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Vibration Suppression of a Handheld Tool Using Active Force Control with Crude Approximation Method

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Abstract- **This paper presents a simulation study involving the application of an active force control (AFC) strategy to suppress vibration on the rear handle of a handheld tool. The research was carried out to investigate the performance in terms of vibration reduction capability of a feedback controller employing an AFC-based scheme on a selected powered portable machine (Hedge Trimmer Maruyama model Ht230D). Two types of control schemes were closely examined and compared involving the classic proportional-integral-derivative (PID) control scheme and the proposed AFC with crude approximation (AFCCA) method. Inherent vibration was measured from the real operation of the handheld tool through operational deflection shape (ODS) experiment. This data was later used in the simulation work to test the robustness of the control scheme. Results show that the AFCCA scheme is able to suppress the vibration at the rear handle much better compared to the conventional PID control scheme.**

I. INTRODUCTION

Undesired vibration can disturb our comfort, damage to structures, reduction of tools performance and machinery noise level. One of the effects of undesired vibration is the Hand-arm Vibration Syndrome (HAVS). HAVS is a disease that involves circulatory disorders (e.g. vibration white finger), sensory and motor disorders, and musculoskeletal disorders, which may occur in workers who use vibrating handheld tools [1]. Undesirable vibration need to be controlled and suppressed because it is wasting energy and creating unwanted sound (noise). In 2002, European Commission reveals that 17% of the European workers were reportedly being exposed to vibration from handheld tools or machinery for at least half of their working time and the Spanish scenario reveals that 22.8% of the workers that use portable electric and pneumatic tools were reportedly being exposed to vibration [1]. In order to protect the workers from HAVS, different countries have proposed and developed their own criteria. For the example, the European Union (EU) and Sweden National Institute for Working Life, and OPERC database have formed Human Vibration Directive and Hand-Arm Vibration Database respectively. Their main objective is to measure a vibration level of handheld powered tools, reveals the effect of HAVS to human body, etc. Different frequencies of vibration interfere with different body parts and systems where whole body vibration occurs at frequencies from l-30 Hz, while segmental vibration, which interferes with the hand-arm system, is between 30-100 Hz [2]. Above 100 Hz threshold, the hand in particular is affected [3]. Other figures suggest a greater overlap; from 2-100 Hz for whole-body vibration and from 8-1500 Hz for segmental vibration and the current NIOSH guideline recommends measuring vibration up to 5000 Hz [4]. In vibration control, there are typically two approaches of tackling vibration problems; passive and active vibration control. Passive vibration control is based on the damping using viscoelastic materials while active vibration control (AVC) involves the use of active elements (actuators) along with sensors and controller (analog or digital) to produce an out-of-phase actuation to cancel the disturbances. When the dynamics of the system and/or frequencies of the disturbance vary with time, passive vibration control becomes ineffective and less functional. Then, AVC is more efficient to be introduced into the systems. There are many control strategies that have been developed by researchers in AVC, mostly involving industrial machineries, vehicles and motors and shafts applications. However, AVC applied to powered portable tools is very seldom found in literature. Researchers are more concern about improving the powered portable's design where natural frequencies are shifted from the operating frequencies of the system instead of controlling the vibration that generated by its motor or driven. Therefore, AVCs were looked from various kinds of engineering fields and controlling techniques. One of the interesting and successful control techniques is called electromagnetic exciter. Electromagnetic exciter has been applied in active vibration control scheme for controlling transverse vibration of a rotor shaft due to unbalance [5]. In this study, four electromagnetic exciters were mounted on the stator at a plane where it will reduce unbalances response amplitude and also help to increase the stability limit speed of the rotor-shaft system. The underlying principle is achieved by varying the control current in the exciters depending upon a proportional and derivative (PD) control law applied to the displacement of the rotor section fed back by pick-ups with respect to the non-rotating position of the section taken as the reference. Yildrim has presented a neural network scheme for controlling a bus suspension system and comparing it to PID, PI and PD control schemes [6]. The finding shows that neural network scheme gives better robust performance compared to other control schemes. An AVC scheme called Linear Quadratic Gaussian (LQG) control has been implemented in woodcutting machine by Chen to reduce saw blade vibration and sawdust [7]. He found that vibration of the saw blade is the key reason for poor wood recovery. This technique involves the use of magnetic actuator which produces a counter force to suppress the vibration of the saw blade.

II. PROPORTIONAL-INTEGRAL-DERIVATIVE (PID)

In a typical feedback control system, the controller takes the error signal (difference between the desired and measured signals) and processes it. The output of the controller is passed as an input to the process. One type of controller which is widely used in industrial applications is the proportional integral derivative (PID) controller. The proportional part of this controller multiplies the error by a constant. The integral part of the PID controller integrates the error. Finally the derivative part mathematically differentiates the error. The output of the controller is the sum of the above three signals. The transfer function of this controller is

$$G_c(s) = K_P + \frac{K_I}{s} + K_D s \tag{1}$$

where K_P, K_I and K_D are the controller gains related to the proportional, integral and derivative terms, respectively. Taking into account the error signal, $e(t)$, the control signal can be written as

$$h(t) = K_P e(t) + K_I \int e(t)dt + K_D \frac{de(t)}{dt} \tag{2}$$

If the PID controller gains are chosen incorrectly, the controlled process input can be unstable. Tuning a control loop is the adjustment of its control gains to the optimum values for the desired control response. There are several methods for tuning a PID loop. The most effective methods generally involve the development of some form of process model, and then choosing P, I, and D based on the dynamic model parameters. Sometime, manual tune by feel methods can be inefficient.

III. ACTIVE FORCE CONTROL (AFC)

Active Force Control (AFC) strategy has been introduced and applied by Hewit in the late 70s for controlling a dynamic system [8]. A number of experimental studies has verified that this strategy gave good stability, robustness and effectiveness to the system even in the presence of known/unknown disturbances, uncertainties and varied operating condition. Basically, AFC operates by computing the estimated disturbance force, F^* via measurement of the mass acceleration, a and actuator force F_a and an appropriate estimation of the estimated/virtual mass, M^* as given by

$$F^* = F_a - M^* a \tag{3}$$

The basic schematic of the AFC scheme applied to a dynamic system is illustrated in Fig. 1. As mentioned previously, two physical quantities which are required to be measured by the sensing elements are the actuating force and acceleration of the system while the system operates. Then the estimated mass (for translational system while for the rotational system, it is the inertial parameter) of the system with the presence of the disturbances that contributes to the acceleration should be acquired appropriately by using suitable techniques such as crude approximation (CA) method or other intelligent methods (like fuzzy logic, neural network or iterative learning). In this study, the crude approximation method is employed and we called the system, AFCCA.

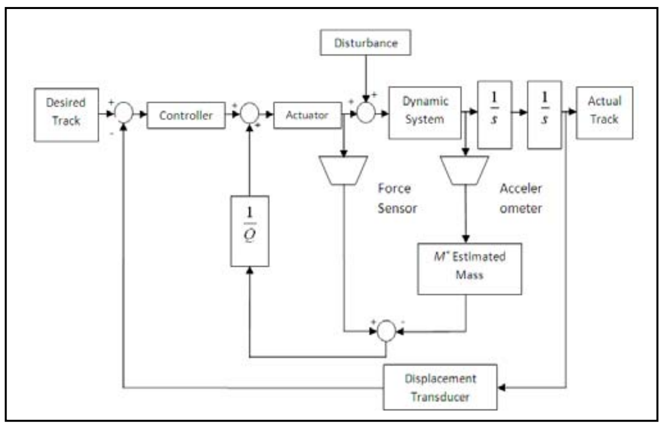

Fig. 1. Basic schematic of an AFC scheme

Kwek *et al.* [9] have investigated the efficiency of active force control (AFC) strategies and proved that this method is robust and stable. Three different control strategies have been applied to the biped robot; PD control scheme, PD-AFC-Crude Approximation (PD-AFCCA) and PD-AFC-Iterative Learning (PD-AFCAIL) The performance of the biped robot is measured by making the biped walk on a horizontal flat surface. It is found that for both proposed AFC-based schemes, the performance of the dynamic systems are found to be robust and stable even under the influence of disturbances. Another intelligent method to estimate the inertial parameter is through the use of fuzzy logic (FL) as done by Mailah and Rahim [10]. The FL element is embedded in AFC to control a robot arm where its main function is to computes the inertial parameter, IN automatically and continuously while the robot arm is in operation. Comparison has been made between the PD and AFC-FL schemes where it was analyzed through trajectory track errors. A practical AFC scheme applied to a mechanical suspension has been studied by Mohamad *et al.* in order to control unwanted vibration generated by varied disturbances [11]. In this study, three different schemes were compared; PID and AFC-CA scheme as well as a passive system. The suspension rig was built using the MATLAB/Simulink with Real Time Workshop (RTW) tool that was interfaced with a suitable data acquisition card via a personal computer as the main controller. The study verified the excellence of the proposed AFC scheme in suppressing vibration at various disturbance levels.

IV. DYNAMIC AND DISTURBANCE MODEL

For controlling purpose, dynamic model of the system (part of the portable handheld machine as shown in Fig. 2) has to be modelled mathematically prior to the simulation work. Dynamic model of the rear handle of the machine can be represented by a spring-mass-damper system, similar to the system considered in a previous study [12]. Equation (4) is related to a single degree-of-freedom (1-DOF) passive system where disturbance of the system comes from the effect of engine's operation.

$$m\ddot{x} + c\dot{x} + kx = F \tag{4}$$

where

m	=	mass of the handle
c	=	damping coefficient
k	=	spring coefficient
F	=	force

Fig. 2. Rear handle of hedge trimmer

The actuator plays a main role in order to actively control the dynamic system, e.g. a pneumatic actuator like the one described in [13] and piezo actuator. Desired signal is channelled to the actuator based on what has been computed by the controller/s. In the study, a piezo actuator was proposed because it is considered to be more reliable and can give a response up to nano scale displacement. The dynamic equation of a piezo actuator can be represented as a spring-mass-damper system as follows:

$$F_a = m\ddot{x} + c\dot{x} + k_T x \tag{5}$$

where k_T is the spring stiffness. The piezo actuator stiffness is an important parameter for calculating the force generation, resonant frequency, full-system behavior, etc compared to its mass and damping parameter. a force. By taking account piezo actuator stiffness only, it becomes linear and can be represented as follows:

$$F_a = k_T x \tag{6}$$

By adding an actuator to the system, the resulting system is transformed to an active system and the equation becomes:

$$m\ddot{x} + c\dot{x} + kx - F_a = F \tag{7}$$

The parameters (c and k) of the proposed model were obtained through an impact test. It requires a few vibration equipments where stiffness and damping coefficients were calculated using basic natural frequency formula and 3dB method respectively. Two types of disurbances were introduced to the system. The first disturbance is an internal disturbance due to the engine's operation measured from Operational Deflection Shape (ODS) experiment. This internal disturbance was introduced to the system during the tuning process of the controller parameters. An appropriate external disturbance was modelled and introduced to the system after tuning process was completed in order to examine whether the system could perform robustly in the presence of external disturbance or conversely. Figs. 3 and 4 show the internal and external disturbances respectively.

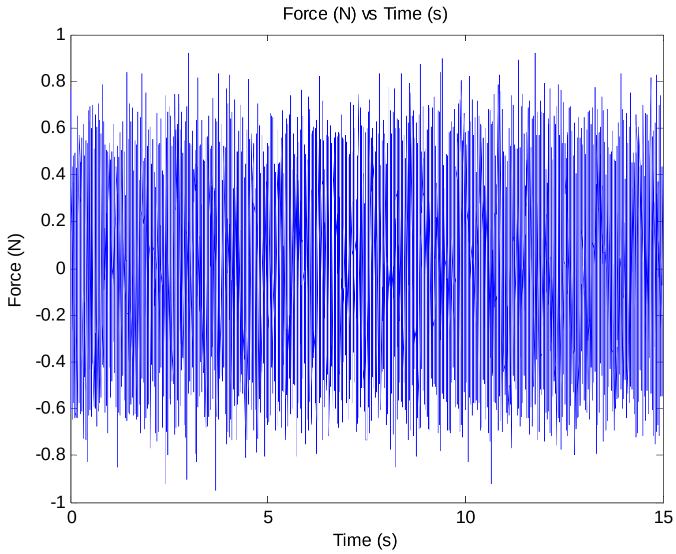

Fig. 3. Internal disturbance due to engine's operation

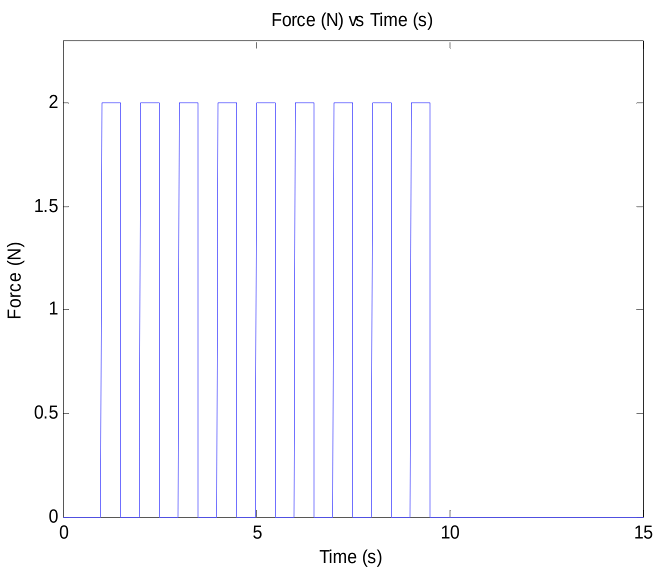

Fig. 4. External disturbance

V. SIMULATION RESULTS AND DISCUSSION

The tuning process of the controller gains is an important task that needs to be done to ensure the control action gives desirable results. The tuning was done by feel technique and a step input signal was used as a reference input signal because the results obtained are easy to evaluate and discuss in terms of overshoot, steady state error, rise and settling time etc. The forcing frequency of the system constituted from the rotational speed of the engine's operation while the piezo actuator stiffness is 350N/m. During the tuning process, a number of combinations with respect to the PID controller gains were tried. It is found that the system could not be effectively controlled by implementing the P element only (i.e., P controller) because the system would experience high vibration for any K_P given to the system. Other combinations such as PI and PD also show the same performance. The best combination of the PID-gain is determined as K_P = 200, K_D = 5 and K_I = 40. For AFC, a crude approximation technique was used to estimate the appropriate estimated/virtual mass (M*) and it is a constant value. The best estimated mass is found to be 13 kg where overshoot is very small, rise and settling time is faster while steady state error is almost zero compared to the others M* as shown in Fig. 5. Fig. 6 shows the comparative response between the PID and AFCCA schemes. For AFCCA scheme, steady state error, rise and settling time is better than the PID control scheme. AFCCA gives the smooth response with hardly any vibration occurred. From a closed-up view as shown in the box of Fig. 5, the PID scheme did not exhibit an overshoot compared to AFCCA but obviously, the vibration or small ripple occurred could not be eliminated. This signifies that by using PID control scheme alone, it will still affect operators hand-arm when operating the portable machine (e.g.. a hedge trimmer).

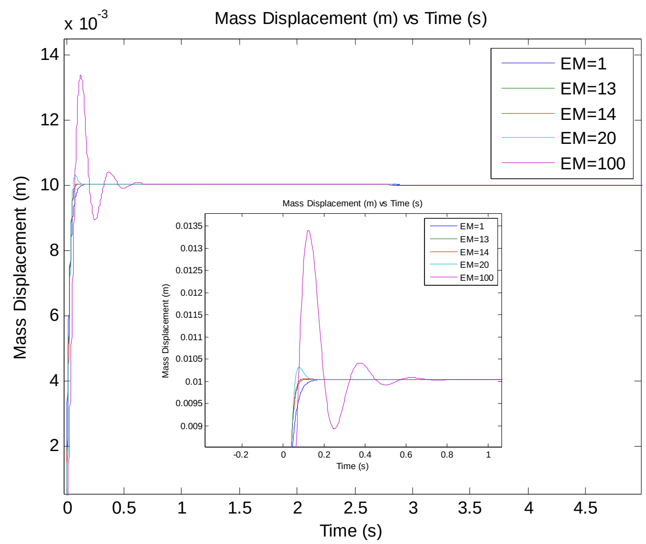

Fig. 5. Mass displacements for different values of the estimated mass (M*) with a step reference input signal

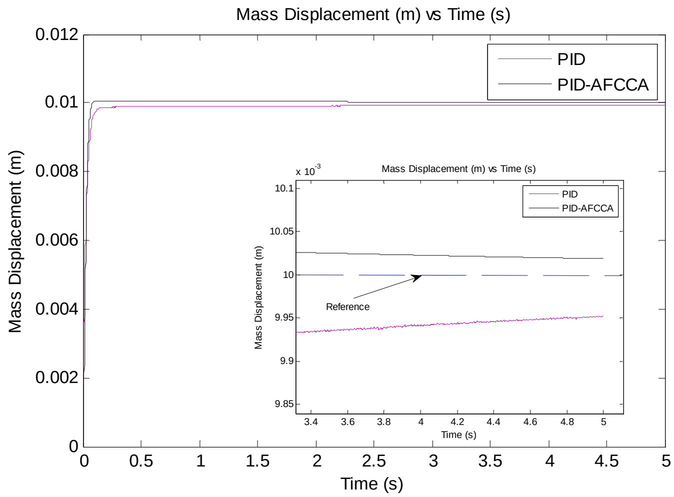

Fig. 6. Mass displacements for the PID and PID-AFCCA control schemes with a step reference input signal

The parameters obtained from tuning process as discussed above have been used as the selected parameters to investigate the AFC performance in suppressing the vibration that generated at rear handle or not. From Fig. 7, PID control scheme shows the conspicuous vibration (relatively large displacement) while the AFC-based scheme managed to suppress the vibration excellently in which a zero-level vibration is almost achieved.

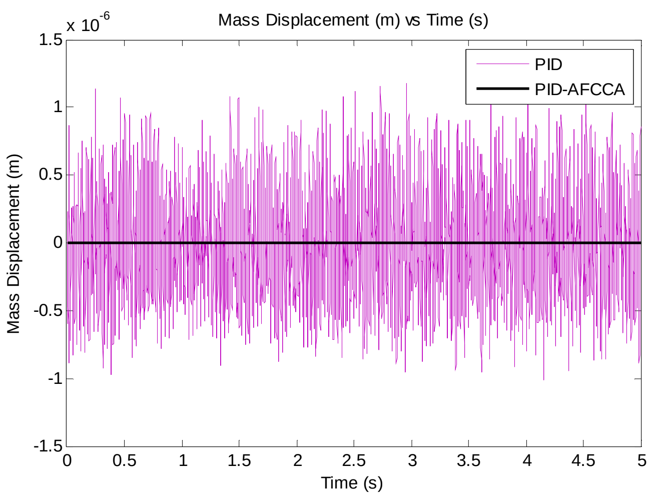

Fig. 7. Mass displacement for PID and PID-AFCCA control scheme with a zero reference input signal

When the external disturbance is introduced to the system, AFC still give the same response as shown in previous figure. AFC effectively and robustly compensates the external disturbance but not the PID control scheme. Fig. 8 shows the comparative response between the two controllers in the presence of external disturbance.

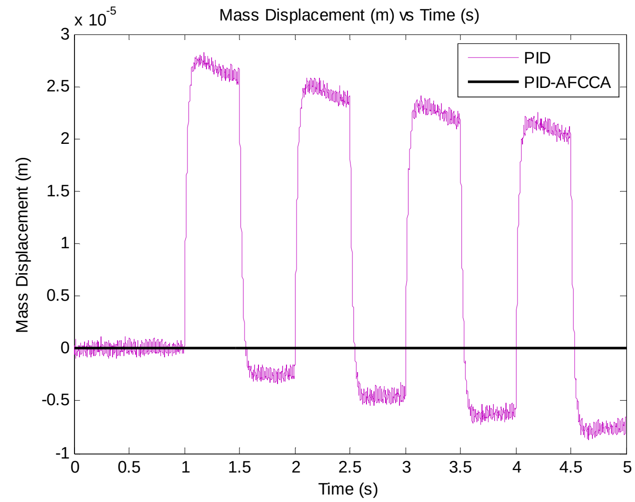

Fig. 8. Mass displacements for the PID and PID-AFCCA control schemes in the presence of external disturbance with zero reference input signal

VI. CONCLUSION

Vibration that is produced from the rear handle model of a portable handheld machine was effectively suppressed by applying the AFC strategy. From the simulation study, AFC gives the best responses compared to the conventional PID control scheme, by providing a much more robust performance. The crude approximation method in computing the estimated mass is found to be sufficient in producing the desired performance. A more comprehensive simulation study should be investigated, taking into account complex model involving higher DOF system. Other intelligent methods such as neural network, genetic algorithm and expert system can be tried in the AFC strategy to estimate the virtual mass. Development of an experimental rig using the data/results from the simulation study is also highly recommended to verify and validate the simulated findings.

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