraft nosegear is developed in [9] and effects of design parameters such as energy absorption coefficient of the shimmy damper, the location of the center of gravity of the landing gear, shock strut elasticity, tire compliance, friction between the tire and the runway surface and the forward speed on shimmy are investigated. It is shown in [26] that dry friction is one of the principal causes of shimmy.

Bifurcation analysis of a nosegear with torsional and lateral degrees of freedom is performed in [21]. Similarly, bifurcation analysis of a nosegear with torsional, lateral and longitudinal modes is performed in [27]. In a more mathematical study, incremental harmonic balance method is applied to an aircraft wheel shimmy system with Coulomb and quadratic damping [28] and amplitudes of limit cycles are predicted.

Theoretical research on shimmy has a long history, with the initial focus on tire dynamic behavior because tires play an important role in causing shimmy instability. Theories on tire models can be divided into stretched string models and point contact models. In the stretched string model proposed by von Schlippe, the tire centerline is represented as a string in tension, the tire sidewalls are represented by a distributed spring where the string rests and the wheel is represented by a rigid foundation for the spring. Pacejka has proposed replacing the string by a beam. The point contact method assumes the effects of the ground on the tire act at a single contact point and is much easier to implement in an analytical model.

V. Mathematical Mode

a) *Landing gear model*

In this study, dynamics of a landing gear model with torsional degree of freedom and torsional freeplay is analyzed. The nonlinear mathematical shimmy model presented in [11], [29] and [30] describes the torsional dynamics of the lower parts of a landing gear mechanism and stretched string tire model. Figures 1 a and b show the physical and mathematical nose landing gear models. Dynamics of the lower part of the landing gear is described by a second order ordinary differential equation for the yaw angley about the vertical axis z, while the dynamics of the tire modeled with respect to the stretched string tire model is described by a first order ordinary differential equation for the lateral tire deflection y.

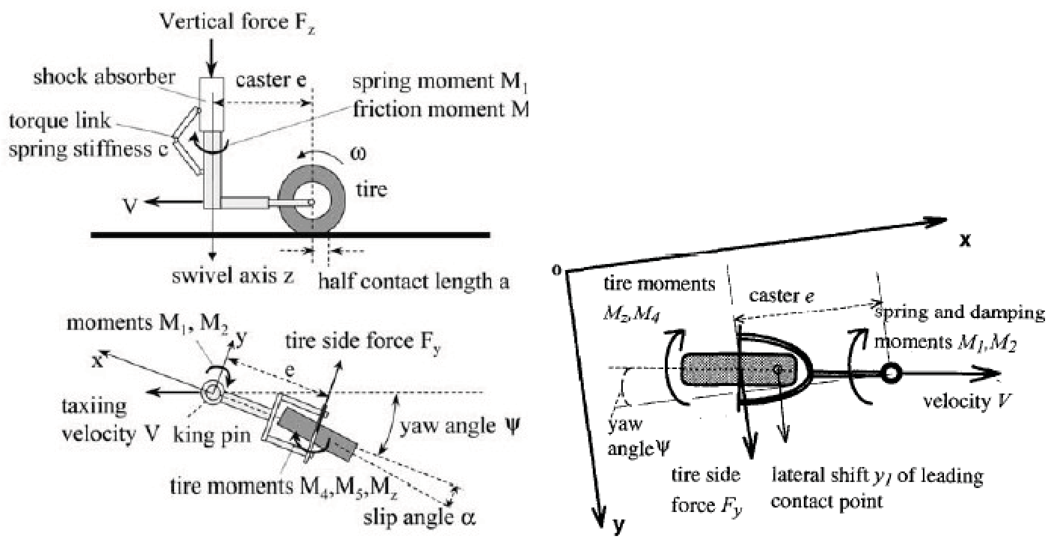

Figure 1: a. Nose landing gear model [30], b. shimmy dynamics model [29].

$$I_z\ddot{\psi} = M_1 + M_2 + M_3 + M_4 \qquad (1)$$

where I_z is the moment of inertia about the z axis,

M_1 is the linear spring moment between the turning tube and the torque link,

M_2 is the combined damping moment from viscous friction in the bearings of the oleo–pneumatic shock absorber and from the shimmy damper,

M_3 is the tire moment about the z axis and

M_4 is the tire damping moment due to tire tread width..

M_1 and M_4 are external moments.

M_3 and M_4 are caused by lateral tire deformations due to side slip.

M_3 is composed of M_z, tire aligning moment about the tire center, and tire cornering moment eF_y. F_y is the wheel cornering force or the sideslip force acting with caster e as lever arm.

$$M_1 = k\dot{\psi} \qquad (2)$$

$$M_2 = c\psi \qquad (3)$$

$$M_3 = M_z - eF_y \qquad (4)$$

$$M_4 = \frac{\kappa}{v}\dot{\psi} \qquad (5)$$

where k is the torsional spring rate, c is the torsional damping constant, v is the taxiing velocity and k is the tread width moment constant defined as [29]

$$\kappa = -0.15\, a^2 c_{F\alpha}\, F_z \qquad (6)$$

F_y and M_z depend on the vertical force F_z and slip angle . Tire sideslip characteristics are nonlinear. Cornering force F_y and vertical force F_z are related as

$$F_y/F_z = c_{F\alpha}\alpha, \qquad \text{for } |\alpha| \le \delta \tag{7}$$

$$F_y/F_z = c_{F\alpha}\delta \, sign(\alpha), \qquad \text{for } |\alpha| > \delta \tag{8}$$

Where δ is the limiting slip angle or the limit angle of tire force and $sign(\alpha)$ is the sign function defined as

$$sign(\alpha) = \begin{cases} 1, \text{if } \alpha > \delta \\ -1, \text{if } \alpha \le \delta \end{cases} \tag{9}$$

Slip angle may be caused by either pure yaw or pure sideslip. Pure yaw occurs when the yaw angle ψ is allowed to vary while the lateral deflection y is held at zero. Pure sideslip, on the other hand, occurs when the lateral deflection y is allowed to vary as the yaw angle ψ is held at zero [11].

An expression is given for the nonlinear sideslip characteristic in the widely used Magic Formula [7, 11, 17] as the following

$$F_y = D \sin\left[C \arctan\left\{B\alpha - E\left(B\alpha - \arctan(B\alpha)\right)\right\}\right] \tag{10}$$

where B,C, D and E are functions of the wheel load, slip angle, slip ratio and camber. B and E are related to vertical force F_z, C is the shape factor and D is the peak value of the curve.

Plots of F_y/F_z versus α will not be presented here due to lack of space, but they have similar characteristics when obtained using either (7) and (8) or the Magic Formula, thus the simple approximations given by (7) and (8) are used instead of the complicated Magic Formula. Only force and moment derivatives are needed as parameters for (7) and (8).

Aligning moment M_z is defined using a half–period sine. M_z/F_z is approximated by a sinusoidal function and the constant zero given by (11) and (12).

$$M_z/F_z = c_{M\alpha}\frac{\alpha_g}{180}\sin\left(\frac{180}{\alpha_g}\alpha\right), \qquad \text{for } |\alpha| \le \alpha_g \tag{11}$$

$$M_z/F_z = 0, \qquad \text{for } |\alpha| > \alpha_g \tag{12}$$

where α_g is the limiting angle of tire moment.

b) Tire model

Tire is modeled using the elastic string theory. Lateral deflection of the tire is described as [11,29]

$$\dot{y} + \frac{v}{\sigma} y = v\psi + (e-a)\dot{\psi} \tag{13}$$

Ground forces are transmitted to the wheel through the tire, and these forces acting on the tire footprint deflect the tire. Elastic string theory states that lateral deflection y of the leading contact point of the tire with respect to tire plane can be described as a first order differential equation given by (13). This equation is derived as follows.

Tire sideslip velocity V_t is expressed as

$$V_t = \dot{y} + \frac{y}{\tau} \tag{14}$$

Where $\tau = \dfrac{\sigma}{V}$ is the time constant, σ is the relaxation length, which is the ratio of the slip stiffness to longitudinal force stiffness. The tire also undergoes yaw motion, leading to a yaw velocity V_r which is approximated as

$$V_r = v\psi + (e-a)\dot{\psi} \tag{15}$$

As the wheel rolls on the ground,

$$V_t = V_r \tag{16}$$

Substituting (14) and (15) into (16) yields (13).

An equivalent side slip angle caused by lateral deflection is used to compute cornering force F_y and aligning moment M_z and is approximated as

$$\alpha \approx \arctan\alpha = \frac{y}{\sigma} \tag{17}$$

Equations (1), (13) and (17) constitute the governing equations of the torsional motion of the landing gear and include nonlinear tire force and moment. Parameters of a light aircraft used in the computations are given in table 1.

Table 1: Parameters used in the torsional dynamics.

Parameter	Description	Value	Unit
v	velocity	0…80	m/s
a	half contact length	0.1	m
e	caster length	0.1	m
I_z	moment of inertia	1	kg m^2
F_z	vertical force	9000	N
c	torsional spring rate	-100000	Nm/rad
$c_{F\alpha}$	side force derivative	20	1/rad
$c_{M\alpha}$	moment derivative	-2	m/rad
k	torsional damping constant	0…-50	Nm/rad/s
κ	tread width moment constant	-270	Nm^2/rad
$\sigma = 3a$	relaxation length	0.3	m
α_g	limit angle of tire moment	10	deg
δ	limit angle of tire force	5	deg

c) Linearization

In order to use linear analysis tools, nonlinear landing gear model has to be linearized. Following this, linear stability analysis will be performed.

within a small range of the side slip angle α, cornering force F_y and the ratio M_z / F_z can be approximated proportional to the side slip angle. Based on this assumption, substituting equations (7), (8), (11) and (12) into (4) yields (20), the complete expression for the tire moment M_3

$$F_y = \begin{cases} c_{F\alpha}\alpha F_z, & |\alpha| \le \delta \\ c_{F\alpha}\delta\, sign(\alpha), & |\alpha| > \delta \end{cases} \tag{18}$$

$$M_z = \begin{cases} c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z, & |\alpha| \le \alpha_g \\ 0, & |\alpha| \ge \alpha_g \end{cases} \tag{19}$$

$$M_3 = \begin{cases} -ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \alpha \le -\alpha_g \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & -\alpha_g < \alpha < -\delta \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, F_z, & -\delta < \alpha < \delta \\ c_{M\alpha}\dfrac{\alpha_g}{180}\sin\left(\dfrac{180}{\alpha_g}\alpha\right)F_z - ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \delta < \alpha < \alpha_g \\ ec_{F\alpha}\delta\, sign(\alpha)\, F_z, & \alpha > \alpha_g \end{cases} \tag{20}$$

Substituting (17) into (20) and expressing M_3 in the neighborhood of a = 0 or y = 0 yields

$$M_3 = c_{M\alpha}\frac{\alpha_g}{180}\sin\left(\frac{180}{\alpha_g}\frac{y}{\sigma}\right)F_z - ec_{F\alpha}\frac{y}{\sigma}F_z \tag{21}$$

M_3 can be linearized using the Taylor series expansion as

$$\left.\frac{\partial M_3}{\partial y}\right|_{y=0} = c_{M\alpha}\frac{\alpha_g}{180}\cos\left(\frac{180}{\alpha_g}\frac{y}{\sigma}\right)F_z\frac{180}{\alpha_g\sigma} - ec_{F\alpha}\left.\frac{1}{\sigma}F_z\right|_{y=0} = \frac{F_z}{\sigma}\left(c_{M\alpha} - ec_{F\alpha}\right) \tag{22}$$

Defining the state variables as $(\psi, \dot{\psi}, y)$ gives the linearized model as three ordinary differential equations of first order as

$$\begin{bmatrix}\dot{\psi}\\ \ddot{\psi}\\ \dot{y}\end{bmatrix} = \begin{bmatrix}0 & 1 & 0\\ c_1 & c_2 & c_3\\ v & c_4 & c_5\end{bmatrix}\begin{bmatrix}\psi\\ \dot{\psi}\\ y\end{bmatrix} \tag{23}$$

where

$$c_1 = \frac{c}{I_z} \tag{24}$$

$$c_2 = \frac{k}{I_z} + \frac{\kappa}{vI_z} \tag{25}$$

$$c_3 = \frac{(c_{M\alpha} - ec_{F\alpha})F_z}{I_z\sigma} \tag{26}$$

$$c_4 = e - a \tag{27}$$

$$c_5 = \frac{-v}{\sigma} \tag{28}$$

VI. Freeplay

Freeplay is a type of concentrated structural nonlinearity inherent in many mechanical systems. Such concentrated structural nonlinearities, such as cubic, freeplay and hysteresis stiffnesses, have significant effects on aeroelastic responses of airfoil surfaces. Freeplay gives the most critical flutter condition among the three and is inevitable for control surfaces due to wear and manufacturing errors. It exists in the hinge part of the control surfaces of most flight vehicles and is generated from loose or worn hinge connections, joint slippage and manufacturing tolerances. Freeplay may couple with aerodynamic effects and cause limit cycle oscillations during flight, leading to structural damage due to fatigue. Thus, it is crucial to incorporate freeplay into the equations of motion and to predict its effects in advance. Freeplay nonlinearity causes structural stiffness to become piecewise continuous. A spring is often used in literature to represent worn or loose control surface hinges. Most of the literature considering the effect of freeplay concentrates on problems of aeroelasticity. Missile control surfaces, moveable aircraft lifting surfaces such as horizontal tails or rotatable pylons on aircraft with variable sweep exhibit freeplay that can be potentially dangerous from an aeroelastic perspective, in terms of flutter conditions. It is found that limit cycle oscillations in the case of freeplay nonlinearity occur below the linear flutter speed boundary, which means the critical flutter speed is below that of the system without freeplay. Additionally, freeplay may cause instabilities both above and below the flutter speed predicted by the linear theory. Responses to freeplay include nonlinear phenomena such as limit cycle oscillations and even chaotic responses. Limit cycle oscillations are likely to occur in the presence of freeplay nonlinearities, leading to fatigue and damage in the long run. The possibility of even small freeplay angles leading to severe instabilities dangerous fatigue conditions are shown in literature [31–34,35,36,37].

Cyclic loading occurs during taxi due to runway surface irregularities, which may lead to wear in mechanical components of the landing gear, including freeplay in the rack and pinion of the steering system, interlinkages of the torque link, fuselage attachment points, steering collar and wheel axle [38].

Freeplay is hard to avoid in loose or old joints. Its existence may affect the system response, even leading to chaos, however harmful results can be avoided if possible limit cycle oscillations or chaotic motion are known in advance. Therefore, it is important to determine the possibility of the existence of such motions before they occur [31–33]. Additionally, freeplay will have an effect on the response of the system to a control law that was initially designed for the linear model [39]. Although freeplay is often linearized or ignored in calculations, it is necessary to compare the responses of the systems with and without freeplay. Amount of freeplay present in the studies mentioned here are in the range 0.1°–2.12°.

Various parts of the landing gear move with respect to each other during landing impact and when retracted. Freeplay at the wheel axle due to the contributions from various connections are less then one degree in yaw and in the order of millimeters in the lateral and fore/aft directions. It has been verified experimentally that the amount of freeplay is a function of the shock absorber deflection. Free play will increase with the number of flights. Application of tight tolerances might help in solving shimmy problems in the prototype phase, however, the problem will reoccur when the aircraft is in service, due to wear [7].

VII. Literature Survey On Freeplay

A literature survey on freeplay reveals that freeplay has been considered mostly by researchers working on the fluid–structure interaction problem. Flutter analysis of airfoils with freeplay nonlinearities in pitch degree of freedom subject to incompressible flow have gained some attention. Limit cycle oscillations of airfoils having two degrees of freedom and freeplay nonlinearities in pitch, placed in transonic and supersonic flows are investigated in [31] and [32], respectively. Similar numerical studies investigating the same model are [40], where the model is placed in subsonic and transonic flows, [37], where the model is placed in transonic and low supersonic flows, and [41], where the model is placed in turbulent flow. Bifurcation analysis of the same system with two degrees of freedom is conducted in [42].

Mathematical analysis of the behavior of a two dimensional aeroelastic system with freeplay nonlinearity is presented in [43] and two formulations are developed. Formulations are extended for a hysteresis model in [44]. Unlike a freeplay model which consists of three linear subsystems, a hysteresis model consists of six.

Bifurcation analysis an airfoil having two degrees of freedom with both freeplay and cubic stiffness nonlinearities in pitch placed in supersonic flow has been conducted in [35]. Bifurcation analysis of an aircraft with freeplay nonlinearity is conducted in [45]. Limit cycle oscillations of an airfoil with two degrees of freedo having freeplay in the pitching degree of freedom are examined experimentally and theoretically in [34]. An experimental delta wing model with freeplay at the attachment points is designed and tested in [46], and its gust response is investigated in [47]. Effect of freeplay on the aerodynamic response, such as limit cycle flutter, has been examined. It has been found that the amplitude and position of the limit cycle varies with the magnitude of freeplay. Effects of variations in parameters have been examined for both the damped and limit cycle oscillations. Critical flutter speeds are predicted.

Hinges of control surfaces often demonstrate freeplay nonlinearity. [48] is a study examining the limit cycle oscillations of a combination of an airfoil and an aileron, resulting in three degrees of freedom, with freeplay in the aileron hinge. Aeroelastic response of other two dimensional systems having control surface freeplay nonlinearity are studied using the harmonic balance approach in [49] and both numerically and experimentally in [39]. A dissertation was presented to Duke University in 2000, covering the dynamics of a two dimensional aeroelastic system with control surface freeplay nonlinearity, both experimentally and mathematically [50]. Limit cycle oscillations are observed. The system is very similar to the one given in [48], a combination of an airfoil with an aileron.

A three dimensional control surface with play is investigated in [51] to demonstrate the effects of angle of attack and Mach number. Flutter analysis of a missile wing having freeplay in it the rotation degree of freedom of the wing control mechanism is conducted in [33] by investigating limit cycles and chaotic motion. Results state that the system response depends on the amount of freeplay and initial conditions.

A study on a mechanical system exhibiting freeplay nonlinearity is studied both numerically and experimentally in [36] where the problem of developing a mathematical model and performing a simulation of the dynamics of systems exhibiting freeplay nonlinearity is addressed. Contact due to freeplay is considered, constraints are formulized and the stability of an aircraft wing displaying freeplay in the hinge supporting a control surface is investigated. Freeplay is considered as one of the rotor faults in the simulation of helicopter structural damages in [52].

Freeplay model used in this study is based on the ones in [31] and [38]. Dynamics of a landing gear mechanism with freeplay in the torsional degree of freedom is analyzed in [38], while dynamic behavior of a two dimensional airfoil with freeplay in pitch, oscillating in pitch and plunge directions, subjected to inviscid, transonic flow is analyzed in [31]. Both freeplay nonlinearities are modeled using the same principle and formulation, although the two studies are in two very distinct disciplines. Same formulation as in [31] is employed in [40,93], and mathematical models given in [32,33,35,37,41–43,48] are also similar .

Freeplay is modeled as a nonlinear spring as in figure 2, where some deflection is possible before a force develops and the spring force is zero if the amplitude remains within the freeplay band. Formulations have been suggested in literature to determine an equivalent linear stiffness.

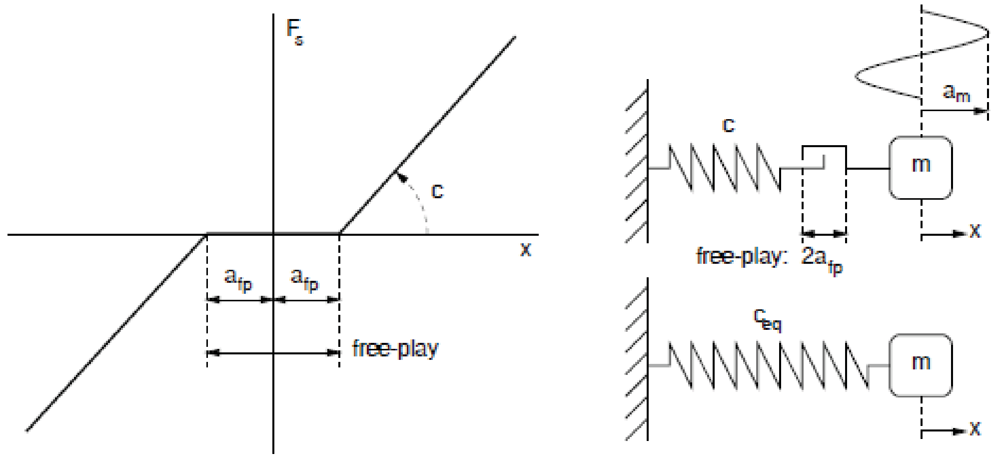

Figure 2 : Modeling of freeplay [7].

Equation 34 gives the piecewise continuous restoring moment function similar to the one used in [38] to describe the concentrated nonlinearity at the torsional degree of freedom.

$$M(\psi)=\begin{cases} K_\psi(\psi-\psi_{fp}) & \text{if } \psi\geq\psi_{fp} \\ 0 & \text{if } -\psi_{fp}\leq\psi\leq\psi_{fp} \\ K_\psi(\psi+\psi_{fp}) & \text{if } \psi\leq-\psi_{fp} \end{cases} \tag{29}$$

Torsion is denoted by ψ , K_ψ is the stiffness coefficientand ψ_{fp} is the freeplay angle.

VIII. Incorporation of Freeplay into the Landing Gear Model

Torsional freeplay is incorporated into the equations of motion of the landing gear. Results are displayed for various values of the freeplay angle ψ_{fp} within the range 0º–2º, as this is the range employed in literature. Freeplay has been incorporated into the equations of motion of landing gear mechanisms in very few studies literature [38].

Freeplay model given in (29) can be incorporated in the equations of motion in two ways. One of them, is to linearize the model as in (23)–(28) and substitute (29) into $c_1\psi$ in (23). This way, the only nonlinearity in the model is freeplay nonlinearity such that the second equation in (23) becomes

$$\ddot{\psi}=M(\psi)+c_2\dot{\psi}+c_3y \tag{30}$$

Second way of incorporating freeplay nonlinearity in the model is to obtain a more realistic model by substituting (29) directly into the nonlinear model. This is the approach taken here. Nonlinear equations are integrated using the fourth order Runge–Kutta algorithm.

IX. Results

Effects of freeplay are observed by obtaining time histories of the torsion angle and lateral tire deformation and limit cycles. Freeplay angles of 0º, 0.5º, 1º and 1.5º are incorporated. Amplitudes and frequencies of oscillations of the time histories of the torsion angle and lateral tire deformation are presented in tables 2 and 3, respectively.

a) *Effect of freeplay on the torsion angle*

Time histories of the torsion angle are presented for freeplay angles of 0º, 0.5º, 1º and 1.5º in figures 3–6 for $\psi(0) = 0.1$. Amplitudes and frequencies of oscillations of the time histories of the torsion angle are presented in table 2.

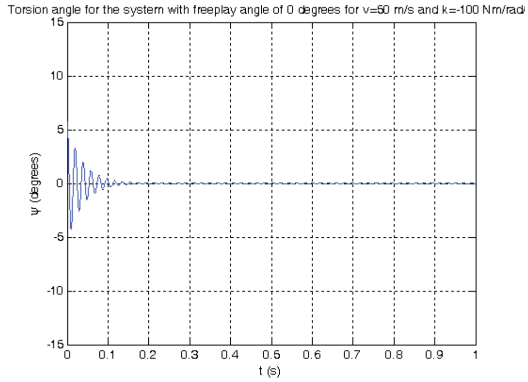

Figure 3 : Torsion angle for ψ_{fp} of 0°.

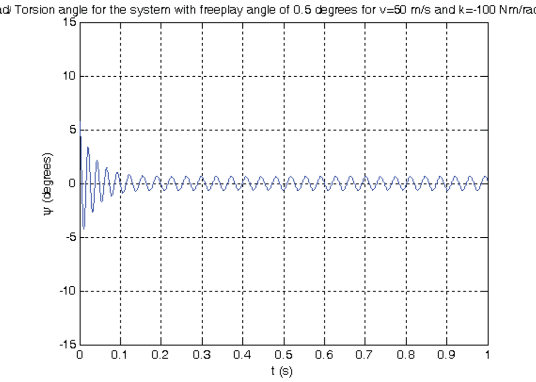

Figure 4 : Torsion angle for ψ_{fp} of 0.5°.

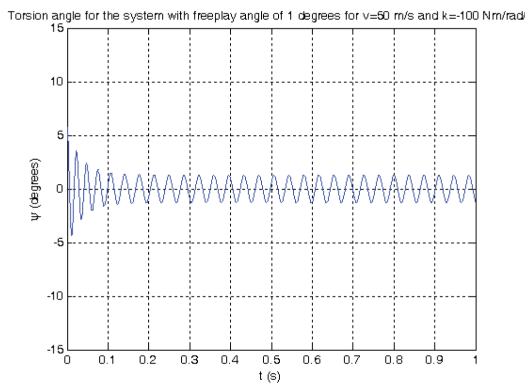

Figure 5 : Torsion angle ψ_{fp} of 1°.

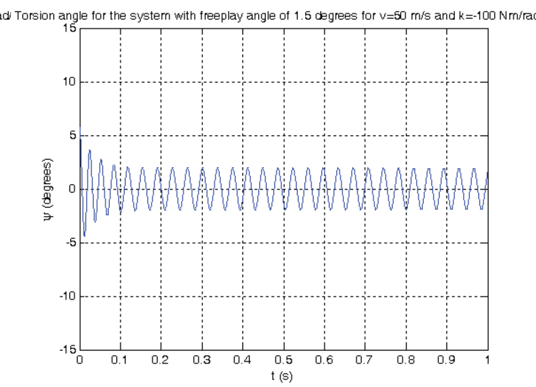

Figure 6 : Torsion angle for ψ_{fp} of 1.5°.

Table 2 : Amplitudes and frequencies of the torsion angle for various

ψ_{fp}	amplitude	frequency
0°	oscillation decays after 0.2 s	-
0.5°	1°	29 Hz
1°	2°	28 Hz
1.5°	2.5°	27 Hz

b) Effect of freeplay on the lateral tire deformation

Time histories of the lateral tire deformation are presented for freeplay angles of 0°, 0.5°, 1° and 1.5° in figures 7–10 for $\psi(0) = 0.01$ and in figures 11–14 for $\psi(0) = 0.1$. Amplitudes and frequencies of oscillations of the time histories of the lateral tire deformation are presented in table 3 for $\psi(0) = 0.01$ and in table 4 for $\psi(0) = 0.1$.

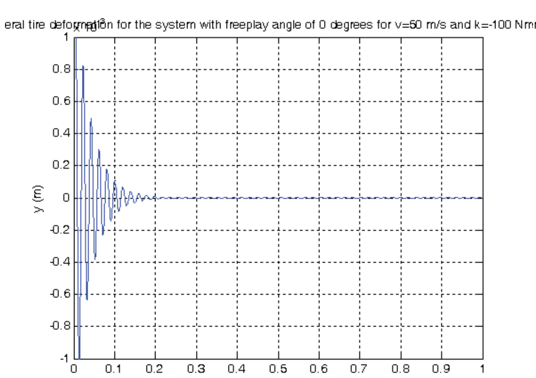

Figure 7 : Lateral tire deformation for ψ_{fp} of 0°.

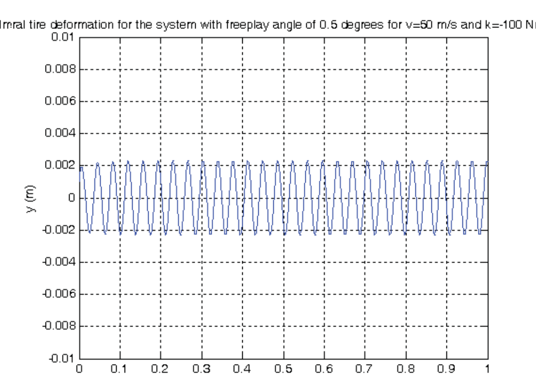

Figure 8 : Lateral tire deformation for ψ_{fp} of 0.5°.

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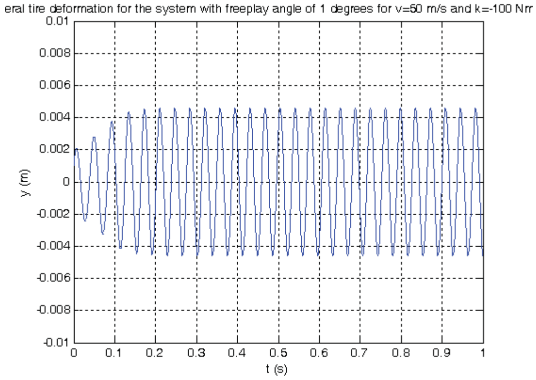

Figure 9 : Lateral tire deformation for ψ_{fp} of 1°.

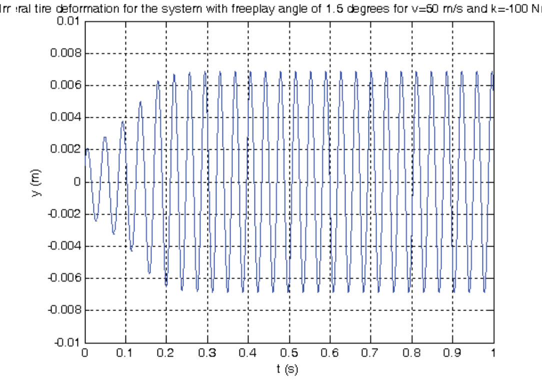

Figure10 : Lateral tire deformation for ψ_{fp} of 1.5°.

Table 3 : Amplitudes and frequencies of the lateral tire deformation for various ψ_{fp} and $\psi(0) = 0.01$.

ψ_{fp}	amplitude	frequency
0°	oscillation decays after 0.2 s	-
0.5°	$2 * 10^{-3}$ m	27 Hz
1°	$4.5 * 10^{-3}$ m	26 Hz
1.5°	$7 * 10^{-3}$ m	26 Hz

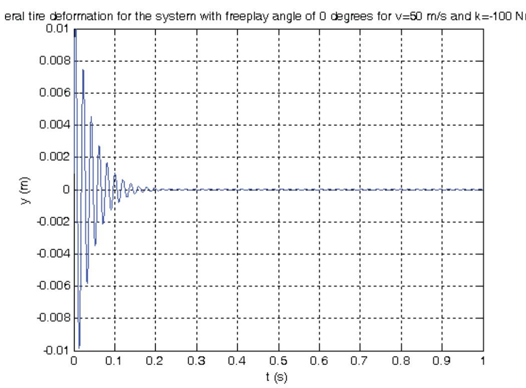

Figure11: Lateral tire deformation for ψ_{fp} of 0°.

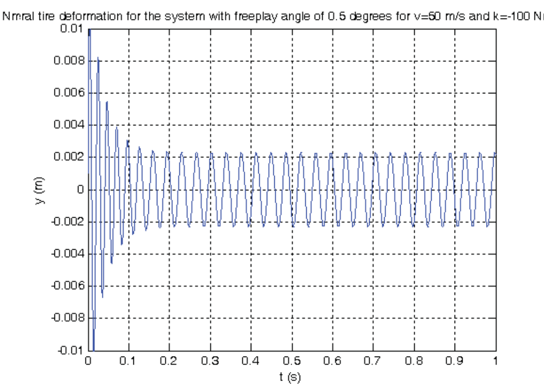

Figure 12 : Lateral tire deformation for ψ_{fp} of 0.5°.

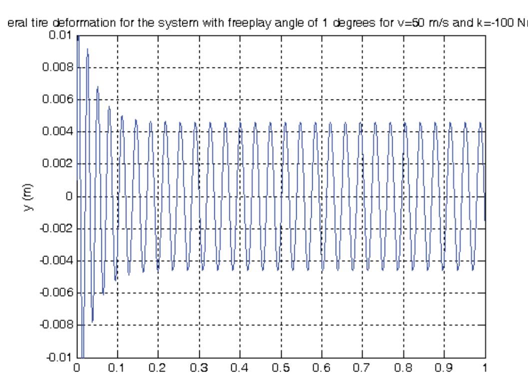

Figure13 : Lateral tire deformation for ψ_{fp} of 1°.

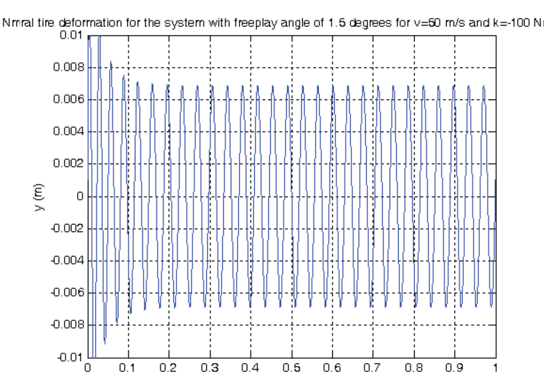

Figure14 : Lateral tire deformation for ψ_{fp} of 1.5°.

Table 4 : Amplitudes and frequencies of the lateral tire deformation for various ψ_{fp} and $\psi(0)=0.1$.

ψ_{fp}	**amplitude**	**frequency**
0°	oscillation decays after 0.2 s	-
0.5°	$2.2 * 10^{-3}$ m	29 Hz
1°	$4.5 * 10^{-3}$ m	28 Hz
1.5°	$7 * 10^{-3}$ m	28 Hz

c) Effect of freeplay on limit cycles

Limit cycles of the torsion angle are obtained for $\psi(0)=0.01$ in figures 15–18 and for $\psi(0)=11$for figures 19–22.

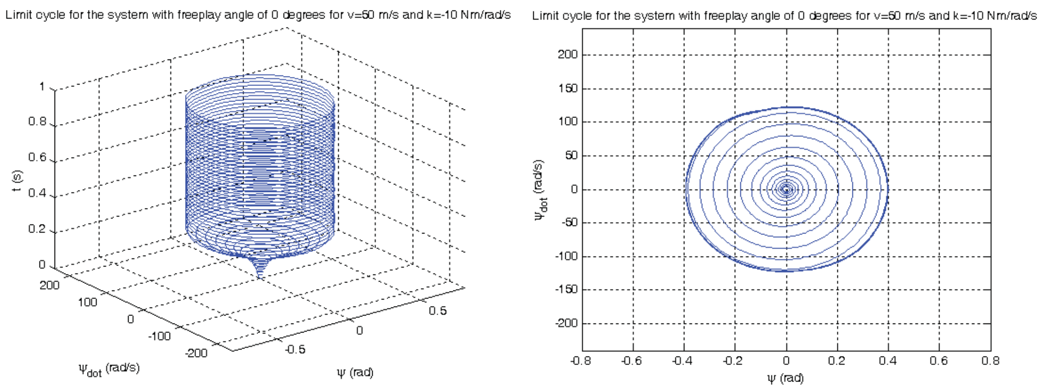

Figure 15 : Limit cycle for ψ_{fp} of 0° and $\psi(0)=0.01$.

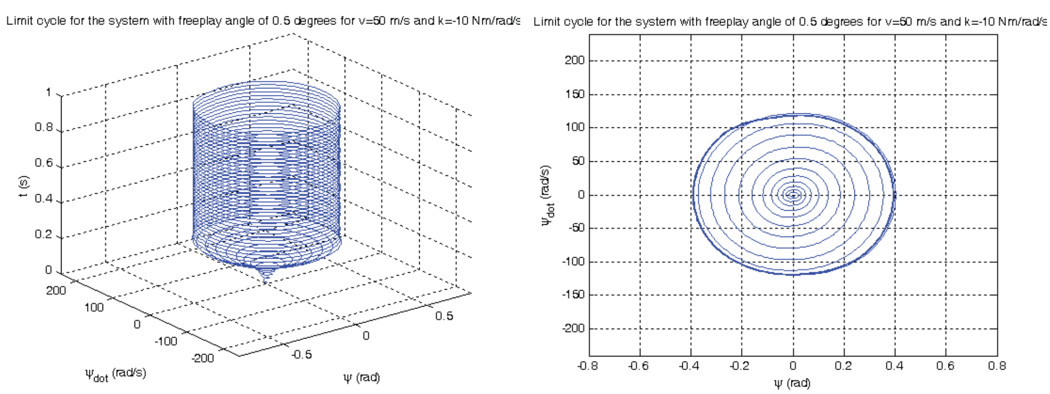

Figure 16 : Limit cycle for ψ_{fp} of 0.5° and $\psi(0)=0.01$.

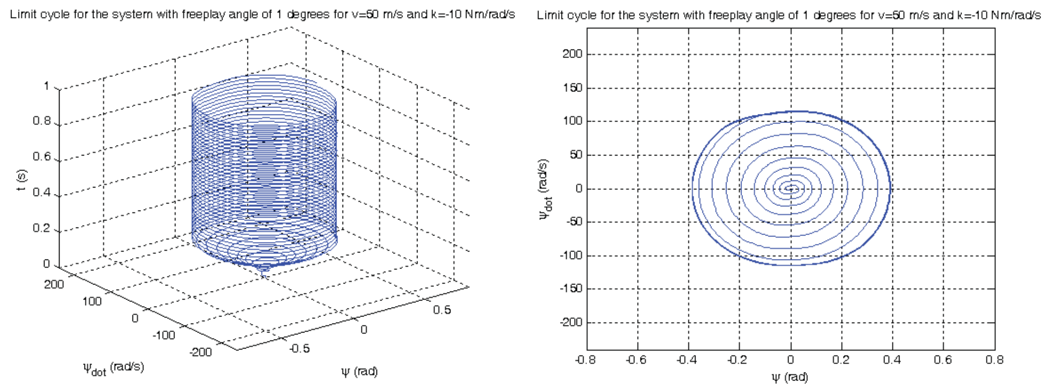

Figure 17 : Limit cycle for ψ_{fp} of 1° and $\psi(0) = 0.01$.

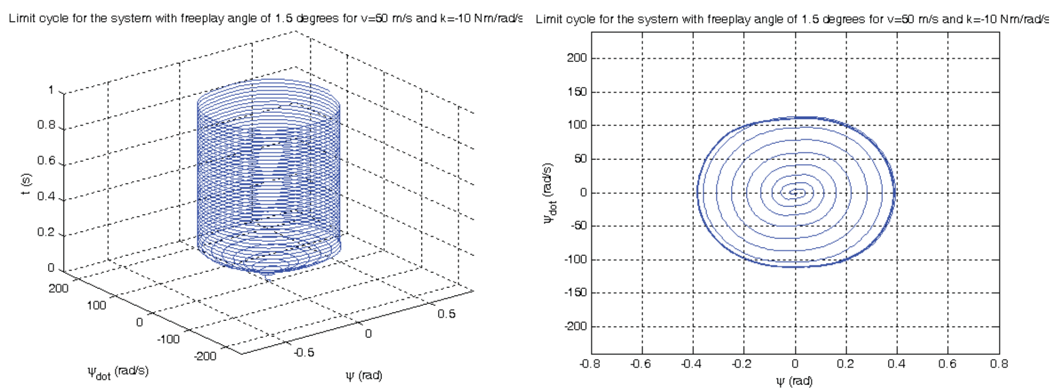

Figure 18 : Limit cycle for ψ_{fp} of 1.5° and $\psi(0) = 0.01$.

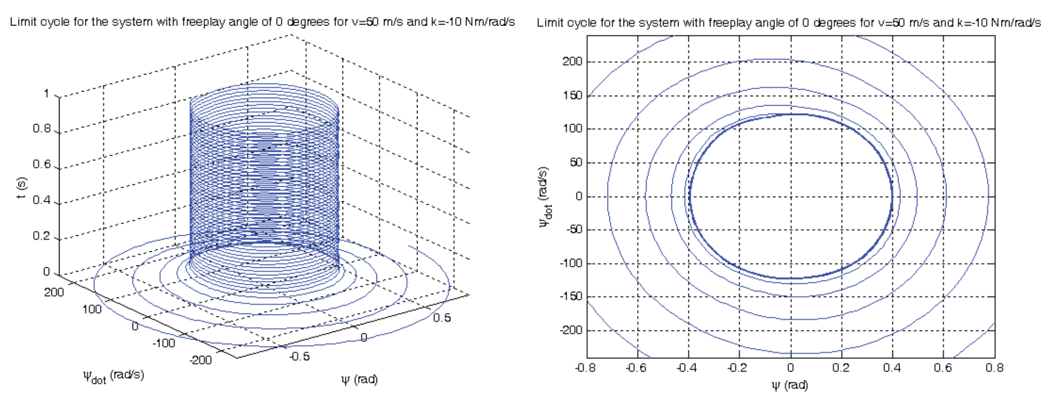

Figure 19 : Limit cycle for ψ_{fp} of 0° and $\psi(0) = 1$.

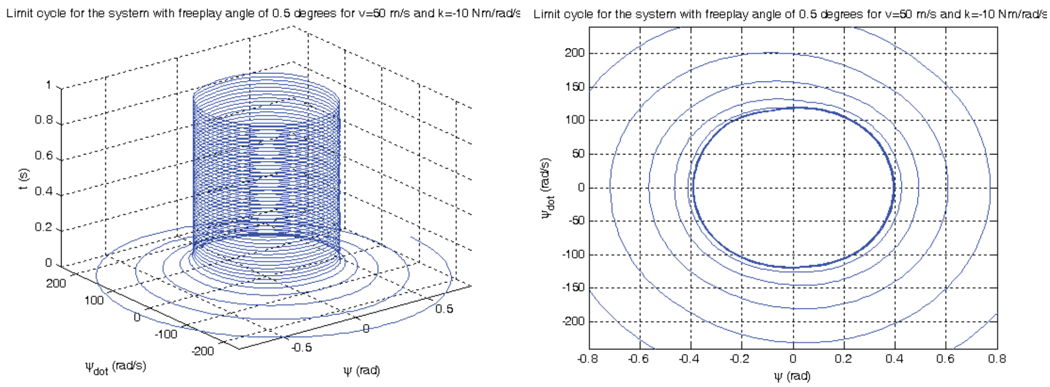

Figure 20 : Limit cycle for ψ_{fp} of 0.5° and $\psi(0)=1$.

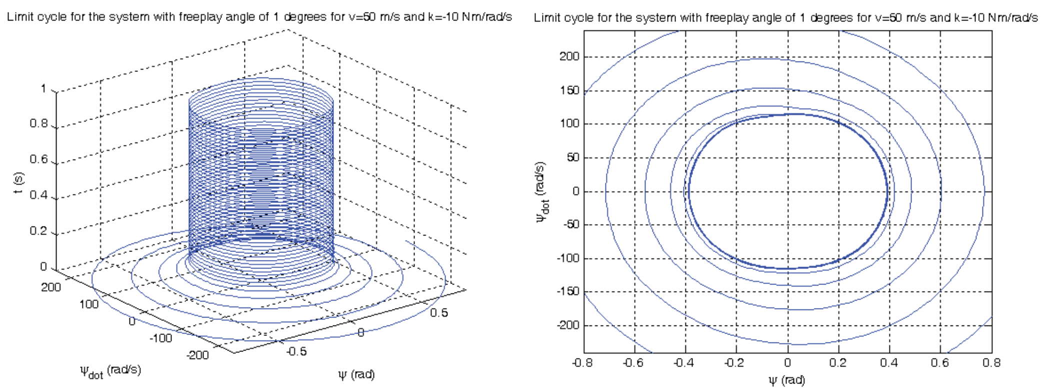

Figure 21: Limit cycle for ψ_{fp} of 1° and $\psi(0)=1$.

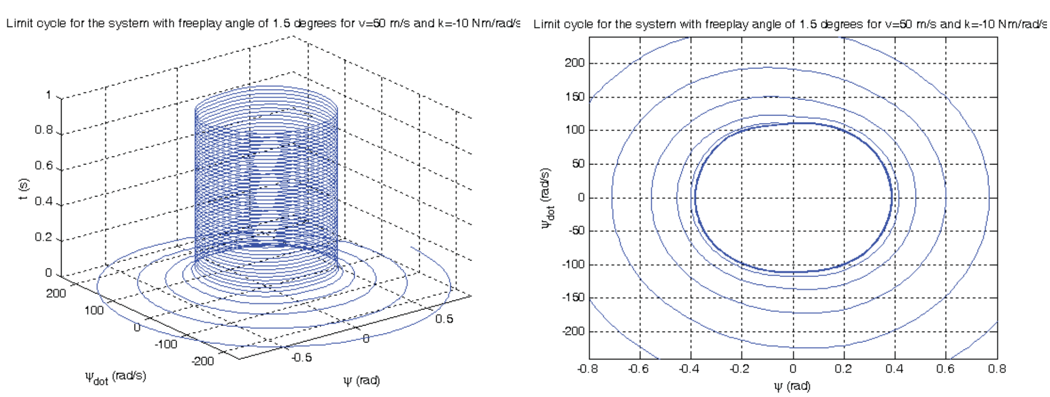

Figure 22 : Limit cycle for ψ_{fp} of 1.5° and $\psi(0)=1$.

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Effect of freeplay on the torsion angle and lateral tire deformation are observed. By observing tables 2–4 it can be stated that the existence of a freeplay angle prevents shimmy damping of the system with the same physical parameters. The increase in the freeplay angle increases shimmy amplitude. A 0.5^{o} increase of the freeplay angle from 0.5^{o} to 1^{o} doubles the amplitude in all 3 cases. Another 0.5^{o} increase in the freeplay angle from 1^{o} to 1.5^{o} causes a 25% increase in the amplitude of the torsion angle and a 55% increase in the amplitude of the lateral tire deformation.

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21. Arrangement of information: Each section of the main body should start with an opening sentence and there should be a changeover at the end of the section. Give only valid and powerful arguments to your topic. You may also maintain your arguments with records.

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27. Refresh your mind after intervals: Try to give rest to your mind by listening to soft music or by sleeping in intervals. This will also improve your memory.

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Approach:

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Methods and Procedures	Clear and to the point with well arranged paragraph, precision and accuracy of facts and figures, well organized subheads	Difficult to comprehend with embarrassed text, too much explanation but completed	Incorrect and unorganized structure with hazy meaning
Result	Well organized, Clear and specific, Correct units with precision, correct data, well structuring of paragraph, no grammar and spelling mistake	Complete and embarrassed text, difficult to comprehend	Irregular format with wrong facts and figures
Discussion	Well organized, meaningful specification, sound conclusion, logical and concise explanation, highly structured paragraph reference cited	Wordy, unclear conclusion, spurious	Conclusion is not cited, unorganized, difficult to comprehend
References	Complete and correct format, well organized	Beside the point, Incomplete	Wrong format and structuring

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