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Vibration Suppression of Milling Machine Tools by the Inerter

Fu-Cheng Wang, *Member, IEEE*, and Chung-Hao Lee

***Abstract*—This paper discusses vibration suppression of milling machine tools by the inerter. There are two traditional methods to repress the cutting vibration of milling machines: the passive and the active approaches. The former normally applies mechanical networks consisting of the masses, dampers and springs. However, the mass element is not a genuine two-terminal network element and might restrict the achievable performance of the mechanical networks. Therefore, the inerter is invented to substitute the mass element. This paper investigates the benefits of the inerter in improving the vibration responses of milling machines. First, we conduct identification experiments to obtain the model of a milling machine. Second, we design three passive suspension layouts to illustrate the benefits of the inerter in suppressing the cutting vibration. Last, we conduct experiments to demonstrate the effectiveness of the inerter in improving the manufacturing performance of the milling machine.**

I. INTRODUCTION

Milling machines are widely used in traditional industry, aerospace industry, automobile industry, biomedical engineering, electrical industry, and so on. The common requirements for machine production include dimensional accuracy, surface finish, and high material remove rate (MRR) in the manufacturing process. Generally speaking, increasing the spindle speed that can improve the performance of surface finish because feed pre tooth decreases. On the other hand, increasing cutting feed and the depth can achieve high MRR but usually result in bad surface finish because of vibration problems. Therefore, many researches applies the passive and active techniques to solve cutting vibration problems, as discussed in [1]. The active techniques usually apply sensors to detect the vibration signals and controller to operate actuators accordingly. For example, Munoa et al. [2] proposed a biaxial inertial actuator consisting linear motors and voice coils to the milling spindle. Rashid and Nicolescu [3] applied piezo-transducers (PZTs) to set palletized work holding systems to repress vibration in cutting processes. On the other hand, the passive techniques aimed to reduce system vibration by changing system dynamics. For instance, Wang and Liu [4] investigated the tool materials can cause the amplitude of vibration. Aguiar et al. [5] used different slenderness-ratio tools to discuss the influences of tools in reducing vibration. Many studies applied mechanical network layouts, which consisted of the masses, dampers and springs, to reduce system vibration. For instance, Yang et al. [6] proposes a fourth-order tuned-mass damper (TMD) system to repress the vibration in machining processes. Rashid and Nicolescu [7] proposed tuned viscoelastic dampers to predict the real cutting vibration by analyzing the amplitude of frequency response function (FRF).

The inerter was invented by observing the imperfect analogy between the mechanical and electrical systems. Because the mass element is not real two-terminal network element, the achievable performance of traditional mechanical systems is inherently restricted [8]. The inerter is a genuine two-terminal element and is proposed to substitute the mass element in the passive networks. It has been successfully applied to several mechanical systems, such as vehicle suspensions [9], motorcycle steering [10], train suspension lateral control [11, 12], optical tables [13], and building [14]. This paper extends these ideas to discuss the potential performance improvement of milling machine tools by the inerter.

The paper is arranged as follows: Section II introduces a milling machine and obtains the model by experiments. Section III applies the model was applied to discuss the effects of three suspension layouts consisting of the inerter, damper, and spring. Section IV defines four indexes to evaluate the cutting performance by the suspension layouts. We also conduct experiments to verify the performance improvement by the inerter. Last, we draw conclusions in Section V.

II. SYSTEM DESCRIPTION AND IDENTIFICATION

We consider the milling machine, as shown in Fig. 1, which is a computer numerical control (CNC) planer type milling machine. The maximum travels are 400 mm, 400 mm, and 80 mm in the X-, Y-, and Z- axes, respectively. The maximum milling cutter diameter is 10 mm. The suspension base is designed to reduce vibration and to implement suspension networks. The workpiece is fixed on the base while the suspension elements can be connected to the base to modify system dynamics. We implement a microphone [15] to detect the cutting air press, an acceleration [16] to measure the cutting vibration.

We apply the impact hammer model testing method to obtain the vibration model of the milling machine. As shown in Fig. 2(a), we use a hammer to hit the test specimen and record the hitting impulse signal and the acceleration signal for system identification. The frequency response function (FRF) of the system is shown in Fig. 2 (b), which has two peaks at about 600 Hz and 3700 Hz. Therefore, we can simplify the system as a fourth-order model, as illustrated in

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Fu-Cheng Wang is with the Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan (phone: +886-2-3366-2680; e-mail: fcw@ntu.edu.tw).

Chung-Hao Lee is with the Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan (e-mail: dennisli8246@gmail.com).

Fig. 2(c), where m_w is the upper base with a displacement of y_w, m_b is lower base with a displacement of y_b, y_r is the displacement of the ground, $Q(s)$ is the suspension connecting the ground and m_w, F_w is the cutting force when the cutting tool contacts the workpiece.

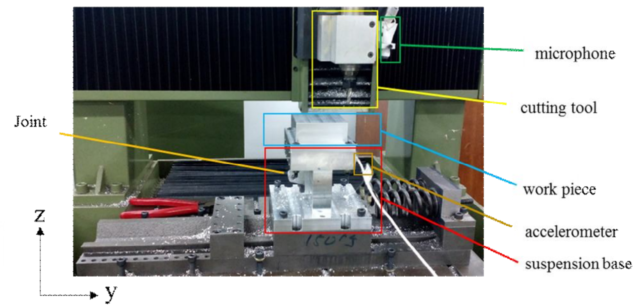
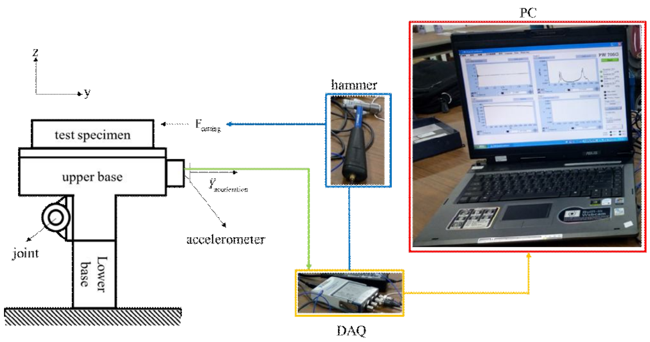
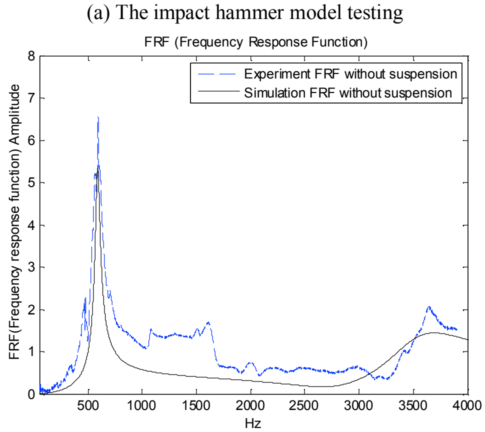


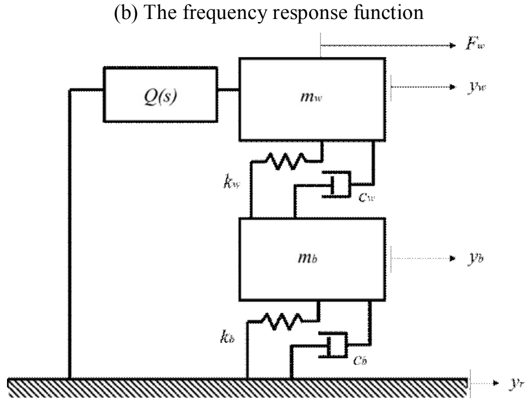
Fig. 1 The experimental setup



(a) The impact hammer model testing



(b) The frequency response function



(c) The approximated model

Fig. 2 System identification

The masses of m_w and m_b can be calculated from the dimension of the test specimen, the upper base, and the lower base. Then we can estimate the values of the dampers and springs by minimizing the errors between the experimental and theoretical responses at the concerned frequency ranges, as in the following:

$$\{k_w,c_w,k_b,c_b\}=\arg\min\left(\sum_{\omega_i=500\text{Hz}}^{700\text{Hz}}\left|FRF_{\exp}(\omega_i)-FRF_{sim}(\omega_i)\right|^2+\sum_{\omega_i=3500\text{Hz}}^{4000\text{Hz}}\left|FRF_{\exp}(\omega_i)-FRF_{\exp}(\omega_i)\right|^2\right)$$

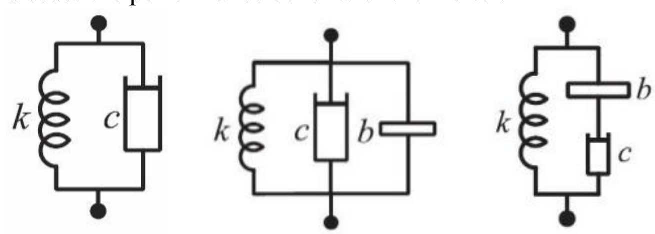
where $FRF_{\exp}$ and FRF_{sim} represent the experimental and the theoretical FRF, respectively. The derived system parameters are illustrated in Table I.

Table. 1 the values of parameters

Parameters	Value
m_w (kg)	1.5
c_w (kN • s/m)	2.953
k_w (kN/m)	31.813×10^4
m_b (kg)	1.2
c_b (kN • s/m)	0.775
k_b (kN/m)	3.819×10^4

III. Suppression Layouts and Design

From Fig. 2(c), it is possible to adjust system dynamics and to improve system performance by tuning the suspension $Q(s)$. In this paper, we consider three suspension layouts, as shown in Fig. 3, with the corresponding admittances to discuss the performance benefits of the inerter.

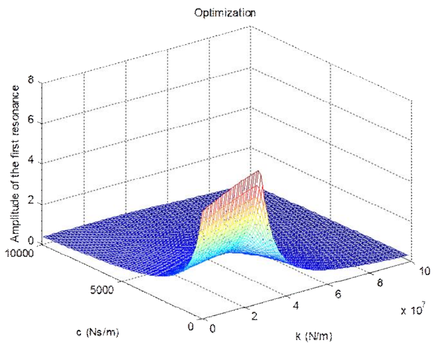


(a) S1: $Q_{S1}(s)=c+\frac{k}{s}$ (b) S2: $Q_{S2}(s)=bs+c+\frac{k}{s}$ (c) S3: $Q_{S3}(s)=\frac{bcs}{bs+c}+\frac{k}{s}$

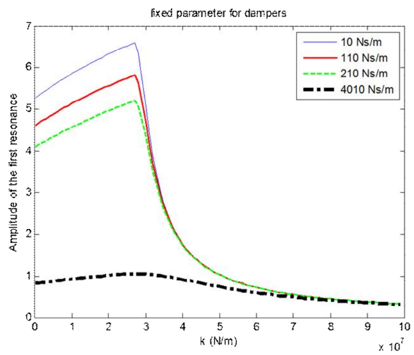
Fig. 3 Suspension layouts

A. S1 optimization

Setting $Q(s)=Q_{S1}(s)$, the magnitudes of the first FRF resonance peak with corresponding values of c and k are shown in Fig. 4(a). If we fix the damping rate c, the results are illustrated in Fig. 4(b) where the first resonance peak decreases quickly when k is greater than 3.7×10^7 N/m. Similarly, if we fix the value of k, the first resonance peak can also decrease as c increases.



(a)

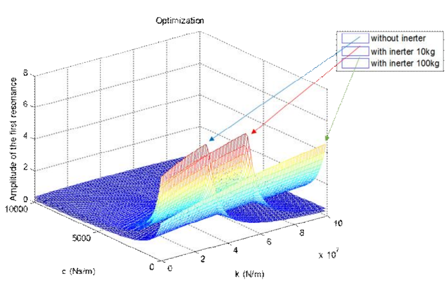


(b)

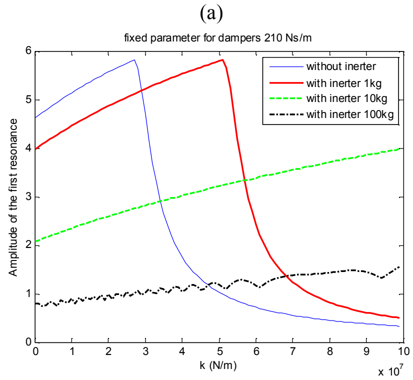
Fig. 4 S1 optimization

B. S2 optimization

Setting $Q(s) = Q_{S2}(s)$, the magnitudes of the first FRF resonance peak are shown in Fig. 5(a), where we set b=0, 10, and 100kg to compare their influences. The results indicate that the first resonance peaks usually decrease when adding the inerter. For example, setting b=10 or 100kg can effectively reduce the first resonance peaks, as shown in Fig. 5(b). In addition, the optimal peak resonance is improved by S2 compared with that by S1.



(a)



(b)

Fig. 5 S2 optimization

C. S3 optimization

Setting $Q(s) = Q_{S3}(s)$, the first resonance peak of system FRF is shown in Fig. 6. We note that the curves with different settings of b are almost the same, e.g. the inerter in the series layout does not have significant effects on the resonance peak. Therefore, we will conduct the experiments and focus the discussion on the S2 layout.

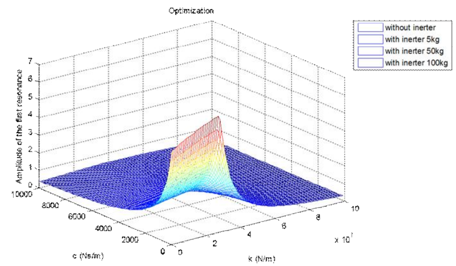


Fig. 6 S3 optimization

D. Experimental verification

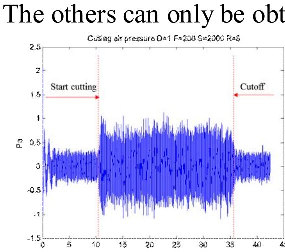
We construct an experiment platform, as shown in Fig. 1, to verify the effects of different suspension elements on system performance. We choose three springs, three dampers, and two inerters, as illustrated in Table II, to test the performance improvement by these individual elements. The coefficients of these elements are obtained by experiments. The experimental responses are similar to the simulation, but with larger variation that might be caused by model simplification and disturbances. However, the simplified model can basically predict system responses. The quantitative comparison will be discussed in the next section.

TABLE II. SELECTED SUSPENSION ELEMENTS

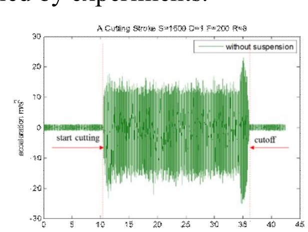
Springs (N/m)			
	k_1=94587.09	k_2=223714.8	k_3=367012.4
Dampers (Ns/m)			
	c_1=587	c_2=619	c_3=651
Inerters (kg)			
	b_1=231.5	b_2=411.8	

IV. PERFORMANCE IMPROVEMENT BY SUSPENSIONS

We define the following four performance indexes to evaluate the effects on vibration suppression by the suspension layouts: (1) 2-norm of the acceleration signals (see Fig. 7(a)); (2) 2-norm of the air pressure signals (see Fig. 7(b)); (3) surface roughness; (4) magnitudes of the first resonance. Note that the simulation model can only derives the fourth index to compare with the experimental results. The others can only be obtained by experiments.



(a) Acceleration



(b) Air pressure

Fig. 7 Air pressure and acceleration signals

A. *Individual effects of springs, dampers, and inerters*

We first evaluate the individual effects of the springs, dampers, and inerters, as illustrated in Table III. First, adding these elements can help improve system performance. Second, the improvement tends to be more notable with larger values in these cases. Therefore, further improvement is possible by elements which are not tested in the present experiments.

TABLE III. INDIVIDUAL EFFECTS OF SUSPENSION EXPERIMENTS

Q(s)	Spring effects			
	-	k_1	k_2	k_3
Acceleration(m/s^2)	1655.1	1524.0	1279.7	1263.8
(% improvement)	-	(7.92%)	(22.68%)	(23.64 %)
Air press (Pa)	111.9185	113.8408	95.0944	97.1910
(% improvement)	-	(-1.72%)	(15.03%)	(13.16%)
Roughness (μm)	3.8138	3.4372	3.1422	3.3342
(% improvement)	-	(9.88%)	(17.61%)	(12.58%)
|resonant| (sim)	5.3495	5.2994	4.9869	4.7893
(% improvement)	-	(0.94%)	(6.78%)	(10.47%)
|resonant| (exp)	**5.4**	**4.7**	**4.1**	**3.6**
(% improvement)	**-**	**(12.96)**	**(24%)**	**(33.33%)**
	Damper effects			
	-	c_1	c_2	c_3
Acceleration(m/s^2)	1592.2	1536.6	1399.9	1390.1
(% improvement)	-	(3.49%)	(12.08%)	(12.7%)
Air press (Pa)	120.2322	115.7427	101.5096	111.5212
(% improvement)	-	(3.73%)	(15.57%)	(7.25%)
Roughness (μm)	2.899	2.5401	2.24924	2.0614
(% improvement)	-	(12.38%)	(22.41%)	(28.89%)
|resonant| (sim)	5.3495	2.8955	2.8256	2.7585
(% improvement)	-	(45.87%)	(47.18%)	(48.43%)
|resonant| (exp)	**5.5**	**4.3**	**3.7**	**3.2**
(% improvement)	**-**	**(21.81%)**	**(32.72%)**	**(41.81%)**
	Inerter effects			
	-	b_1	b_2	
Acceleration(m/s^2)	1560.0	1444.4	1291.8	
(% improvement)	-	(7.41%)	(17.19%)	
Air press (Pa)	117.1645	113.7163	107.3257	
(% improvement)	-	(2.94%)	(8.40%)	
Roughness (μm)	2.899	2.429	2.2848	
(% improvement)	-	(16.21)	(21.18)	
|resonant| (sim)	5.3495	0.446	0.2888	
(% improvement)	-	(91.66%)	(94.60%)	
|resonant| (exp)	**5.581**	**4.48**	**3.971**	
(% improvement)	**-**	**(19.72%)**	**(28.84%)**	

B. *Combining effects of suspension layouts*

We now consider the combining effects of the suspension layouts: S1 by one spring and one damper; S2 by one spring, one damper, and one inerter. The improvement in the resonance peak by simulation are illustrated in Table IV and V, respectively. First, the combination of components can further improve the performance, compared with the results in Table III. Second, adding the inerter can greatly reduce the resonance peak.

Table IV. Improvement by the S1 Layout (simulation)

|resonant| (sim) (% improvement)	k_1	k_2	k_3
c_1	46.17%	47.44%	48.66%
c_2	52.45%	53.45%	54.41%
c_3	56.02%	56.87%	57.69%

TABLE V. IMPROVEMENT BY THE S2 LAYOUT (SIMULATION)

|resonant| (sim) (% improvement)	no spring	k_1	k_2	k_3
c_1 & b_1	93.81%	94.89%	95.03%	95.46%
c_2 & b_1	94.36%	94.44%	95.40%	95.91%
c_3 & b_1	94.49%	94.57%	95.49%	95.97%
c_1 & b_2	95.20%	95.68%	96.76%	97.34%
c_2 & b_2	96.64%	96.02%	96.94%	97.45%
c_3 & b_2	96.75%	96.11%	97.45%	97.48%

C. *Experimental verification of S1 and S2*

The experiment set is shown in Fig. 8, where the suspension have three joints to connect three elements.

1. S1 layout: we choose two combination c_1+k_1 and c_3+k_3 from Table IV for experimental demonstration, because they represent the worst and the best cases of S1 using the available elements. The experimental results are illustrated in Table VI, where S1 with c_3 + k_3 does have better effects than with c_1+k_1, as predicted by the simulation (see Table IV). Second, the percentage improvement is better by c_1+k_1 than the simulation, but it is worse by c_3+k_3. These might be caused by model simplification and system disturbances.

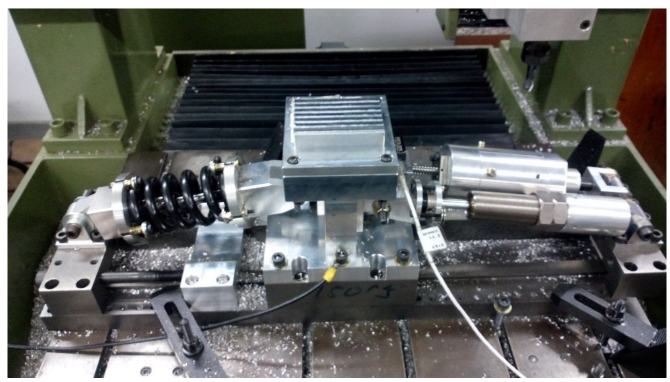


Fig. 8 Experiments with S1 and S2 layouts

TABLE. VI EXPERIMENTS WITH THE S1 LAYOUT

Q(s)	No suspension	c_1 +k_1	c_3 +k_3
Acceleration	1592.2	1297.6	1194.9
(% improvement)	-	(18.51%)	(24.95%)
Air press (Pa)	104.4520	88.2520	81.5769
(% improvement)	-	(15.51%)	(21.90%)
Roughness (μm)	2.899	2.5035	2.23
(% improvement)	-	(13.64%)	(23.07%)
resonant| (exp)	5.4	2.697	2.439
(% improvement)	-	50.05%	50.84%

For experiments with the S2 layout, we choose three cases: b_1+c_1, b_2+c_3, and b_2+c_3+k_3, which respectively indicate the worst, the medium, and the best performance (see Table IV). The experimental results are shown in Table VII. The performance ranking is the same as in the simulation, where b_2+c_3+k_3 gives the best performance. Though the percentage improvements are not as great as the simulation, we can predict the best suspension layouts for reducing vibration and improving manufacturing performance.

TABLE. VII EXPERIMENTS WITH THE S2 LAYOUT

Q(s)	No suspension	$b_1 + c_1$	$b_2 + c_3$	$b_2 + c_3 + k_3$
Acceleration	1592.2	1335.7	1143.8	664.5655
(% improv.)	-	(16.11%)	(28.16%)	(58.26%)
Air press (Pa)	104.4520	89.1276	80.3212	58.9747
(% improv.)	-	(14.67%)	(23.10%)	(53.325%)
Roughness (μm)	2.899	2.5035	2.0408	1.733
(% improv.)	-	(13.64%)	(29.60%)	(40.20%)
resonant| (exp)	5.4	3.272	2.62	2.144
(% improv.)		39.4%	51.48%	60.30%

V. CONCLUSION

This paper has demonstrated the improvement on suppression vibration of milling machine tools employing the inerter. First, we applied the impact hammer testing to obtain a two-mass model to represent the system dynamics. Second, we applied the model to discuss the performance benefits by three passive suspension layouts employing the inerter, the damper and the spring. The results indicated that the parallel inerter layout S2 is most beneficial in reducing system vibration. Last, we conducted experiments to verify the performance improvement. In the future, more complex inerter layouts can be considered to explore further potentials by the inerter.

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