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Research Article

Optimal PID Controller Design Based on Proportional Gain for Quarter Vehicle Model

Abdullah Turan[1*], Huseyin Aggumus[2]

[1*] Sirnak University, Departmant of Mechanical and Metal Technologies, Sirnak, Turkey, (ORCID: 0000-0002- 0174-2490), abdullahturan@sirnak.edu.tr

[2] Sirnak University, Departmant of Mechanical and Metal Technologies, Sirnak, Turkey, (ORCID: 0000-0002- 7158-677X), haggumus@sirnak.edu.tr

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Abstract

In this study, an effective and new design method was used to determine the parameters of the PID controller used in order to improve the performance of a vehicle's active suspension system and to suppress vibrations in the vehicle. In this method, the PID controller is designed based on the optimal proportional gain k_p setting, taking into account the settling time and maximum overshoot of the quarter vehicle system. This method is based on obtaining the other parameters of the controller by adjusting the k_p to minimize the settling time and maximum overshoot error in a stable cycle. The obtained simulation results were evaluated by comparing the uncontrolled suspension system and the suspension system in which the PID controller whose parameters were adjusted with the proposed effective design method. It suppressed the system responses of the PID controller more effectively than the passive suspension system.

Anahtar Kelimeler: Quarter vehicle model, PID controller, Active control.

Çeyrek Taşıt Modeli için Oransal Kazanca Dayalı Optimum PID Kontrolör Tasarımı

Öz

Bu çalışmada, bir aracın aktif süspansiyon sisteminin performans iyileştirilmesi ve araçta meydana gelen titreşimlerin bastırılması amacıyla kullanılan PID kontrolöre ait parametrelerinin belirlenmesinde etkin ve yeni bir tasarım yöntemi kullanılmıştır. Bu yöntemde PID kontrolör, çeyrek taşıt sistemin yerleşme süresi ve maksimum aşması dikkate alınarak optimal oransal kazanç ayarına dayalı olarak tasarlanmıştır. Bu yöntem, kararlı bir döngüde yerleşme süresini ve % aşım hata oranını en aza indirgemek için optimum oransal kazancı (k_p) ayarlayarak kontrolörün diğer parametrelerini elde etmeye dayanmaktadır. Elde edilen simülasyon sonuçları, kontrolsüz süspansiyon sistemi ile önerilen etkin tasarım yöntemiyle parametreleri ayarlanan PID kontrolörün uygulandığı süspansiyon sisteminin karşılaştırılmasıyla değerlendirilmiştir. PID kontrolörün sistem cevaplarını pasif süspansiyon sisteminden daha etkili bir şekilde bastırmıştır.

Keywords: Çeyrek taşıt modeli, PID kontrolör, Aktif kontrol.

[*] Corresponding Author: abdullahturan@sirnak.edu.tr

1. Introduction

Suspension systems, which have a direct effect on the driving dynamics due to being between the vehicle body and the wheel, are one of the most important parts of the vehicle dynamics (Cao et al., 2011). Today, three different types of suspension designs have been proposed by researchers as passive, semi-active and active. Due to the high costs of semi-active and active suspensions, manufacturers prefer passive suspension systems (Kararsız and Basturk, 2018). However, passive suspensions compromise comfort or handling performance due to internal limitations (Aly and Farhan, 2013).

In vehicles, semi-active and active suspension systems have been proposed by researchers to provide comfort and handling performance at the same time. In semi-active systems, it is desired to prevent the effect of disturbances by using variable damping that can be adjusted according to road conditions. Here, although the power required by the system is low, the desired comfort cannot be reached due to the dampers with variable damping ratio used in semi-active suspension systems operating in the low frequency range (0-1 Hz). To overcome this situation, active suspension systems are used. Here, it is aimed not to transmit the disruptive effect to the vehicle body by using drive elements instead of adaptive dampers. The disadvantage of the system is that it requires more energy than semi-active systems. However, the driving element used does not reflect the effects of the disturbances in the range of approximately 0-10 Hz to the vehicle body (Koch et al., 2010).

In this study, active suspension control with an optimal PID controller designed by applying a new and effective design method on the quarter vehicle model is proposed. In the literature, researchers have been working on many different control design methods for active suspension systems. Examples of these are LQR (Taghirad and Esmailzadeh, 1998; Altun, 2017), fuzzy logic (Guclu, 2005), floating type control (Guclu and Yagız, 2004), backstepping approach (Karlsson et al., 2001), adaptive nonlinear control (Lin and Kanellakopoulos, 1996), H_∞ (Onat et al., 2005) and the most common PID controller (Guclu and Ates, 2005; Kuo et al., 1999). PID controller is widely used in industrial applications due to the simplicity of the control structures, easy to understand, easy to maintain, and low cost (Denizci and Ulu, 2020), and the Ziegler-Nichols method (Ziegler and Nichols, 1942), gain and phase margin (Ho et al., 1995) for determining the optimum parameters. Many methods such as Cohen-Coon internal model control (Cohen and Coon, 1953), error-integral criterion adjustment formulas (Astrom et al., 1993; Astrom and Hagglund, 1995) have been applied. However, closed loop responses of these controllers may not be at the desired level in some cases. In studies on the development of these methods, the desired answers are not always obtained (Zhuang and Atherton, 1993). In addition, the recently proposed weighted geometric center method (Turan et al., 2019), gain scheduling (Onat et al, 2017) has been successfully applied. Therefore, the studies carried out to determine the optimum controller parameters are still up-to-date today. The maximum performance to be obtained by applying the PID controller to a system depends on the optimum setting of its parameters.

In this study, the active control performance of the PID controller, which gives successful results in structural systems (Turan and Aggumus, 2021a; 2021b) and in the quarter vehicle model for semi-active control (Turan and Aggumus, 2022), on the quarter vehicle model, whose optimum parameters are obtained with a new approach, is investigated. In this method, the PID controller design is based on the optimum k_p based on the settling time requested from the system and the maximum overshoot. The infrastructure of the technique is based on obtaining other PID controller parameters by setting the optimum k_p that minimizes the t_s and M_p error rate in a stable loop, and this process allows the calculation of optimum controller parameters by creating a loop in the stable area.

The main motivation of the study is the application of the PID controller designed with the proposed optimal method to the quarter vehicle model for the first time.

2. Material and Method

2.1. Quarter Vehicle Model

The quarter vehicle model used in this study is shown in Figure 1. The equations of motion of the system are as follows.

$$m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = -U \quad (1)$$

$$m_2\ddot{x}_2 - c_1(\dot{x}_1 - \dot{x}_2) - k_1(x_1 - x_2) + k_2(x_1 - x_r) = U \quad (2)$$

Quarter vehicle parameters are m_1=338.5 kg, m_2=59 kg, k_1=15000 N/m, k_2=190000 N/m, c_1=600 Ns/m (Mahala et al., 2009).

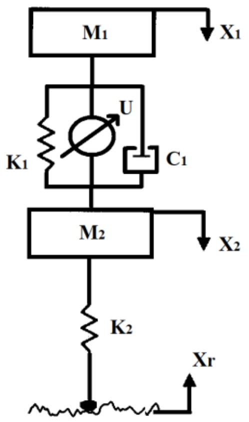

Figure 1. Quarter Vehicle Model

The disruptive road input applied to the vehicle model is shown in Figure 2. Here x and R denote amplitude and path length, respectively. Class A-B road is taken into account according to ISO_8806 standards (Agostibacchio et al., 2014).

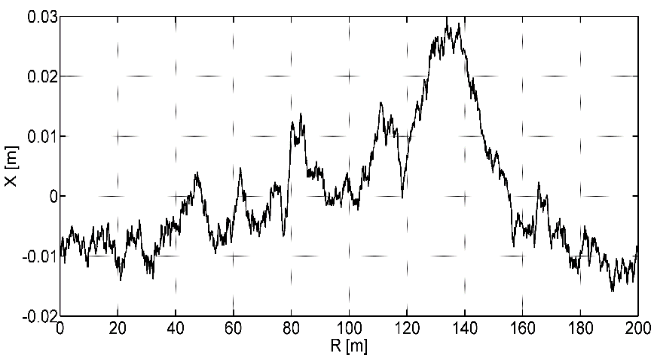

Figure 2. The excitation acting on the system

2.2. Methodology

Figure 3 shows the block diagram of the feedback PID control system.

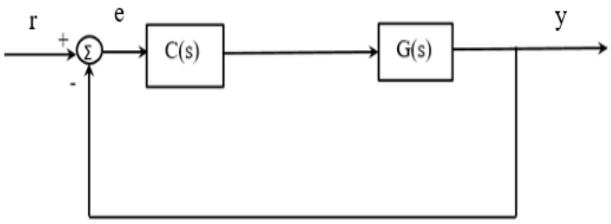

Figure 3. Feedback PID control system

The input, error, and output of the system are denoted by r, e, and y, respectively.

The controller equation of the system is as follows.

$$C(s) = \frac{(k_d s^2 + k_p s + k_i)}{s} \tag{3}$$

Accordingly, the general unit feedback loop system T(s) of the system is given in Eq (4).

$$T(s) = \frac{C(s)G(s)}{1+(C(s)G(s))} \tag{4}$$

Substituting Eq (3) in Eq (4) gives Eq (5).

$$T(s) = \frac{G_N(s)(k_d s^2 + k_p s + k_i)}{G_D(s)s + G_N(s)(k_d s^2 + k_p s + k_i)} = \frac{T_N(s)}{T_D(s)} \tag{5}$$

$T_D(s)$ is the denominator of the system, in other words its characteristic equation. The degree of the system is determined from the characteristic equation. t_s and M_p values are determined. The damping ratio and natural frequency of the system are calculated and written in the target polynomial of the closed loop system given in Eq. (6).

$$\Delta(s) = s^2 + 2\zeta w_n s + {w_n}^2 \tag{6}$$

Here, a difference polynomial R(s) is defined since Δ(s) is of order 2. In addition, R(s) should contain as many variables as the degree difference (m) between $T_D(s)$ and Δ(s) Eq. (7).

$$R(s) = \begin{cases} s + a, & m = 1 \\ s^2 + a_1 s + a_2, & m = 2 \\ s^3 + a_1 s^2 + a_2 s + a_3, & m = 3 \\ & \vdots \\ s^n + a_1 s^{n-1} + a_2 s^{n-2} + \cdots a_n s^{n-m}, & m = n \end{cases} \tag{7}$$

The a_1, a_2, $a_3 \ldots a_n$ expressed in Eq. (7) must be ∈R. The condition to be satisfied here is seen in Eq. (8).

$$(\Delta(s)R(s))_{coeff} \equiv T_D(s)_{coeff} \tag{8}$$

To equalize the number of equations in Eq. (8), k_p is subtracted from the variables and a solution set is obtained.

It is sufficient for the variables in R(s) to be positive for the stability of the system. The flow chart of finding the optimum PID parameters is given in Figure 4. M_p and t_s values are determined for k_p in the value range that makes the system stable, and e_1 and e_2 variables are assigned to the error rates, respectively.

$$e_1 = \frac{M_p - M_{p_{ans}}}{M_p} \tag{9}$$

$$e_2 = \frac{t_s - t_{s_{ans}}}{t_s} \tag{10}$$

Therefore, both error values are expressed in one equation as shown in Eq. (11).

$$err = xe_1 + ye_2 \tag{11}$$

Here, x, y are the coefficients affecting the total error, and x and y values are selected according to the importance expected from the system and x+y=1. The err value obtained is also added to the loop and PID controller parameters are calculated according to the err_{min} value.

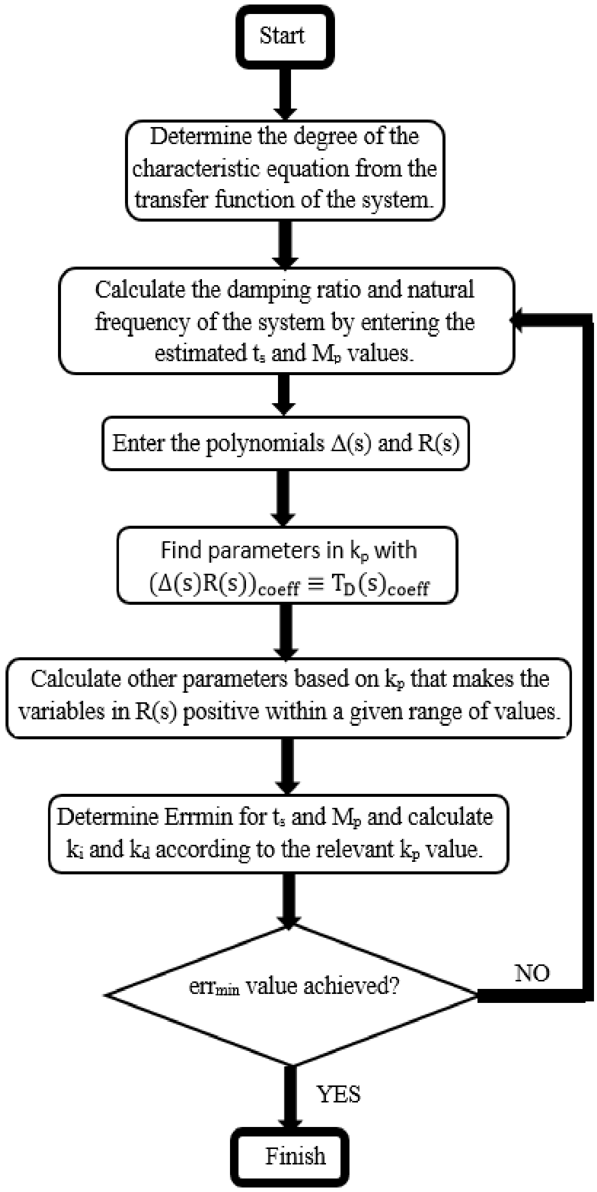

Figure 4. Flowchart of Parameter Setting Algorithm

2.3. PID Controller Design

In order to investigate the effectiveness of the proposed method, the t_s, M_p, peak time t_p and rise time t_r criteria of the system were evaluated. The transfer function of the quarter vehicle model given in Figure 1 is given in Eq. (12).

$$G(s) = \frac{-0.0029586s^2 - 3.8747e{-16}s - 9.5276}{s^4 + 11.9446s^3 + 3518.955s^2 + 5716.5781s + 142914.4519} \tag{12}$$

First of all, the expected performance values from the system were determined as 0.001% M_p and 0.5 s t_s. According to the obtained m=3 value, the difference polynomial is considered as in Eq. (13).

$$R(s) = s^3 + as^2 + bs + c \tag{13}$$

Therefore,

$a = 8.663 - 0.0001849*k_p$, $b= 3303.0 - 2.422e-17*k_p$

$c= - 0.5955*k_p - 7013.0$

expressions were obtained with the proposed method, For the stability criterion, it is sufficient for the variables a, b, c to be positive. k_p = [-60000, 0] is selected. Eq. (14) was obtained so that the total error value would be at the same rate according to the M_p and t_s and was written in its place within the loop. Accordingly, the PID controller parameters in the line with the err_{min} value are the optimum parameters to be reached.

$$Err = 0.5e_1 + 0.5e_2 \tag{14}$$

The PID controller parameters calculated according to the err_{min} result obtained as a result of the loop and the performance criteria in the step response of the system with the controller are given in Table 1.

Table 1: PID control parameters and performance measures of the system

PID controller parameters			**Performance criteria of the system**			
k_p	k_i	k_d	t_s (s)	M_p (%)	t_p (s)	t_r (s)
-37420	-123790	-6637.6	1.0405	0.0995	0.1275	0.0823

3. Results and Discussion

3.1. Simulation Results

Simulations were carried out in Matlab-Simulink environment to test the performance of the PID controller designed for the quarter vehicle model in Figure 1. Displacement in Figure 5, displacement RMS with maximum displacement in Figure 6, and % performances of these values in Figure 7 are shown. In the figures, the situation where there is no control application is defined as passive, and the situation with control application is defined by PID.

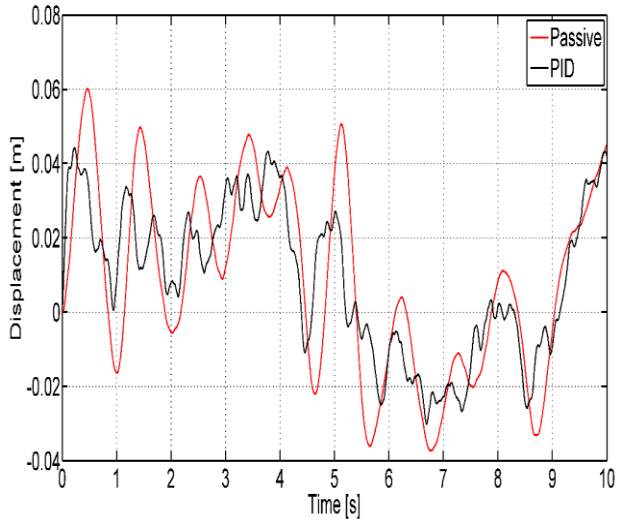

Figure 5. Displacement time response of the system

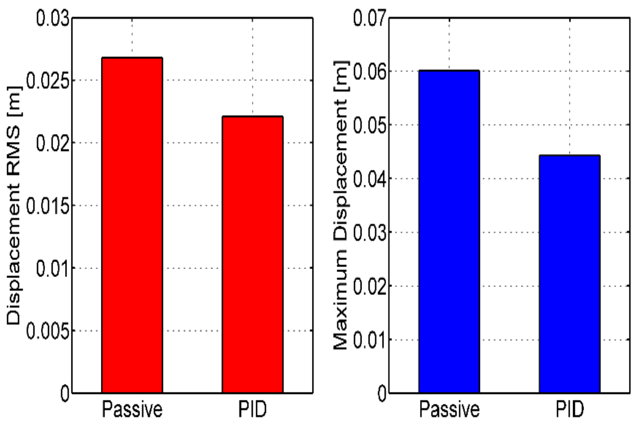

Figure 6. Maximum displacement and displacement RMS values of the system

As can be clearly seen in Figure 5 and Figure 6, the PID control performance of the system in time response, maximum response and RMS responses is higher than the passive state. Considering the % performance improvements compared to the passive state in Figure 7, there is a performance increase of 22% in maximum responses and 18% in RMS responses.

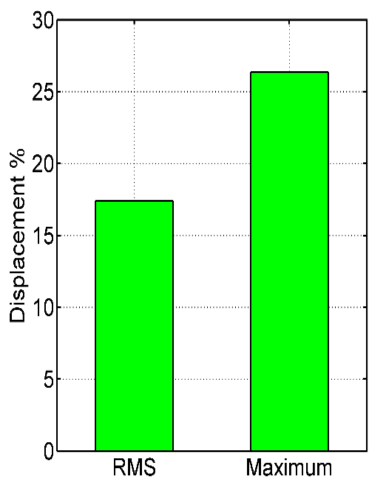

Figure 7. The performances % of the system's maximum displacement and displacement RMS values

4. Conclusion

In this study, the performance of the PID controller, whose optimum parameters were obtained with a new approach, on the quarter vehicle model was investigated. In this method, the PID controller design is based on the optimum k_p based on the settling time requested from the system and the maximum overshoot. The background of the technique is based on obtaining other PID controller parameters by setting the optimum k_p that minimizes the t_s and M_p error rate in a stable loop, and this process allows the calculation of optimum controller parameters by creating a loop in the stable area. The system responses were obtained by comparing the passive control application with the active control application to the vehicle model. With the determined parameters, the PID controller improved the performance of the examined vehicle model. Applying the present method for the multi-degree-of-freedom system is recommended for the next study.

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