Passive voice coil feedback control of closed-box subwoofer systems

C-Y Chen[1], G T-C Chiu[2], C-C Cheng*[1] and H Peng[3]
[1]Department of Mechanical Engineering, National Sun Yat-Sen University, Kaohsiung, Taiwan
[2]School of Mechanical Engineering, Purdue University, West Lafayette, Indiana, USA
[3]Department of Mechanical Engineering and Applied Mechanics, University of Michigan, Ann Arbor, Michigan, USA

Abstract: In this paper, the feasibility of using a voice coil back-electromotive voltage (back-e.m.f.) signal as feedback information for closed-loop control was investigated. A dual voice coil closed-box subwoofer system is used to demonstrate the effectiveness of back-e.m.f. feedback control. A second-order filter is developed to extract the velocity information from the coil back-e.m.f. signal. A proportional-plus-derivative (PD) controller is well suited for reducing the harmonic distortion and extending the subwoofer bass response. Experimental results verified that the proposed control scheme effectively extended the bass response of the subwoofer system by one octave and at the same time reduced harmonic distortion by more than 6 dB. The proposed feedback and control scheme can be easily implemented using inexpensive analogue components, which can further reduce the cost and complexity of the system.

Keywords: bandwidth, feedback control, harmonic distortion, loudspeaker, passive coil

NOTATION

$A(s)$	Laplace transform of cone acceleration, $\ddot{x}(t)$
b	effective damping coefficient of the subwoofer
c	forcing factor or back-e.m.f. constant of the electromechanical system
$C(s)$	transfer function of the controller
C_m	compliance of the speaker box (m/N)
d	feedforward path gain
F_{CB}	resonant frequency for the closed-box loudspeaker
F_S	resonant frequency for the open-air loudspeaker
$G_p(s)$	transfer function from input voltage to passive coil terminal voltage
$G_s(s)$	transfer function from input voltage to cone acceleration
i_1	active coil current
i_2	passive coil current
k	equivalent suspension stiffness of the subwoofer
K_d	differential gain of the proportional-plus-derivative (PD) controller
$K_{d.c.}$	d.c. gain of the filter
K_p	proportional gain of the PD controller
K_v	d.c. gain offset of the dominating system
L	self-inductance of the two coils for the subwoofer
m	equivalent mass of the subwoofer
M	mutual inductance of the two coils for the subwoofer
R	d.c. voice coil resistance for the subwoofer
S_D	effective piston area of the cone (m^2)
$U(s)$	Laplace transform of the input voltage $U(t)$
$U(t)$	applied voltage supplied by the audio amplifier
V	enclosed air volume (m^3)
V_{AS}	equivalent volume of compliance, which specifies a volume of air having the same compliance as the suspension system of the loudspeaker
V_B	volume of the sealed enclosure
$V_s(s)$	Laplace transform of the passive coil terminal voltage $V_s(t)$
$V_s(t)$	voltage across the passive coil
$x(t)$	moving coil position of the subwoofer
ξ	damping ratio of the dominating system
τ	time constant of the high-frequency first-order dynamics
ω_n	resonant frequency of the dominating system

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**Corresponding author: Department of Mechanical Engineering, National Sun Yat-Sen University, Kaohsiung, 80424 Taiwan.*

1 INTRODUCTION

Although the quality of audio electronics has advanced dramatically, the loudspeaker system is still the weaker link in reproducing live audio. One of the main issues in reproducing a realistic audio experience is the bass response of the speaker. In this paper, a report will be given of a novel and cost effective closed-loop control approach to improve the bass response of a closed-box subwoofer system by extending its range as well as reducing harmonic distortions.

The lowest tone that can be generated by a musical instrument and can be heard by the human ear is around 16 Hz [1]. A reasonable subwoofer loudspeaker should not miss that threshold value by more than an octave or two (i.e. 32–48 Hz). To model speaker systems and predict their response, the Theile–Small model [2] is almost exclusively used by manufacturers. This model uses a high-pass filter model to predict the loudspeaker response. In fact, the response curve of an open-air loudspeaker and a closed-box loudspeaker system can be modelled as a third-order high-pass filter. The bass response is dictated by the low resonant frequency induced by the speaker coil mass and the compliance of the surrounding suspension structure (see Fig. 1). Although a subwoofer may have a desirable low-frequency response (desirable low resonant frequency) when it is measured in open-air, mounting it in a sealed enclosure raises the resonant frequency owing to the added stiffness of the enclosed air. However, some kind of enclosure is necessary. Since the pressure wave from the front of the speaker cone is in antiphase of that in the rear of the speaker, for waves with long wavelength (as in bass sounds), two pressure waves tend to diffract around the woofer and cancel each other out. In essence, bass sounds are likely to cause an acoustic short circuit around the open-air woofer. To extend the bass response of closed-box woofers, an obvious thought is to make the resonant frequency in open-air lower to compensate for the increase caused by the speaker enclosure. However, the efficiency of a speaker is proportional to the cube of its resonant frequency. Hence, there is always the trade-off between efficiency and response bandwidth.

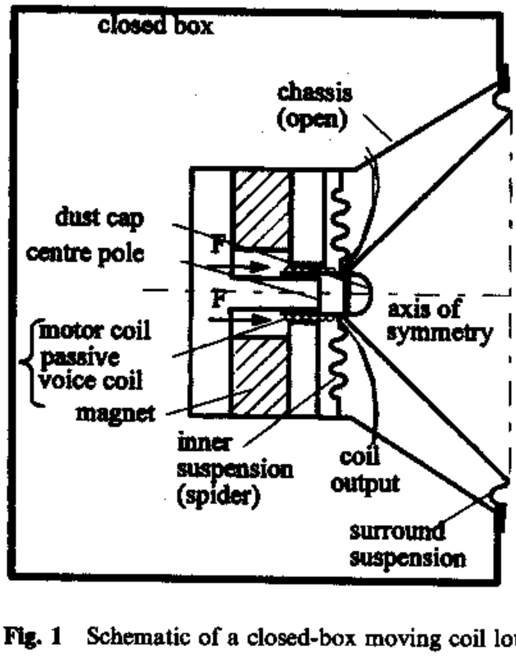

Fig. 1 Schematic of a closed-box moving coil loudspeaker

Non-linear distortion is another major performance specification for loudspeaker systems. Total harmonic distortion (THD) is the result of the non-linearities in the loudspeaker structure. Electromagnetic non-linearities in the force generation of the voice coil actuator and the non-linearities in the speaker cone suspensions (spider) are principal contributors to speaker distortion. Careful mechanical design can minimize the distortion induced by the electromagnetic structures. In a closed-box loudspeaker system, the compliance of the enclosure generally dominates the speaker compliance. This tends to reduce the effect of compliance-related non-linearities, since the compliance of the enclosure is, in general, 'more linear' than the speaker compliance. However, even the most carefully designed and manufactured loudspeakers still fall short of low distortion operation much of the time.

To improve the subwoofer bass response and reduce harmonic distortion, both open-loop and closed-loop compensations have been pursued. The open-loop approach typically employs a bandpass or high-pass filter to equalize the speaker response. To achieve this, a precise model of the loudspeaker system is needed. Over the years, much work has been done in developing a speaker model [3–6]. Once a precise speaker and amplifier model has been developed, bandpass and high-pass filters can be designed to equalize the subwoofer response [7]. One difficulty in the open-loop approaches is the availability of an accurate set of speaker, amplifier and enclosure model. As the environment and enclosure of the speaker vary, it is very difficult to tune open-loop compensators that will produce acceptable responses for a wide range of applications. Kuriyama and Furukawa [8] proposed a least mean square (LMS) algorithm in designing a loudspeaker system that can adapt to different environments and enclosures. In their work, a microphone is used as the feedback sensor. In 1959, Novak [9] showed that for a speaker with a diameter less than one-third of its highest operating frequency, its far-field sound pressure level (SPL) is non-directional and proportional to the cone acceleration. This result opened the potential of using feedback control to improve loudspeaker response without having to rely on the existence of microphones. Various loudspeaker acceleration feedback control schemes [10–12] were introduced. Although an accelerometer can be mounted within the speaker, the cost of the accelerometer some-times exceeds the cost of the speaker itself. In addition, when using acceleration feedback, the compensator is often a variation of the proportional-plus-integral (PI) type. The d.c. drift characteristics of the accelerometer makes the implementation of these PI compensators difficult. The major obstacles to wide implementation of closed-loop speaker control are the added cost and complexity of the system.

In this paper, the feasibility of using the back-electromotive voltage (back-e.m.f.) of the voice coil actuator as the feedback information to improve the dynamic range and to reduce harmonic distortion for closed-box subwoofer systems will be demonstrated. This approach has the advantages of closed-loop speaker control but without the drawback of requiring additional sensors. In addition, the back-e.m.f. signal is clean and requires only a simple second-order filter to recover the necessary velocity information. The actual control is done with a simple proportional-plus-derivative (PD) algorithm in the feedback path. The proposed approach has the potential of being implemented in a very compact analogue circuit with minimum cost increase. Experimental results confirmed that the back-e.m.f. signal of a passive coil can be used to improve the bass response of a closed-box subwoofer system as well as reduce its harmonic distortion.

2 MODELLING OF A LOUDSPEAKER SYSTEM

2.1 Dynamic model of a loudspeaker

A closed-box loudspeaker system consists of an enclosure (closed-box) which is sealed airtight except for a single aperture in which the loudspeaker is mounted. A schematic diagram of a closed-box loudspeaker is shown in Fig. 1. The subwoofer used in this study is an 8 in, dual voice coil subwoofer with a nominal open-air resonance at 40 Hz. It consists of a chassis, which serves as the electrical ground and houses the other components. The permanent magnet generates a constant magnetic field. The audio amplifier generates electrical current which is passed through the voice coil (an inductance element). As the amplifier voltage varies with the input sound signal, the current in the coil changes accordingly, which creates a time-varying (a.c.) magnetic field. The interaction of the two fields generates a force that acts on the cone. The motion of the cone, in turn, pushes air and generates localized sound pressure waves.

If the amplifier drives one of the dual coils (active coil), a back-e.m.f. voltage will be induced in the idle (passive) coil that is proportional to the velocity of the cone. However, owing to the electrical transformer coupling effect [13, 14], the voltage measured at the passive coil terminals will not be proportional to the cone velocity. From stationary magnetically coupling circuit analysis [14], and assuming that these two coils are identical, the coupling characteristics can be described as

$$U(t) = c\dot{x} + Ri_1 + L\frac{\mathrm{d}i_1}{\mathrm{d}t} + M\frac{\mathrm{d}i_2}{\mathrm{d}t}$$

$$V_s(t) = c\dot{x} + Ri_2 + M\frac{\mathrm{d}i_1}{\mathrm{d}t} + L\frac{\mathrm{d}i_2}{\mathrm{d}t}$$

where $U(t)$ is the applied voltage supplied by the audio amplifier, $V_s(t)$ is the voltage across the passive (sensing) coil, i_1 and i_2 represent the currents in each coil, L and M are the self-inductance and mutual inductance of the two coils respectively, and c is the forcing factor or back-e.m.f. constant (in V s/m or N/A). Figure 2 shows the electrical schematic of this coupled induction effect.

If it is assumed that the motion of the cone can be approximated by one-dimensional vibration of a single mass–spring–damper system, then the motion equation can be written as

$$m\ddot{x}(t) + b\dot{x}(t) + kx(t) = ci_1(t)$$

where m, b and k represent the equivalent mass, effective damping coefficient and suspension stiffness respectively. The electrical and mechanical parts of the speaker are linked by the electromechanical transformer shown in Fig. 3. Since a high impedance device is used to measure the voltage across the passive coil, it is further assumed that $i_2 \approx 0$ and $\mathrm{d}i_2/\mathrm{d}t \approx 0$. Therefore, the differential equations that describe the dual voice coil speaker are

$$\begin{aligned} U(t) &= c\dot{x} + Ri_1 + L\frac{\mathrm{d}i_1}{\mathrm{d}t} \\ V_s(t) &= c\dot{x} + M\frac{\mathrm{d}i_1}{\mathrm{d}t} \\ m\ddot{x}(t) + b\dot{x}(t) + kx(t) &= ci_1(t) \end{aligned} \tag{1}$$

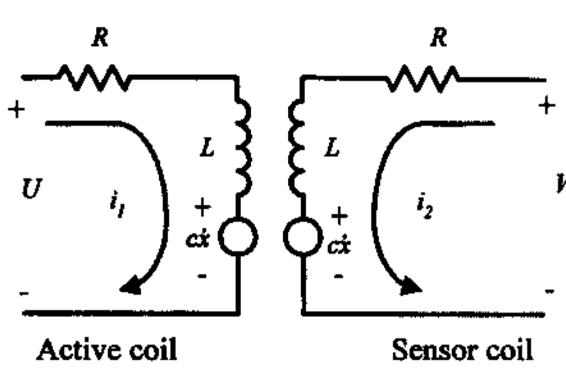

Fig. 2 Electromechanical interaction in loudspeakers

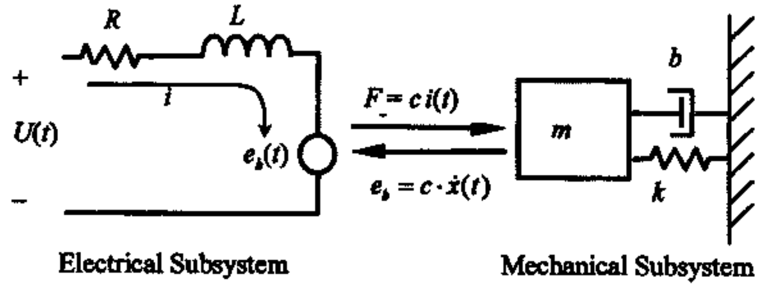

Fig. 3 Electromechanical interaction in loudspeakers

From equations (1), the transfer function from the input voltage to the cone acceleration can be written as

$$G_s(s) = \frac{A(s)}{U(s)}$$
$$= \frac{cs^2}{Lms^3 + (Lb + Rm)s^2 + (Rb + Lk + c^2)s + Rk}$$
$$= \frac{cs^2}{K_v[(1/\omega_n^2)s^2 + (2\xi/\omega_n)s + 1](\tau s + 1)} \quad (2)$$

where $U(s)$ and $A(s)$ are the Laplace transform of the input voltage $U(t)$ and cone acceleration $\ddot{x}(t)$, ω_n represents the resonant frequency of the dominating system, ξ is its damping coefficient, τ is the time constant of the high-frequency first-order dynamics and K_v is the d.c. gain offset. Since the bass response of the system is of interest and is significantly smaller in practice, in the subsequent design, the dominant dynamics of the speaker will be considered as the second-order response.

2.2 Passive coil back-e.m.f. feedback

Assume that the passive coil is used to measure the back-e.m.f. induced by the cone motion. From equations (1), the transfer function between the input voltage and passive coil terminal voltage is

$$G_p(s) = \frac{V_s(s)}{U(s)}$$
$$= \frac{s[Mms^2 + Mbs + (Mk + c^2)]}{LMs^3 + (Lb + Rm)s^2 + (Rb + Lk + c^2)s + Rk}$$
$$= \frac{s[Mms^2 + Mbs + (Mk + c^2)]}{K_v[(1/\omega_n^2)s^2 + (2\xi/\omega_n)s + 1](\tau s + 1)} \quad (3)$$

Comparing equations (2) and (3), note that $V_s(s)$ is related to the velocity of the speaker cone, $(1/s)A(s)$, through a second-order filter, i.e.

$$\frac{1}{s}A(s) = \frac{1}{s}G_s(s)U(s)$$
$$= \frac{c}{Mms^2 + Mbs + (Mk + c^2)}G_p(s)U(s)$$
$$= \frac{c}{Mms^2 + Mbs + (Mk + c^2)}V_s(s) \quad (4)$$

2.3 Effect of loudspeaker enclosure

In equation (2), assuming the coil inductance L is small compared with the effective parameters of the mechanical system, the resonance frequency of the speaker can be approximated by the effective speaker stiffness and the effective mass. The nominal parameter values of the open-air speaker can be estimated from the manufacturer specifications, as shown in Table 1. Knudsen *et al.* [4] proposed an empirical method in identifying the system parameters. However, note that, when a loudspeaker is put in a sealed enclosure, the effective compliance of the system changes. The change is mainly due to the compliance caused by the compressibility of the trapped air inside the enclosure. In most instances, the compliance of the enclosed air dominates. The com-

Table 1 Nominal parameters for the 8 in subwoofer

Definition	Value
Moving mass, m	25.98 g
D.c. voice coil resistance, R	5.5 Ω
Voice coil inductance at 1 kHz, L	1 mH
Flux density or force factor, c	5.17 V s/m or N/A
Mechanical suspension compliance, $1/k$	561.38 μm/N
Piston area, S_D	0.0214 m^2
Open-air resonance frequency, F_S	45 Hz
Nominal impedance	4–8 Ω
Mechanical Q factor, $Q_{ms} = \sqrt{mk}/b$	3.26
Total Q factor, Q_T	0.59
Accelerometer gain	0.0102 mV/(m s^2)

pliance of the air in a sealed enclosure largely depends on the enclosure size. The relation between the compliance of the enclosed air C_m and the enclosed air volume V is given by Alden [1]

$$V = 1.4 \times 10^5 \times C_m \times S_D^2 \tag{5}$$

where V is the enclosed air volume (m^3), 1.4×10^5 (N/m^2) is a constant representing the air density and the speed of sound in air, C_m is the compliance of the speaker box (m/N) and S_D is the effective piston area of the cone (m^2). The effective stiffness k in equation (1) is the inverse of the compliance C_m. Equation (5) provided a method for expressing the loudspeaker effective stiffness (compliance) as a volume of air. For an open-air speaker, the suspension stiffness is represented through an *equivalent volume of compliance*, V_{AS}, which specifies a volume of air having the same compliance as the suspension system of the loudspeaker.

When a speaker is mounted in a sealed enclosure, the effective stiffness of the combined speaker/enclosure system increases. This will increase the resonant frequency. The 'new' resonant frequency of a closed-box speaker can be estimated by

$$\frac{F_{CB}}{F_S} = \sqrt{\frac{V_{AS}}{V_B} + 1} \tag{6}$$

where V_B is the volume of the sealed enclosure and F_{CB} and F_S are the resonant frequencies for the closed-box loudspeaker and the open-air loudspeaker respectively. Table 2 lists the estimated resonant frequencies and the measured resonant frequencies of three closed-box subwoofer systems, each with a different volume. The actual resonant frequencies were identified by an HP 35670A dynamic signal analyser, which supplied swept sinusoidal voltage and measured the vibration signal from an accelerometer located at the duct cap of the speaker.

As shown in Table 2, there is very good correlation between the predicted resonant frequency, by equation (6), and the actual measured resonant frequency. In addition, from equation (6) it is established that, as the enclosure volume decreases, the resonant frequency increases. This is not desirable for the subwoofer, where accurate bass sound reproduction is important.

Table 2 Effect of enclosure volume and resonant frequency

	Length (in)	Height (in)	Depth (in)	Volume (m^3)	Predicted F_{CB} (Hz)	Measured F_{CB} (Hz)
Box 1	7	10	12.9375	0.01180	90.56	93.676
Box 2	9.5	9.5	9.5	0.01868	76.98	75.364
Box 3	12	12	12	0.02417	71.00	67.598

Note: $V_{AS} = 0.036\,m^3$.

2.4 Model validation

To validate the speaker models, $G_s(s)$ and $G_p(s)$, the frequency response of an 8 in, dual voice coil subwoofer mounted in box 2 (Table 2) is measured. Two frequency responses, $G_s(j\omega)$ and $G_p(j\omega)$, were taken, as shown in Figs 4 and 5. The frequency response from the amplifier input to the acceleration output of the speaker cone, $G_s(j\omega)$, is taken by attaching an accelerometer at the tip of the speaker cone. A third-order model

$$G_s(s) = \frac{0.4217s^2}{(s^2 + 324.256s + 2.0808 \times 10^5)(s + 5302.7)} \tag{7}$$

is fitted to the measured frequency response. As can be seen in Fig. 5, the third-order model fits the experimental data very well in the low-frequency range (40–550 Hz) where subwoofers are designed to operate. The accelerometer transfer function revealed many high-frequency resonances that are not modelled in a simplified model as described in equation (7). These resonances are the higher structure vibration modes due to the complex geometry and suspension structure of a loudspeaker. The audio amplifier possesses extra-low-frequency dynamics below 20 Hz, which affects the frequency response. Figure 5 shows the frequency response from the amplifier input to the back-e.m.f. measurement from the passive (idle) coil. A third-order model that has the same structure as described in equation (3) and the same characteristic polynomial as in equation (7) is constructed:

$$G_p(s) = \frac{0.0222s(s^2 + 287.825s + 1.229 \times 10^6)}{(s^2 + 324.256s + 2.0808 \times 10^5)(s + 5302.7)} \tag{8}$$

As shown in Fig. 5, the model fits well with the experimental data, except around the second-order zero positions. This can be attributed to the assumption used in equation (3) and consequently in equation (8). In actual implementation, a better model can be fitted to the measured passive coil back-e.m.f. response. Similar to the acceleration response, low-frequency amplifier dynamics below 30 Hz is seen, affecting the overall frequency response. Comparing the acceleration frequency response and the back-e.m.f. response (see Fig. 6), it can be seen that the induced back-e.m.f. measured from the passive voice coil is proportional to the cone speed up to 100 Hz. This is evident from the 90° phase difference between the two frequency responses as well as the 20 dB/decade magnitude difference. At frequencies higher than 100 Hz, the proportionality fails owing to the mutual inductance effect between the two coils, manifesting itself as two complex zeros in the complex

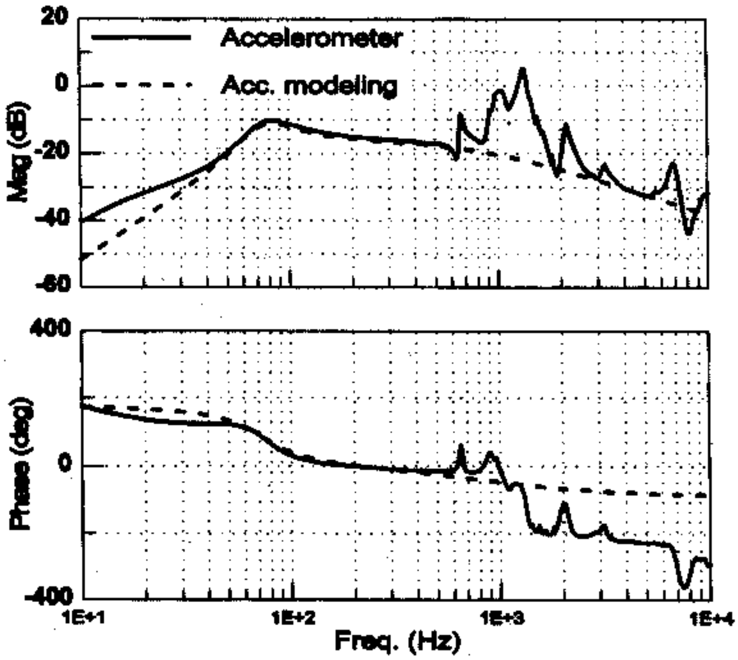

Fig. 4 Speaker acceleration frequency response

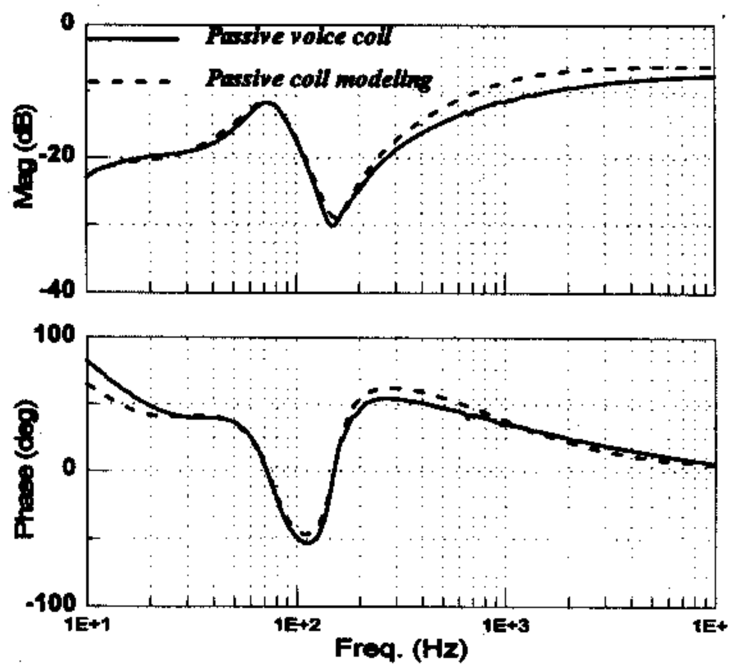

Fig. 5 Speaker passive coil back-e.m.f. frequency response

plane. This is exactly what is expressed in equations (3) and (4).

3 FEEDBACK CONTROLLER DESIGN

A typical control problem for the closed-box subwoofer system can be posed as follows:

Given the two plant models, equations (7) and (8), using the back-e.m.f. measurement from the passive coil, design a controller $C(s)$ such that the closed-loop response between the amplifier input $U(s)$ and the acceleration output $A(s)$ has extended lower resonant frequency below 40 Hz and a pass band up to 400 Hz with gain variation of less than 3 dB.

The specification of 40 Hz is to recover the open-air performance of the subwoofer. It is obvious that a

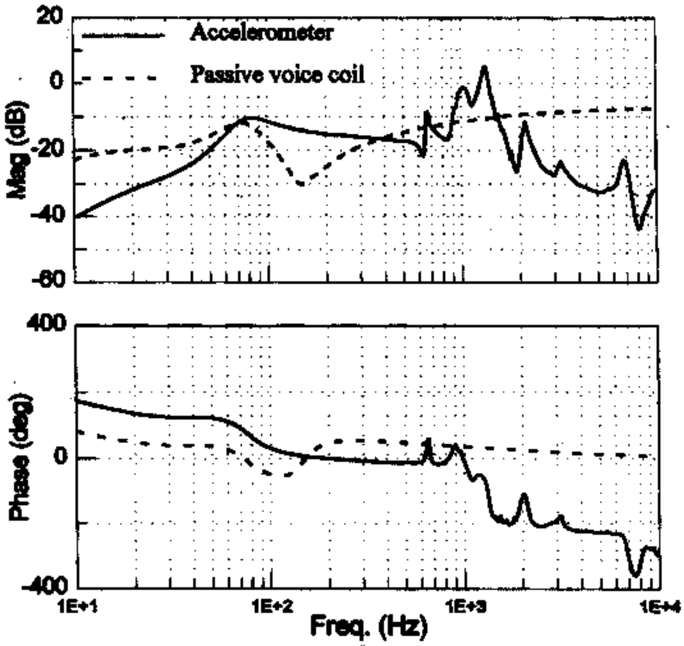

Fig. 6 Acceleration and passive coil back-e.m.f. frequency responses

simple pole placement control scheme that places the dominant poles below 40 Hz with a properly designed damping ratio will achieve the design specification. If speaker cone acceleration feedback is used, a simple PI control scheme will be able to place the dominant poles and adjust the damping to achieve the desired specification. Of course, the controller needs to provide enough roll-off so as not to excite the higher-frequency unmodelled dynamics. This is essentially the control scheme used in reference **[10]**. If passive coil back-e.m.f. is used as the feedback signal, assuming that this signal is proportional to the speaker cone velocity, then a simple PD control will be able to achieve the same goal. However, from equation (4) a desired filter is needed to extend the frequency range of the proportionality between the passive coil back-e.m.f. signal and the speaker cone velocity.

3.1 Filter design for velocity feedback

In order to make passive voice coil more representative of the speaker velocity, the coupling effect of the extra second-order zeros should be considered and eliminated by a second-order low-pass filter. An empirical approach was used to identify the second-order filter. Firstly, the actual velocity response from the measured acceleration response was constructed by augmenting an integrator to the acceleration frequency response and factoring out the accelerometer gain. Secondly, the back-e.m.f. response of the passive coil was divided by the voice coil constant c to match the d.c. gain of the velocity response. The difference between the above two velocity frequency responses is shown in Fig. 7. It is obvious that the poles of the filter should be designed to cancel the second-order zeros in the back-e.m.f. response. Therefore, a filter to extract the speaker cone velocity information from the voice coil back-e.m.f. signal is

$$F(s) = \frac{K_{\text{d.c.}}}{s^2 + (b/m)s + k/m + c^2/(Lm)}$$

where $K_{\text{d.c.}}$ is the d.c. gain of the filter to make sure the filter has unity d.c. gain. For the experimental closed-box subwoofer system, the filter is designed to be

$$F(s) = \frac{9.212 \times 10^5}{s^2 + 305.6s + 9.212 \times 10^5} \tag{9}$$

The frequency response of the filter is shown by the dashed line in Fig. 7. It maintained a good fit to the desired filter response up to 600 Hz. Figure 8 shows the comparison between the measured acceleration response and that of the derivative of the filtered passive coil back-e.m.f. signal. A good estimation of the speaker cone velocity (up to 600 Hz) can be obtained by filtering the passive coil back-e.m.f. signal with the filter designed in equation (9).

3.2 Proportional-plus-derivative velocity feedback

With the filtered back-e.m.f. signal (filtered to obtain an estimated speaker cone velocity) available for feedback, a simple PD control can be designed to extend the bass

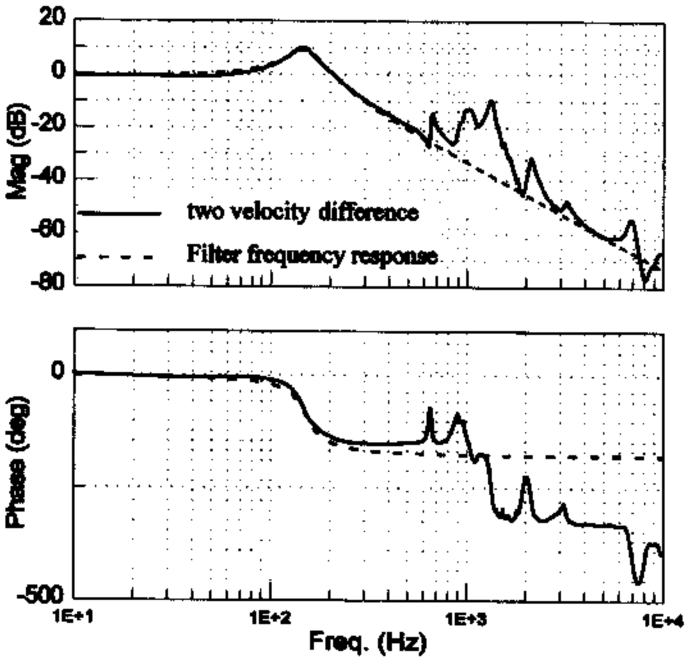

Fig. 7 Desired velocity filter frequency response

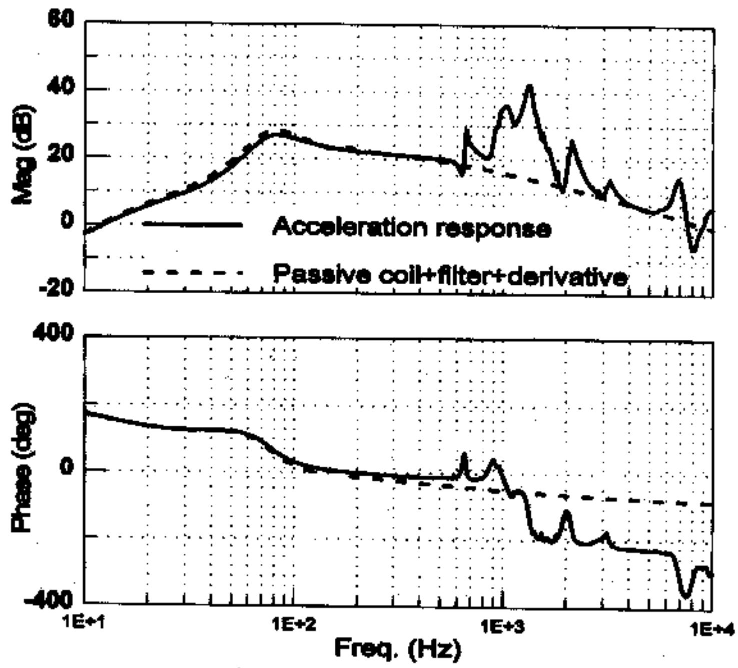

Fig. 8 Acceleration frequency response

response of the closed-box subwoofer. Assuming that the mechanical system dynamics dominates in the low-frequency range, i.e. in equation (2) $\tau \ll \omega_n$ (inductance is relatively small), the transfer function from the amplifier voltage input $U(s)$ to the speaker cone velocity output $V(s)$ can be obtained by ignoring the inductance L of the primary (active) winding in equation (2):

$$\frac{V(s)}{U(s)} = \frac{cs}{Rms^2 + (Rb + c^2)s + Rk} = \frac{b_1 s}{s^2 + a_1 s + a_2} \quad (10)$$

Equation (10) is a second-order system with velocity as the output. It is obvious that if a PD controller of the form

$$C(s) = K_d s + K_P$$

is placed in the feedback path together with a feedforward path gain d, the bass response of the system can be extended by placing the closed-loop pole near a desired lower frequency limit. In this case, the closed-loop transfer function from the amplifier input voltage to the speaker cone velocity output is

$$T(S) = \frac{V(s)}{U(s)} = \frac{db_1 s}{(1 + b_1 K_d)s^2 + (a_1 + b_1 K_P)s + a_2} \quad (11)$$

The closed-loop poles can be determined by solving for K_P and K_d using the closed-loop characteristic polynomial in equation (11) for the desired dominant resonant frequency and damping ratio. In actual implementation, the controller and filter combination will be combined to form a compensator in the feedback path. The output of the compensator will be subtracted from the gain amplified audio signal and sent to the audio amplifier, as shown in Fig. 9. The controller is simple enough to be easily implemented with inexpensive analogue components.

4 EXPERIMENTS

4.1 Experimental set-up

The experimental set-up is shown in Fig. 10. A Radio Shack 8 in, dual voice coil subwoofer is mounted in a sealed particle board enclosure. An accelerometer with a gain of 0.0102 mV/(m/s^2) is attached at the tip of the speaker cone. The active coil of the subwoofer is driven by a QSC audio amplifier with a frequency response of 20–20 kHz. The passive coil is connected to the second analogue-to-digital (A/D) channel of a TMS C31-based processor board. The filter $F(s)$ and the PD controller

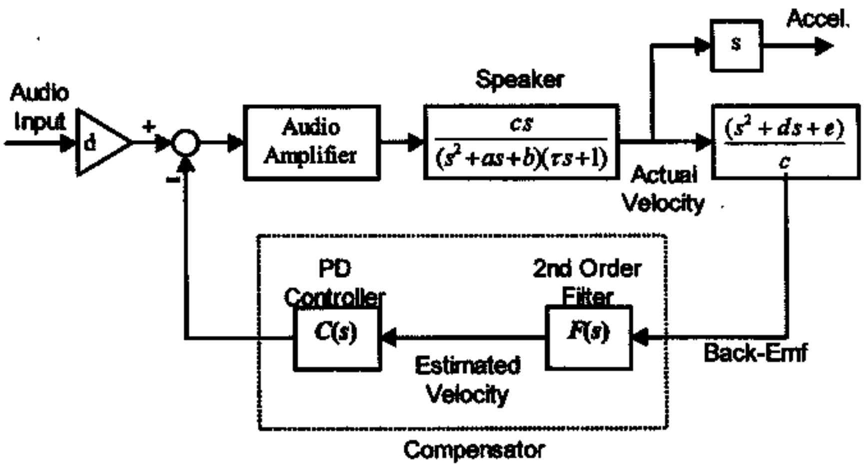

Fig. 9 Closed-loop speaker control using passive coil back-e.m.f. feedback

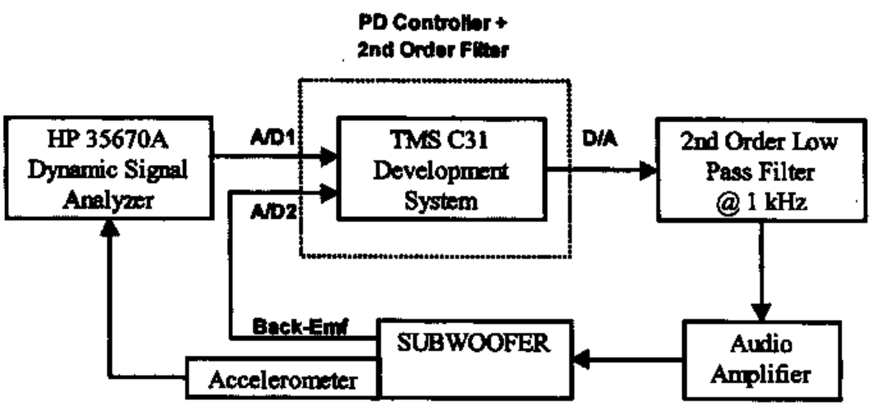

Fig. 10 Experimental set-up

$C(s)$ are digitally implemented in the C31 processor. The audio signal is connected to a second A/D channel. The controlled input to the amplifier is sent out through a 12 bit D/A. The overall sampling rate of the system is set at 5 kHz. An achievable bandwidth of around 2 kHz, which is well within the operating range of the subwoofer system (40–400 Hz), is given. The zero-order hold (ZOH) effect of the D/A will excite the higher resonance in the speaker system and will create a high pitch tone at the sampling frequency. To smooth out the staircase type D/A output, the D/A output is filtered through an analogue second-order low-pass filter with the cut-off frequency set at 1 kHz. This filter will not be needed if the filter/controller combination is implemented with clean analogue components.

4.2 Experimental results

The effectiveness of the proposed closed-loop control approach will be evaluated by looking at the range of the bass response and the relative magnitude of the harmonic distortion of the subwoofer system. Table 1 showed that the free air response for the 8 in subwoofer used in the experiment is 45 Hz. When it is mounted to the speaker box (box 2 in Table 2), the response degrades to 75 Hz. In addition, the gain variation (distortion) in the pass band (70–500 Hz) is larger than 6 dB (the solid line in Fig. 11). Using the simple PD control with back-e.m.f. feedback discussed in the previous section, the bass response of the closed-box speaker can be improved to 35 Hz. The pass band gain variation from 35 to 600 Hz is maintained within 3 dB. The closed-loop controlled speaker system still exhibited the high resonant mode of the original system. This is anticipated since the back-e.m.f. signal does not capture the higher-order dynamics in the suspension of the speaker. However, in actual audio application, crossover filters will be used to filter out the high-frequency audio signals and only pass through low-frequency bass audio signals to the subwoofer [1]. With high-frequency input being filtered by the crossover filters, the internal feedback system does not generate high-frequency signals to excite these dynamics. This is another reason for low-pass filtering of the D/A output.

Harmonic distortion is another important measure of loudspeaker performance. When excited by a single-frequency audio signal (pure tone), the speaker should generate a sound pressure wave at the input frequency. Owing to the non-linear effects in the speaker actuation, high-order harmonics will also be present. A 'good' speaker generates signal spectra that have low distortion and harmonics well below that of the primary signal. Since the far-field SPL is proportional to the speaker cone acceleration [9], the acceleration spectrum of the closed-box subwoofer system subjected to pure tone (single-frequency) input signals will be used to measure the effect of harmonic distortion. Figure 12 shows the acceleration spectrum for 40 Hz input. By increasing the bass response using back-e.m.f. feedback, the magnitude of the fundamental frequency component was increased by 10 dB. In addition to the 10 dB increase in the fun-

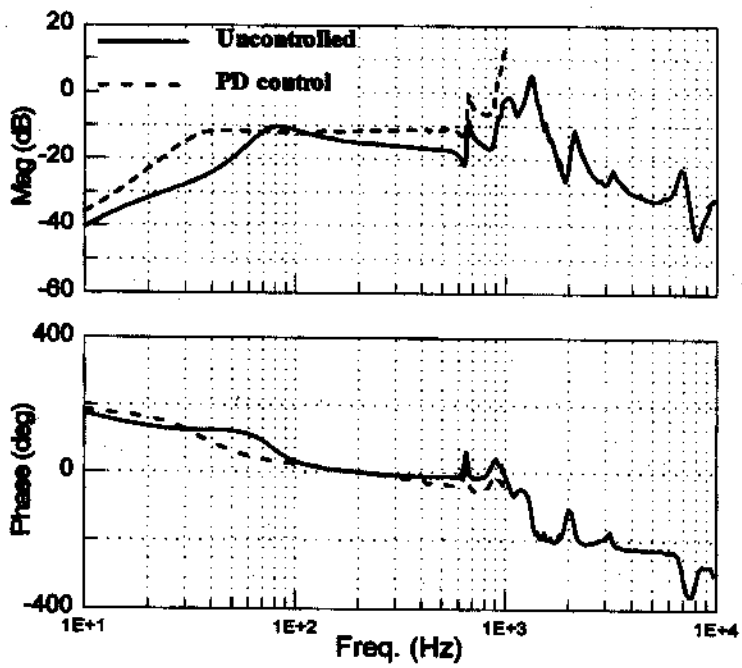

Fig. 11 Subwoofer acceleration frequency response

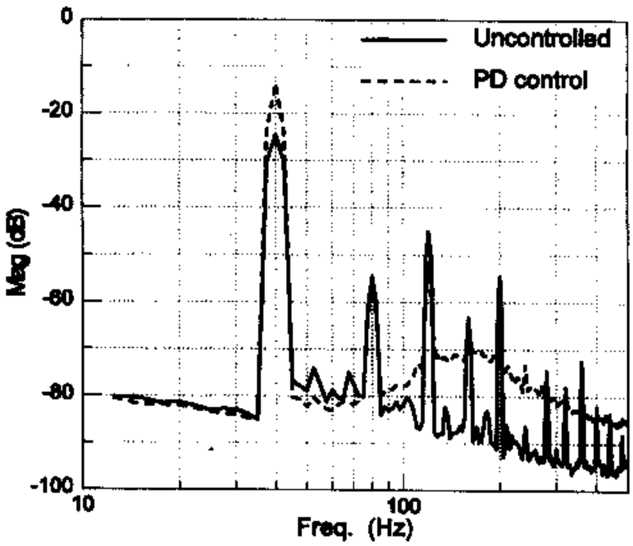

Fig. 12 Acceleration power spectrum for 40 Hz input

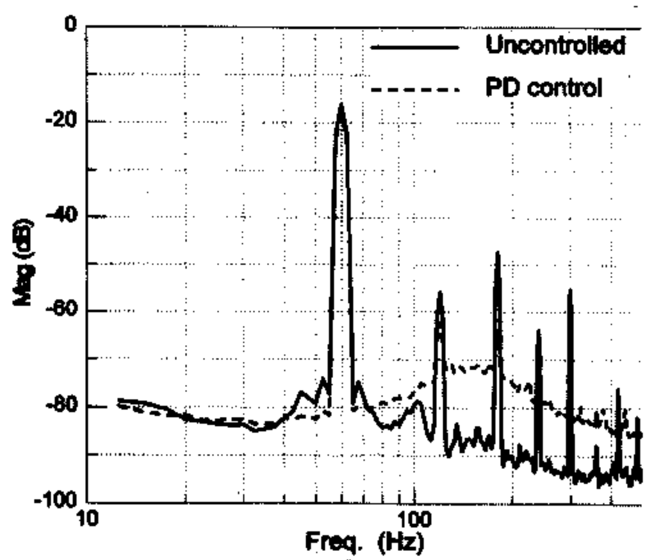

Fig. 13 Acceleration power spectrum for 60 Hz input

damental frequency, there is also a 3 dB decrease in the second harmonic and a 6 dB reduction in the third harmonic. For 60 Hz input, Fig. 13 depicts a 7 dB drop in the second harmonic and a 17 dB reduction in the third harmonic.

5 CONCLUSIONS

In this paper, the feasibility of using a voice coil back-e.m.f. signal as feedback information for closed-loop control was investigated. A dual voice coil closed-box subwoofer system is used to demonstrate the effectiveness of back-e.m.f. feedback control. A second-order filter needs to be developed to extract the velocity information from the coil back-e.m.f. signal. A PD controller is well suited for reducing the harmonic distortion and extending the subwoofer bass response. Experimental results verified that the proposed control scheme could extend the bass response bandwidth by one octave and at the same time reduce harmonic distortions by more than 6 dB. The proposed feedback and control scheme can be easily implemented using inexpensive components. Analogue implementation of the proposed algorithm is both economical and simple. Unlike acceleration feedback, the coil back-e.m.f. signal is more cost effective. In the case of dual-voice coil speakers, the back-e.m.f. feedback is obtained virtually without any extra cost. Although acoustic applications have been discussed in this paper, the design concept can be extended to the velocity feedback control of virtually all electromagnetic actuated systems. Work is currently in progress to extract back-e.m.f. information from an actively driven coil.

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