Model of Agricultural Vehicle Operator's Seat with PID Control

Iksan Khosiya Rohman

January 12, 2024

Model of Agricultural Vehicle Operator's Seat with PID Control

Iksan Khosiya Rohman[1] Second Author[2]

[1] University of Al Azhar Indonesia, South Jakarta, Indonesia
[2] Springer Heidelberg, Tiergartenstr. 17, 69121 Heidelberg, Germany
ihsankhosyiarohman@gmail.com

Abstract. Seat needs to be made as comfortable as possible, especially seating in a vehicle, especially in vehicles that always run on uneven surfaces such as fields, there will be many disturbance inputs that cause discomfort or even make the body unhealthy. The use of MSDS as a cushion on the seat is the right choice to reduce disturbance input such as a sudden impact, then the damping contained in it makes vibration and resonance can be reduced properly, but the system will be more if the system is stabli. PID control is one of the control systems that can be added to stabilize this system. In this study, we tested 3 MSDS systems and gave them PID control so that the system became stable and safe.

Keywords: Mass-Spring-Damper System, Simulation, Control, PID Control.

1 Introduction

Comfort when driving, especially when driving equipment or transportation, is generally found in the suspension of each system, but not only that, comfort is also found in the seat occupied by the operator, so to increase comfort it is necessary to provide springs, especially those that can be controlled to reduce disturbances. or bumping from uneven surfaces optimally[1].

The human body can be exposed to vibrations in various transportation environments, such as cars, buses, tractors, mining machinery, trains, monorails, ships and more. Exposure to these vibrations can cause discomfort and health problems. Evidently, many experiences show that vibrations can be potentially harmful to the human body, depending on intensity, frequency, exposure time, sitting posture, body type and other factors[2].Vibration from vehicles is also something that affects comfort when driving apart from vehicle shaking caused by uneven roads. Vibrations can cause bodily injury and deterioration, cause fatigue and increase a person's chances of having an accident[3]. Such is the case with tractors. Tractor drivers experience discomfort when exposed to excessive low-frequency vibrations while doing a lot of work on the farm, causing tractor driver performance to be impaired and leading to underutilization of tractor power[4] and heavy truck, heavy truck drivers experience whole body vibration (WBV) because they spend most of their time driving for long distances, which will result in driver discomfort. Various elements can affect the driver's driving comfort such as road roughness, vehicle suspension system, and seat suspension system. In addition, the noise and vibration generated from the internal combustion engine also have an effect on driver discomfort. As a result, the human body subjected to vibrations can experience adverse health disorders such as fatigue, back pain, motion sickness, nerve disorders, spline fractures, and fatigue[5].

Several suspensions have been created, namely with the application of a mass spring damper system[6] and its development to date and most seating system designs can be described dynamically as linear systems. However, the human body greatly deviates from simple rigid mass but has characteristics such as springs and damping.

PID control has been widely used in all industrial fields at this time[7]. The three PID controls have their own advantages such as proportional control has the advantage of fast risetime, then integral control has the advantage of minimizing error, and derivative control has the advantage of minimizing error and reducing overshoot/undershoot. The elements of P, I and D controllers each overall aim to accelerate the reaction of a system, produce offsets and produce large initial changes.

In this research will model the mass spring damper system for operator seating in agricultural vehicles with the 1-DOF, 2-DOF and 3-DOF concepts along with PID control applied to each model to obtain stability when operating. The mass parameter of the operator is assumed to be the average mass of operators with male gender whose average mass ranges from 60-80 kg, so the middle value taken is 70 kg as the research sample.

2 Literature Review

2.1 Modelling with Single Degree Freedom

The mechanical system that will be applied in the first study is with the concept of single degree of freedom to model the operator's seat on an agricultural vehicle. Assumption $m_1$mass of the operator with mass m is mounted on the solid frame of Figure 1 with a spring and a damper. The mass of the spring in the system can be neglected first. The system is excited harmonically by a variable force F_1 and moves linearly in the direction of the spring axis and the damper axis.

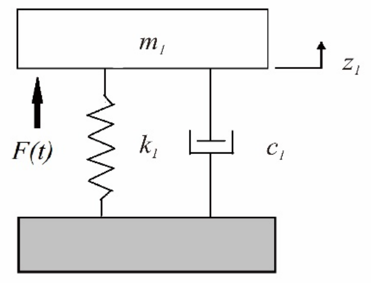

Fig. 1. System with Single Degree of Freedom

Dynamic equation of motion which applies to the mechanical system Figure 1 in its vector form:

$$m\,a = F_r + F_d + F(t) \tag{1}$$

Substituting the forces equation of the motion.

$$m\,\ddot{x} = -F_r - F_d + F(t) \tag{2}$$

Where *m* the mass of the operator.
m_1 = mass
a, $\ddot{x}$ = mass acceleration.
F_r= reaction force on the spring.
F_d= damping force.
$F(t)$ = total force.
The reaction force on a spring can be expressed as follows.

$$F_r = k.x \tag{3}$$

And the damping force can be expressed as follows,

$$F_d = c.\dot{x} \tag{4}$$

Total force.

$$F(t) = F_0 \sin(\omega t) \tag{5}$$

k = spring coefficient.
c = damping coefficient
x = spring displacement
ω = natural frequency
Then substituting equations (3), (4), and (5) into equation (2), the equation for a single degree of freedom mechanical system is obtained.

$$\ddot{x} = -\frac{c}{m}\dot{x} - \frac{k}{m} + \frac{1}{m}F(t) \tag{6}$$

From these equations, kinematic values such as displacement, velocity, and acceleration of the mechanical system can be obtained against time. The SDOF framework can be used for modeling the machine. In addition, visualization and analysis are very simple[9].

2.2 Modelling with Two Degree of Freedom

A mechanical system with two degrees of freedom is a system consisting of two mass bodies m_1 and m_2 which is mounted on a frame with springs having a spring constant k_1 and a damper with a linear damping coefficient b_1 just like SDOF which contain of two SDOF[10]. The two objects are bound to each other by a spring with a spring constant of k_2 and a damper with a linear damping coefficient b_2 with negligible spring masss[11].

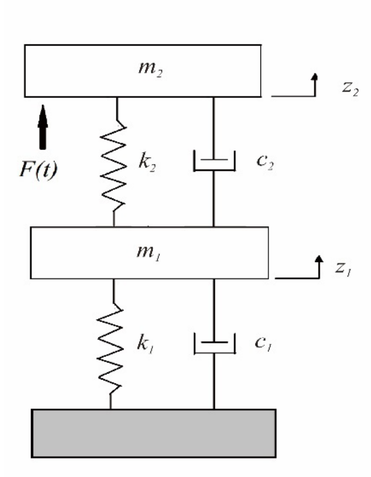

Fig. 2. System with Two Degree of Freedom concept

The second-order differential equation of motion of the system associated with the mechanical system is depicted in Figure 2. With the equation that can be written as follows.

$$m_1 \ddot{z}_1 = -k_1 z_1 - c_1 \dot{z}_1 + k_2(z_2 - z_1) + c_2(\dot{z}_2 - \dot{z}_1) + F(t) \quad (7)$$

$$m_2 \ddot{z}_2 = -k_2(z_2 - z_1) - c_2(\dot{z}_2 - \dot{z}_1) \quad (8)$$

Then processed to solve in MATLAB and get the results of the first-order differential by substituting.

$$x_1 = y_1 \quad (9)$$

$$x_2 = \dot{y}_1 \quad (10)$$

$$x_4 = \dot{y}_2 \quad (11)$$

Then transform equations (7) and (8) into four first-order differential equations.

$$\dot{x}_1 = x_1$$

$$\dot{x}_1 = \frac{1}{m_1}[-(c_1 + c_2)x_2 + c_2 x_4 - (k_1 + k_2)x_2 + k_2 x_3 + F(t)] \quad (12)$$

$$\dot{x}_4 = \frac{1}{m_2}[c_2 x_2 - c_2 x_4 + k_2 x_3 + k_2 x_1] \quad (13)$$

2.3 Modelling with Three Degree of Freedom

The three degree of freedom system is the same as the previous two systems with the addition of one mass with springs and dampers in series. The equation can be written as follows.

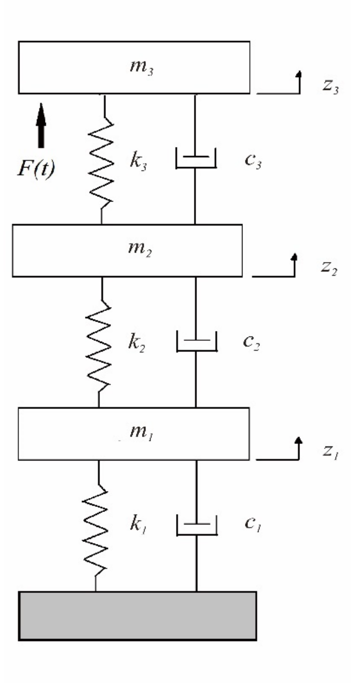

Fig. 3. System with Three Degree of Freedom

$$m_1\ddot{z}_2 + c_1\dot{z}_1 + k_1 - c_2(\ddot{z}_2 - \dot{z}_1) - k_2(z_2 - z_1) = 0 \tag{14}$$

$$m_1\ddot{z}_2 + c_2(\dot{z}_2 - \dot{z}_1) + k_2(z_2 - z_1) - c_3(\dot{z}_3 - \dot{z}_2) - k_3(z_3 - z_2) = 0 \tag{15}$$

The above equation can be transformed into a matrix as below:

$$\begin{bmatrix} m_1 \\ m_2 \\ m_3 \end{bmatrix}\begin{Bmatrix} \ddot{z}_1 \\ \ddot{z}_2 \\ \ddot{z}_3 \end{Bmatrix} + \begin{bmatrix} (k_1 + k_2) & -k_2 & 0 \\ -k_2 & (k_2 + k_3) & -k_3 \\ 0 & +k_3 & -k_3 \end{bmatrix}\begin{Bmatrix} z_1 \\ z_2 \\ z_3 \end{Bmatrix} +$$
$$\begin{bmatrix} (c_1 + c_2) & -c_2 & 0 \\ -c_2 & (c_2 + c_3) & -c_3 \\ 0 & +c_3 & -c_3 \end{bmatrix}\begin{Bmatrix} \dot{z}_1 \\ \dot{z}_2 \\ \dot{z}_3 \end{Bmatrix} = 0 \quad (16)$$

Then it can be simplified as follows:

$$[m]\{\ddot{z}\} + [k]\{z\} + [c]\{\dot{z}\} = 0 \tag{17}$$

2.4 PID Control

PID control is a feedback control method used to regulate a dynamic system. The abbreviation "PID" comes from the three key components in this method namely Proportional (P), Integral (I), and Derivative (D). The purpose of PID control is to adjust the output variable of a system to approach the desired or reference value as effectively as possible, taking into account the difference between the desired and actual values and changes in time [12]. The components of PID control are as follows:

- Proportional Control: serves to accelerate the response
- Integral Control: serves to eliminate steady error
- Derivative Control: serves to improve as well as accelerate the transient response[13].

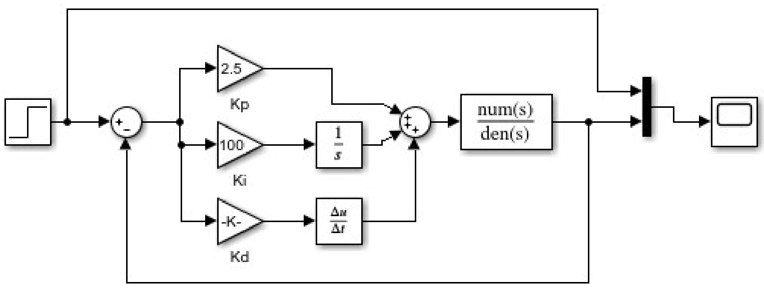

Fig. 4. PID control simulation example used in the experiments

Before using PID control, tuning is needed first in order to get an initial reference which is then carried out further tuning so that it can maximize control.

In this study, the first tuning to get an initial reference is using the Ziegler-Nichols method then after that proceed with fine tuning.

2.5 PID Tuning with Ziegler-Nichols closed loop method

John Ziegler and Nathaniel Nichols developed the Ziegler-Nichols open- loop tuning method in 1942, and it remains a popular technique for tuning controllers that use proportional, integral, and derivative actions in the industrial sector [14].

In the Ziegler-Nihcols tuning method, there is a procedure to perform this tuning so as to obtain the values needed when performing advanced tuning. The process is as follows:

- Select proportional control K_p alone.
- ncrease the value of the proportional gain k_p until the point of instability is reached (sustained oscillations), the critical value of gain (K_{cr}), is reached.
- Measure the period of oscillation to obtain the critical time constant (P_{cr}).

- Once the values for K_{cr} and P_{cr} are obtained, the PID parameters can be calculated, according to the design specifications as shown in table[15].

Controller Type	K_p	T_i	T_d
P	$0.5K_{cr}$		
PI	$0.45K_{cr}$	$1/1.2P_{cr}$	
PID	$0.6K_{cr}$	$0.5P_{cr}$	$0.125P_{cr}$

After PID tuning using the Ziegler-Nichols method, fine tuning is then performed to optimize the PID control output of the system.

Fine tuning is done to optimize the PID control output of a system by changing the required values obtained from the previous tuning.[17] As in this research the first tuning uses the Zigler-Nichols method.

3 Research Method

3.1 System Parameters

PARAMETERS	SYMBOL	VALUE
Operator mass	m_1	70 kg
Spring mass 1	m_2	10 kg
Spring mass 2	m_3	20kg
Spring coefficient 1	k_1	150 N/m
Spring coefficient 2	k_2	150 N/m
Spring coefficient 3	k_3	150 N/m
Damping coefficient	c_1	25 Ns/m
Damping coefficient	c_2	25 Ns/m
Damping coefficient	c_3	25 Ns/m

3.2 Modeling System

SDOF System Model

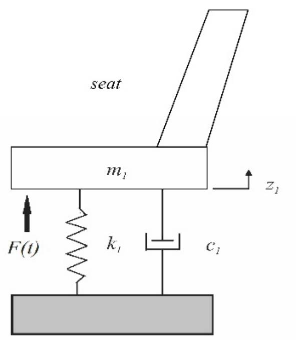

Fig. 5. SDOF Seat Model

The SDOF system model can be modeled as follows.

$$F(t) = m_1\ddot{x}(t) + c_1\dot{x}(t) + kx(t) \quad (18)$$

Then transformed into Laplace form into.

$$ms^2X(s) + csX(s) + kX(s) = \frac{1}{s} \quad (19)$$

Then we can obtain the transfer function.

$$H(s) = \frac{H(s)}{U(s)} = \frac{1}{ms^2+cs+k} \quad (20)$$

Then we substitute the parameters as follows.

$$\frac{H(s)}{U(s)} = \frac{1}{70s^2+25s+150} \quad (21)$$

2-DOF Model

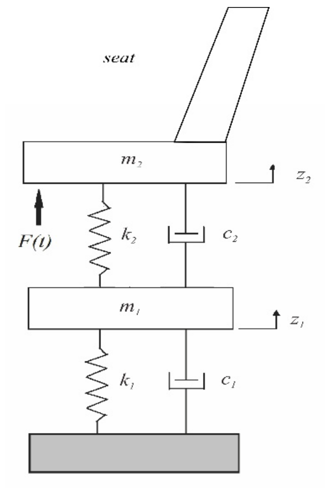

Fig. 6. 2-DOF Seat Model

The 2-DOF system model can be modeled with the approach of substituting the two parameters that have been given. Then the model can be expressed as follows.

$$m_1\ddot{x}(t) + c_1\dot{x}(t) + kx(t) = -c_1\dot{u} - k_1u \quad (22)$$

$$m_2\ddot{x}(t) + c_2\dot{x}(t) + kx(t) = c_2\dot{u} - k_2u \quad (23)$$

Then transform it into Laplace form.

$$m_1 s^2 X_1(s) + c_1 s X_1(s) + k_1 X_1(s) = -c_1 s U(s) - k_1 U(s) \tag{24}$$

$$m_2 s^2 X_2(s) + c_2 s X_2(s) + k_2 X_2(s) = c_2 s U(s) - k_2 U(s) \tag{25}$$

Then we can obtain the transfer function by converting it into a matrix first as follows.

$$\begin{bmatrix} m_1 s^2 + c_1 s + k_1 & 0 \\ 0 & m_2 s^2 + c_2 s + k_2 \end{bmatrix} \begin{bmatrix} X_1(s) \\ X_2(s) \end{bmatrix} = \begin{bmatrix} -c_1 s - k_1 \\ c_2 s - k_2 \end{bmatrix} U(s) \tag{26}$$

Then the transfer function can be written as follows.

$$H(s) = \frac{\begin{bmatrix} X_1(s) \\ X_2(s) \end{bmatrix}}{U(s)} = \begin{bmatrix} \frac{-c_1 s - k_1}{m_1 s^2 + c_1 s + k_1} \\ \frac{c_2 s - k_2}{m_2 s^2 + c_2 s + k_2} \end{bmatrix} \tag{27}$$

Then we substitute the parameters as follows.

$$H(s) = \frac{\begin{bmatrix} X_1(s) \\ X_2(s) \end{bmatrix}}{U(s)} = \begin{bmatrix} \frac{-c_1 s - k_1}{70 s^2 + 25 s + 150} \\ \frac{c_2 s - k_2}{10 s^2 + 25 s + 150} \end{bmatrix} \tag{28}$$

3-DOF Seat Model

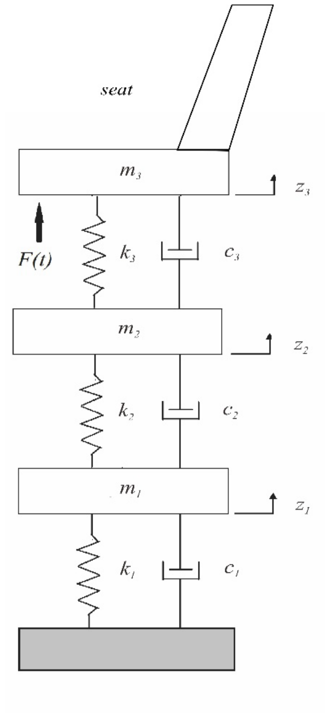

Fig. 7. 3-DOF Seat Model

The 3-DOF system model can be modeled using the same approach as before with the incorporation of parameters. Then the model can be written as follows.

$$m_1\ddot{x}(t) + c_1\dot{x}(t) + kx(t) = -c_1\dot{u} - k_1u \quad (29)$$

$$m_2\ddot{x}(t) + c_2\dot{x}(t) + k_2x(t) = c_1\dot{x}_1 - c_2\dot{u} + (k_1 - k_2)x_1 - k_2x_2 \quad (30)$$

$$m_2s^2X_2(s) + c_2sX_2(s) + k_2X_2(s) = c_1sX_1(s) - c_2sU(s) + (k_1 - k_2)X_1(s) - k_2X_2(s) \quad (31)$$

$$m_3s^2X_3(s) + c_3sX_3(s) + k_3X_3(s) = c_2sX_2(s) - c_3sU(s) + (k_2 - k_3)X_2(s) - k_3X_3(s) \quad (32)$$

Then we can obtain the transfer function by converting it into matrix form first.

$$\begin{bmatrix} m_1s^2 + c_1s + k_1 & 0 & 0 \\ 0 & m_2s^2 + c_2s + k_2 & 0 \\ 0 & 0 & m_3s^2 + c_3s + k_3 \end{bmatrix} \begin{bmatrix} X_1(s) \\ X_2(s) \\ X_3(s) \end{bmatrix} = \begin{bmatrix} -c_1s - k_1 \\ c_1s - c_2s + (k_1 - k_2) \\ c_2s - c_3s + (k_2 - k_3) \end{bmatrix} \quad (33)$$

Then the transfer function becomes.

$$H(s) = \begin{bmatrix} \frac{-(c_1 s + k_1)}{m_1 s^2 + c_1 s + k_1} \\ \frac{c_1 s - c_2 s + (k_1 - k_2)}{m_2 s^2 + c_2 s + k_2} \\ \frac{c_2 s - c_3 s + (k_2 - k_3)}{m_3 s^2 + c_3 s + k_3} \end{bmatrix} \tag{34}$$

Then we substitute the parameters as follows.

$$H(s) = \begin{bmatrix} \frac{-(c_1 s + k_1)}{70 s^2 + 25 s + 150} \\ \frac{c_1 s - c_2 s + (k_1 - k_2)}{10 s^2 + 25 s + 150} \\ \frac{c_2 s - c_3 s + (k_2 - k_3)}{20 s^2 + 25 s + 150} \end{bmatrix} \tag{35}$$

3.3 PID Tuning

PID Tuning for SDOF

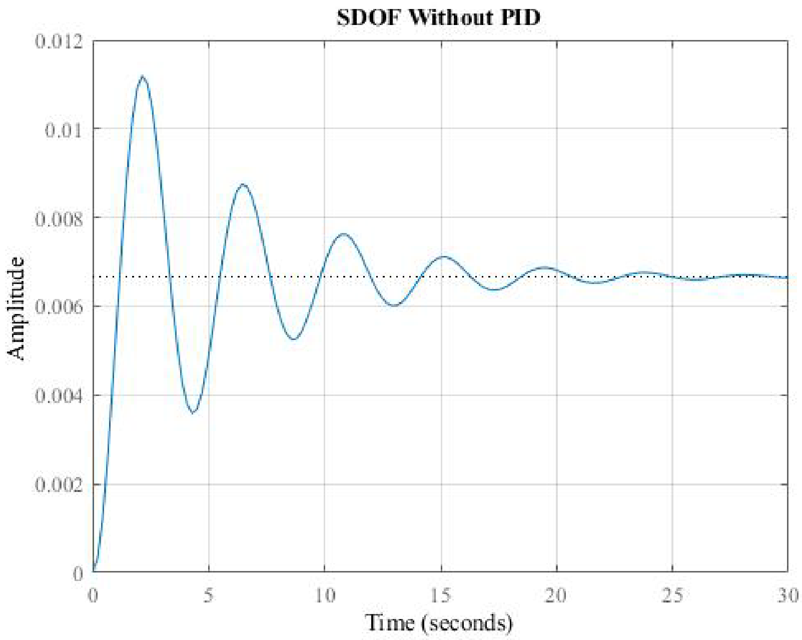

Fig. 9. SDOF Output without PID control

In the SDOF model system is stable, but the response time given tends to be long and the amplitude is very small, therefore this system is given an automatic PID with the syntax in the MATLAB software then the parameters obtained are:

Table 1. PID Parameters

PARAMETERS	VALUE
k_p	1.26
k_i	639
k_d	432

After obtaining PID parameters for this system, the system can be given PID control according to the previously obtained parameters.

PID Tuning for 2-DOF

In the 2-DOF model system, it is found that after being given a step input, the system gives an output that looks stable, but when given several parameters for PID tuning, the system gives an unstable output.

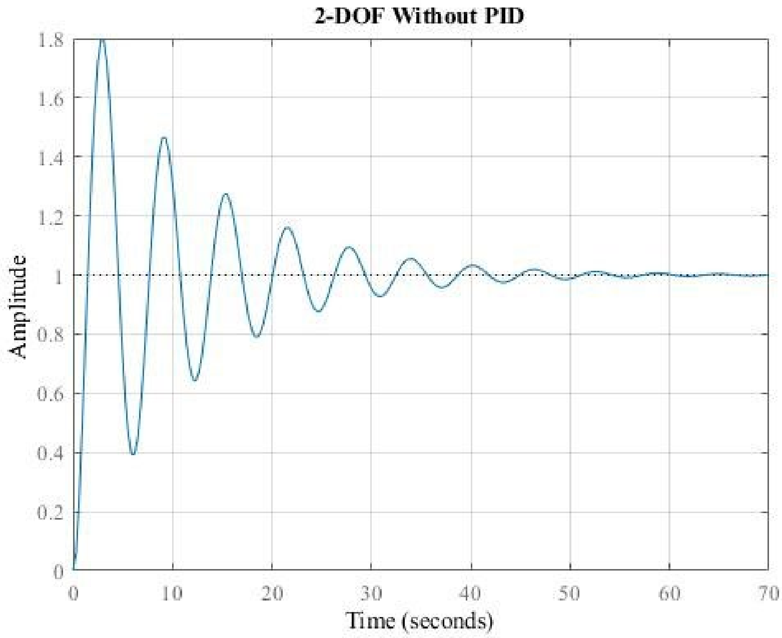

Fig. 10. 2-DOF without PID Control

Therefore, it is necessary to do tuning for this system in order to be given optimal PID control. By using the Ziegler-Nichols method, the required parameters can be obtained as follows.

Table 2. PID Parameters

Parameter	Value
K_{cr}	13.5
P_{cr}	1.153 s

After obtaining the PID parameters, the system can be given PID control to stabilize according to the parameters that have been obtained so that further tuning can be done.

PID Tuning for 3-DOF

In the 3-DOF model system when this system is given a step input the system gives a response that is quite long but with a very high amplitude, this indicates that this system is not stable according to the experiments that have been carried out.

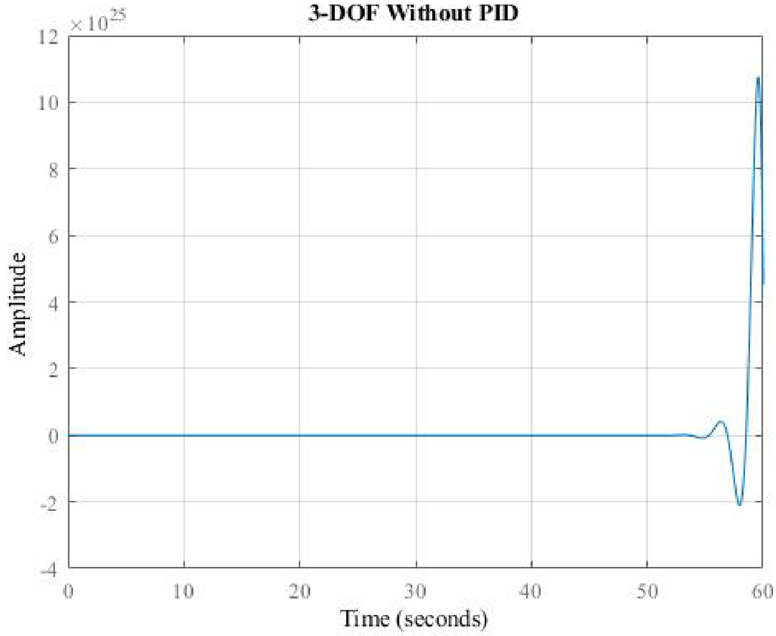

Fig. 11. 3-DOF Output without PID Control

Before PID control is given to this system, tuning is required first as was done in the previous system using the Ziegler-Nichols method. After tuning, the required parameters were successfully obtained in this experiment as follows.

Table 3. PID Parameters

Parameter	Value
K_{cr}	1158
P_{cr}	0.193 s

After obtaining the required parameters, the system can be given PID control to stabilize the system with advanced tuning.

4 Result

After all systems have been given PID control, all systems now provide the output as desired, namely with a faster response and stabilized at the desired set point of 1 value.

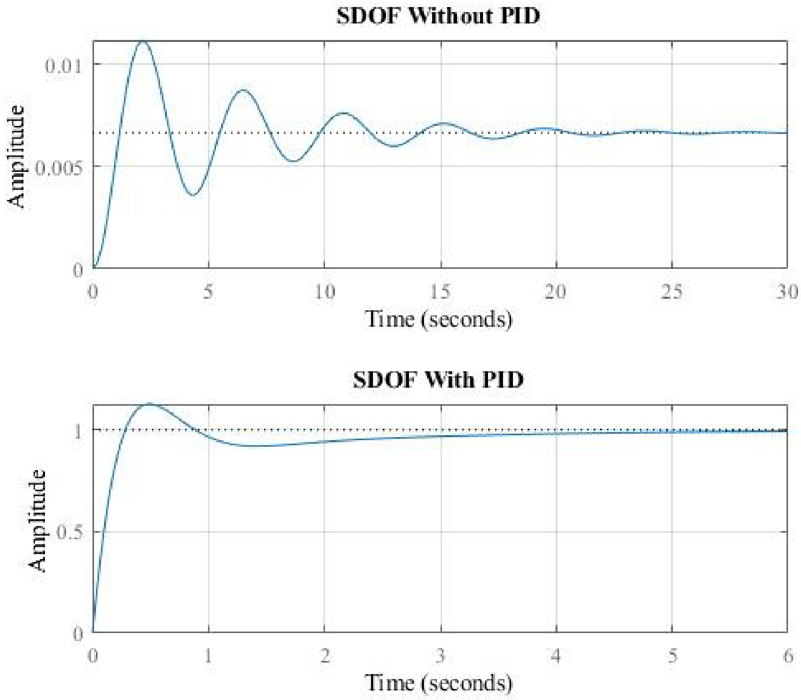

Fig. 12. SDOF Comparison with and without PID

The SDOF system that has been given PID control can produce the desired output. Before being given PID this system tends to have a fast time response but oscillations occur long enough that it will make the operator uncomfortable when using it and the amplitude tends to be very small. After being given PID control with PID parameters that have been tuning the system not only has a fast time response, but also dampens the oscillations that occur when given input and the system is able to reach the desired set point quickly.

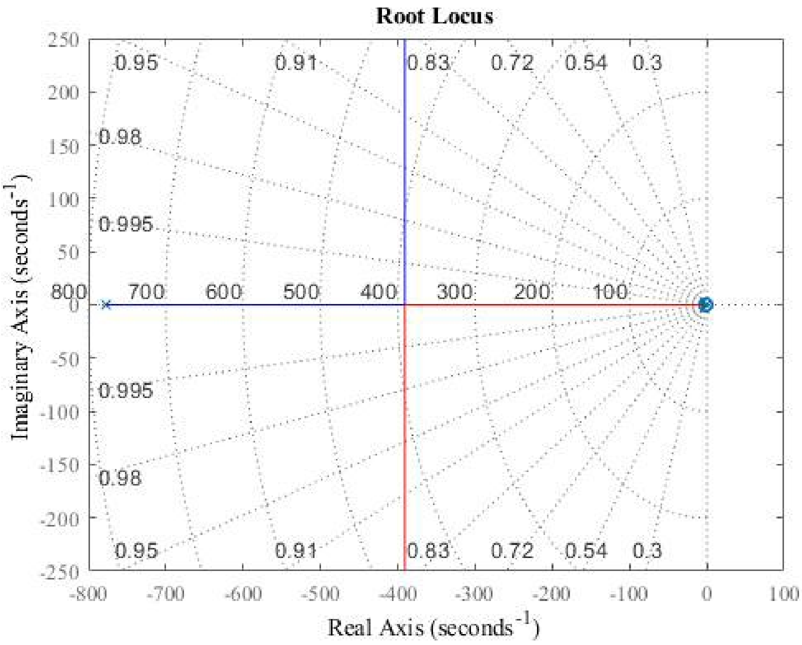

Fig. 13. SDOF Root locus after PID control

In the root locus graph, it can be concluded that the value produced by the system does not point to the right of the real axis 0, indicating that the system has become a stable system both after and before being given PID control for this system.

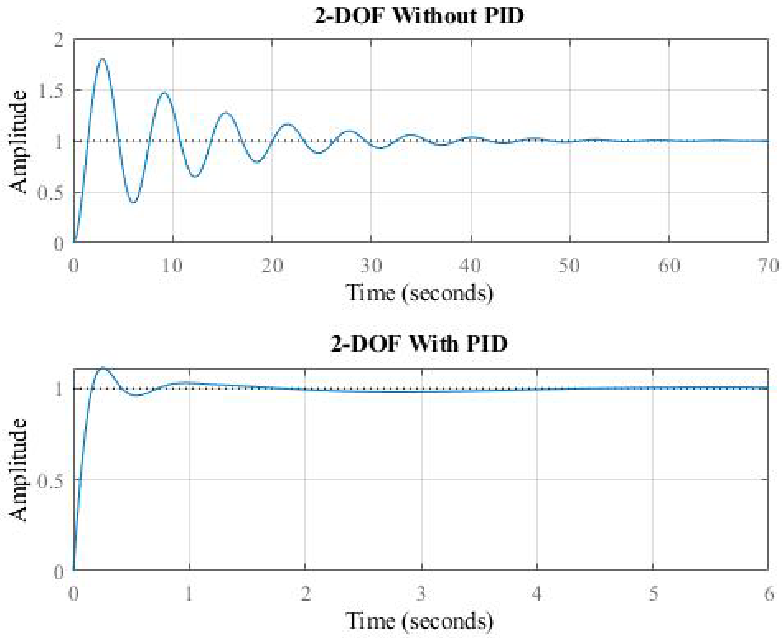

Fig. 14. 2-DOF comparison with and without PID control

In the 2-DOF system when the system has not been given PID control, this system produces a fairly stable output, but the case is similar to the SDOF system where the system looks stable but when tested by changing some parameters on this system then this system will turn unstable, then oscillations that occur after being given input with a long enough duration which is not good for the operator when using it.After being given PID control on this system, the system becomes more stable where the response time is faster, reduces overshoot and oscillations that occur can be dampened.

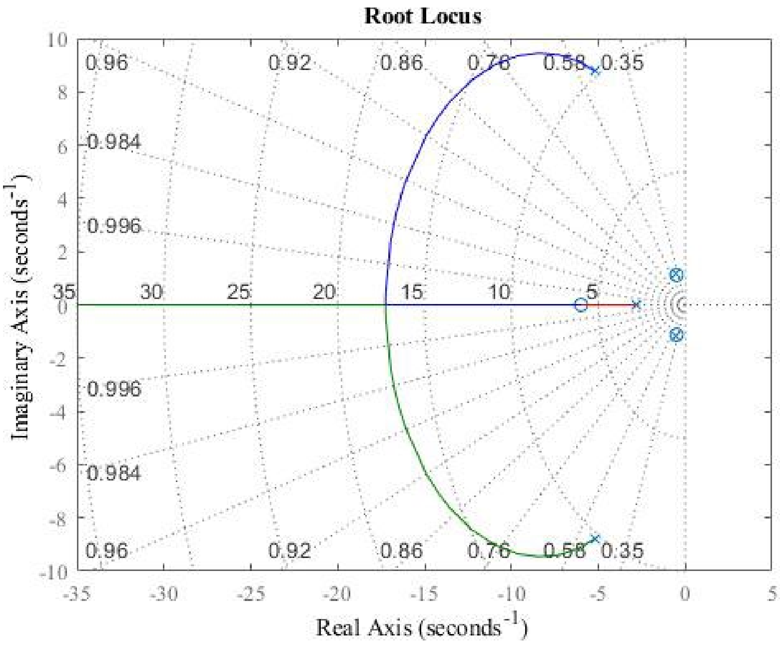

Fig. 15. 2-DOF Root locus after PID control

In the root locus graph, it can be concluded that this system has become a stable system after being given PID control characterized by the resulting value not going to the right of the real axis 0 so that the system is safe to use in real life.

Then in the 3-DOF system, this system when given input produces a very unstable response where the time response is very slow then high overshoot and oscillations that get bigger over time.

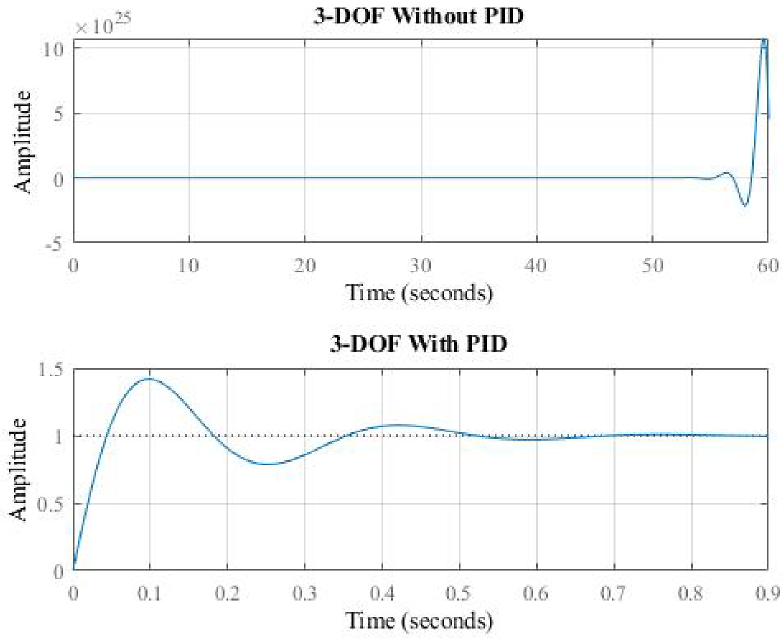

Fig. 16. 3-DOF comparison with and without PID

After being given PID control this system is able to produce better output than before significantly where the time response becomes very fast compared to before then the overshoot is reduced very drastically and the oscillations are damped making this system stable.

After being given PID control this system is able to produce better output than before significantly where the time response becomes very fast compared to before then the overshoot is reduced very drastically and the oscillations are damped making this system stable.

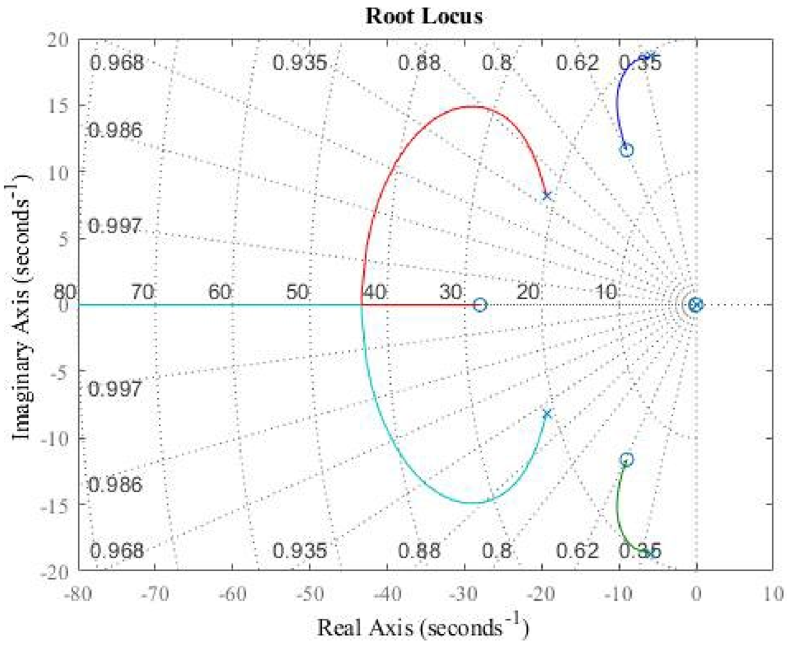

Fig. 17. 3-DOF Root locus after PID control

In the root locus graph, it can be concluded that this system is stable after being given PID control which can be characterized by the resulting value not going to the right of the real axis 0 so that this system is safe for use by operators.

5 Conclusion

Based on the research that has been done on 3 mass-spring-damper system models, it can be concluded that a system that looks stable is not necessarily fully stable, it would be nice to be modified or adjusted parameters or even given control so that the system can produce the output as we want. In addition, a stable system has an advantage for its users because it can provide comfort and safety when used with a long enough span of time, moreover when the system can be controlled according to what we specify then we will feel more comfortable when using it so that it can create comfort when using it.

References

1. V. K. Tewari and N. Prasad, "Three-DOF modelling of tractor seat-operator system," *J Terramech*, vol. 36, no. 4, pp. 207–219, 1999, doi: 10.1016/S0022-4898(99)00008-7.

2. P. Kumbhar, N. Li, P. Xu, and J. Yang, "Optimal seat dynamic parameters determination for minimizing virtual driver's fatigue," in *SAE Technical Papers*, SAE International, 2014. doi: 10.4271/2014-01-0877.
3. C. W. Suggs, C. F. Abrams, and L. F. Stikeleather, "Application of a Damped Spring-Mass Human Vibration Simulator in Vibration Testing of Vehicle Seats," *Ergonomics*, vol. 12, no. 1, pp. 79–90, 1969, doi: 10.1080/00140136908931030.
4. E. Zheng, Y. Fan, R. Zhu, Y. Zhu, and J. Xian, "Prediction of the vibration characteristics for wheeled tractor with suspended driver seat including air spring and MR damper," *Journal of Mechanical Science and Technology*, vol. 30, no. 9, pp. 4143–4156, Sep. 2016, doi: 10.1007/s12206-016-0826-x.
5. M. Al-Ashmori and X. Wang, "A systematic literature review of various control techniques for active seat suspension systems," *Applied Sciences (Switzerland)*, vol. 10, no. 3. MDPI AG, Feb. 01, 2020. doi: 10.3390/app10031148.
6. R. Sharma, P. Gaur, and A. P. Mittal, "Performance analysis of two-degree of freedom fractional order PID controllers for robotic manipulator with payload," *ISA Trans*, vol. 58, pp. 279–291, Sep. 2015, doi: 10.1016/j.isatra.2015.03.013.
7. H. Wu, W. Su, and Z. Liu, "PID controllers: Design and tuning methods," in *2014 9th IEEE Conference on Industrial Electronics and Applications*, 2014, pp. 808–813. doi: 10.1109/ICIEA.2014.6931273.
8. I. I. Gorial, "Dynamical analysis and controllers performance evaluation for single degree-of-freedom system," *International Journal on Smart Sensing and Intelligent Systems*, vol. 13, no. 1, pp. 1–12, 2020, doi: 10.21307/ijssis-2020-018.
9. D. Wang, T. Ji, Q. Zhang, and E. Duarte, "Presence of Resonance Frequencies in a Heavily Damped Two-Degree-of-Freedom System," *J Eng Mech*, vol. 140, no. 2, pp. 406–417, Feb. 2014, doi: 10.1061/(asce)em.1943-7889.0000668.
10. P. Frankovský, D. Hroncová, I. Delyová, and I. Virgala, "Modeling of Dynamic Systems in Simulation Environment MATLAB / Simulink – SimMechanics," no. January, 2013, doi: 10.12691/ajme-1-7-26.
11. R. P. Borase, D. K. Maghade, S. Y. Sondkar, and S. N. Pawar, "A review of PID control, tuning methods and applications," *International Journal of Dynamics and Control*, vol. 9, no. 2. Springer Science and Business Media Deutschland GmbH, pp. 818–827, Jun. 01, 2021. doi: 10.1007/s40435-020-00665-4.
12. I. Boiko, "Non-parametric tuning of PID controllers," in *Advances in Industrial Control*, no. 9781447144649, Springer International Publishing, 2013, pp. 9–23. doi: 10.1007/978-1-4471-4465-6_2.
13. E. Joseph and O. O. O, "Cohen-coon PID Tuning Method; A Better Option to Ziegler Nichols-PID Tuning Method," *International Journal of Recent Engineering Research and Development (IJRERD) www.ijrerd.com* ||, vol. 02, no. 11, pp. 141–145, 2017, [Online]. Available: www.ijrerd.com
14. V. Kumar and A. Patra, "Analysis of Power Transformer using fuzzy expert and neural network system APPLICATION OF ZIEGLER-NICHOLS METHOD FOR TUNING OF PID CONTROLLER." [Online]. Available: https://www.researchgate.net/publication/327816155
15. F. Hassan, A. C. Zolotas, and T. Smith, "Optimized Ziegler-Nichols based PID control design for tilt suspensions."
16. B. Verma and P. K. Padhy, "Robust Fine Tuning of Optimal PID Controller with Guaranteed Robustness," *IEEE Transactions on Industrial Electronics*, vol. 67, no. 6, pp. 4911–4920, Jun. 2020, doi: 10.1109/TIE.2019.2924603.