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AN ACTIVE CONTROL METHOD TO REDUCE FRICTION INDUCED VIBRATION CAUSED BY NEGATIVE DAMPING

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ABSTRACT

In this paper, a novel approach to reduce the effect of negative damping that causes friction induced vibration (FIV) is proposed by applying an active force control (AFC) based strategy to a simplified two degree-of-freedom disk brake model. At first, the model is simulated and analyzed using a closed loop pure PID controller. Later, it is integrated with AFC and simulated under similar operating environment. After running several tests with different sets of operating and loading conditions, the results both in time and frequency domains show that the PID controller with AFC is much more effective in reducing the vibration, compared to the pure PID controller alone.

Keywords: *Active force control, disk brake, friction induced vibration, negative damping.*

1.0 INTRODUCTION

The problem of friction induced vibration exists in almost all mechanisms that are utilizing sliding surfaces, and in order to keep the mechanism under smooth operation, it is required to reduce this problem as much as possible. Disk brake system is one of the mechanisms that gets affected and causes noise due to friction induced vibration.

Disk brake noise in an automotive system is always taken into consideration by the designers and manufacturers due to the fact that it may cause discomfort to the passengers and adversely affect their perceptions of the quality and reliability of the vehicle. Thus, noise generation and suppression have become an important factor in the design and manufacture of brake components. Indeed, as noted by Abendroth and Wernitz [1], a large number of manufacturers of brake pad materials spend up to 50% of their engineering costs on issues related to noise, vibration and harshness. Brake noise vibration phenomena are described by a number of terminologies that are sometimes interchangeably used such as squeal, groom, chatter, judder, moan and hum [2]. Even to this day, there is no precise or conclusive definition of brake squeal that has gained complete acceptance. Basically, there are four main mechanisms that contribute to the generation of noise in brake system, namely negative damping [3], sprag slip [4], modal coupling [5] and moving load [6]. More explanation of these mechanisms can be found in [2]. In

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this paper, the effect of negative damping is considered. It occurs when the friction coefficient decreases by the increase of the relative velocity between the rotor and the pad in the disk brake system.

Normally there are three major methods to study and reduce brake squeal, namely through mathematical modelling, experimental and finite element (FE) methods. A recent study that proposed a method for reducing brake noise using FE can be found in reference [7], where the authors created a dynamic FE model of the brake system. In their analysis, they showed that pad design changes can be used in the FE model to determine the potential improvements in the dynamic stability of the system and also in noise reduction. Wagner *et al.* presented a new mathematical rotor based model of a brake system that is suitable for noise analysis [8]. In their work, a brief description of the previous mathematical models that have been developed by other researchers was also given. Besides, there is also an active control method known as dither control which makes use of high frequency disturbance signal for the suppression of the automotive disc brake squeal [9]. Through this scheme, the dither signal stabilizes friction induced self-oscillations in the disc brake using a harmonic vibration, with a frequency higher than the squeal frequency generated from a stack of piezoelectric elements placed in the caliper piston of the brake system. The results seem to be fine in reducing the noise, how ever other operating factors such as wear, temperature, friction and speed are required to be considered for evaluating the effectiveness of the system.

This paper presents a closed loop control method employing Active Force Control (AFC) with PID element applied to a brake model described in [10] in order to suppress the brake noise and squeal. The main advantage of the AFC technique is its ability to reject disturbances that are applied to the system through appropriate manipulation of the selected parameters. In addition, the technique requires much less computational burden and has been successfully demonstrated to be readily implemented in real-time. AFC as first proposed by Hewit and Burdess [11] is very robust and effective in controlling a robot arm. Mailah and co-researchers [12, 13] have successfully demonstrated the application of the technique to include many other dynamical systems with the incorporation of artificial intelligence (AI) methods.

2.0 THE SIMPLIFIED BRAKE

The effect of negative damping in a disc brake system was investigated taking into account a two degree-of-freedom (DOF) model based on the one described in [10]. The model as shown in Figure 1 represents the disk brake element in the form of a two DOF system which is connected through a sliding friction interface. The motion of the mass, m_1 may represent the tangential motion of the pad, and the second mass, m_2 may represent the in-plane motion of the disc. The normal force, N is a constant load that can be computed from pressure P multiplied by the surface area S. The dynamic friction coefficient is considered to reduce by the corresponding increase of the relative velocity as shown in Figure 2. The relative velocity in the figure consists of a constant velocity of the disk, v_0 added to the variable velocity of the disk and subtracted from the variable velocity of the pad. The friction coefficient or the slope α is known to be influential in contributing to the vibration in the system.

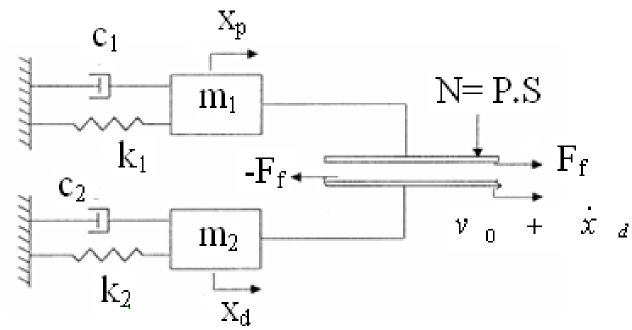


Figure 1: Two DOF model of a disc brake system

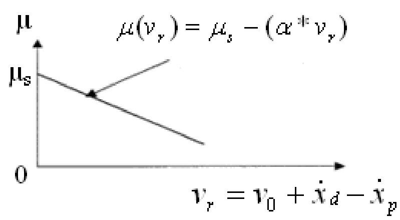


Figure 2: Dynamic friction coefficient

Equation (1) shows the linear function of the dynamic friction coefficient:

$$\mu_k = -\alpha v_r + \mu_s \tag{1}$$

The dynamic equations of motion are written as follows [10]:

$$\left.\begin{aligned} m_1 \ddot{x}_p + c_1 \dot{x}_p - N\alpha(\dot{x}_p - \dot{x}_d) + k_1 x_p &= N(\mu_s - \alpha v_0) \\ m_2 \ddot{x}_d + c_2 \dot{x}_d - N\alpha(\dot{x}_d - \dot{x}_p) + k_2 x_d &= -N(\mu_s - \alpha v_0) \end{aligned}\right\} \tag{2}$$

3.0 CONTROL STRATEGY

Upon obtaining the model of the disc brake and its related equations of motion, it is required to control the vibration in both directions x_p and x_d, by having actuators which can apply forces that are parallel the given axes. Thus the equations of motion after applying the actuators can be obtained as follows:

$$\left.\begin{aligned} m_1\ddot{x}_p + c_1\dot{x}_p - N\alpha(\dot{x}_p - \dot{x}_d) + k_1 x_p &= N(\mu_s - \alpha v_0) + A_p \\ m_2\ddot{x}_d + c_2\dot{x}_d - N\alpha(\dot{x}_d - \dot{x}_p) + k_2 x_d &= -N(\mu_s - \alpha v_0) + A_d \end{aligned}\right\} \qquad (3)$$

A control strategy is proposed here employing an Active Force Control (AFC) based scheme that is used in conjunction with the conventional PID controller. The PID controller was first tuned with *Ziegler-Nichol*'s method and then manipulated for good performance. Later, the AFC part was incorporated into the system to provide the compensation of the disturbances that are inherent in the brake system. Figure 3 shows the AFC scheme applied to a dynamic translation system (disc brake). AFC scheme is shown to be very effective provided the actuated force and body acceleration are accurately measured and at the same time the estimated mass property appropriately approximated [12, 13]. The essential AFC equation can be related to the computation of the estimated disturbance force, F_d as follows:

$$F_d = F - M'a \qquad (4)$$

where F is the measured actuating force, M' is the estimated mass and a is the measured linear acceleration.

This parameter is then fed back through a suitable inverse transfer function of the actuator to be summed up with the PID control signal. The theoretical analysis including the stability of the proposed AFC method has been sufficiently described in [14]. A number of methods to estimate the mass have been proposed in previous studies such as through the use of artificial intelligence (AI) and crude approximation techniques [12-14]. In this study, the use of crude approximation method to approximate the estimated mass is deemed sufficient. The main challenge of the AFC method is to acquire appropriate estimation of the mass needed to compute the disturbance F_d in the feedback loop. A conventional PID that is used with the AFC scheme can be typically represented by the following equation:

$$G_c(s) = K_p + K_i/s + K_d\, s \qquad (5)$$

where K_p, K_i and K_d are the proportional, integral and derivative gains respectively.

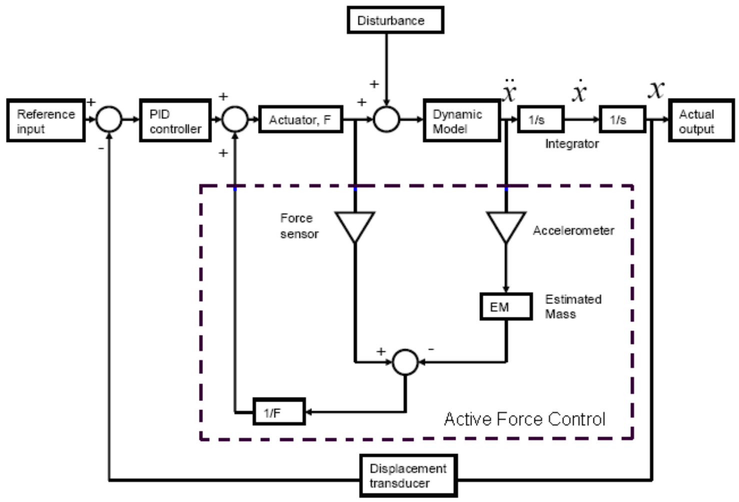


Figure 3: Schematic diagram of AFC strategy

4.0 SIMULATION

MATLAB, Simulink and Control System Toolbox (CST) software were used to simulate the brake model with the controllers. The actuators are assumed to be a linear type with a suitable constant gain. They provide the necessary external energy to suppress vibration in the model. The parameters used in this study were taken from the previous researches [8], [10] and [15]. However, some of them need to be adjusted to suit the application in the simulation. The detailed parameters are given as follows:

Minimal disc brake model parameters:

- Body masses, m_1 = 0.3 kg, m_2 = 2.5 kg
- Spring stiffness, k_1 = 26000 N/m, k_2 = 38000 N/m
- Damping coefficient, c_1 = 2 Ns/m,
  c_2 = 3.5 Ns/m
- Static friction coefficient, μ_s = 0.6
- Normal preload, N = 100 N

Actuators:

- Actuators gain: Q(pad) = 0.4 , Q(disk) = 0.6

Reference value:

- Reference input = 0.00 m (i.e. no vibration)

In this work, several types of operating conditions are deliberately introduced to the disc brake system to evaluate the robustness of the control system. The Simulink diagram of the passive disc brake system model is shown in Figure 4. The schematic block diagram was constructed from equation (2).

In order to have an active disc brake system, two actuator forces for compensating the disturbances (that are actually inherent in the two DOF brake system) are required. The actuator forces are controlled by two individual PID controllers which typically involve two negative feedback loops. Hence, there are two inputs to the dynamic disc brake system which are the inputs to actuator forces. Figure 5 shows the active disc brake system.

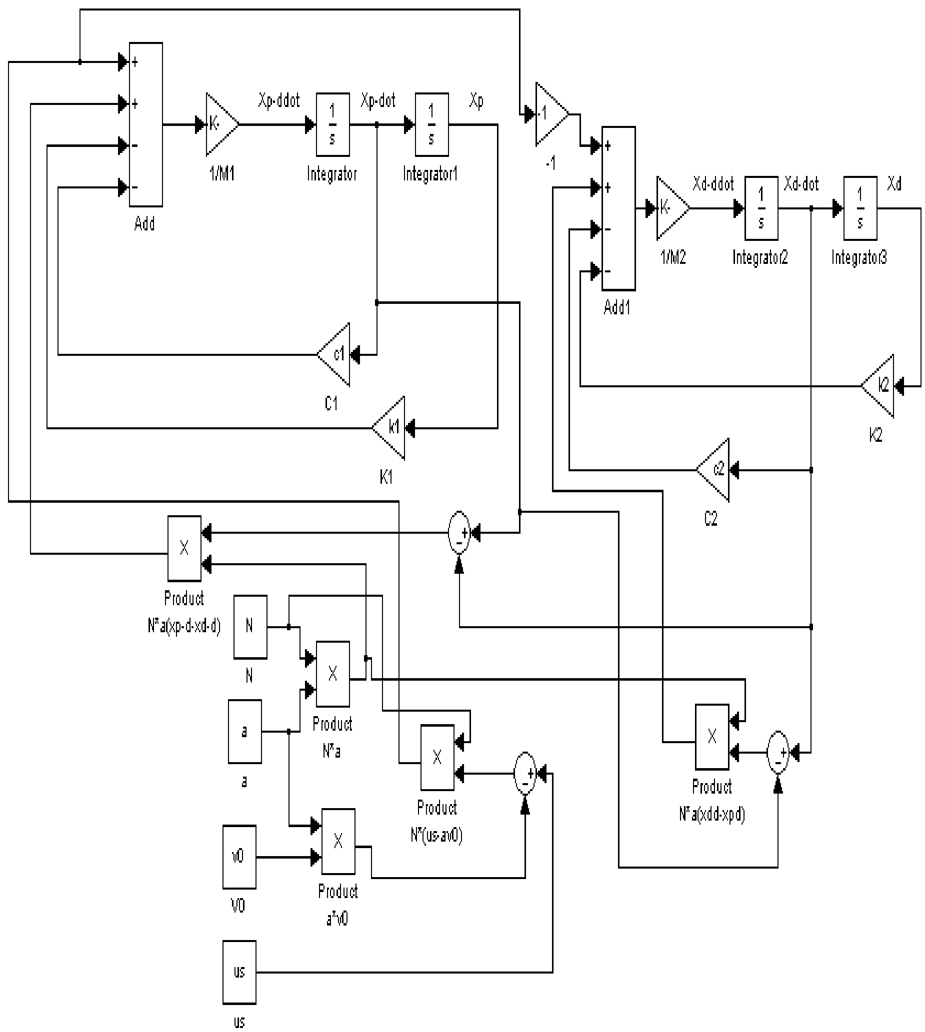


Figure 4: Schematic diagram of passive brake system

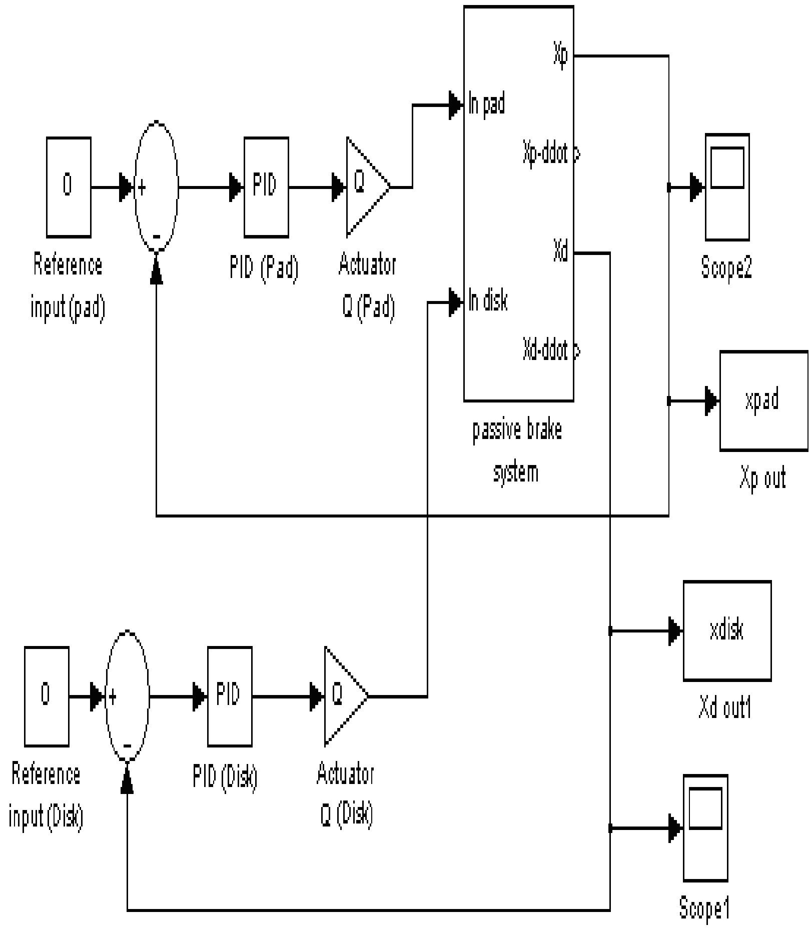


Figure 5: Schematic diagram of active brake system

A better overall performance of the brake system could be obtained when a AFC is 'added' to the pure PID controller. The AFC Simulink diagram includes the estimated mass and the parameter *1/Q*. The input to the AFC control is the mass acceleration and the output is summed with the PID controller output and then multiplied with the actuator gain which subsequently generates the actuator force. In order to get the effective results using this method, it is required to acquire an appropriate mass estimation combined with the best tuning of the PID controller gains. Figure 6 shows an AFC scheme used in the study to suppress the vibration and noise of the brake model that is mainly caused by the inherent disturbances in the system. It can be seen that each PID controller is equipped with an AFC loop.

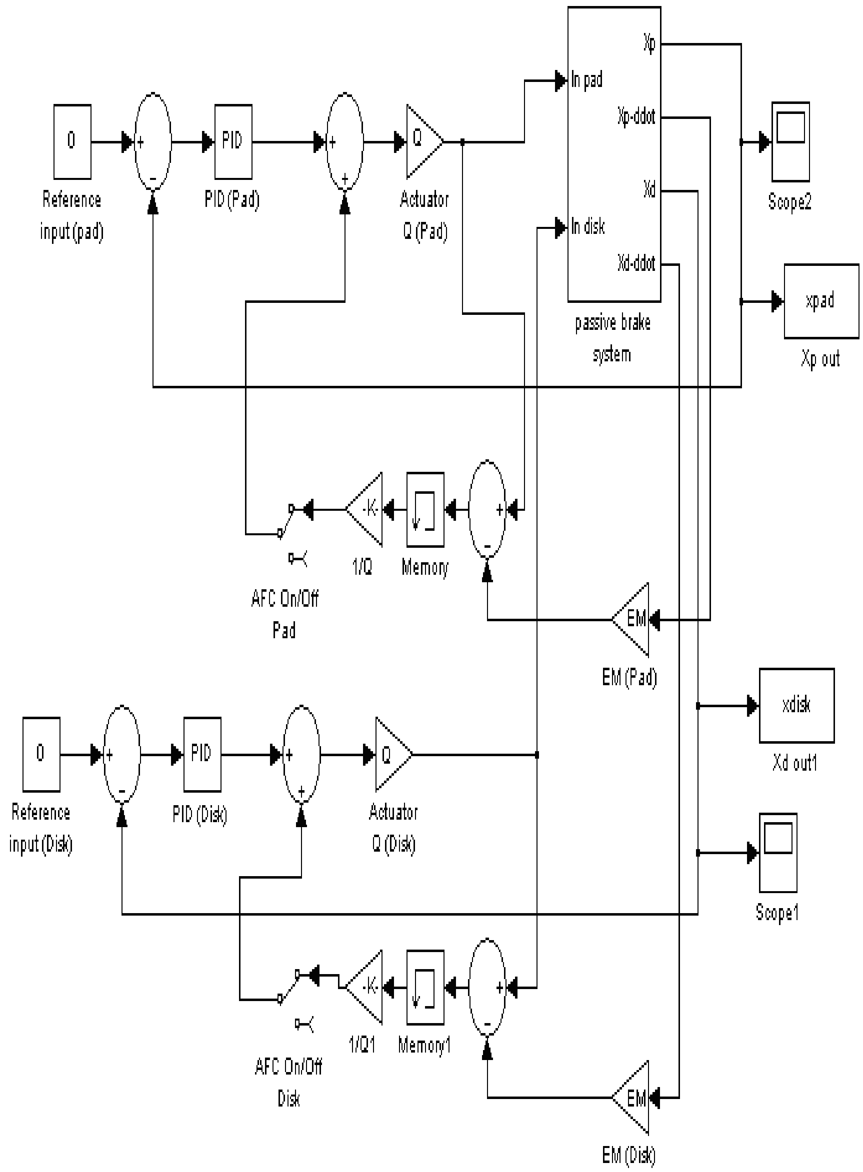


Figure 6: Proposed AFC model

To tune the PID controllers, *Ziegler-Nichol*'s method with trial and error changes was used and the results are tabulated as shown in Table 1. The estimated masses for the AFC loops were obtained by crude approximation method in which they are shown in table 1, and the percentage of AFC used is 100% (switch: ON), implying that the AFC loop employs full AFC implementation.

Table 1: The values for the estimated mass and the PID parameters

<table>
<tr><td>PID Parameters</td><td>K_p</td><td>K_i</td><td>K_d</td><td rowspan="3">NA</td></tr>
<tr><td>PID (pad)</td><td>0.03</td><td>0.015</td><td>0.07</td></tr>
<tr><td>PID (disk)</td><td>0.8</td><td>0.4</td><td>1.3</td></tr>
<tr><td colspan="5">Estimated Mass (kg)</td></tr>
<tr><td colspan="3">E.M (pad)</td><td colspan="2">1</td></tr>
<tr><td colspan="3">E.M (disk)</td><td colspan="2">1.3</td></tr>
</table>

5.0 RESULTS AND DISCUSSION

The simulation was executed for a period of 5 seconds. At first, the dynamic friction coefficient or the slope was set to 0.015 (negative damping) and with a constant velocity of 100 km/h with simulation initially performed using only the pure PID controller. The results of this process can be seen in Figures 7 and 8. It can be observed that the vibration that may result in producing noise, is relatively high especially in the first 2 seconds though it converges to a steady state condition towards the end of the simulation period. Next, the simulation was carried out again but this time considering 100% AFC mode plus the PID controller. Contrary to the previous scheme, the vibration almost disappeared instantaneously and the system seems to operate very smoothly throughout the simulation period.

In the next simulation test, the slope with negative damping was increased to 0.02 with the same constant velocity; the results of this simulation are shown in Figures 9 and 10. It can be seen that with a pure PID controller, the vibration of the pad is not reduced significantly though the trend is converging, but when the mode is switched to AFC with the PID scheme, the vibrations are again very much reduced to a steady state condition almost immediately. Frequency response analyses were also carried out and the results are shown in Figures 11 and 12. It can be seen that the PID controllers obviously produce considerable sharp spikes with high amplitude at around 45 Hz and 20 Hz while the AFC-based schemes hardly produces any peaks, thus verifying the robust system performance.

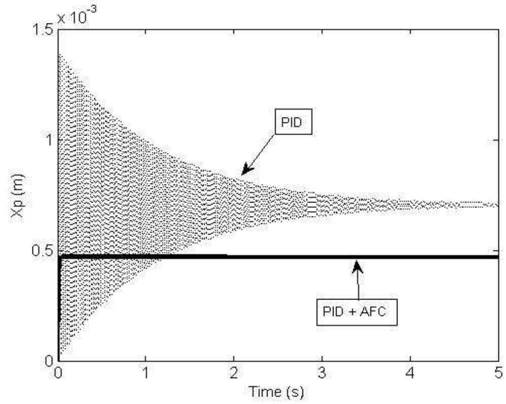


Figure 7: Response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.015, v_0 = 100 km/h$

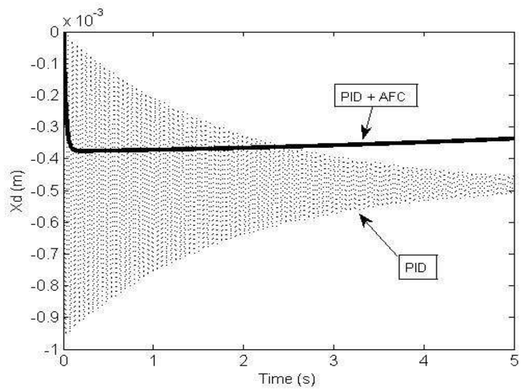


Figure 8: Response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.015, v_0 = 100\ km/h$

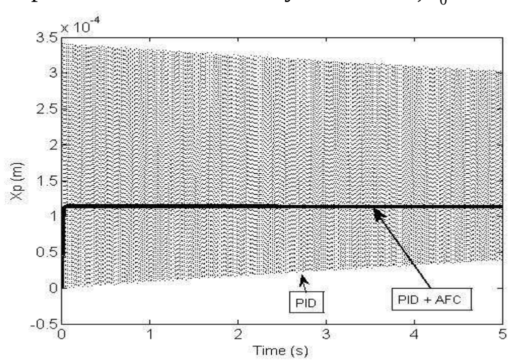


Figure 9: Response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.02, v_0 = 100\ km/h$

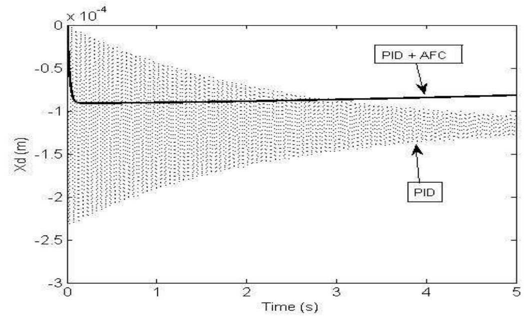
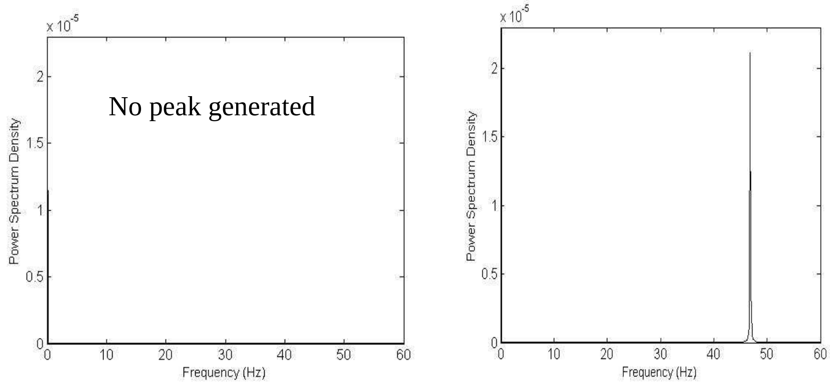


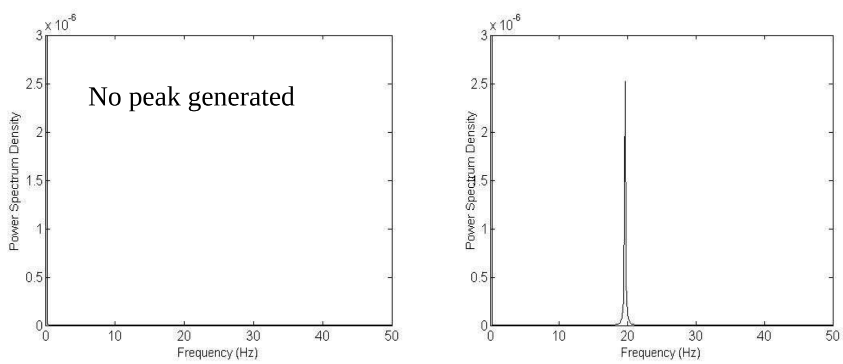
Figure 10: Response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.02, v_0 = 100\ km/h$



a) PID + AFC b) PID

Figure 11: Frequency domain response of the pad (direction x_p) in the brake model having the slope and constant velocity:

$\alpha = 0.02, v_0 = 100\ km/h$



a) PID + AFC b) PID

Figure 12: Frequency domain response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.02, v_0 = 100 km/h$

By increasing the slope to 0.027, it can be seen from Figures 13 and 14, the system can no longer be stabilized with a PID controller as the vibrations are increasing in amplitude with the increase in time. On the other hand, after applying AFC to the PID controller, the vibrations in the brake system are significantly and consistently reduced to nearly zero level. Figures 15 and 16 shows the frequency responses through this simulation procedure.

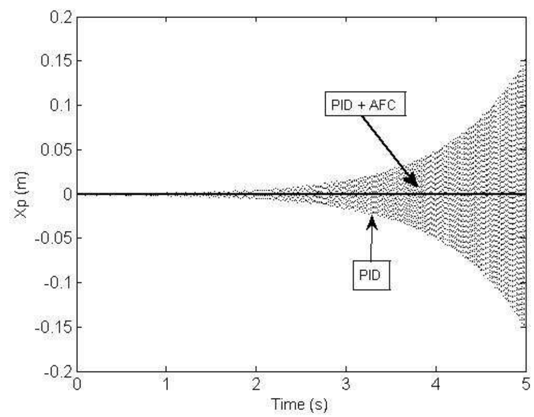


Figure 13: Response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 100 km/h$

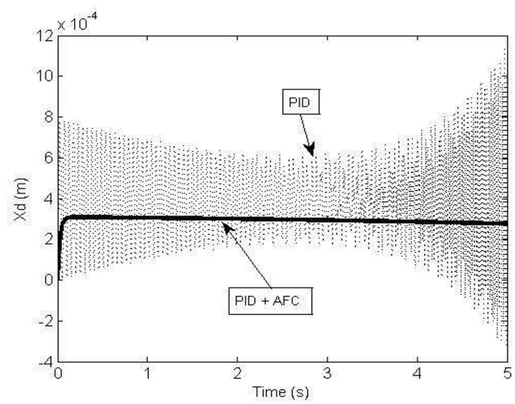
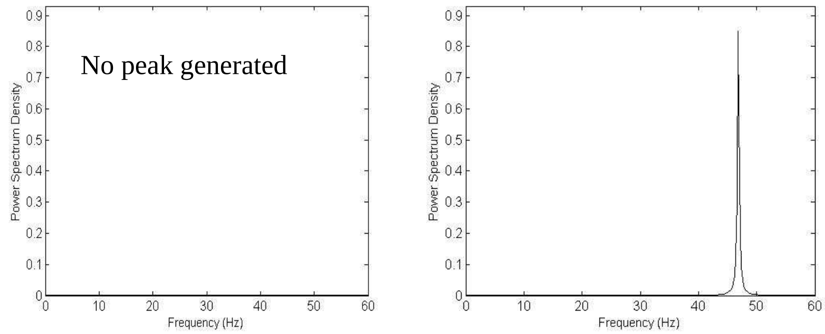
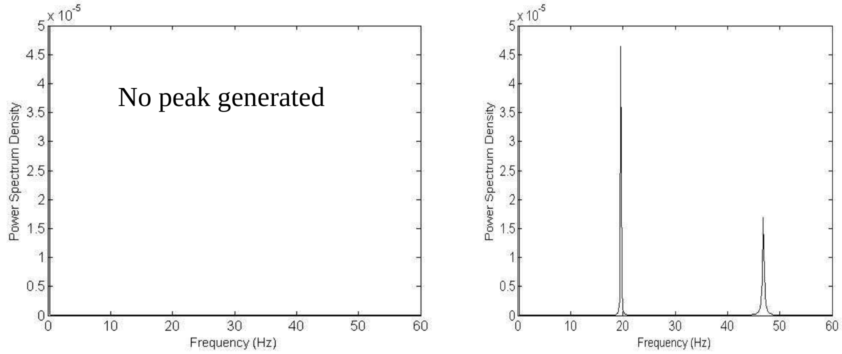


Figure 14: Response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 100 km/h$



a) PID + AFC b) PID

Figure 15: Frequency domain response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 100 km/h$



a) PID + AFC b) PID

Figure 16: Frequency domain response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 100 km/h$

In this study, the effect of negative damping at low speed was also taken into consideration, by running the simulation under a constant velocity of 25 km/h and keeping the slope with negative damping to 0.027. The results of this simulation are shown in Figures 17 and 18. The vibrations in the friction induced vibration system were shown to have increased significantly in magnitude compared to the case where the speed was set at 100 km/h. Thus, it can be safely deduced that the negative damping has more adverse effect on the system at low speed than at the other end. Again, when AFC was applied to the PID controller, the result was as predicted in which the vibrations are largely suppressed. Figures 19 and 20 show the frequency responses of this simulation.

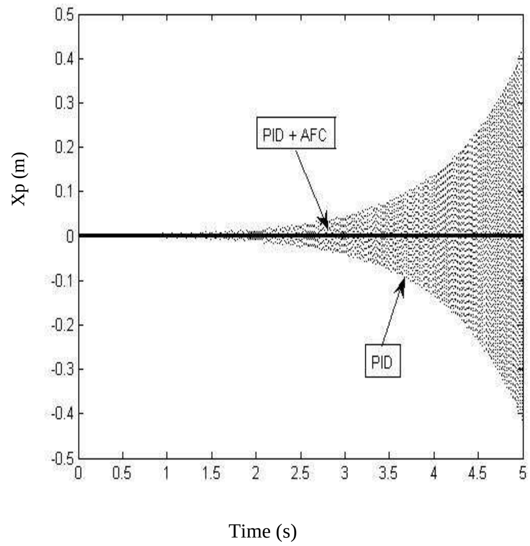


Figure 17: Response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 25\ km/h$

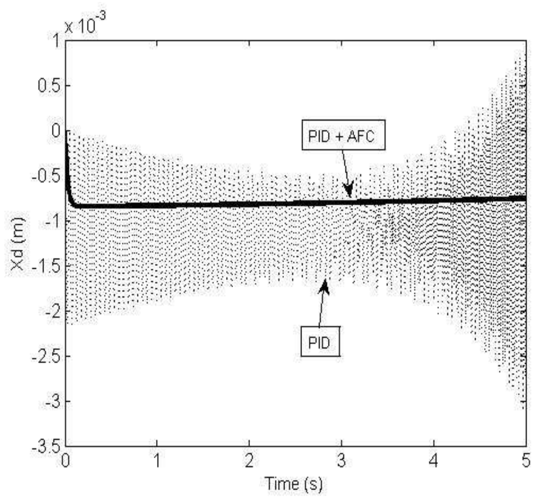
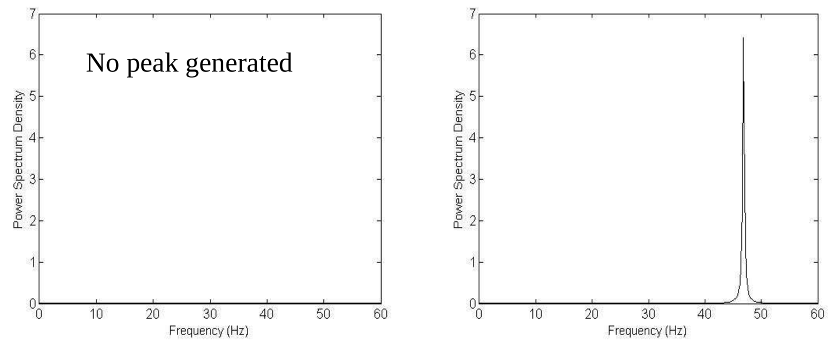
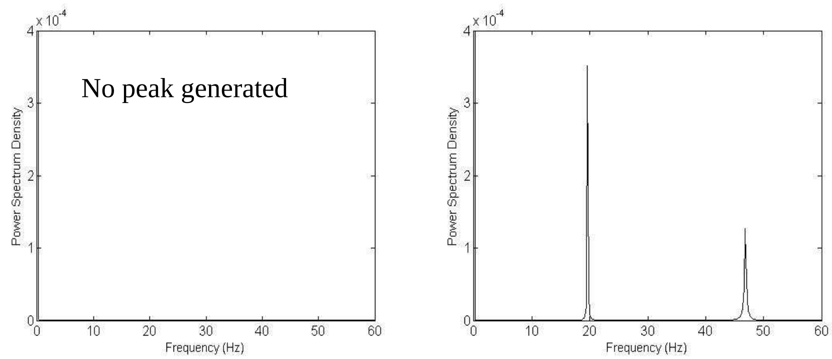


Figure 18: Response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 25\ km/h$



(a) PID + AFC (b) PID

Figure 19: Frequency domain response of the pad (direction x_p) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 25\ km/h$



a) PID + AFC b) PID

Figure 20: Frequency domain response of the disk (direction x_d) in the brake model having the slope and constant velocity: $\alpha = 0.027, v_0 = 25\ km/h$

6.0 CONCLUSION

A novel proposed AFC-based scheme has been shown to reduce the vibration that is caused due to friction induced vibration by negative damping in the disk brake model. According to the simulation results, it is obvious that when a pure PID controller is applied to the FIV system, vibration and noise are in fact gradually reduced but still a noticeable amount of them remains specially in the first two seconds of the simulation period, and in some cases the PID controller was not able to reduce the vibration at all. However, upon applying the AFC-based technique, the vibration and noise are considerably reduced and approaching a zero datum, thereby implying the effectiveness of the proposed strategy in countering the adverse operating and loading conditions. Also, it was noticed that negative damping shows a more severe effect, when the brake model is operated at a low constant velocity. For future work it is planned to design an experimental rig with controllers installed, and run a few experiments based on the simulations that were done.

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