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Simulink and Simelectronics based Position Control of a Coupled Mass-Spring Damper Mechanical System

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ABSTRACT

This paper presents the use of Simelectronics Program for modeling and control of a two degrees-of freedom coupled mass-spring-damper mechanical system.The aims of this paper are to establish a mathematical model that represents the dynamic behaviour of a coupled mass-spring damper system and effectively control the mass position using both Simulink and Simelectronics.The mathematical model is derived based on the augmented Lagrange equation and to simulate the dynamic accurately a PD controller is implemented to compensate for the oscillation sustained by the system as a result of the complex conjugate pair poles near to the imaginary axis.The input force has been subjected to an obstacle to mimic actual challenges and to validate the mathematical model a Simulink and Simelectronics models were developed, consequently, the results of the models were compared. According to the result analysis, the controller tracked the position errors and stabilized the positions to zero within a settling time of 6.5sec and significantly reduced the overshoot by 99.5% and 99. 7% in Simulink and Simelectronics respectively. Furthermore, it is found that Simelectronics model proved to be capable having advantages of simplicity, less time-intense and requires no mathematical model over the Simulink approach.

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Nomenclatures

m_1, m_2	Translating inertial elements (masses)
k_1, k_2	Spring stiffness coefficients
b_1, b_2	Viscous damping coefficients
L	Lagrange's function
T	Kinetic energy
U	Potential energy
D	Rayleigh's dissipative function
x_1, x_2	Generalized coordinates displacement
Q_i	Generalized forces
$\dot{x}_1, \dot{x}_2$	Velocities of the masses m_1 and m_2
$\ddot{x}_1, \ddot{x}_2$	Accelerations of the masses m_1 and m_2

Abbreviations

DOF	Degree of freedom
EOM	Equations of Motion
MSD	Mass Spring Damper

1. INTRODUCTION

In a mechanical systems, accurate control of motion (such as position, velocity) is a fundamental concern to control engineers. Thus, the position control tries to adjust the dynamic of the mass while achieving the constraints imposed by the force positioning the mass. Consequently, the control problems involve finding suitable mathematical models that describe the dynamic behaviour of the physical mass spring dampers (MSD) model to permit suitable controller design and allow corresponding control strategies to realize the expected system response and performance. Mass-spring-damper systems (MSD) are widely used in robot manipulator control [1], [2],vehicle suspension systems for shock absorption in automobiles [3]-[7], Mechatronic application especially in piezoelectric for vibration energy harvester [8]-[10] and motion control application [11]-[12]. Recently, MSD systems are in increasing demand for hybrid vehicle suspension to increase passenger ride comfort and vehicle stability over cracks and uneven pavement.

Tarik *et al* [1] developed a mass spring damper model with MATLAB graphical user interface which permits flexibility in the choice of control strategies on the model through model parameters adjustment. Sivák and Hroncová [13] presented equations of motion (EOM) of a mechanical system with two degrees of freedom in MATLAB/Simulink using state space and transfer function. The conclusion of their work is that Newton's law and Lagrange's equation resulted in the same solution. Furthermore, in [14], the author presented Simmechanics and Simulink models for car automobile suspension and implemented proportional-integral-derivative (PID) control strategy to minimize body acceleration. In [15], [16] , the authors established mass spring damper models based on Newton's law of motion to derive a state-space model for numerical computation with the aids of MATLAB and Simulink. An author in [17] examined the effect of the position of the damper in systems with multi-degree of freedom. Their research works concluded that displacement of an oscillator that affected by the force show more energy absorption for low displacement. In [18]-[20], the authors presented mathematical modeling of a mass spring damper system in MATLAB and Simulink. The author in [21], presented control of coupled mass spring damper system using polynomial structures approach. Malas and co-worker [22] presented a novel control strategy for inducing sif-sustained oscillation of a single degree-of-freedom MDS mechanical systems. The researchers concluded in their work that the proposed control model is capable of generating stable self-excited oscillation at the natural frequency regardless of the value of the control gain.

The objectives of this paper are to establish a mathematical model that represent the dynamic behaviour of a coupled mass spring damper systems and effectively control the mass position using both Simulink and Simelectronics as simulation tools

2. RESEARCH METHOD

We derived the equations of motion (EOM) of a coupled mass spring damper systems using second-order, ordinary differential equations and to simulate dynamic accurately [23] the Lagrange's equation was adopted. The motivation for chosen Lagrange's equation over Newton's law or D'Alembert principle is that it allows significant simplification of the geometry of the system motion for solving large complex systems and also eliminates explicit rewriting all forces acting on the body. The mathematical model is formulated based on energy property of Lagrange approach and the control strategy and Simulink simulation are expanded on the derived mathematical model. However, the langrage's equation does not improvise for dissipative (damping) force in the mechanical system, hence, Rayleigh's dissipation function is introduced into Lagrange's equation to account for dissipative force in the model and we refer to this as augmented Lagrange's equation. In order to describe the physical motion of a coupled mass spring damper systems, we need to choose a set of variables or coordinates which are often referred to as generalized coordinates. Thus, the displacement of the masses is chosen as the generalized coordinates.

2.1. Mathematical modeling

The dynamic of the mass spring damper systems with two degrees of freedom (DOF) movement is explicitly derived based on Lagrange's equation to expound the problems involved in dynamic modeling. Figure 1 depicts a coupled mass spring damper systems, where two masses m_1 and m_2 are linked to a parallel spring-damper configuration with spring stiffness coefficients k_1 and k_2 and viscous damping coefficients b_1 and b_2 for mass m_1 and m_2 respectively. The force, $F(t)$, acts on mass 1 and the energy is transmitted to mass 2 via the springs and some part of energy is absorbed by the dampers

The systems performs linear motion along the axes of the springs and dampers. For simplicity, we assume the following assumptions in the model:

a. The springs are linear
b. The dampers are linear and
c. The weight of the springs are assumed to be negligible.

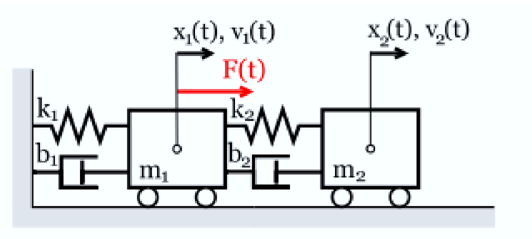

Figure 1. A coupled mass spring damper systems adapted from [8]

The equations of motion (EOM) for a mechanical system with 2-DOF can be derived by an augmented.

Lagrange's equation in the form

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right)-\left(\frac{\partial L}{\partial q_i}\right)+\left(\frac{\partial D}{\partial \dot{q}_i}\right)=Q_i \tag{1}$$

$$q_i=\left(x_1,x_2\right) \tag{2}$$

Where, q_i is the generalized coordinates to describe the displacement of the masses.

First we calculate the kinetic energy, T, and the (augmented) potential energy, U, of the system, the kinetic energy of a coupled mass spring damper systems as function mass velocity is expressed as:

$$T=\frac{1}{2}m_1\,\dot{x}_1^2+\frac{1}{2}m_2\,\dot{x}_2^2 \tag{3}$$

The augmented potential energy is expressed as:

$$U=\frac{1}{2}k_1x_1^2+\frac{1}{2}k_2\left(x_1-x_2\right)^2 \tag{4}$$

$$U=\frac{1}{2}k_1x_1^2+\frac{1}{2}k_2x_1^2-k_2x_1x_2+\frac{1}{2}k_2x_2^2 \tag{5}$$

Rayleigh dissipative function account for damping force in the mechanical system and it is expressed as:

$$D=\frac{1}{2}b_1\,\dot{x}_1^2+\frac{1}{2}b_2\left(\dot{x}_1-\dot{x}_2\right)^2 \tag{6}$$

$$D=\frac{1}{2}b_1\,\dot{x}_1^2+\frac{1}{2}b_2\,\dot{x}_1^2-b_2\,\dot{x}_1\,\dot{x}_2+\frac{1}{2}b_2\,\dot{x}_1^2 \tag{7}$$

Generalized forces:

$$Q_1=F,Q_2=0 \tag{8}$$

The Lagrange formulation defines the behaviour of a dynamic systems in terms of work and energy stored in the system [24].

The augmented Lagrange function L is denoted as:

$$L = T - U \tag{9}$$

$$L = \frac{1}{2}\left(m_1 \dot{x}_1^2 + m_2 \dot{x}_2^2\right) - \left(\frac{1}{2}k_1 x_1^2 + \frac{1}{2}k_2 x_1^2 - k_2 x_1 x_2 + \frac{1}{2}k_2 x_2^2\right) \tag{10}$$

We evaluate the following derivatives based on equation Equation (7) and Equation (10):

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_1}\right) = m_1 \ddot{x}_1 \tag{11}$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_2}\right) = m_2 \ddot{x}_2 \tag{12}$$

$$\left(\frac{\partial L}{\partial x_1}\right) = -k_1 x_1 - k_2 x_1 + k_2 x_2 \tag{13}$$

$$\left(\frac{\partial L}{\partial x_2}\right) = -k_2 x_2 + k_2 x_1 \tag{14}$$

$$\left(\frac{\partial D}{\partial \dot{x}_1}\right) = b_1 \dot{x}_1 + b_2 \dot{x}_1 - b_2 \dot{x}_2 \tag{15}$$

$$\left(\frac{\partial D}{\partial \dot{x}_2}\right) = b_2 \dot{x}_1 + b_2 \dot{x}_2 \tag{16}$$

For generalized coordinate x_1 , the Lagrange's equation is:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_1}\right) - \left(\frac{\partial L}{\partial x_1}\right) + \left(\frac{\partial D}{\partial \dot{x}_1}\right) = Q_1 \tag{17}$$

After substitution of the derived derivatives in Equation (11), Equation (13), and Equation (15) in Equation (17), we obtain equation of motion for mass 1 in this form:

$$m_1 \ddot{x}_1 + (b_1 + b_2)\dot{x}_1 - b_2 \dot{x}_2 + (k_1 + k_2)x_1 - k_2 x_2 = F \tag{18}$$

Similarly, for the generalized coordinate x_2, the Lagrange's equation is:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_2}\right) - \left(\frac{\partial L}{\partial x_2}\right) + \left(\frac{\partial D}{\partial \dot{x}_2}\right) = Q_2 \tag{19}$$

In the same vein, substitution of the derived derivatives in Equation (12), Equation (14), and Equation (16) in Equation (19), we obtain equation of motion for mass 2 in this form

$$m_2 \ddot{x}_2 - b_2 \dot{x}_1 + b_2 \dot{x}_2 - k_2 x_1 + k_2 x_2 = 0 \tag{20}$$

The EOM in Equation (18) and Equation (20) are re-arrange to facilitate the implementation of the equations in Simulink as:

$$\ddot{x}_1 = \frac{-(b_1+b_2)}{m_1}\dot{x}_1 + \frac{b_2}{m_2}\dot{x}_2 - \frac{(k_1+k_2)}{m_1}x_1 + \frac{k_2}{m_1}x_2 + \frac{F}{m_1} \tag{21}$$

$$\ddot{x}_2 = \frac{b_2}{m_2}\dot{x}_1 - \frac{b_2}{m_2}\dot{x}_2 + \frac{k_2}{m_2}x_1 - \frac{k_2}{m_2} \tag{22}$$

The derived EOM is further written in terms of mass, damping, stiffness matrices and *F, x, x, x* are force, displacement, velocity and acceleration vectors respectively and they are presented as:

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} b_1+b_2 & -b_2 \\ -b_2 & b_2 \end{bmatrix} + \begin{bmatrix} k_1+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} = \begin{bmatrix} F \\ 0 \end{bmatrix} \tag{23}$$

In the light of Equation (23), the EOM obtained by using Lagrange's equation for linear systems revealed that inertial, stiffness matrices as well as damping matrix if modeled by Rayleigh's function are symmetric

2.2. State space representation of the model

A state space representation is a time domain approach of modeling muiltiple input multiple output system. However, complex system with many degree-of-freedom, description of such systems with differential equations are often time intense and burdensome. So, state space representation of the systems serves as an alternative approach to alleviate the challenges. Also, the state space representation of a system replaces the higher-order differential equations with a single first-order matrix differential equation that gives an expedient and concise way to model and analyze systems with multiple inputs and outputs. The state model is notably advantageous when applied to simulation. Hence, the state and output equations are given in [25] as:

$$\begin{aligned} \dot{x}(t) &= A..x(t) + B.u(t) \\ y(t) &= C.x(t) + D.u(t) \end{aligned} \tag{24}$$

Where, x, y, u, A, B, C, D are the state vector, output vector, input vector, system matrix, input matrix, output matrix and feedback matrix respectively.

Let define,

$$\frac{d^2x_1}{dt^2} = \frac{dv_1}{dt}, \frac{d^2x_2}{dt^2} = \frac{dv_2}{dt}$$

$$\frac{dx_1}{dt} = v_1, \frac{dv_2}{dt} = v_2$$

So that x_1, x_2, v_1, v_2 are selected as state variables and equation Equation (21) and Equation (22) are presented in state space equation in vector matrix form as:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{v}_1 \\ \dot{x}_2 \\ \dot{v}_2 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ -\frac{(k_1+k_2)}{m_2} & \frac{k_2}{m_1} & \frac{-(b_1+b_2)}{m_1} & \frac{b_2}{m_1} \\ 0 & 0 & 0 & 1 \\ \frac{k_2}{m_2} & \frac{-k_2}{m_2} & \frac{b_1}{m_1} & \frac{-b_2}{m_2} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ v_1 \\ v_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{m_2} \\ 0 \\ 0 \end{bmatrix} F(t) \tag{25}$$

And the output are the displacement x_1 and x_2 of masses m_1 and m_2 respectively. Hence,

$$y(t) = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{26}$$

3. MODELING OF A COUPLED MASS SPRING DAMPER SYSTEMS USING SIMELECTRONICS AND SIMULINK

A coupled mass spring damper system is modeled and simulated using SimElectronics toolbox in MATLAB software as shown in Figure 2. In MATLAB software, SimElectronics is a components libraries and special simulation features for modeling and stimulating physical system in the Simulink environment. The distinct feature of SimElectronics is the use of physical network approach to model electronic and mechatronic systems which mimic the physical system. Therefore, the use of physical connection permits a bidirectional flow of energy between components. The SimElectronics permit the use of Simulink library in modeling in which Simulink-PS-converter block converts the Simulink signal to physical system signal and PS-Simulink converter block does the reverse [26]. However, Simulink is a block diagram environment embedded in the MATLAB that allows modeling and simulation of multi-domain dynamics systems.

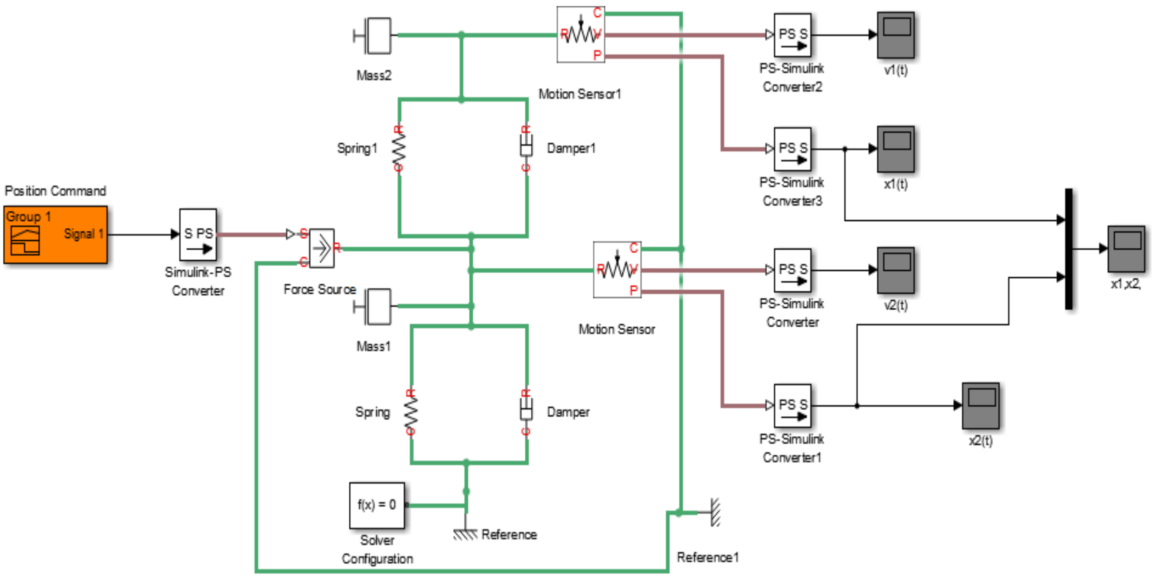

Figure 2. Simelectronics model for a coupled mass spring damper system

Furthermore, it provides the detailed function of each block that represents the mathematical model of the dynamic system and such model is often reduced to first order differential equation to simulate the dynamic accurately. It is dominant among other software in the engineering field because it enables rapid design, simulation, verification, testing and debugging of virtual prototypes of a model prior to real-time implementation. As a result of flexibility and efficient in use, it permits conversion of MATLAB code to other source codes such as C, C^{++} for real-time implementation especially in embedded systems and robotic.

Simulink model of a coupled mass spring damper system is prepared with the mathematical model presented through Equations (21) and (22). PD controller is implemented to track the masses position and Figure 3 represents a schematic of the Simulink model.

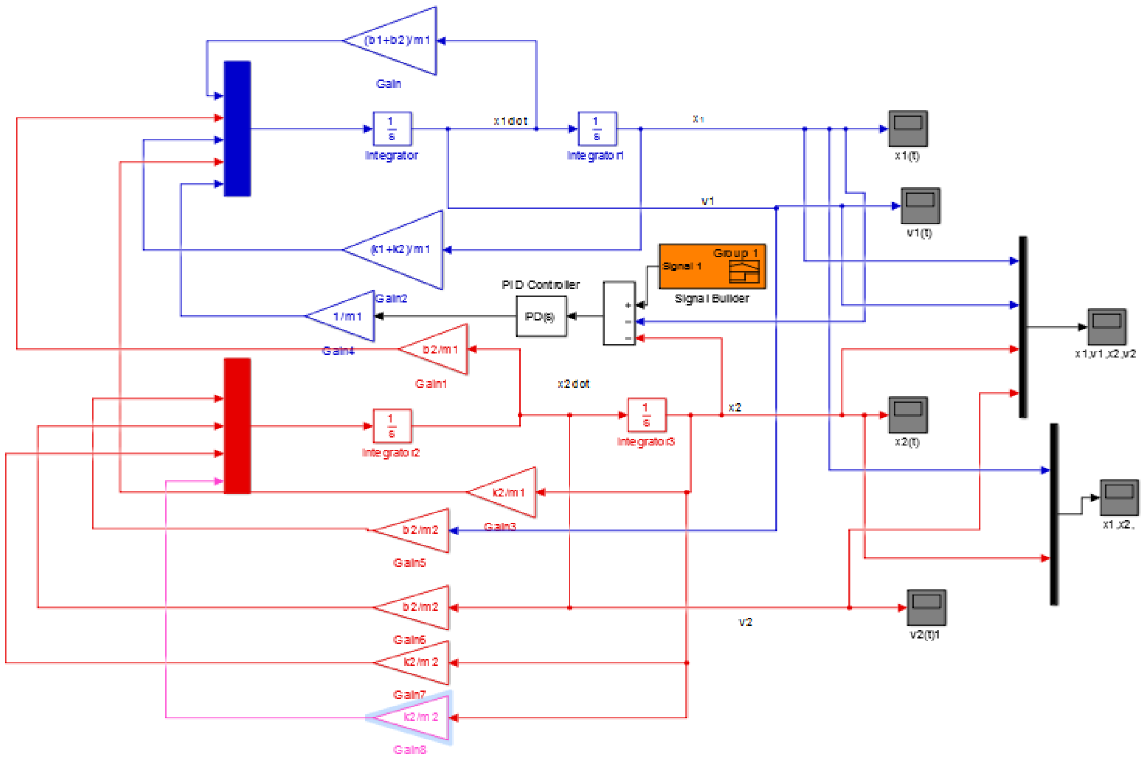

Figure 3. Simulink model for a coupled mass spring damper system

4. PD CONTROLLER IMPLEMENTATION

The control goal is to stabilize the position of the masses by minimizing the error and to achieve this stated objective a proportional-Derivative controller is implemented in such a way that the gain of the proportional controller**,** **k**, is high to produce a fast system and the derivative gain $\mathbf{k_d}$, is select in a manner to decrease the oscillation. The PD controller algorithm combines the P-action and D-action to adjust the system. It is a two-term controller that is coined from the PID controller by setting the integral action to zero. The term domain expression for PD controller is given as:

$$u_c(t)=k_p e(t)+k_D(t)\frac{de(t)}{dt} \tag{26}$$

In the Simulink and Simelectronics models, a PD controller is implemented from the PD block in the Simulink toolbox library and converted to a physical system with Simulink-PS converter for the Simelectronics. The tuning of control parameters is done using PID tuner and the best performance of the controller parameter values are selected.

5. RESULTS AND ANALYSIS

This paper proposed a novel approach of simulating the system dynamics of a couple mass spring damper system and compared the performance with the Simulink approach.

The Simulink model was established based on the derived mathematical model and the force that mimic an obstacle commonly experienced are build with a signal builder block. Parameters used in the model were m_1 = 250kg, m_2 = 300kg, k_1 = 80,000N/m, k_2 = 5000N/m, b_1 = 200N-s/m, b_2 = 15,000N-s/m and to validate the mathematical model a Simelectronics and Simulink models were compared and the simulation was then tested with and without a controller. The response of the mass positions are illustrated in Figure 4 and Figure 5 .The curve reveals that, without the implementation of the controller the positions of the masses overshoot sharply and then sustained an oscillation for about 7.6 secs before settling to zero at 10sec.

Figure 6 shows the performance of both models in the absence of control strategy, hence, the curve shows that the system has complex conjugate poles near the imaginary axis and this dominates the transient response of the mass-spring-damper system which eventually resulted in oscillation and this led to system instability.

However, it is challenging to stabilize or control the model with the proportional controller alone irrespective of the value of gain K chosen, hence, a combination of proportional and the derivative controller is implemented to compensate for fast response and steady error respectively. Figure 7 and Figure 8 show the responses of the displacements of the masses with the implementation of PD controller. According to the Figure 7, Figure 8, the PD-action damped the oscillation and stabilized the positions by compensating for the steady error. In Figure 9, both models were compared under the influence of PD controller and the controller compensates for the error in the position and stabilized the positions to zero within a settling time of 6.5sec and significantly reduced the overshoot by 99.5% and 99.7% in Simulink and Simelectronics respectively. It can also be noted from the graph that, the responses of both model show a similar result. Furthermore, the results revealed that, modeling of the mathematical equation in Simulink and assembling of the physical system in Simelectronics give a similar result. In much related research work [27], the dynamics are transformed into transfer function model using Laplace transformation to aids system analyze in the frequency domain as well as allows motion controller implementation. Thus, this is quite complicated and tedious for a complex system with several degree-of-freedom. However, the novel Simelectronics approach provides a quick and fast approach to model, integrate complex systems and control multi-domain systems by eliminating rigorous mathematical formulation of the system's dynamics. It is also suitable for fresh engineering students' who need to perform laboratory works on systems control and related engineering courses that require modelling and simulation.

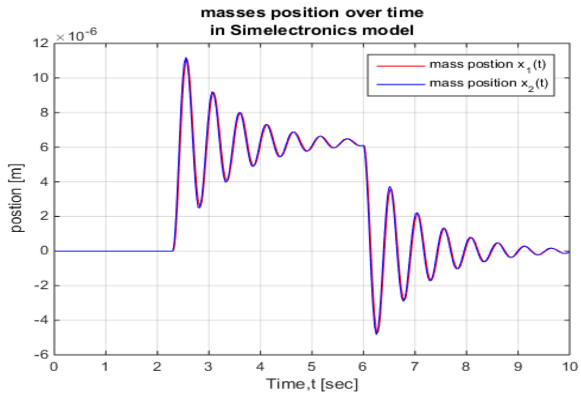

Figure 4. Displecements without controller in simelectronics model

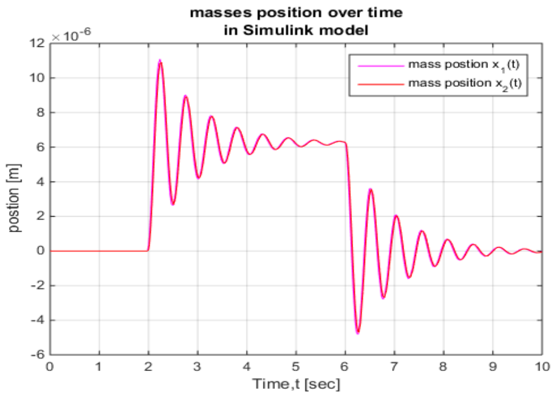

Figure 5. Postions without controller in simulink model

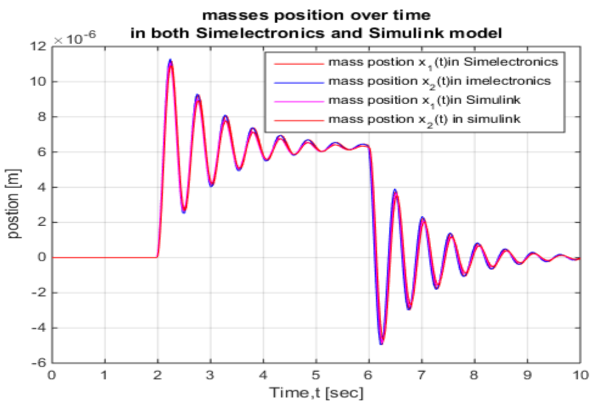

Figure 6. Postions in both simelectronics and simulink models without controller

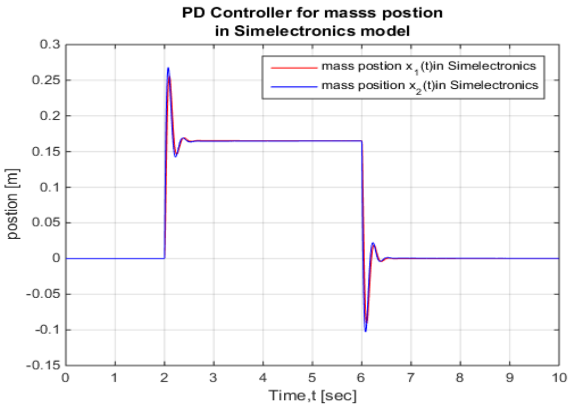

Figure 7. Positions response with PD controller in Simelectronics model

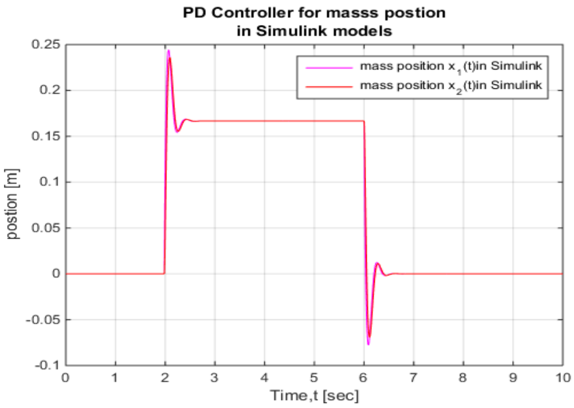

Figure 8. Positions response with PD controller in Simulink model

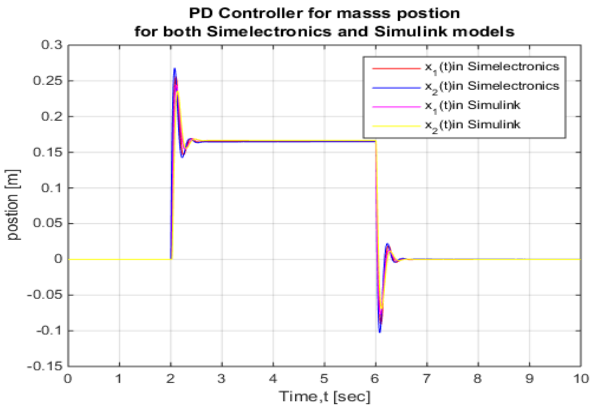

Figure 9. Position responses of both Simelectronics and Simulink model

6. CONCLUSION

In this paper, Simulink and Simelectronics model for position control of a coupled mass-spring-damper system were developed and presented. The mathematical model was formulated based on energy property of Lagrange approach and Rayleigh's dissipation function to account for dissipative force in the model, hence, the control strategy and Simulink simulation were expanded on the derived mathematical model while the physical system was set up in the Simelectronics to stimulate the dynamic. A proportional-Derivative (PD) controller was implemented for both the models. It can be concluded from the study that both models produced the same results, but the response time of the positions was slightly shorter in Simulink than that of the Simelectronics. It can also be deduced that it was simple, easy, less time-intense and requires no mathematical model to model in the Simelectronics than Simulink, although, Simulink provided the advantage of a sdetailed representation of the mathematical model. In our future work, we would like to (1) take into account the impact of nonlinearity of the spring and damper in our model; (2) to implement state feedback controller and LQR and compare the performance; (3) analyze the effect of the spring stiffness and damping coefficient parameters by varying these parameters.

The contribution of this work is significantly expedient in the field of mechatronics and Control systems and also provides a novel approach to simulation of the mechanical system in a concise and precise method.

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