International Review of Mechanical Engineering (I.RE.M.E.), Vol. 2, N. 2
March 2008

Praise Worthy Prize

Vibration Isolation of Hand-Held Tools to Prevent Human Hand-Arm Vibration Syndrome

A. M. Abu Hanieh

Abstract – *This paper discusses two main ideas for vibration isolation of hand-held tools to avoid health syndrome of men working on these machines. The first technique is based on using integral force feedback active control in parallel with passive damping. Simulation results of this technique have shown high performance for this technique on transmitted vibrations. The second technique is based on frequency reduction for the purpose of vibration isolation using proportional plus integral feedback active control technique. Simulation and some experimental results on a similar system are shown at the end of the paper.*

Keywords: Hand-Arm Vibrations, Vibration isolation, Integral Force Feedback, Proportional plus Integral Controller

Nomenclature

F_d	Excitation harmonic force
m	Disturbance source dirty body mass
x_d	Disturbed body displacement
M	Isolated clean body mass
x_c	Clean body displacement
k	Passive elastomer stiffness
c	Passive elastomer damping
F	Reading of the force sensor
δ	Displacement of piezoelectric actuator
k_a	Spring stiffness of the actuator
rad/s	Radians per second
Hz	Hertz
PI	Proportional plus Integral
FRF	Frequency Response Function
B&K	Bruel and Kjaer company
g	Integral gain
ga	Proportional gain
ω_n	Natural frequency
ω_c	Corner frequency

I. Introduction

Vibration isolation is becoming more and more stringent as the mechatronic systems are advancing and developing in different ground and space applications. Vibrations of rotating hand-held machines can be considered as one of the most serious problems for human workers because it can influence the nerves, blood vessels, muscles and joints of the hand and arm. To avoid the effect of these vibrations, active and passive ways can be used to damp or isolate the perturbations coming from the body of the machine and prevent them from transmitting to the worker [1]-[3].

This research discusses different effective ways to reduce the influence of vibrations transmitted to the workers hand from a rotating hand-held machine. This can be done either by attenuating the vibration modes of the system or by reducing the corner frequency to increase the isolation performance. In order to obtain high performance from the vibration isolator, the corner frequency should be as low as possible [4]. This leads the researchers to reduce the frequency by the following ways:

- Increasing the mass of the equipment needed to be isolated or to reduce the stiffness to have soft isolation mounts either in single or two stages [5].
- Using a hybrid, soft and stiff mounts, with velocity feedback using phase lag and phase lead compensators on the unity gain points [6].
- Using totally stiff mount isolator with proportional plus integral feedback compensator [7].

Increasing the mass or inertia is problematic specifically in hand-held machine applications to enable workers to hold the tool with the least effort. On the other hand, reducing the amount of stiffness leads to difficulties in stability of the systems under normal loads and reduces the effect of the working force exerted on the tool to make the job.

II. Active-Passive Isolator Using Simple Integrator

Fig. 1 shows a schematic drawing of a hand-held drilling machine where the drilling bit acts with a force F_d on the mass m of the mechanical and electrical parts of the drilling machine causing a displacement x_d.

Manuscript received and revised February 2008, accepted March 2008

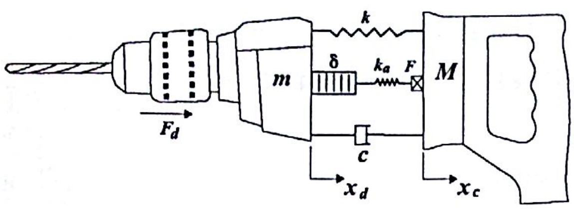

Fig. 1. Schematic drawing of active-passive drilling machine

The clean body handle of the machine represented by the mass M is also excited by this force with the displacement x_c. In order to reduce the effect of vibrations transmitted from the disturbance source to the clean handle, an active-passive isolator is suggested to be integrated in the system in the form of active-passive interface that consists of a passive elastomer (rubber) and an active strut in parallel. The passive elastomer is represented in Fig. 1 by a spring with a constant k and a damper with a damping coefficient c. The active strut here consists of a force sensor to measure the force signal F and piezoelectric actuator; the actuator is represented by the extension displacement of its piezoelectric stacks δ and the stiffness of its internal spring k_a.

II.1. Modeling and Simulation

Equations of motion governing the system are shown as follows. Disturbance equation (where s is Laplace variable):

$$ms^2x_d = k_a\left(x_d - x_c + \delta\right) + k\left(x_d - x_c\right) + cs\left(x_d - x_c\right) + F_d \qquad (1)$$

Isolated body equation:

$$Ms^2x_c = k\left(x_d - x_c\right) + cs\left(x_d - x_c\right) + F \qquad (2)$$

Force sensor equation:

$$F = k_a\left(x_d - x_c + \delta\right) \qquad (3)$$

The system has been simulated using Matlab software and Fig. 2 depicts the frequency response function between the extension of the actuator as an input and the force measured by force sensor as an output. The distance between the zero and the pole of the system shows a good controllability of the system when increasing the closed-loop gain.

Fig. 3 shows the root locus of the system with a simple integrator compensator connected to the feedback control loop.

This root locus proves that the system is unconditionally stable and damping can be introduced to the system by increasing the control gain.

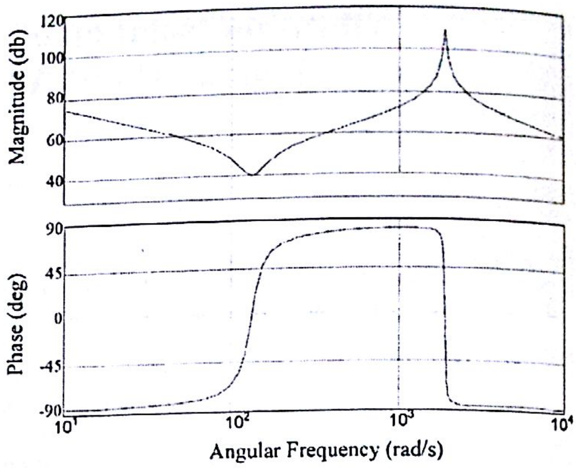

Fig. 2. Open-loop transfer function between the extension of the piezoelectric actuator and the force measured by the force sensor

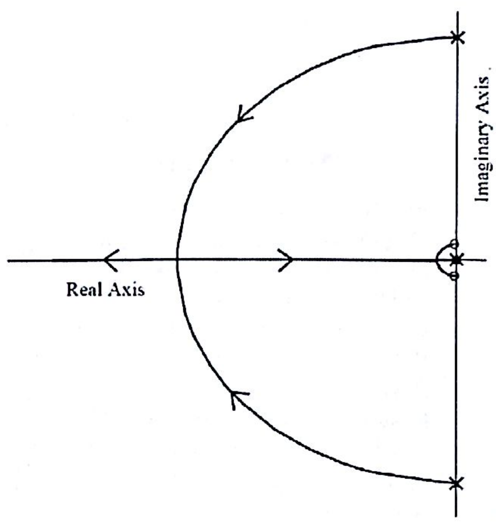

Fig. 3. Root locus of the system

In the shown situation critical damping can be reached, theoretically, by increasing the gain to a specific value but in practical situation this is difficult to apply because the gain value is restricted by the high-pass filters need to be added to the control loop to prevent low frequency signals from propagating into the system deteriorating the control performance.

Another restriction for this controller is the difficulty to push the transmission zeros of the system to low values because the system needs a minimum amount of static stiffness to withstand the static loads in the case of active system failure. This justifies the reason for adding a passive isolator (elastomer) in parallel with the active strut, although it is known that passive isolation decreases the roll-off slope of the high frequency attenuation of any vibration isolation system [8]. This controller is similar to the sky-hook damper technique discussed by Karnopp is [9] with the difference that it uses the force signal in the feedback control loop instead of the absolute acceleration.

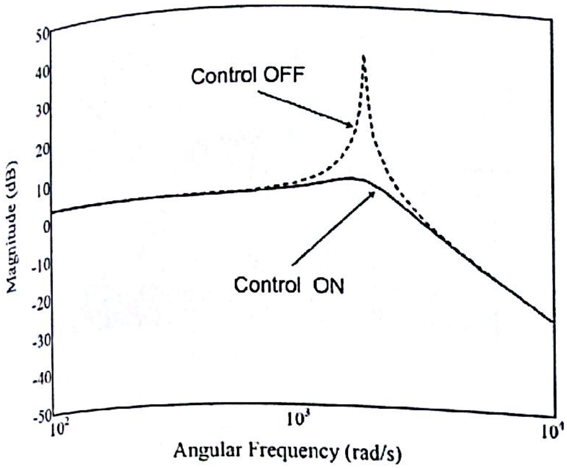

Fig. 4. Transmissibility FRF between the disturbed mass displacement and the clean mass displacement, with control (dashed line) and without control (solid line)

This technique proved robustness and stability over the traditional sky-hook technique when the clean body (payload) is flexible [4], [8].

Fig. 4 shows the transmissibility frequency response function between the displacement of the disturbed mass x_d and the displacement of the clean body mass x_c. The dashed line shows the passive isolation effect on the system where the disturbance is amplified near the corner frequency of the system while it is well isolated at high frequency.

The influence of the active damping is clear in the solid line where the overshoot at the corner frequency is reduced significantly with keeping the high frequency attenuation.

III. Single Axis Isolator Using Proportional plus Integral Compensator

Consider the schematic drawing shown in Fig. 5. This figure represents a vibration isolation interface with the disturbance source, the sensitive clean body, a force sensor and a piezoelectric actuator. PI feedback controller is used here to reduce the corner frequency and improve the response.

This system can be used as an active strut for the previously mentioned drilling machine where frequency reduction can create an adaptive structure that can change its resonance frequency instantaneously to avoid being excited when the excitation frequency approaches the resonance of the system.

This schematic drawing represents the active strut of the isolator in the drilling machine shown in Fig. where the active feedback controller is applied by acquiring the signal measured by the force sensor and feeding it back to the piezoelectric actuator after being filtered and compensated using a PI compensator.

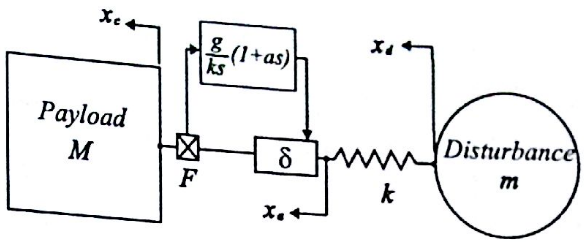

Fig. 5. Single axis piezoelectric isolator with PI feedback

The governing equation of motion for the system in Laplace transform is given by:

$$Ms^2 x_c = -ms^2 x_d = k\left(x_d - x_a\right) = F \tag{4}$$

Knowing that:

$$\delta = x_c - x_a \tag{5}$$

The open-loop FRF between the extension of the piezoelectric stack in the piezoelectric actuator δ and the output of the force sensor F reads:

$$\frac{F}{\delta} = k\frac{Mms^2}{Mms^2 + k\left(M+m\right)} \tag{6}$$

Applying a force feedback control strategy using a proportional plus integral compensator, the control law reads:

$$\delta = \frac{g}{ks}\left(1+as\right)F \tag{7}$$

or:

$$\delta = \frac{1}{k}\left(\frac{g}{s} + ga\right)F \tag{8}$$

Here ga is the proportional gain and g is the integral gain. The root locus for the closed-loop poles of this system is shown in Fig. 6; it shows that increasing the loop gain decreases the frequency of the closed-loop poles. If the proportional term is used alone, the poles will move on the imaginary axis towards the origin but this means the risk of destabilizing the system at any instant.

The use of the integral controller here pushes these poles deeper to the left half plane increasing the stability.

From the analytical calculation, the intermediate displacement x_a is:

$$x_a = \frac{sx_c + g\left(as+1\right)x_d}{s + g\left(as+1\right)} \tag{9}$$

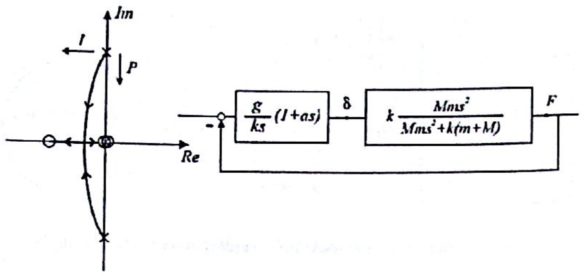

Fig. 6. Root locus of single axis piezoelectric isolator with PI feedback

From the foregoing equations, one can calculate the transmissibility FRF between the disturbance displacement and the payload displacement, and is equal to:

$$\frac{x_c}{x_d}=\frac{1}{s^2\left[(1+ga)/\omega_n^2\right]+s\left[g/\omega_n^2\right]+1} \tag{10}$$

where ω_n is the natural frequency of the system.

This implies that the corner frequency ω_c of the system is determined by the proportional gain of the compensator:

$$\frac{1}{\omega_c^2}=\frac{1+ga}{\omega_n^2} \tag{11}$$

The damping of the system is determined by the gain g of the compensator

$$\frac{g}{\omega_n^2}=\frac{2\xi}{\omega_c} \tag{12}$$

If ω_n is much larger than ω_c then

$$\frac{\omega_n^2}{\omega_c^2}\approx ga$$
$$\frac{ga}{k}=\frac{1}{M\omega_c^2}=\frac{1}{k^*} \tag{13}$$

Here $(1/k^*)$ is the closed-loop flexibility of the system and is proportional to the gain. From the foregoing analysis, one can see that the closed-loop stiffness of the system is inversely proportional to the control gain; in other words, if one increases the proportional gain, the stiffness is reduced.

III.1. Simulation Results

The system shown in Fig. 5 has been simulated using Matlab software. The simulation was based on the previous analysis of the system taking the mass m as 1.1 kg, the mass M as 1.7 kg and the stiffness of the piezoelectric actuator k as 1×10^7 N/m.

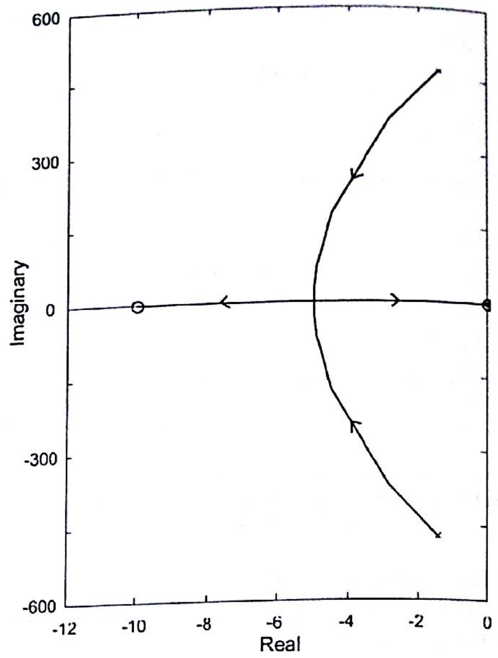

Fig. 7. Root locus prediction

Fig. 7 depicts the root locus prediction that the poles should follow when the control loop is closed. The root locus shows that the poles will remain in the left hand side of the s-plane which means that the system is unconditionally stable. On the other hand, the loop of the plot is not moving in a circular shape which means that when the gain of the controller is increased, the distance between the pole and the origin will be shorter leading to slow down the poles or to reduce the frequency of the corresponding mode.

Theoretically, the poles will move till reaching critical damping but in real time work this is impossible as will be shown in the experimental verification part. Fig. 8 shows the transmissibility FRF (x_c / x_d) before and after stiffness reduction using PI controller.

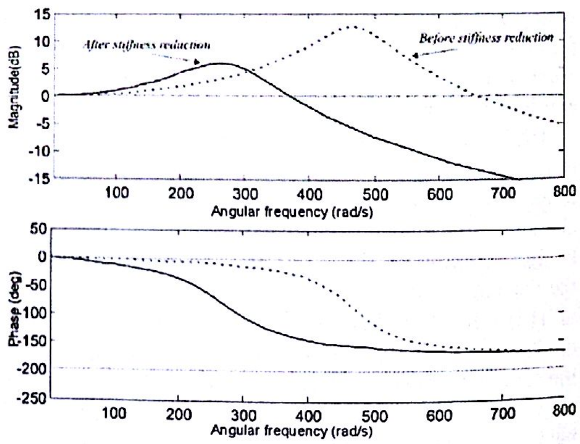

Fig. 8. Predicted transmissibility from simulation results

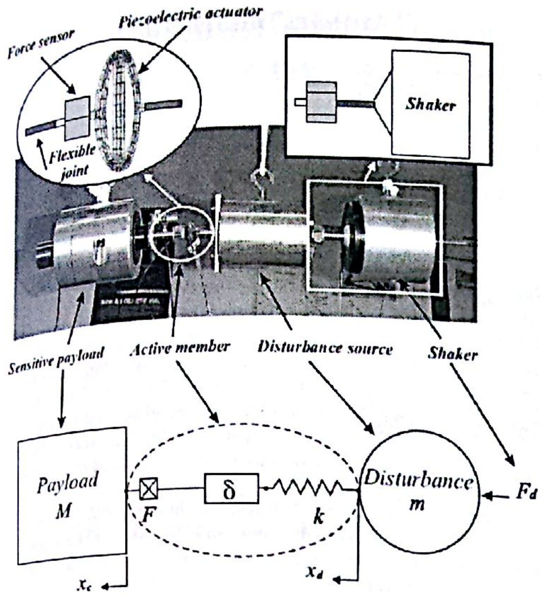

Fig. 9. Experimental setup of single axis piezoelectric isolator

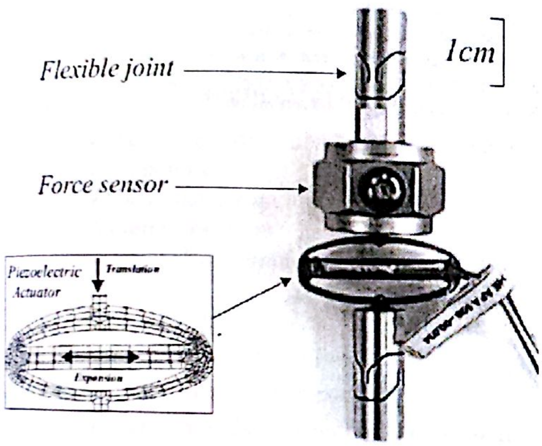

Fig. 10. Active member consists of piezoelectric actuator, a force sensor and two flexible joints

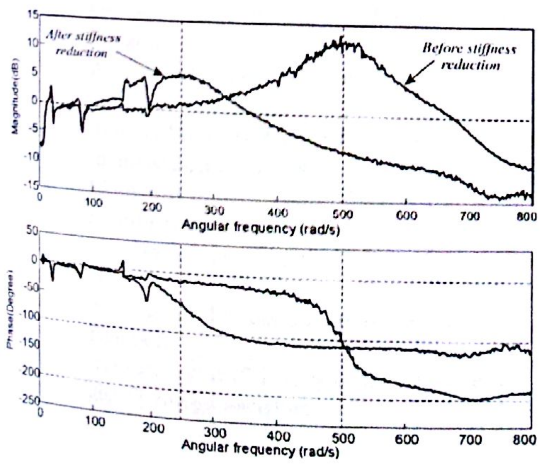

Fig. 11. Experimental transmissibility (x_c / x_d)

III.2. Experimental Verification

Discussion in this part will be concentrated on a practical setup based on using the PI controller to reduce the stiffness of the structure as discussed theoretically in the previous section. Consider the experimental set-up shown in Fig. 9. It consists of two masses connected to each other by an active member.

The active member (see Fig. 10) consists of a CEDRAT APA50 piezoelectric actuator, a B&K 8200 piezoelectric force sensor with charge output and two flexible joints to avoid the side effect of the lateral modes of the system by decoupling these modes mechanically from the axial studied mode.

Using an external shaker, the system has been excited with a random signal ranging from 1 to 800 Hz and the transmissibility FRF between the displacement of the disturbance source body and that of the clean body mass is measured (see Fig. 11). The resonance of the system is found at 500 Hz. A feedback system with a PI control law is applied to the system and the same FRF measured again. Fig. 12 shows the two measured FRFs: the open-loop (before stiffness reduction) and the closed-loop (after stiffness reduction). The natural frequency of the system has been reduced by 50%; from 500 Hz to 250 Hz. The maximum reduction has been obtained by increasing the gain of the proportional part of the compensator, but this leads to the risk of walking along the imaginary axis which can lead to instability if the surrounding conditions change slightly. Thus, there is a need to increase the integral gain too at the same time to increase the stability margin of the system and to reduce the overshoot in the resonance vicinity.

IV. Discussion and Conclusions

The foregoing text discussed two techniques of active vibration isolation for the hand-held rotating tools to avoid the syndrome and bad effect of these vibrations on the human workers. The first technique is based on using integral force feedback technique which is a type of sky-hook damper but with force feedback. This type of control proved having a high authority on single degree of freedom systems by reducing, significantly, the amplification near the natural frequency of the system but it does not isolate signals at low frequency. Adding proportional term to the controller in addition to the integrator has a great influence on reducing the corner frequency of the isolator which enables the isolator to have higher impact on the low frequency vibrations as shown in the experimental work. The natural frequency of the system has been reduced by 50%; from 500 Hz to 250 Hz. The maximum reduction has been obtained by increasing the gain of the proportional part of the compensator, but this leads to the risk of instability. Therefore, increasing the integral gain of the controller results in modifying the stability margin of the system.

Acknowledgement

The author would like to thank the Active Structures Laboratory in the Free University of Brussels for supporting the experimental work.

References

[1] Sanna Pöyhönenl, Pedro Jover and Heikki Hyötyniemil, *Independent component analysis of vibrations for fault Diagnosis of an induction motor*, Proceedings of the IASTED International Conference CIRCUITS, SIGNALS, AND SYSTEMS, Cancun, Mexico, May 19-21, 2003.

[2] J. Kiciński, The flutter effect in rotating machines, *Bulletin of the polish academy of sciences Technical sciences*, Vol. 52, No. 3, 2004.

[3] Ihab Wattar, Wassim Hafez and Zhiqiang Gao, *Model-Based Diagnosis of Chaotic Vibration Signals*, IEEE – IEC 1999 conference, 1999.

[4] A. Preumont. *Vibration Control of Active Structures*. 2nd ed. Netherlands: Kluwer Academic publishers; 2002.

[5] P. Nelson, An Active Vibration Isolation System for Inertial reference and Precision Measurement, *Rev. Sci.Instrum, American Institute of Physics*, Vol. 62, No. 9, Sep. 1991.

[6] TMC, *Technical Background, Technical report*, Technical Manufacturing corporation.

[7] A. Abu Hanieh. *Active Isolation and Damping of Vibrations via Stewart Platform*. PhD thesis, Brussels: Free University of Brussels; 2003.

[8] A. Preumont, M. Horodinca, I. Romanesco, B. de Marneffe, M. Avraam, A. Deraemaeker, F. Bossens and A. Abu Hanieh, A six-axis single-stage active vibration isolator based on Stewart platform, *Journal of Sound and Vibrations*, Volume 300, Issues 3-5, March 2007, pp 644-661.

[9] D.C. Karnopp, A.K. Trikha, Comparative study of optimization techniques for shock and vibration isolation, *Transactions of the ASME, Journal of Engineering for Industry*, Series B 91, 1969, 1128–1132.

Authors' information

Mechanical Engineering Department,
Faculty of Engineering,
Birzeit University,
P.O Box 14, Birzeit,
Palestine.
Tel./Fax: +97022982115 / +97022982984,
E-mail: ahanieh@birzeit.edu

Ahmed M. Abu Hanieh was born in Palestine on the 21st of April 1966. He earned the first degree in Mechanical Engineering from Birzeit University in Palestine in the year 1993 and obtained the Masters and PhD degrees in Mechatronics from the Free University of Brussels (ULB) in Belgium in the years 2001 and 2003, respectively. The major field of study and research is active control of vibrations based on mechanical engineering background.

He published four reviewed articles in international journals like Journal of Sound and Vibrations and Journal de Phisique and participated in the proceedings of more than 15 international specialized conferences. He worked and still holding research in the field of active control and isolation of vibrations for mechanical and mechtronic systems.

Dr. Abu Hanieh is Assistant professor at the Mechanical Engineering Department (Birzeit University) and he is working as a consultant for the National Palestinian Committee of Science and Technology since 2006. He reviewed several specialized papers for the following Journals: Control Engineering Practice, Robotics and Autonomous Systems and the IEEE Transactions on Control Systems Technology.