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Control Mass-Spring-Damper Based on Tuning Trade-off PID Controller

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ABSTRACT

This paper discusses the control mass-spring-dumper (MSD) system used in vehicle suspensions. The vehicle suspension consists of mass, coil (spring), and shock absorber (dumper). MSD provided a shock effect when the vehicle was caused by the frictional force on the load. The difficulty to achieve the stability of the suspension and the following of set-point tracking. Therefore, Therefore, the proportional-integral-derivative (PID) controller, tuning 1 and 2 degrees of freedom (1-2 DOF), and tuning trade-off PID controller were proposed for stability when the disturbance occurs. The MSD equation was obtained by using the Laplace transform and validated in Matlab Simulink. The result shows that the tuning trade-off PID control reduces disturbance rejection by the smaller set point tracking peak amplitude of 1.01, overshoot of 0.682%, and settling time of 0.318 seconds. The PID controller achieved set point tracking and disturbance rejection with a peak amplitude of 1.52, overshoot of 51.7%, and settling time of 2.23 seconds, and the tuning 1-2 DOF PID controller achieved set point tracking and disturbance rejection with a peak amplitude of 1.07, overshoot 6.53%, settling time 0.524 seconds. The tuning trade-off PID control has the best performance than tuning 1-2 DOF PID and PID controller.

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1. INTRODUCTION

Mass-spring-damper (MSD) suspension systems are used in automobiles and motorcycles [1]. The system consists of mass, coil (spring), and shock absorber (dumper). MSD creates a shock effect caused by friction between the wheels and the load [2]. The vehicle load is used as a support for the suspension of the steering system [3], wheels, and chassis [4]. Important components that must be considered in the suspension such as transmission force, torque, comfort, and stability [5]. The Suspension has three classifications such as passive, semi-active, and active systems [6]. A Semi-active system has the characteristics of stability, balance [7], and low energy consumption [8]. The passive system limits the movement of the car body and wheels at speed for a comfortable ride [9]. The active system consists of actuators, mechanical springs, dampers, actuators, and mechanical springs [10]. The difference between active and passive systems is the controlling force. The controlling force reduces the vibrations that are affected by the entire surface of the vehicle and produce more comfort for the occupant than a driver is comfortable [11]. The comfort of the steering system and road estimation is a concern for controlling the suspension system. The greater the suspension system is disturbed, the MSD will experience vibrations felt by the driver and passengers [12].

The conventional active suspension is mounted in parallel with the spring dumper, hydraulic, and actuator to generate vertical force between the sprung (chassis) [13] and unsprung (wheels) [14]. Active suspension with variable geometry is proposed to reduce mass unsprung gradually and power demand is limited [15]. The adaptive neural scheme is applied to active suspension with vertical mass sprung replacement limits and actuator saturation. Then radial basis function neural networks (RBFNNs) are used to predict the indeterminate body mass and pneumatic spring. So the prescribed performance function (PFF) control is designed for fault tracking from constrained sprung changes [16]. Ride height control (RHC) functions for semi-active air suspension and processes air in or out of the spring. RHC plays a role in improving vehicle performance. Non-linear predictive model control and proportional-integral-derivative are used to evaluate the performance of the car system [17]. Car system performance is analyzed by fuzzy logic for active suspension systems. The suspension system is reviewed for unknown parameters, actuator errors, and displacement limitations. Fault Tolerance Control (FTC) is proposed as an estimation control for external disturbances. External disturbances are analyzed using fault approximation techniques, and actuator errors. FTC is designed to address the output performance of a vehicle's suspension. The proposed control can overcome the problems of fault tolerance and tracking errors [18]. Control Electronic Suspension (ESC) is applied to the suspension to improve the yaw-roll-pitch motion of the vehicle. The vehicle pitch movement is used as a control for each degree of freedom (DOF) movement. The control algorithm consists of an integrated vehicle observer (IVO) for condition estimation, an integrated target generator (ITG), an integrated vehicle controller (IVC), and an optimal distribution controller (ODC). ITG is used as a target for roll, pitch, and yaw angles. IVC detects roll, pitch, and yaw states. The ODC determines the damping force to be applied to each method. The results show that the proposed algorithm can improve roll, yaw, and pitch vehicle movements [19]. Adaptive control is implemented for active suspension vehicles with nonlinearities (spring, and damper). Performance that describes the convergence, maximum overshoot, and steady-state error of the vertical and pitch motion control design. Vertical control performance and transient angle changes, and vehicle steady-state result in error changes [20].

The performance of the mass-spring-damper from transient to steady state is influenced by the spring force and mass transfer to return slowly when shocks occur. This is caused by inertia and vehicle body displacement systems and DOF motion [21]. A single-degree-of-freedom MSD has one mode of oscillation, the mass of which is connected by a single spring. But a single mass connected to two parallel springs causes oscillations back and forth at high frequencies. Meanwhile, the natural frequency experiences more oscillations per second, because a combination of springs arranged in parallel has greater stiffness than a single spring. Parallel springs experience the same displacement, but the resulting force is not the same. MSD system displacement with DOF in dynamic vehicles [22]. In dynamic vehicles, the system response must reach the initial point condition before the disturbance occurs [23], [24]. This study designed a mass-spring-dumper control system for vehicle suspension to dampen oscillations caused by disturbance. The system is designed with a proportional-integral-derivative (PID) control trade-off of 1-DOF and 2-DOF for tracking disturbance rejection. Tracking disturbance using PID tuning.

PID provides reliable control and stability in both linear and non-linear systems. Non-linear systems with proportional (Kp), integral (Ki), and derivative (Kd) constants have limitations in obtaining parameters that can adjust to changes in load [25]. A tuning control is proposed to control an unstable system with a design difference in increased closed-loop response [26].

The development of PID control is fast, but the control produces delays and disturbances that cannot be detected quickly [27]. PID tuning is used to regulate Kp, Ki, and Kd constants in achieving the values required by linear loads [28]. The linear PID system provides a tracking reference and reduces interference. Meanwhile, an unstable system can affect tracking reference and interference. PID also has a large overshoot and settling time [29]. Complex processes such as vehicle suspension require a PID tuning control to overcome the oscillatory response [30]. So, the PID controller control scheme, 1-2 DOF PID, and PID trade-off tuning for setpoint tracking and disturbance rejection are proposed. The control scheme is designed to reduce the oscillatory response of the mass-spring-damper under load and

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without interference. System testing was carried out on Matlab software to validate the design and the resulting response.

2. RESEARCH METHOD

In this section, you should explain how the research was conducted, including research design, research procedure (in the form of algorithms, Pseudocode or other), how to acquire the data and how to perform any test. The description of the course of research should be supported by references, so the explanation can be accepted scientifically.

2.1. PID Controller

The research method uses proportional-integral-derivative (PID) control. PID control corrects the error value between input and output to achieve the response required by the load. If the input and output responses are not the same, then the controller will correct them with a feedback system [31]. The PID control feedback is tuned to disturbance rejection or fast response with good attenuation to change the input set point. Figure 1 describes the P(s), and C(s) process control and transfer functions respectively. On the system r (s) set point, u (s) control output, d (s) output signal, d (s) load disturbance, and y (s) controlled variable [32] described in formula (1).

$$u_{(s)} = P_{(s)}(r(s) - y(s)) \tag{1}$$

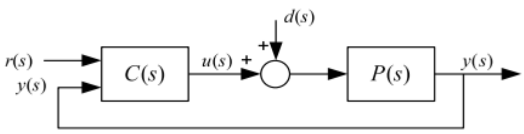

Figure 1. PID Controller [32]

2.2. Tuning 1-DOF dan 2-DOF PID

The 1-DOF and 2-DOF PID controls are used to set the value of the proportional constant (Kp), integral constant (Ki), and derivative constant (Kd) parameters based on variations in the input set point in a closed loop system [26]. Changes in the value of the set point parameter will be monitored (tracking) by reducing the resulting oscillation response (disturbance rejection). Then, the response results are fed back to the u control to find out the error value. The error value is obtained by calculating the difference between the reference input and the response output. The calculation of the error value is explained as follows [29]. 1-DOF control relates to input set point (r), disturbance (d), plant (Po), and output (y). Set point (r) and disturbance (d) have an impact on the error value (u). If (r) and (d) vary in the same way, then control (K) can be selected to reduce the small error value. Input (r) and disturbance (d) have different properties in the control system to provide tracking and disturbance rejection responses [33].

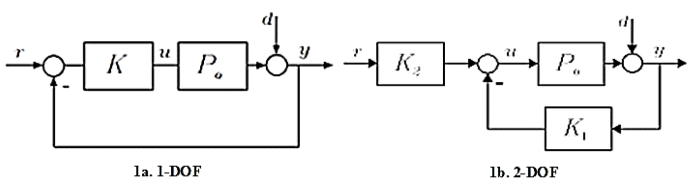

Figure 2. Degree of Freedom (DOF) [33]

The 2-DOF control is explained as input set point (r) on control (K2), error value (u), plant (Po), disturbance (d), output (y), and control feedback (K1). The 2-DOF compensator has a trade-off between feedback and reference tracking. The feedback control system is ensured with a degree of freedom, tracking reference indicated on the control pre-filter. Then the degree of freedom is determined by the open loop of the input reference [34]. The conventional control approach uses feedback to regulate the desired response. in a linear control system, the use of a reference model will determine the desired response from the controlled system [35].

$$u = K\begin{bmatrix} r \\ y \end{bmatrix} = K_2 r - K_1 y \tag{2}$$

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2.3. Tunning Trade-Offs PID

The PID tuning trade-offs are designed to achieve good tracking and fast disturbance rejection at the same time. Assuming a constant control bandwidth, disturbance rejection requires a lot of gains (amplifier) so that the slope of the crossover can be achieved. If the slope is greater, the response will overshoot at the set point. Equation (3) is described as a tuning control where r is the system input, y is the actual output, d is the disturbance, e is defined as the error value, and u is the error signal [36]. The trade-off PID tuning controller can reduce noise quickly without significantly increasing overshoot in set-point tracking. The PID controller trade-off is also useful to reduce the influence of changes in the reference signal on the control signal. Figure 2 shows the control architecture with a trade-off PID controller [37]. The relationship between the trade-off control output (u) and the two inputs (r and y) can be represented in parallel or standard form. The two forms differ in the parameters used to express proportional, integral, and derivative actions. The relationship between the trade-off PID control output (u) and the two inputs (r and y) can be represented by the following equation [38]. The MSD plant system with PID controller tuning control, 1-DOF and 2-DOF PID tuning, and PID trade-off tuning are described in Figure 3. The system was designed using PID tuning parameters and compared based on tracking and disturbance rejection and frequency to obtain the correct overshoot. small in set point tracking and can reduce changes in the reference signal.

$$u = K_p(br - y) + \frac{K_i}{s}(r - y) + \frac{K_d s}{T_f s+1}(cr - y) \tag{3}$$

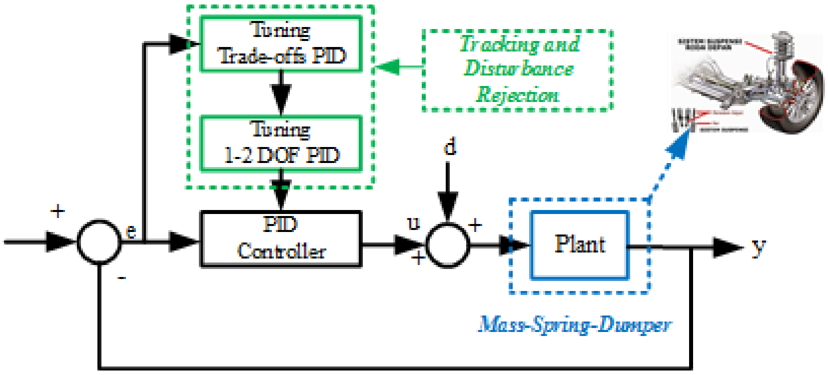

Figure 3. MSD Plant Control System

3. RESULTS AND DISCUSSION

Figure 4 a, b, and c are the MSD (mass-spring-dumper) output responses, where the MSD response on a stable road has a rise time of 2.55 seconds and a settling time of 4.48 seconds. The response to reach a stable 2.5 seconds requires a rise time and a long settling time. This is influenced by the mass-spring to return to its original point when a shock occurs. The output starts with zero to 2.5 seconds describing the state of the wheel and spring shock absorbers. The MSD is installed between the wheels and the vehicle body to reduce shaking and shock. The MSD system is applied to open-loop control. When an MSD disturbance occurs, the system does not provide feedback in the form of variables which will be processed according to the magnitude of the load and physical characteristics. The sudden resistance generated by the response will start to rise and slow down the vehicle according to the decrease in the response and inertia of the car.

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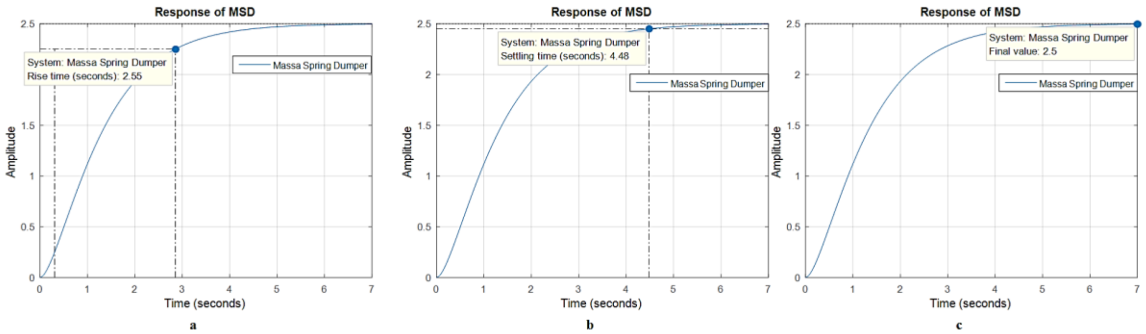

Figure 4. (a) Rise Time, (b) Setling Time, and (c) Steady State Respon Open Loop MSD

The control system uses feedback control in the comparison of tracking and disturbance rejection which is explained in Figures 5 a and b. The control system is designed to reduce interference that will occur and will return to the desired tracking setpoint. The tracking error at peak gain is 1.68 abs and the frequency is 8.12 rad/s. The maximum loop gain of 3.45 dB with a frequency of 6.25 rad/s. Meanwhile, the minimum loop gain occurs at a frequency of 0.00697 rad/s, and a single value of 63 dB. Open-loop tracking and gain are used to determine the frequency domain tracking between predetermined inputs and outputs. The purpose of the setting will determine the maximum error (error) value of the reference input to the tracking error value.

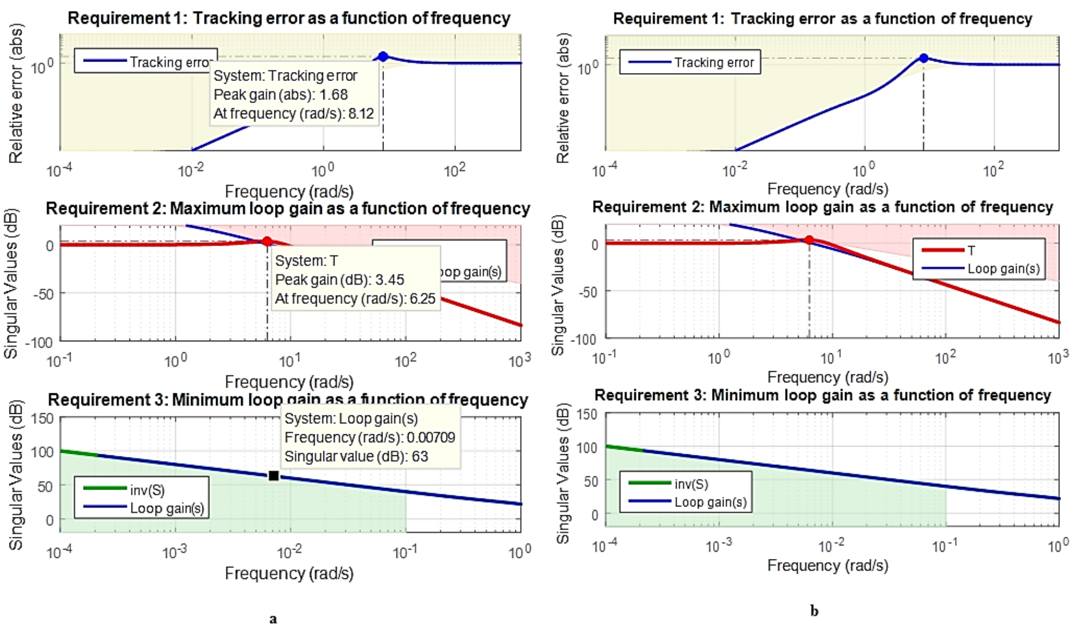

Figure 5. (a) Tracking and (b) Disturbance Rejection

Figure 6a and b show a comparison between the response of setpoint tracking using a PID controller, 1-2 DOF PID tuning, and PID trade-off tuning. The open loop response with a wide frequency is [0, 0, 1] rad/second by adding the value of α. The value of α is the frequency amplifier which is set from 1, 2, and 4 to increase disturbance rejection at a certain point. Setpoint Tracking (a) The PID controller has a settling time of 1.19 seconds, a 1-2 DOF PID tuning of 1.95 seconds, and a PID trade-off tuning of 2.23 seconds. Figure (b) describes the PID controller tracking setpoint overshoot, 1-2 DOF PID tuning, and PID trade-off tuning. The results of the overshoot setpoint tracking using the PID controller have an overshoot of 30.6%, and a peak amplitude of 1.31. Tuning 1-2 DOF PID has an overshoot of 43.3%, a peak amplitude of 1.43. While the PID trade-off tuning gets 51.7% overshoot and 1.52 peak amplitude. The PID controller's settling time response is faster due to the search for Kp, Ki, and Kd parameters that can adjust the load requirements when a disturbance occurs. However, the 1-2 DOF PID tuning and the PID trade-off tuning experienced a delay with the PID parameter search of

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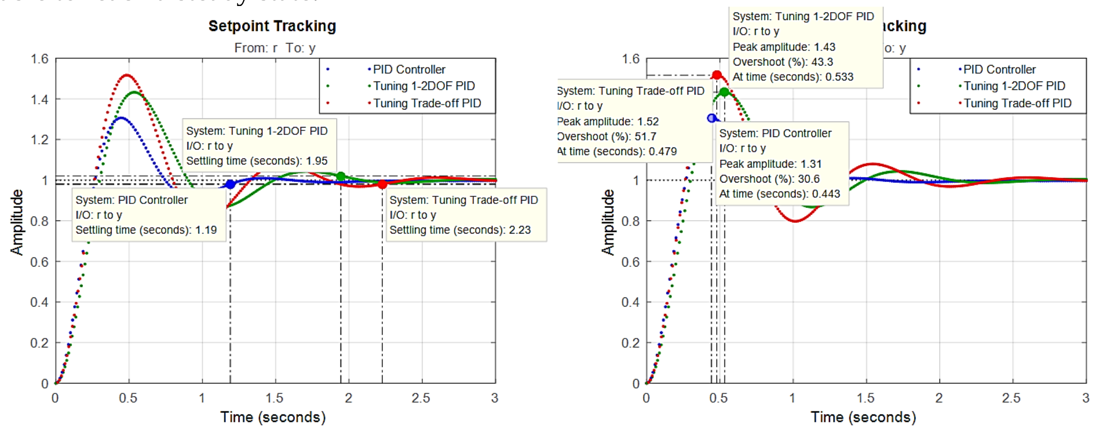
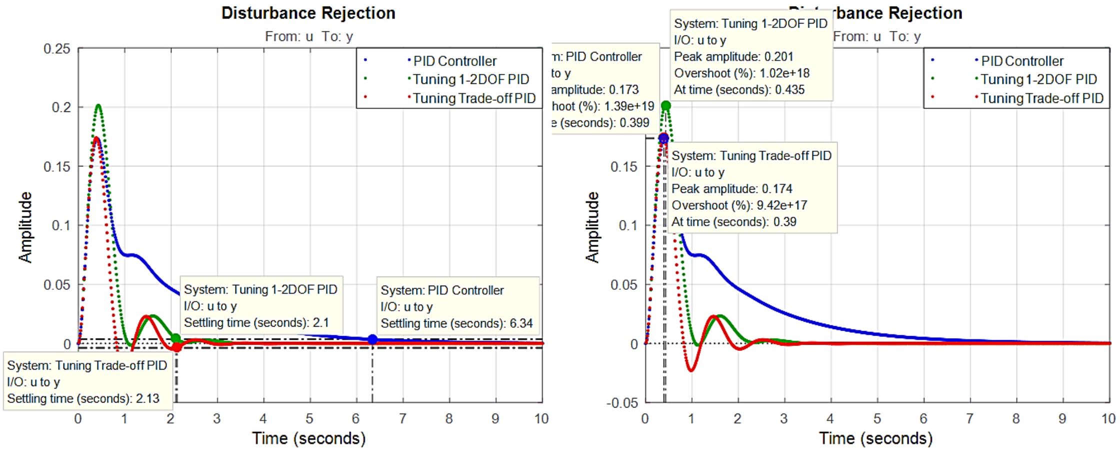
0.443 seconds, 0.533 seconds, and 0.479 seconds. In addition, the overshoot on the three controls affects the MSD to return to its initial state before the disturbance occurs. This indicates that the system is unstable to reach a steady state.



a b

Figure 6. (a) Settling Time and (b) Overshoot Set Point Tracking

Figure 7 (a) and (b) explain the comparison of the PID controller's disturbance rejection response, 1-DOF PID and 2-DOF PID tuning, as well as the PID trade-off tuning to increase shock absorption when the driver is at high speed and can return to the point start stably in case of trouble on the road. The open loop response is affected by changes in the mass-spring dumper load. Disturbance rejection Figure (a) PID controller has a setting time of 6.34 seconds, 1-2 DOF PID tuning with a setting time of 2.1 seconds, and a PID trade-off settling time tuning of 2.13 seconds. Figure (b) PID controller disturbance rejection has a peak amplitude of 0.087, an overshoot of 3.132%. Tuning 1-2 DOF PID has a peak amplitude of 0.201, overshoot of 1.022%, and tuning trade-off PID has a peak amplitude of 10.174, overshoot of 0.39%. This is caused by changes in load and search for the values of the tuning parameters Kp, Ki, and Kd to obtain parameters that match the load requirements.



a b

Figure 7. (a) Settling Time Disturbance Rejection, (b) Overshoot Disturbance Rejection

Figure 8 (a) and (b) show a comparison of the PID controller setpoint tracking control, 1-2 DOF PID tuning, and PID trade-off tuning. The results of the comparison of the PID controller in figure (a) produce a settling time of 2.23 seconds, 0.524 seconds of 1-2DOF PID tuning, and 0.318 seconds of PID trade-off tuning. While figure (b) shows a comparison of the results of the three control setpoint tracking with a PID controller overshoot of 51.7% and a peak amplitude of 1.52, 1-2DOF PID overshoot tuning of 6.53%, and a peak amplitude of 1.07. Meanwhile, the PID trade-off tuning achieves an overshoot of 0.682% and a peak amplitude of 1.01. The time to reach a steady state condition at the tracking setpoint achieved by the PID trade-off tuning control is faster than the other controls. This

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proves that the proposed control can reduce disturbance rejection to adjust to changes in mass-spring-dumper load.

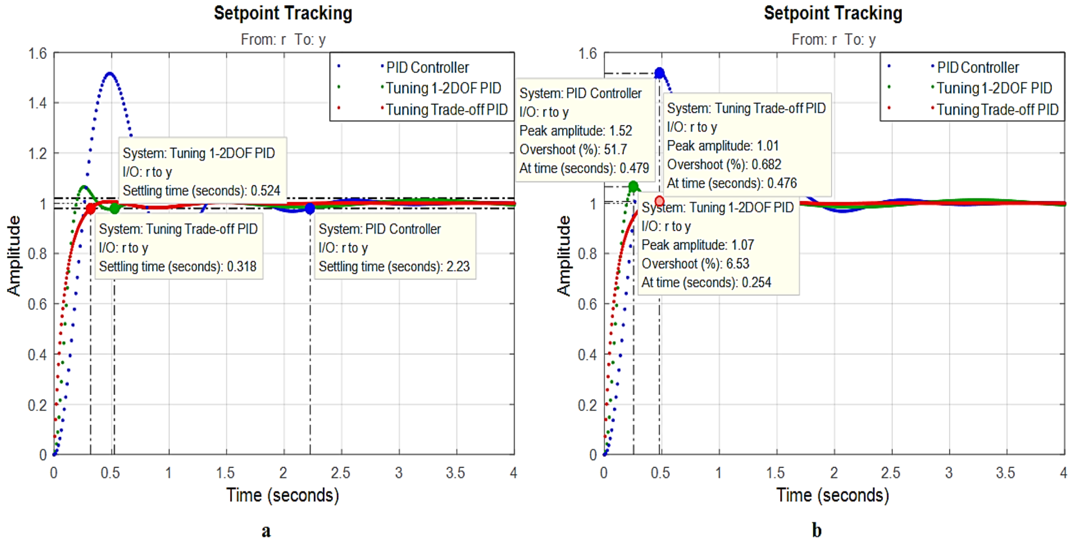

Figure 8. Set Point Tracking 1-DOF, 2-DOF, and Proposed Method

3.1. PID Value Comparison Table

The comparison of PID controller tuning parameters is described in Table 1. The comparison of PID control parameters, 1-2DOF PID tuning, and PID trade-off tuning with each proportional gain control, (K_p) integral gain (K_i), and derivative gain (K_d) can affect the peak amplitude, settling time, and overshoot response in the event of a disturbance. The calculation of the parameters PID controller, 1-2 DOF PID, and tuning trade-off PID is obtained from the equation below. The PID controller parameter is Kp + Ki * s + Kd * Tf*s+1 with Kp = 5.67, Ki = 12.2, Kd = 7.21, Tf = 72.1, The 1-2-DOF PID tuning parameter is u = Kp (b*r-y) + Ki*s (r-y) + Kd*Tf*s+1(c*r-y) with Kp = 0.0615, Ki = 20.2, Kd = 1.85, Tf = 0.106, and the trade-off PID tuning is u = Kp (b*r-y) + Ki*s (r-y) + Kd*Tf*s+1 (c*r-y) with Kp = 9.18, Ki = 45.5, Kd = 0.969, Tf = 2.14e-05.

Table 1. The Comparison of PID Controller Tuning Parameters

PID	The Controller PID Tuning		
	PID Controller	1-DOF dan 2-DOF PID	Trade-off PID
Kp	5.67	0.0615	9.18
Ki	12.2	20.2	45.5
Kd	7.21	1.85	0.969

4. CONCLUSION

MSD (mass-spring-dumper) suspension is used in car or motorcycle vehicle systems. Models of vehicle suspension in the form of mass, coil (spring), and shock absorber (dumper). MSD provides a shock effect when the vehicle is running which is caused by frictional forces on the load. The response must quickly return to a stable point when shocks occur. The MSD equation is obtained by Laplace transform, then validated in Matlab Simulink. Three PID controllers, 1-2 DOF PID tuning, and PID trade-off tuning are proposed to obtain the set point tracking and disturbance rejection. The simulation results show that the PID trade-off tuning control can reduce disturbance rejection to a smaller extent with a set point tracking peak amplitude of 1.01, an overshoot of 0.682%, and a setting time of 0.318 seconds. While the PID controller has a set point tracking and disturbance rejection each larger with a peak amplitude of 1.52, overshoot of 51.7%, settling time of 2.23 seconds, and tuning of 1-2 DOF PID has a peak amplitude of 1.07, overshoot of 6.53%, settling time of 0.524 seconds.

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REFERENCES

[1] S. M. Savaresi, C. Poussot-Vassal, C. Spelta, O. Sename, and L. Dugard, *Semi-Active Suspension Control Design for Vehicles*, 1st Ed. Oxford: Elsevier Ltd, 2010.

[2] H. Arrosida and A. Sudaryanto, “Perancangan Metode Kontrol Linear Quadratic Integral Tracking (LQIT) untuk Pengendalian Posisi Sistem Suspensi Sederhana (Mass-Spring-Damper),” *J. Nas. Tek. Elektro*, vol. 6, no. 2, p. 111, 2017.

[3] C. Canudas-de-Wit, F. R. Rubio, and M. A. Corchero, “D-OSKIL: A new mechanism for controlling stick-slip oscillations in oil well drillstrings,” *IEEE Trans. Control Syst. Technol.*, vol. 16, no. 6, pp. 1177–1191, 2008.

[4] B. Amstrong, “A Survey of Models, Analysis Tools and Compensation Methods for the Control of Machines with Friction,” *Automatica*, vol. 56, no. 2, pp. 158–159, 1994.

[5] D. Wang, D. Zhao, M. Gong, and B. Yang, “Research on Robust Model Predictive Control for Electro-Hydraulic Servo Active Suspension Systems,” *IEEE Access*, vol. 6, pp. 3231–3240, 2017.

[6] X. Ma, P. K. Wong, J. Zhao, J. H. Zhong, H. Ying, and X. Xu, “Design and Testing of a Nonlinear Model Predictive Controller for Ride Height Control of Automotive Semi-Active Air Suspension Systems,” *IEEE Access*, vol. 6, pp. 63777–63793, 2018.

[7] Y. Qin, F. Zhao, Z. Wang, L. Gu, and M. Dong, “Comprehensive analysis for influence of controllable damper time delay on semi-active suspension control strategies,” *J. Vib. Acoust. Trans. ASME*, vol. 139, no. 3, 2017.

[8] Y. Qin, C. Xiang, Z. Wang, and M. Dong, “Road excitation classification for semi-active suspension system based on system response,” *JVC/Journal Vib. Control*, vol. 24, no. 13, pp. 2732–2748, 2018.

[9] A. R. Bhise, R. G. Desai, M. R. N. Yerrawar, A. C. Mitra, and D. R. R. Arakerimath, “Comparison Between Passive And Semi-Active Suspension System Using Matlab/Simulink,” *IOSR J. Mech. Civ. Eng.*, vol. 13, no. 04, pp. 01–06, 2016.

[10] A. F. Mohd Riduan, N. Tamaldin, A. Sudrajat, and F. Ahmad, “Review on active suspension system,” *SHS Web Conf.*, vol. 49, p. 02008, 2018.

[11] Y. Zhang, Y. Liu, and L. Liu, “Adaptive Finite-Time NN Control for 3-DOF Active Suspension Systems With Displacement Constraints,” *IEEE Access*, vol. 7, pp. 13577–13588, 2019.

[12] N. Aisyiyah, “Pemodelan Sistem Suspensi Kendaraan Dengan Menggunakan Software Solidwork,” Institut Teknologi Sepuluh November, 2016.

[13] W. Sun, H. Gao, and B. Yao, “Adaptive robust vibration control of full-car active suspensions with electrohydraulic actuators,” *IEEE Trans. Control Syst. Technol.*, vol. 21, no. 6, pp. 2417–2422, 2013.

[14] T. P. J. Van Der Sande, B. L. J. Gysen, I. J. M. Besselink, J. J. H. Paulides, E. A. Lomonova, and H. Nijmeijer, “Robust control of an electromagnetic active suspension system: Simulations and measurements,” *Mechatronics*, vol. 23, no. 2, pp. 204–212, 2013.

[15] M. Yu, C. Arana, S. A. Evangelou, D. Dini, and G. D. Cleaver, “Parallel Active Link Suspension: A Quarter-Car Experimental Study,” *IEEE/ASME Trans. Mechatronics*, vol. 23, no. 5, pp. 2066–2077, 2018.

[16] C. M. Ho, D. T. Tran, C. H. Nguyen, and K. K. Ahn, “Adaptive Neural Command Filtered Control for Pneumatic Active Suspension with Prescribed Performance and Input Saturation,” *IEEE Access*, vol. 9, pp. 56855–56868, 2021.

[17] J. Zhao, P. K. Wong, Z. Xie, C. Wei, and F. He, “Integrated variable speed-fuzzy PWM control for ride height adjustment of active air suspension systems,” *Proc. Am. Control Conf.*, vol. 2015-July, pp. 5700–5705, 2015.

[18] C. M. Ho and K. K. Ahn, “Design of an Adaptive Fuzzy Observer-Based Fault Tolerant Controller

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ISSN 2355-5068 ; e-ISSN 2622-4852
DOI: 10.33019/jurnalecotipe.v10i1.3723

for Pneumatic Active Suspension with Displacement Constraint," *IEEE Access*, vol. 9, pp. 136346–136359, 2021.

[19] W. Cho, J. Suh, and S. H. You, "Integrated Motion Control Using a Semi-Active Damper System to Improve Yaw-Roll-Pitch Motion of a Vehicle," *IEEE Access*, vol. 9, pp. 52464–52473, 2021.

[20] J. Na *et al.*, "Active Adaptive Estimation and Control for Vehicle Suspensions With Prescribed Performance," vol. 26, no. 6, pp. 1–15, 2017.

[21] B. Hirzinger, C. Adam, and P. Salcher, "Dynamic response of a non-classically damped beam with general boundary conditions subjected to a moving mass-spring-damper system," *Int. J. Mech. Sci.*, vol. 185, no. June, 2020.

[22] P. Salcher and C. Adam, "Modeling of dynamic train–bridge interaction in high-speed railways," *Acta Mech.*, vol. 226, no. 8, pp. 2473–2495, 2015.

[23] N. Zhang, H. Xia, W. W. Guo, and G. De Roeck, "A Vehiclebridge Linear Interaction Model and Its Validation," *Int. J. Struct. Stab. Dyn.*, vol. 10, no. 2, pp. 335–361, 2010.

[24] M. Barreau, S. Tarbouriech, and F. Gouaisbaut, "Lyapunov Stability Analysis of a Mass–Spring System Subject to Friction," *Syst. Control Lett.*, vol. 150, p. 104910, 2021.

[25] V. Rajinikanth and K. Latha, "Setpoint weighted PID controller tuning for unstable system using heuristic algorithm," *Arch. Control Sci.*, vol. 22, no. 4, pp. 481–505, 2012.

[26] H. Taguchi and M. Araki, "Two-Degree-of-Freedom PID Controllers — Their Functions and Optimal Tuning," *IFAC Proc. Vol.*, vol. 33, no. 4, pp. 91–96, 2000.

[27] N. Hidayah, J. Juwari, and R. Handogo, "Maximum Peak-Gain Margin 2DOF-IMC Tuning for a 2DOF-PID Filter Set Point Controller Under Parametric Uncertainty," *IPTEK J. Proc. Ser.*, vol. 0, no. 1, pp. 8–16, 2014.

[28] A. Mulyadi, "Desain Sistem Kendali Kecepatan Motor Induksi Pada Cane-Carier Based on Cohen-Coon Method," *J. Zetroem*, vol. 02, no. 01, pp. 16–20, 2020.

[29] M. A. Johnson and M. H. Moradi, *PID Control: New Identification and Design Methods*. 2005.

[30] F. Naz *et al.*, "PID Tuning with reference tracking and plant uncertainty along with disturbance rejection," *Syst. Sci. Control Eng.*, vol. 9, no. 1, pp. 160–166, 2021.

[31] A. Mulyadi, Wijono, and B. Siswojo, "Kecepatan Motor Penggerak Quadcopter," *J. Transm.*, vol. 22, no. 4, pp. 107–116, 2020.

[32] V. M. Alfaro and R. Vilanova, "Optimal robust tuning for 1DoF PI/PID control unifying FOPDT/SOPDT models," *IFAC Proc. Vol.*, vol. 2, no. PART 1, pp. 572–577, 2012.

[33] S. Alcantara, C. Pedret, and R. Vilanov, "A Model Reference Based 2-DOF Robust Observer-Controller Design Methodology," *New Approaches Autom. Robot.*, no. May, 2008.

[34] D. C. Youla and J. J. Bongiorno, "A Feedback Theory of Two-Degree-of-Freedom Optimal Wiener-Hopf Design," *IEEE Trans. Automat. Contr.*, vol. 30, no. 7, pp. 652–665, 1985.

[35] V. M. Alfaro and R. Vilanova, "Simple robust tuning of 2DoF PID controllers from a performance/robustness trade-off analysis," *Asian J. Control*, vol. 15, no. 6, pp. 1700–1713, 2013.

[36] M. Clever, "PID Setpoint Tracking and Disturbance Rejection," *Matwork*, 2012. [Online]. Available: https://www.mathworks.com/help/control/ug/pid-tuning-for-setpoint-tracking-vs-disturbance-rejection.html.

[37] R. Kurokawa, N. Inoue, T. Sato, O. Arrieta, R. Vilanova, and Y. Konishi, "Simple optimal PID tuning method based on assigned robust stability - Trade-off design based on servo/regulation performance," *Int. J. Innov. Comput. Inf. Control*, vol. 13, no. 6, pp. 1953–1963, 2017.

[38] B. Astrom, K.J. & Wittenmark, *Computer-Controlled Systems - Theory and Design*. Prentice-Hall, 1967.