CRONE CONTROL METHODOLOGY FOR A MECHANICAL ACTIVE SUSPENSION SYSTEM

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Abstract

In the last few decades, significant progress has been made in the field of Process Control and instrumentation and offers a unique controller named CRONE, which is a noninteger controller to ascertaining the solution of the system under various model uncertainties. This paper proposed to analyzes the performance of CRONE controllers for a mechanical domain of Active Suspension System. To avoid vibration and providing a comfortable vehicle should design the active suspension system using CRONE controllers. The work reveals the design and implementation of CRONE controllers for a Mechanical Active Suspension System (MASS). The mathematical modeling of the transfer function for MASS is analytically derived and analyzed performance is obtained by MAT lab Simulink. The simulation results of the servo response for the CRONE controller are recorded. The Third Generation of CRONE (TGC) controller performance is analyzed in terms of error indices and time-domain parameters. In addition to that, the conventional ZN-PID controller is designed and compared with the TGC controller. Hence it is concluded that the performance of the TGC controller proves superiority over the ZN-PID controller.

Keywords: CRONE Controller, ZN-PID, TGC, Mechanical active suspension system, Nichols chart.

I. Introduction

In the past decade, significant progress has been made in the process control industry and this makes the controller explored in different ways to exhibit the performance. The mathematical model of active and passive suspension process has been determined and designed using Linear Quadratic Control (LQR). The comparative performance is analyzed between active and passive suspension system using an LQR controller [I]. The robust stability of the full vehicle suspension system has been obtained and designed by the combination of Fuzzy logic and Artificial...
Neural Network. The comparative performance was found between Fuzzy-ANN with PID-ANN and reported the effective performance [II]. The Hα state feedback controller has been developed for four degrees of freedom of the Quarter car model and considering control input at a different time delay to achieve ride comfort [IV]. The adaptive neural network has been designed for the suspension system and implemented to discuss the issues of actuator saturation and model parameter uncertainty [V]. The analysis is carried out for the suspension system either active or passive and is implemented using PID and Fuzzy logic controller. The simulation results of the controllers are compared with applying different input signals in the Matlab environment [VI]. The new type model-free fuzzy logic controller has been used to measure and control the nonlinearity based on particle swarm optimization techniques and compared with intelligent PID controller and classical PID controller through simulation results [VII].

The regenerative active suspension systems with two actuators concerning advanced dynamic damper mechanisms in the electric vehicle have been proposed to improve dynamic parameters and energy conservative performance for the vehicle [VIII]. The transfer function of the pneumatic level system for a two degree of freedom of the quarter car model has been determined and designed by the CRONE controller. The performance has been examined and compared with the Hα controller [IX]. The robust control of the anti-lock braking system has been determined with the need for an active suspension system and it can be achieved by decreasing the braking distance through improvement of tire vertical load [X] under various road conditions. The model-free finite-time tracking control has been considered for an active suspension system to the achievement of motion control of the system. The controller was designed without using plant nonlinearities, where the dynamics are estimated using the time delay estimator and the error can be compensated by sliding mode controller. The results were compared with the state observer-based feedback controller [XI].

The active seat suspension system can be designed with the help of state and disturbance observer and made a brief comparative robust solution between active, passive, and LQR based seat suspension system [XII]. The active suspension system mathematical model has been determined subjected to road profile and implemented using the Sliding mode Controller. The performance of the controller is analyzed and evaluated in a sense of time and root mean square values [XIV]. The mathematical modeling of a car suspension system has been determined and designed using a PID controller given improving the ride quality [XVI]. The CRONE suspension has been designed for a quarter car model by utilizing the CRONE controller. The transfer function of fractional (non-integer) order force-displacement can be defined for the suspension system towards to obtain robust stability of the vehicle due to road handling input disturbance and analyzed its performance in the frequency domain as well as time-domain characteristics [XII, XVII]. The adaptive backstepping based tracking controller has been considered for an active suspension system to measure the uncertainties.

The vertical and pitch movement of the vehicle body measured using the backstepping technique and Lyapunov stability theory based on adaptive control law by applying the virtual control input and reference trajectory and made a comparative performance analysis [XVIII]. It is difficult to acquire ride comfort and stability by

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employing secondary suspension. In this regard, the active secondary and primary suspensions are used mutually with a linear-quadratic-Gaussian (LQG) control for a full-scale railway vehicle [XX]. The sliding mode controller has been employed to control the nonlinearity of the active suspension system. Whereas the fuzzy approach utilized to construct the sliding mode feedback controller and verify results through Simulink for different system perturbations [XXI]. The new variable gain PI controller has been introduced to the quarter car active suspension system. The robust stability performance has examined by the Popov stability criterion, Routh-Hurwitz, and stability boundary locus method for a different order system and made a comparison with the existing controller [XXII].

The passive suspension system is not good to work in rough terrain while moving at high speed and lead to the development of an active suspension system is acquired best results of motion in terrain field. The mathematical model of the hydro-pneumatic suspension is developed for the quarter car suspension model and analysis of open-loop and closed-loop feedback control system were carried out to compare with the performance of the passive suspension system [XXIII]. The mathematical model of the passive suspension system for two degrees of freedom (DOF) quarter car model is analytically derived and the accuracy of modeling has been checked with the state-space model. The system performance is analyzed in Matlab Simulink by applying different excitation signals [XXIV]. The prototype of the VSVD semi-active suspension system has been developed for high-speed railway vehicles consists of four MRE isolators and an MR damper for measuring nonlinearity of the system. The VSVD performances were compared with the other four suspension systems and evaluated in simulation [XXV]. The Air pressure systems have been considered and have determined mathematical modeling by the Worst case modeling method. The implementation of three generations of CRONE controllers has been performed in the Matlab environment and made comparative results [XXVI]. The displacement of the active suspension system has been controlled based on the neural network method for the active suspension system and the order of the controller is eliminated by using the Dynamic surface control technique [XXVII].

In this work, the mathematical modeling of MASS has been considered and designed a control strategy for a TGC controller and ZN-PID controller. These two controllers are implemented in Matlab by using the CRONE CSD box. The evaluated servo performance of CRONE controllers with ZN-PID controllers is analyzed and made a comparative result in terms of error and time indices.

This manuscript is structured as emulates: Section II discusses brief details of modeling of a selected system for a MASS with supporting measuring devices. Section III reveals the design of a control strategy for the TGC controller and the ZN-PID controller. Section IV reviews the implementation and identified controller parameters assisted by the CRONE CSD toolbox. Section V shows the analysis of simulation comparative results of TGC and ZN-PID controllers. At the end of the paper in section, VI disclosed the conclusion and future work in CRONE controllers.

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II. Active Suspension System Model

The purpose of this section deals to determine the mathematical modeling [XXIII] for the MASS. In this paper proposed to achieve passenger sophistication and avoid vehicle vibration due to road disturbance. The displacement of the vehicle \(X_1(s)\) and tire \(X_2(s)\) is measured by using a displacement sensor. This signal can be applied to the controllers and compared with a predefined control signal to find an error signal. The controller determined the control signal for the MASS by applying the error signal to the controller. Thus the control signal is directly applied to the system to acquire the desired system operation. The schematic model for the MASS is depicted in Fig 1.

\[
\begin{align*}
\text{Applying Newton’s second law the force balance equation of sprung mass } m_1 \text{ is,} \\
& u(t) = m_2 \ddot{x}_1(t) + B \dot{x}_1(t) - x_2(t) + K_1[x_1(t) - x_2(t)] \\
& \text{(1)}
\end{align*}
\]

Using the Laplace Transform of the above equation we can get,

\[
\begin{align*}
& u(s) = m_2 s^2 \dot{x}_1(s) + B s x_1(t) - x_2(t) + K_1[x_1(s) - x_2(s)] \\
& \text{(2)}
\end{align*}
\]

From equation 2 yields,

\[
\begin{align*}
& u(s) = x_1(s)\left[m_2 s^2 + B s + K_1\right] - x_2(s)\left[B s + K_1\right] \\
& \text{(3)}
\end{align*}
\]

Similarly consider Newton second law for unsprung mass \(m_2\) is,
\[ 0 = m_2 \frac{d^2 x_2(t)}{dt^2} + B \frac{d[x_2(t) - x_1(t)]}{dt} s + K_2 x_2(t) + K_1 [x_1(t) - x_2(t)] + u(t) \]  

(4)

Apply Laplace transform on both sides we get,

\[ u(s) = m_2 s^2 x_2(s) + B s [x_2(s) - x_1(s)] + K_2 x_2(s) + K_1 [x_1(s) - x_2(s)] + u(s) \]  

(5)

By solving equation 3 and 5, the transfer function of MASS is,

\[ G(s) = \frac{x_1(s)}{u(s)} \]  

(6)

\[ G(s) = \frac{m_1 s^2 + B s + (k_1 + k_2)}{m_1 m_2 s^4 + (m_1 B + m_2 B) s^3 + (K_1 m_1 + K_1 m_2 + K_2 m_1) s^2 + k_2 B s + K_1 K_2} \]  

(7)

**Table 1**: Mechanical active suspension system parameters [XX]

| Parameters                          | Values         |
|-------------------------------------|----------------|
| Vehicle body mass(Sprung), m_1     | 2.45 kg        |
| Tire mass(Unsprung), m_2           | 1 kg           |
| Spring stiffness of the Suspension,K_1 | 900 N/m  |
| Spring stiffness of the Tire,K_2   | 1250 N/m       |
| Damping coefficient of the suspension, B | 7.5 N sec/m |

Based on this model, the design of TGC and ZN-PID controllers are deliberated in the next chapter.

**III. CRONE Control Methodology**

**A. TGC Methodology**

The closed-loop diagram of the TGC controller is depicted in Fig 2. The TGC controller design methodology broadens the design procedure of the second generation CRONE controller under the nonlinearities of the plant.

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The design methodology of the TGC controller can be accomplished in three different methods specifically generalized template, optimal template, and optimization of the open-loop behavior. The generalized unity feedback form of the TGC controller is depicted in Fig 2. In a generalized template, the Nichols locus is defined by straight line segments over \( \omega_{cg} \) in all directions of the template. From the black locus, the phase location is calculated by the real order of ‘a’ and the imaginary order of ‘b’ decides angle of slides based on the placement of vertical directions. The complex form of fractional order integral of TGC controller transfer function is represented a

\[
\beta_T(s) = C_T(s)G(s) = \left( \cosh \left( \frac{b \pi}{2} \right) \right)^{\text{sign}(b)} \left( \frac{\omega_{cg}}{s} \right)^a \left( \cos \left( b \ln \left( \frac{s}{\omega_{cg}} \right) \right) \right)^{-\text{sign}(b)}
\]

(8)

Fig. 3: Generalized template in Nichols plane (Black Locus)
By applying the derivative operator of $\beta(j\omega)$ to determine the magnitude and phase around $\omega_{cg}$. The angle of the generalized template is deliberated in terms of $a$ and $b$,

$$\frac{d\left[\beta_T(j\omega)\right]_{db}}{d\left(\text{phase}\beta_T(j\omega)\right)} = \frac{-20\text{asign}(b)}{\ln(10)b\tanh(b\frac{\pi}{2})}$$

(9)

The band-limited form of the generalized template transfer function expressed as,

$$\beta_T(s) = C^{\text{sign}(b)}\left(\frac{\omega_l}{s} + 1\right)^{n_I} \left(\alpha_0 \frac{1 + s/\omega_h}{1 + s/\omega_l}\right)^{n_I}
X\left(\text{Re}_i\left(\alpha_0 \frac{1 + s/\omega_h}{1 + s/\omega_l}\right)^{ib}\right)^{-q\text{sign}(b)}\left(1 + \frac{1}{\left(1 + s/\omega_h\right)^{n_h}}\right)$$

(10)

Where,

$$C = \cosh\left(b\left(\tan^{-1}\left(\frac{\omega_{cg}}{\omega_l}\right) - \tan^{-1}\left(\frac{\omega_{cg}}{\omega_h}\right)\right)\right)$$

(11)

$$\alpha_0 = \left(1 + \left(\frac{\omega_r}{\omega_l}\right)^2 / 1 + \left(\frac{\omega_r}{\omega_h}\right)^2\right)^{1/2}$$

(12)

$$|b| < \min\left\{\frac{\pi}{2\ln(\alpha_0)}, \frac{\pi}{2 \ln}\left(\frac{\alpha_0 \omega_h}{\omega_l}\right)\right\}$$

(13)

Therefore the fractional-order TGC controller is,

$$C_T(s) = \frac{\beta_T(s)}{G(s)}$$

(14)

The complex open-loop transfer function of $\beta(s)$ has an eight high-level parameter namely $n_I$, $n_F$, $a$, $b$, $\omega_l$, $\omega_h$, $\omega_r$, and $C$. Where $n_I$ and $n_F$ are chosen by the plant designer (Refer Fig 4). The tangency condition is obtained concerning $\omega_r$ and $C$ and these values have been mentioned earlier. The optimal template is defined by the nonlinear optimization algorithm based on four independent parameters and reduces cost.
function $J$ with respect variation of resonant peak and satisfies the shaping constraints:

$$J = M_{r_{\text{max}}} - M_{r_{\text{o}}}. \quad (15)$$

Synthesizing such a template through the optimization of three independent parameters (a tangency is imposed) from the four high-level parameters $a$, $b$, $\omega_l$, and $\omega_h$, is the initial aim of the third generation CRONE control. The general form of the third generation CRONE controller is given in (14).

The equation (14) is appeared in an unrealized form due to fractional order terms. By using a recursive distribution method the rational form of the controller transfer function is achieved and it is represented as,

$$C_R(s) = \frac{B_R(s)}{G(s)} \quad (16)$$

**Fig. 4:** Asymptotic Nichols locus- effect of parameters

**B. ZN-PID Controller**

It is a Classical design tuning technique proposed by John G. Ziegler and Nathaniel B. Nichols in 1942. Ziegler-Nichols has proposed a closed-loop tuning method for obtaining the optimum controller setting for various modes of conventional controller. It has been discussed the transient response of the closed-loop system and restricted to the open-loop system. This tuning method is performed by the determination of Ultimate Gain ($K_u$) and Ultimate period ($P_u$) through which generation of sustained oscillations based on the changing of proportional gain with the removal of integral and derivative modes of the controller. The selection of the proper value of proportional gain provides a transient response and continue increasing gain value until exhibits a sustained oscillation. The values of changing the gain and the period of oscillation for one cycle correspond to the sustained oscillation.
oscillations are ultimate gain (Ku) and Ultimate period (Pu). Finally, substitute these two found values in ZN closed-loop tuning formulae table to find the PID controller optimum controller settings. The general form of ZN-PID controller transfer function represented as,

\[ C(s) = K_C + \frac{K_C K_I}{S} + K_C K_D S \]  \hspace{1cm} (17)

\[ C(s) = 111 + \frac{541.4634}{s} + 5.68875s \]  \hspace{1cm} (18)

Where,

KC=0.6 Ku, KI= 2/Pu and KD=8/Pu

IV. CRONE Controller Implementation by CSD Toolbox

The process of developing and maintaining of Control System Design (CSD) [III] toolbox is organized by the research group of CRONE. This makes the user allow adjusting the plant information with their perturbation, time and frequency domain characteristics, fixed and tuned parameters of open-loop systems, etc. The MASS transfer function model is applied in the CSD toolbox to find user-defined specifications of CRONE control design for further analyses. The changing perturbation gain range is ±20%. Otherwise, the system will incorporate to attain a stable steady state of controlled response and it is lies between the range of 1000<1250<1500.

The obtained user-defined specification of CRONE controller design using a CSD toolbox of active suspension is given in Table 2.

| Required nominal open-loop - \( \omega_{cg} \) | 3 | \( \omega_A / \omega_{cg} \) - ratio | 10 |
| Required nominal phase margin - \( \mu \) | 54.9176 | \( \omega_B / \omega_\mu \) - ratio | 10 |
| Integral order - \( n_i \) | 1 | \( \omega_{\mu} / \omega_A \) - ratio | 1 |
| Low-pass filter order - \( n_r \) | 1 | \( \omega_{cg} / \omega_{(cg)} \) - ratio | 30 |
| Rational Approx. cell no. N | 5 | \( \omega / \omega_{cg} \) - ratio | 30 |

Fractional effect width - \( \omega_A / \omega_B \) ratio - 0.77888

By using the design procedure in section 4 with the need for a CSD toolbox using a mat lab to determine rational order transfer functions of CRONE controllers for TGC is presented in table 3.

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V. Result and Discussion

In this work, the simulation of TGC and ZN-PID controllers is carried out by MATLAB with the need of the CSD toolbox and the servo response results are recorded. The performance of the servo response is analyzed and compared in terms of ISE, IAE, $t_s$, and $t_r$. The simulation of MASS is determined for TGC and ZN-PID controllers for three different operating points of 40%, 50%, and 60% by applying a different perturbation to the system. The ±5% and ±10% of step-change are applied for each operating point and the results are tabulated. Fig 5, 6, and 7 shows obtained simulation output responses of the ZN-PID controller for three different operating points with applying ±5% and ±10% of step-change with different perturbations to the system. Similarly, Fig 8, 9, and 10 demonstrate the simulated responses of the TGC controller with various perturbations for three different operating points, and results are obtained for ±5% and ±10% of step-change input is given to the system. It can be seen that table 4 presented the performance of servo responses for a MASS in terms of error indices such as ISE and IAE for different operating points with perturbations. The 40% operating point and +5% step change is applied to the MASS and provides error performance of ISE value as 553.4 and IAE value is 24.51. Besides by using the ZN-PID controller on the system, the controller ISE value is increased to 3377 and the IAE value is 207.8. Similarly, at 60% operating point control force with +10%, a step change is applied to the MASS. The TGC controller gives ISE value as drastically increased to 7201, and the IAE value is an increment to 269.9. Respectively the same comparative process has to be done for other operating points. It is possible to identify from table 5 demonstrate the time-domain performance in terms of settling time and rise time for TGC and ZN-PID controller for different operating points with their perturbations. In the case of settling time ($t_s$), a 60% operating point is applied to the system. The TGC controller has to be settled at 16 seconds whereas the ZN-PID controller is settled at 29 seconds. In the view of rising time at the same operating point, the TGC controller provides rise

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time \( (t_e) \) as 04 seconds and the ZN-PID controller is 28 seconds. The respective same comparison is made for other operating points.

Finally, it is concluded that from the above discussion indicates that in all operating points set-point tracking performance, the TGC controller proves excellent performance (very low values of error indices and time-domain performance) over the ZN-PID controller.

Fig. 5: Servo response at 50 % Control force using ZN-PID controller

Fig. 6. Servo response at 50 % Control force using ZN-PID controller

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Fig. 7: Servo response at 50% Control force using ZN-PID controller

Fig. 8: Servo response at 40% Control force using TGC controller
Fig. 9: Servo response at 50 % Control force using TGC controller

Fig. 10: Servo response at 60 % Control force using TGC controller

The time and error indices performances of the MASS for three different operating points with ±5% and ±10% step change for TGC and ZN-PID controllers are presented in tables 4 and 5.

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Table 4: Mechanical active suspension system simulation: Servo response error indices performance

| Operating Point | Step Change | ZN-PID | TGC |
|-----------------|-------------|--------|-----|
|                 |             | ISE    | IAE | ISE | IAE |
| 40%             | +10%        | 3228   | 187 | 528.9 | 22.09 |
|                 | +5%         | 3377   | 207.8 | 553.4 | 24.51 |
|                 | -5%         | 3228   | 187 | 528.9 | 22.09 |
|                 | -10%        | 3377   | 207.8 | 553.4 | 24.51 |
| 50%             | +10%        | 5016   | 228.5 | 821.9 | 26.97 |
|                 | +5%         | 5165   | 249.3 | 846.3 | 29.42 |
|                 | -5%         | 5016   | 228.5 | 821.9 | 26.97 |
|                 | -10%        | 5165   | 249.3 | 846.3 | 29.42 |
| 60%             | +10%        | 7201   | 269.9 | 1180 | 31.87 |
|                 | +5%         | 7350   | 290.5 | 1204 | 34.32 |
|                 | -5%         | 7201   | 269.9 | 1180 | 31.87 |
|                 | -10%        | 7350   | 290.5 | 1204 | 34.32 |

Table 5: Mechanical active suspension system simulation: Servo response time-domain indices Performance

| Operating Point | Step Change | ZN-PID | TGC |
|-----------------|-------------|--------|-----|
|                 |             | ts    | tr  | ts  | tr  |
| 40%             | +10%        | 22    | 18  | 7   | 04  |
|                 | +5%         | 18    | 17  | 6   | 03  |
|                 | -5%         | 17    | 16  | 6   | 03  |
|                 | -10%        | 22    | 18  | 7   | 04  |

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The graphical view of mechanical active suspension system simulation set point tracking performances ISE, IAE, ts, and tr at 40%, 50%, and 60% operating points are shown in Fig 11, 12, 13 and 14.

![Graphical view of active suspension system simulation ISE performances](image1)

**Fig. 11:** Graphical view of active suspension system simulation ISE performances

![Graphical view of active suspension system simulation IAE performances](image2)

**Fig. 12:** Graphical view of active suspension system simulation IAE performances
VI. Conclusion

In this work, the MASS has been considered and the mathematical model of the transfer function of the processing system is analytically derived. The simulation of three generations of CRONE controllers for three different operating points of control force input has been executed and examined in the Matlab environment. The implementation of all three CRONE controllers designed using the CSD toolbox and found the servo responses to be recorded with ±5% and ±10% of step change is applied. From the investigation, it is validated that the TGC controller proved superiority than the ZN-PID controller.

Conflict of Interest:

Authors declared: No conflict of interest regarding this article

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