Engineering Design of Machines with Rotating Members

Jiri Vondrich, Evzen Thöndel
Department of Electric Drives and Traction, Faculty of Electrical Engineering, Czech Technical University in Prague, Technicka 2, 166 27 Prague 6, Czech Republic
vondrich@ˇfel.cvut.cz, thondee@fel.cvut.cz

ABSTRACT

The aim of the paper is to acquaint the reader with the design of the incorporated absorber to the vibration machine with rotating members, which makes the reduction of the vibration of the machine to a minimum possible. Vibration in a machine has two effects. First, the very high peak accelerations can mean that the effective weight of the vibration machine increases several-fold, and this may cause its destruction. Secondly, people near machine feel these accelerations, which can be uncomfortable or even dangerous. A vibration absorber is used to protect the machine from steady-state harmonic disturbance. By attaching the absorber to the machine, which is modeled as a SDOF system, the new system becomes a two DOF system. The present article will discuss a method of vibration control LQR (Linear Quadratic Control) for structure using the vibration absorber without damping. Depending on the driving frequency of the original system, the absorber needs to be carefully tuned, that is, to choose adequate values of absorber mass and stiffness, so that motion of the original mass is a minimum.

Keywords: vibration, absorber, Matlab, LQR

1. INTRODUCTION

The tunable vibration absorber is advantageous primarily in that reduces the amplitude of vibrations in the machine by, for example unbalanced rotors, crank gear, clearances in bearings, oscillation of the moving driven parts, transient loading by diverging and coasting of driving motors, etc.

The equivalent model of this machine with the a reduced mass m_1, located on a cushion with coefficient of elasticity k_1 and the affiliate mass m_2 of the absorber, located on a controlled air pressure in an air-operated spring is possible illustrate as two mass system (Figure 1).

The tunable absorber connected with vibration machine is advantageous primarily in that it reduces the amplitude of the vibrations in the system by an oscillating force $F_{(t)}$ acting (alternative 1 and 2):

1) $F_{(t)}=me\omega^2 sin\omega t$, where m is the mass of the unbalanced rotor of machine, e is the eccentricity of unbalanced rotor and ω is the angular velocity of the rotor,
2) square wave course of acting force $F_{(t)}$.

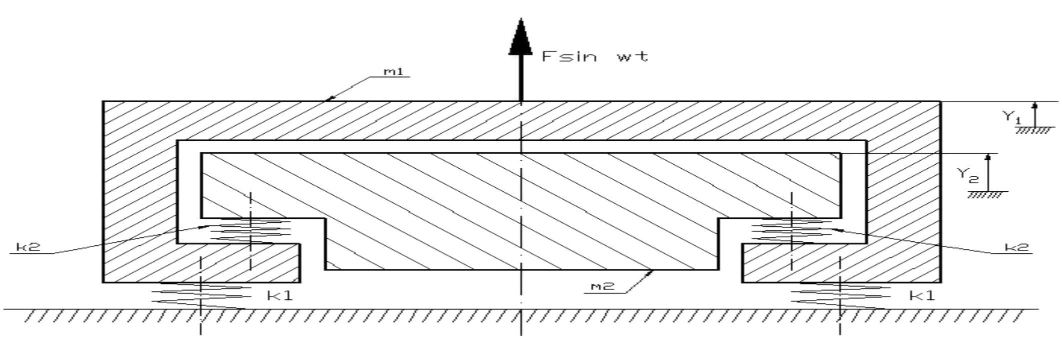

Figure 1: Model of the system with absorber as two mass systems

1. SOLUTION OF MACHINE MODEL WITH ABSORBER

The equations of motion of the model are

$$
\begin{aligned}
m_1 \ddot{y}_1 &= -k_1 y_1 + k_2 (y_2 - y_1) + F(t), \\
m_2 \ddot{y}_2 &= -k_2 (y_2 - y_1).
\end{aligned}
\tag{1}
$$

We want to stop the vibration of mass m_1. If the natural frequency of the absorber is the same as the excitation frequency $\omega_2=\omega$, the machine stop moving. The dimension mass m_2 of the absorber in the above mentioned experimental research is suggested to be

$$m_2 = 0.2 m_1, \tag{2}$$

where m_1 is the reduced mass of the machine.
The result of solution equations (1) is shown in the graph of the function in dimensionless variables of the amplitude characteristic dependence displacements y_1/y_2 and angular velocities ω/ω_2 of the absorber m_2 (Figure 3). The affiliate mass m_1 of the machine is not moveable in the case when the ratio $\omega/\omega_2=1$.
The next result is the phase characteristic. The change of the motion of the affiliate mass m_2 of absorber is 180^o in area when the ratio $\omega/\omega_2=1$ (Figure 4).
The Matlab solution equations (1) for the parameters of machine are in m-file on Figure 2.

```
The constants:%------------------------------------------------------//
w = 5;          %exciting frequency [rad/sec]
m1 = 350;       %dimension reduction mass m1 [kg] of the machine system
k1 = 10000;     %coefficient of elasticity k1 [N/m] on the cushion of the machine system
m2 = 70;        % affiliate masse m2 [kg] of absorber solved on equation (4)
k2 = w^2*m2;    %coefficient of elasticity k2 [N/m] spring of the absorber, solved on equation (2), (3)
b1 = 10;        %damping coefficient b1 [Ns/m] on the cushion of the machine system
b2 = 0.1;       %damping coefficient b1 [Ns/m] spring of the absorber
%x1 = dy1       velocity  [m/s] of the the machine system
%x2 = dy2       velocity  [m/s] of the absorber
%x3 = y1        position y1 [m] of the machine system
%x4 = y2        position y2 [m] of the absorber

%The system without damping
%--------------------------------------------------------//
A = [   -b1/m1  -k1/m1;
        1       0       ];
B = [   1/m1;
        0   ];
C = [   0   1   ];
D = [   0   ];
sys1 = ss(A,B,C,D);

%Bode characteristic
%--------------------------------------------------------//
W = 0.1*w:0.1:10*w;
[mag phase] = bode(sys, W);
[mag1 phase] = bode(sys1, W);
mag = squeeze(mag(1,1,:));
mag1 = squeeze(mag1(1,1,:));
```

```
phase = squeeze(phase(1,1,:));
figure(1);
plot(W./w,mag,'-k',W./w,mag1,':k');
axis([0.4 2 0 0.001]);
xlabel('w/w_h');
ylabel('y/y_h');
title('Amplitude Characteristic');
figure(2);
plot(W./w, phase,'-k');
axis([0 3 -180 0]);
xlabel('w/w_h');
ylabel('Phase [deg]');
title('Phase Characteristic');
```

Figure 2: The Matlab m-file for the solution of the model

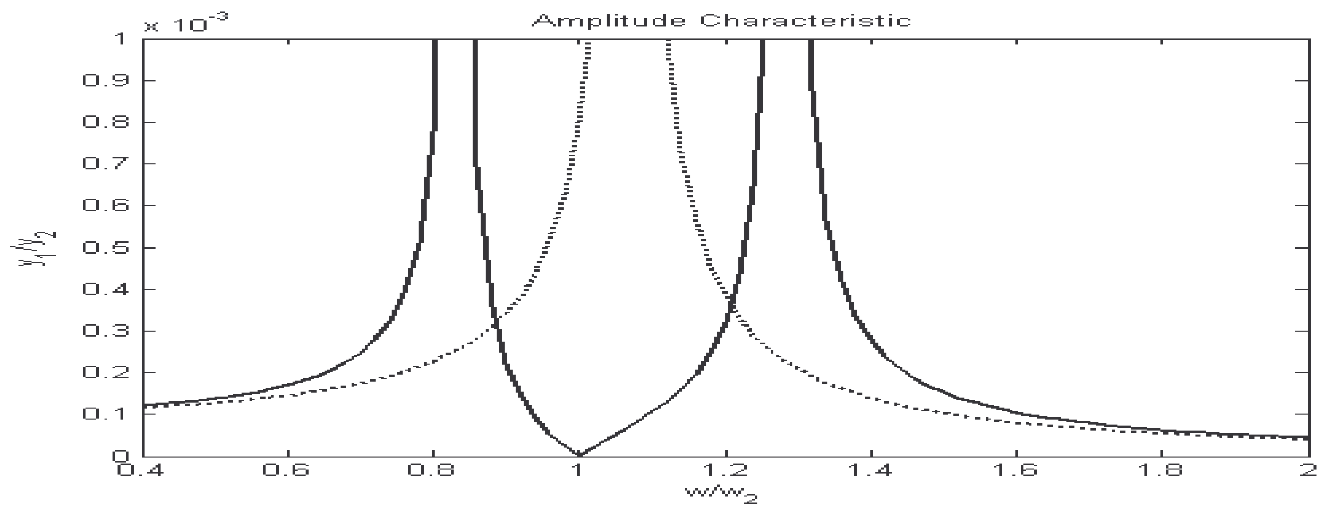

Figure 3: The amplitude characteristic on dimensionless variables (without the absorber is stroke curve)

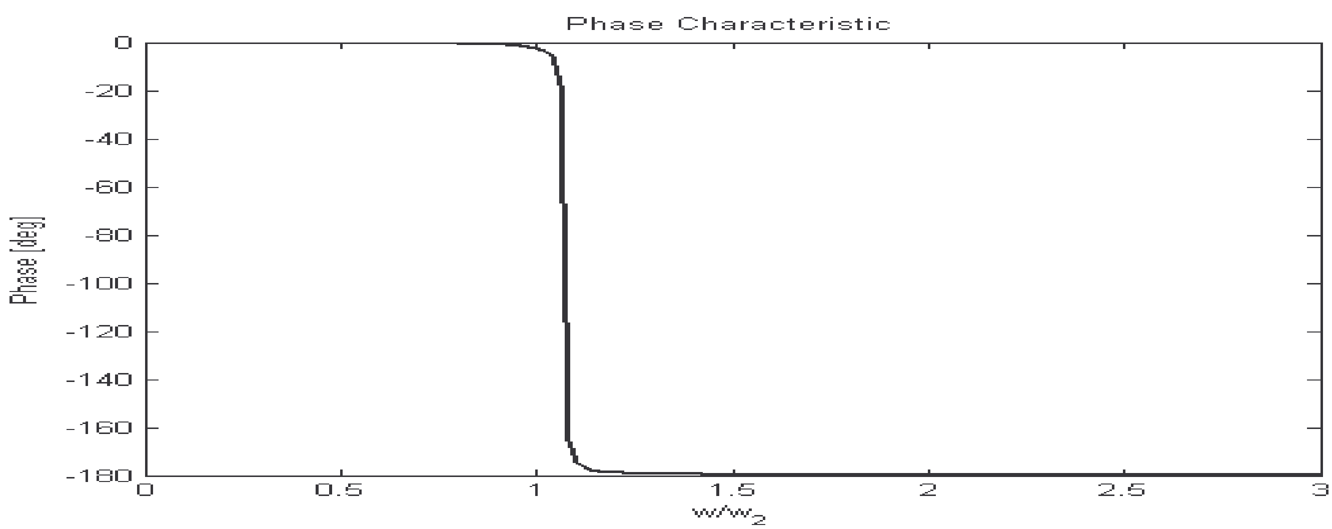

Figure 4: Phase characteristic on the dimensionless variables

2. ONLINE OPTIMIZATION TUNABLE ABSORBER WITH LQR CONTROL

The present article will discuss the method online optimization of the vibration control LQR (Franklin et al., 2001) for a machine by using a vibration tunable absorber. In the method, a variable stiffness vibration tunable absorber is used for controlling the principle mode. The stiffness is possible controlled by the accelerometer under the auto-tuning algorithm for creating an ant resonance state.
The optimal vibration tunable absorber with the air-operated spring is also utilized for controlling higher modes. The analyses and Matlab m-file for the auto-tuning control have been used. It is possible to write the state description system and model in the form

$$\begin{aligned} \dot{x} &= Ax + B_{act} F_{act} + B_{tech} F_{tech}\ ,\ y_1 = Cx + Du\ , \\ \dot{x}_m &= Ax_m + B_{act} F_{act} + B_{tech} F_{tech} + LC\ , \\ y_{1m} &= Cx_m + Du_m\ , \end{aligned} \tag{3}$$

where A is the state matrix, C the state matrix of output, D is the matrix of the coupling between input and output, F_{act} is the control force in the air-operated spring, F_{tech} is the spurious force from the technological process, B_{act} is the matrix of the input a control, B_{tech} is the matrix of input of the spurious force, x_m is the state vector of the model, y_{1m} is the displacement of the model and vector of input u is

$$u = \left[F_{act}, F_{tech}\right]^T . \tag{4}$$

The synthesis of control is make by means of finding the a state feedback

$$F_{act} = -G.x_m, \tag{5}$$

where G is the matrix of the control.
It is possible to obtain this state bond with the help of the minimization of the integral criterion on the LQR control [1-9]

$$J = \int_0^\infty (x^T \cdot Q \cdot x + F_{act}^T \cdot R \cdot F_{act})dt, \tag{6}$$

where Q, R are balance matrices.
The LQR control radiates from complete vector states, which in real life must be not in the feedback to position. In our case, we have to dispose the output parameters from the accelerometer a location on the machine. One-way, which this problem can be solved is use of the so-called state observer formulate with matrix L, where the parameters from the accelerometer are in to reconstructed a state of the system.
If we use the air-operated spring with the changed coefficient of elasticity k_2 with the possibility of regulation of the pressure p_{act} air in dependence to the displacement y_1, it is possible to reduce this displacement y_1 on the minimum.
It is possible change some dependence for a pressure in the rubber bellows in the air-operated spring as the calculated force F_{act}

$$p_{act} = \frac{F_{act}}{S}, \tag{7}$$

where S is the effective cross-sectional are of the bellows for the air-operated spring. The coefficient of elasticity k_2 of the air-operated spring is

$$k_2 = \frac{2\,\gamma\,p_{act}\,S^2}{V}, \quad (8)$$

where γ is the ratio of specific heats for air and V is the volume of the bellows.

3. RESULTS AND CONCLUSION

Vibrations of the machine vanish perfectly at a certain frequency when they have a vibration absorber with very small damping. But if forced frequencies vary from the anti-resonance frequency, their vibration amplitudes increase significantly. Then, the absorber with very small damping cannot be applied to the structure subjected to variable frequency loads or to the loads having high frequency components. The present article discusses a method of vibration control LQR (Franklin RANKLIN G.F., POWELL J.D., WORKMAN M.L., LEWIS F.L.for a model of the machine by using the vibration absorber with very small damping. In the method, a variable stiffness vibration absorber is used for controlling the principle mode. The stiffness is controlled by the accelerometer *a* under the auto-tuning algorithm for creating an anti-resonance state. The optimal vibration absorber with the air-operated spring is also utilized for controlling higher modes. A method to obtain the optimal parameters has been presented for the vibration absorber, which controls higher modes. In order to validate the control method and the analysis, experimental tests will be carried out in the next phase of research.

If we use the air-operated spring with the changed coefficient of elasticity k_2 with the possibility of regulation of the pressure p_{act} for the air in the operated spring in dependence to the displacement y_1 of is possible to reduce this displacement y_1 on the minimum (Figure 5, Figure 7).

On the lecture, the students are on lecture acquainted with knowledge of the kinds of absorbers and their development in technical practice, the possible solution and design of the absorbers. Then the students are acquainted with the kinds of controls of tunable absorbers.

In the case when the frequency ω of the acting force $F_{(t)}$ (alternative 1 and 2) driving the machine is the same as frequency ω_2 of the vibration of the absorber is the displacement of the manufacturing system after starting (t=10 sec) of LQR control show on Figure 5 for the alternative 1 and for the alternative 2 on the Figure 7. The displacements of the absorber are show on the Figure 4 and Figure 6. If we use the air-operated spring with the changed coefficient k_2 with possibility of regulation of the pressure $p_{act}=F_{act}/s$ for the air in the operated spring in the dependence to the displacement y_1 of the machine is possible to reduce this displacement y_1 on the minimum.

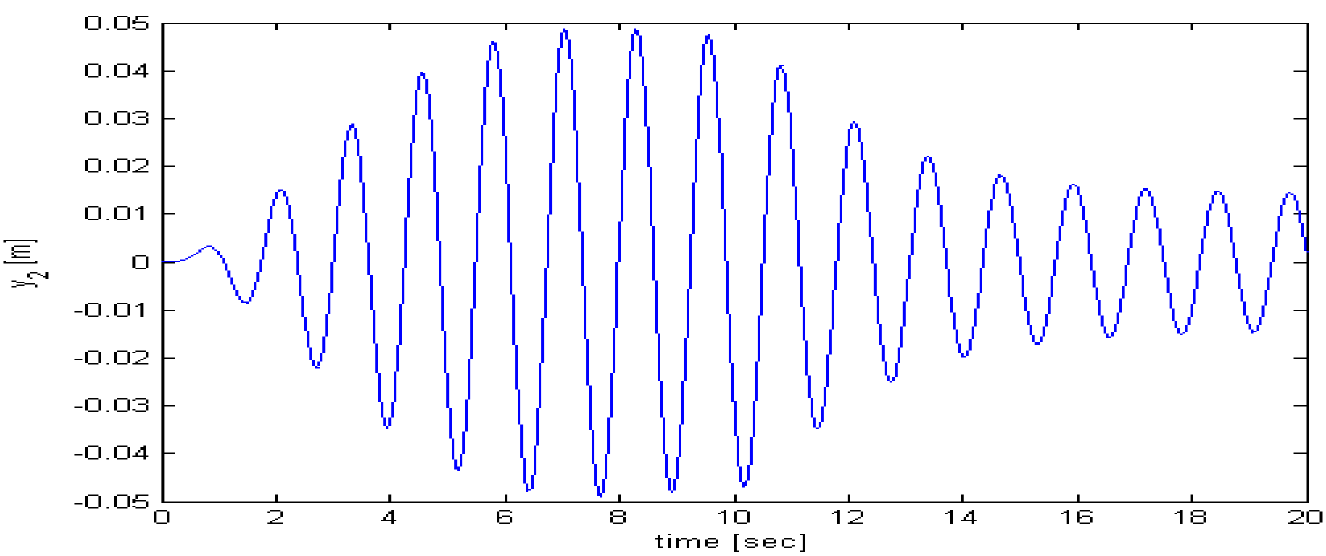

Figure 4: Displacement y_2 of the absorber after starting (10sec) of LQR control (alternative 1)

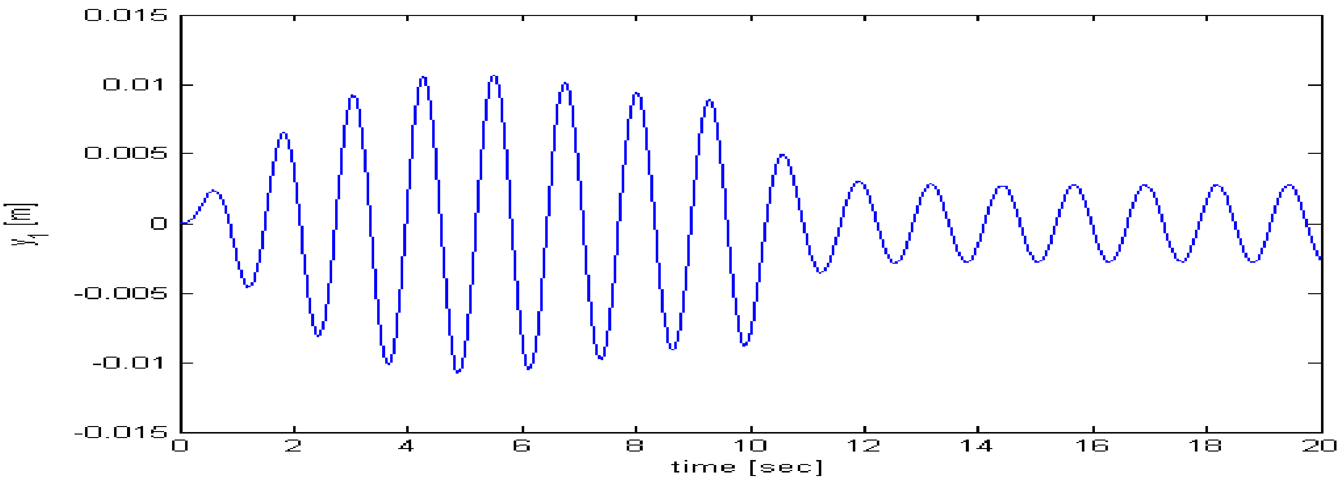

Figure 5: Displacement y_1 of the machine after starting (10sec) Of LQR control (alternative 1)

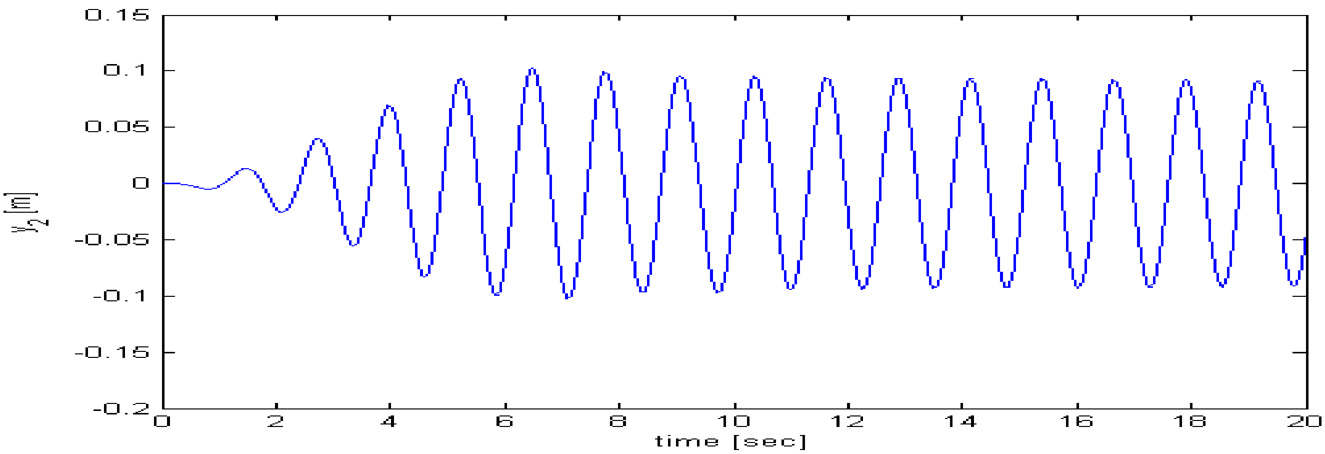

Figure 6: Displacement y_2 of the absorber after starting (10sec) of LQR control (alternative 2)

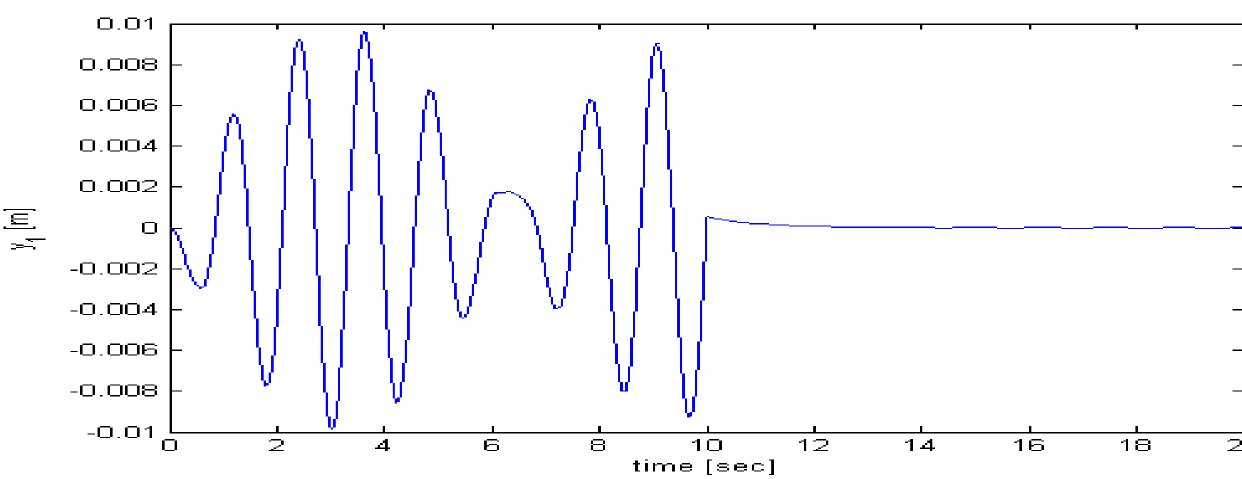

Figure 7 Displacement y_1 of the machine after starting (10sec) Of LQR control (alternative 2)

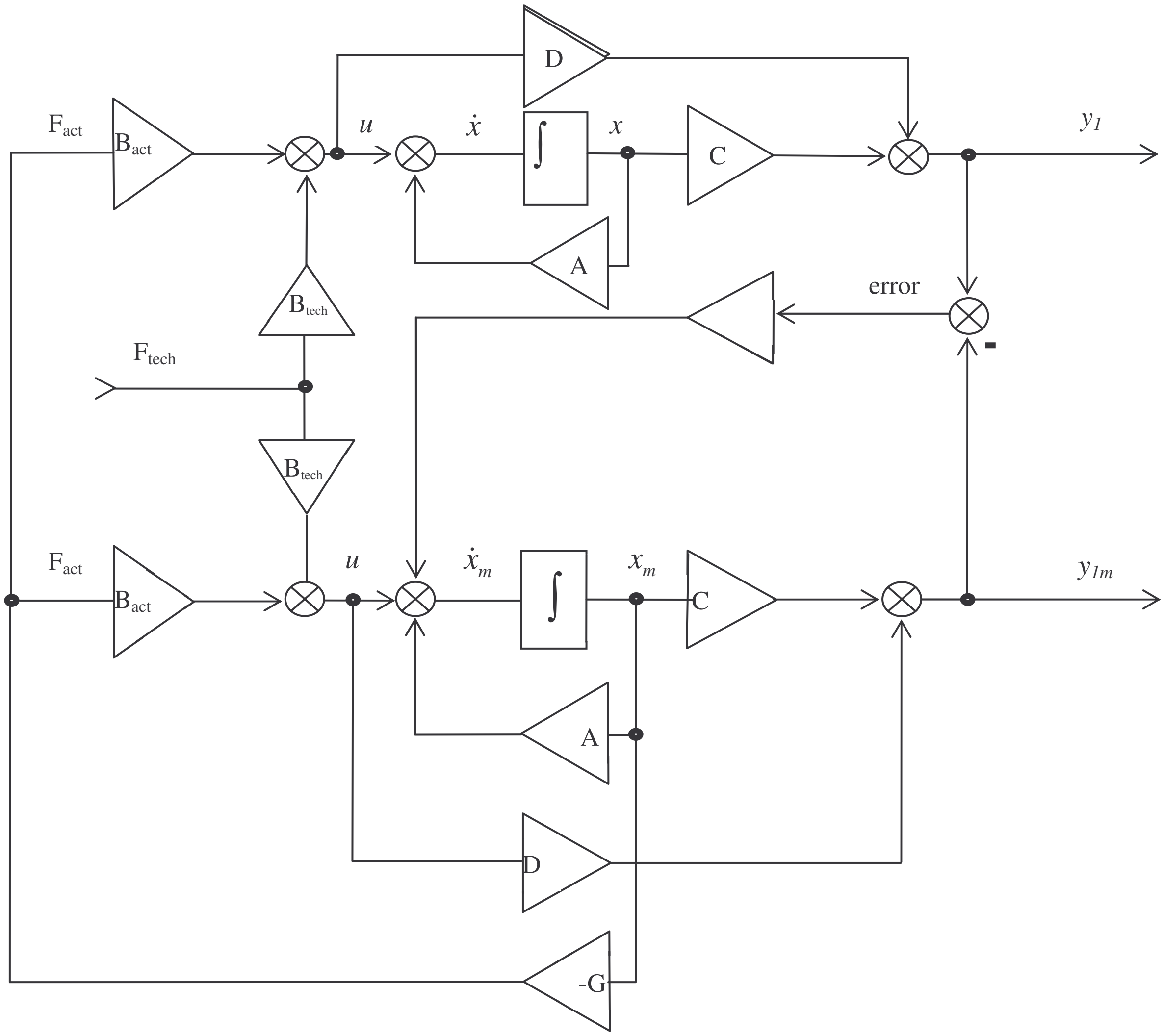

Figure 8: Simulink LQR control block diagram for tunable absorber

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REFERENCES

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