육상중장비용 시트의 진동평가 프로그램 개발

Development of a Seat Vibration Evaluation Program for Earth Moving Machinery

이건명†· 박오철*
Gun-Myung Lee and O-Cheol Park

Abstract

A simulation program has been developed to evaluate operator seat vibration for earth-moving machinery and decide whether a seat meets the requirements imposed by ISO 7096. An operator seat is assumed as a linear system composed of a mass, a spring, and a damper mounted on a platform. The program evaluates the transmissibility at resonance, and the SEAT factors for a light person and a heavy person. The developed program can be utilized effectively in designing a new operator seat.

Key Words : Seat Vibration(시트 진동), Earth Moving Machinery(육상중장비), ISO 7096, SEAT Factor(SEAT 계수), Transmissibility(전달률), Power Spectral Density(파워스펙트럼밀도), Frequency Weighting(주파수 가중)

1. Introduction

The operators of earth−moving machinery are often exposed to a low frequency vibration environment partly caused by the movement of the vehicles over uneven ground and the tasks carried out. The seat constitutes the last stage of suspension before the driver. To be efficient at attenuating the vibration, the suspension seat should be chosen according to the dynamic characteristics of the vehicle. The design of the seat and its suspension are a compromise between the requirements of reducing the effect of vibration and shock on the operator and providing him with stable support so that he can control the machine effectively.

The international standard, ISO 7096[(1)], specifies a laboratory method for measuring and evaluating the effectiveness of the seat suspension in reducing the vertical whole−body vibration transmitted to the operator of earth moving machinery at frequencies between 1 Hz and 20 Hz.

† School of Mechanical and Aerospace Engineering, Gyeongsang National University
TEL: +82-55-751-5313 FAX: +82-55-757-5622
E-mail: gmlee@gnu.ac.kr
*Graduate School, Gyeongsang National University

When a seat of new type is developed, its prototype should be manufactured and tested according to the standard. Based on the test results its design should be modified to meet the requirements of the standard. To avoid this time consuming and costly process, a computer simulation program has been developed to decide prior to manufacturing whether a designed seat can meet the standard. Seats are assumed to be linear systems composed of a mass, a spring and a damper mounted on a platform.

2. Brief description of the standard

The international standard, ISO 7096, defines the input spectral classes, EM1 through EM9 required for earth−moving machines. Each class defines a group of machines having similar vibration characteristics. The standard specifies the input vibration in nine input spectral classes using the PSD(power spectral density) of the acceleration measured on the platform, $G^*_P(f)$. The PSD for each input spectral class is given by an equation. For example, the PSD for spectral class EM9 which includes skid steer loaders with operating mass less than 4500 kg, is given as follows.

$$G^*{}_P(f) = 2.10(HP24)^2(LP12)^2 \quad (1)$$

where

$$(LP12) = 1/(1 + 1.414S + S^2) \quad (2)$$

$$(HP24) = S^4/(1 + 2.613S + 3.414S^2 + 2.613S^3 + S^4) \quad (3)$$

and

$$S = jf/f_c, \; j = \sqrt{-1} \quad (4)$$

with filter cut−off frequency, f_c, equal to 4.0 Hz for LP12 and 3.5 Hz for HP24.

The PSD for class EM9, is shown in Fig. 1 along with its tolerances illustrated by dotted lines. The tolerances are given as follows.

$$G^*{}_{PL}(f) \le G_P(f) \le G^*{}_{PU}(f) \quad \text{for} \quad f_1 \le f \le f_2 \quad (5)$$

where

$$G^*{}_{PL}(f) = G^*{}_P(f) - 0.1 \times \max[G^*{}_P(f)] \quad (6)$$

$$G^*{}_{PU}(f) = G^*{}_P(f) + 0.1 \times \max[G^*{}_P(f)] \quad (7)$$

If the calculated value for the lower bound $G^*{}_{PL}(f)$ in Eq. (6) is less than zero, it is replaced by zero. f_1 and f_2 for EM9 are 0.89 and 17.78 Hz, respectively.

The standard also specifies the rms accelerations on the platform, $a^*{}_{P12}$ and $a^*{}_{P34}$, which represent the rms accelerations between frequencies f_1 and f_2, and f_3 and f_4, respectively. f_3 and f_4 for EM9 are 3.00 and 6.00 Hz, respectively. The rms accelerations between these frequencies can be obtained by passing the measured acceleration signals through band pass filters with cut−off frequencies equal to these frequencies, and calculating the rms values of the filtered signals. These digital band pass filters were designed using the MATLAB signal processing toolbox[(2)]. The target values of $a^*{}_{P12}$ and

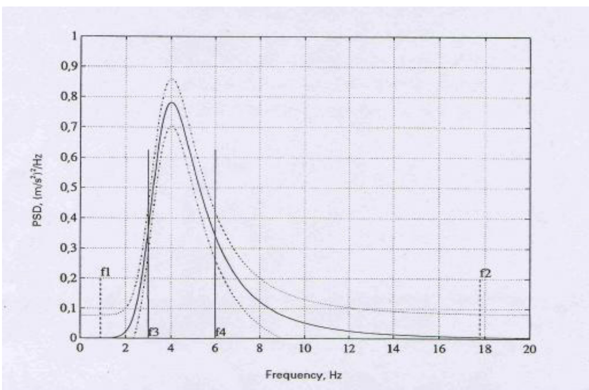

Fig. 1. PSD for input spectral class EM9.

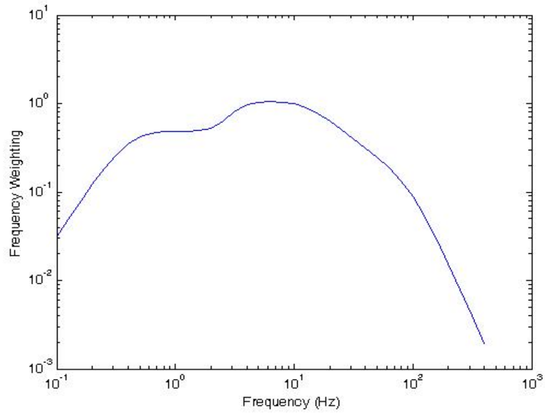

Fig. 2. Frequency weighting curve for the weighting Wk

$a^*{}_{P34}$ are 1.63 and 1.33 m/s2, respectively. The tolerances for rms accelerations are given as follows.

$$0.95 \times a^*{}_{P12} \le a_{P12} \le 1.05 \times a^*{}_{P12} \quad (8)$$

$$0.95 \times a^*{}_{P34} \le a_{P34} \le 1.05 \times a^*{}_{P34} \quad (9)$$

The standard requires two kinds of vibration tests to be performed on seats. One is simulated input vibration test and the other is damping test. In the simulated input vibration test the platform of a seat is excited so that its acceleration signal satisfies the PSD and rms value requirements which are described above. By performing the test, the SEAT(Seat Effective Amplitude Transmissibility) factor defined below is evaluated.

$$SEAT = a_{WS12} / a_{WP12} \quad (10)$$

where a_{WS12} and a_{WP12} represent the weighted rms values of the measured vertical accelerations between frequencies f_1 and f_2 at the seat disk and at the platform, respectively. The frequency weighting shall be in accordance with ISO 2631−1[(3)]. According to the standard the frequency weighting Wk, whose frequency weighting curve is shown in Fig. 2, shall be applied to acceleration signals measured at the seat disk and the platform. The SEAT factors shall be measured for two persons: a light person and a heavy person. The light person shall have a total mass of 52 kg to 55 kg, while the heavy person 98 kg to 103 kg. The standard requires that the SEAT factors be less than the specified values, for example, 0.9 for input spectral class EM9.

In the damping test the transmissibility $H(f_r) = a_S(f_r)/a_P(f_r)$ at resonance along the vertical axis shall be measured. In the above equation $a_S(f_r)$ and $a_P(f_r)$ represent unweighted rms values of the measured vertical accelerations at the resonance frequency f_r at the seat disk and at the platform, respectively. In the damping test the seat shall be loaded with an inert mass of 75 kg. The standard requires that the transmissibility be less than the specified value, for example, 2.0 for input spectral class EM9.

3. Development of the simulation program

In the simulation program the seat is modeled as a mass, a spring, and a damper mounted on a platform as shown in Fig. 3. The system is assumed to be linear. It can be shown that the above mentioned transmissibility $H(f_r)$ is equal to the following displacement transmissibility between the mass and the platform calculated for the frequency ratio $r=1$ [4].

$$T_d = \frac{\sqrt{1+(2r\varsigma)^2}}{\sqrt{(1-r^2)^2+(2r\varsigma)^2}} \tag{11}$$

Hence

$$H(f_r) = \frac{\sqrt{1+(2\varsigma)^2}}{2\varsigma} \tag{12}$$

where ς represents the damping ratio which is equal to $c/2\sqrt{mk}$. For given values of m, c, and k, where m is equal to the mass of the seat plus 75 kg, the transmissibility at resonance $H(f_r)$ is calculated using Eq. (12).

If $z=x-y$ denotes the motion of the mass relative to the platform, the equation of motion can be written as

$$m\ddot{z}+c\dot{z}+kz=-m\ddot{y} \tag{13}$$

If the acceleration of the platform, $\ddot{y}$, is given, z, $\dot{z}$, $\ddot{z}$ can be calculated by solving the above equation using the Runge–Kutta method. Finally, the acceleration of the seat, $\ddot{x}=\ddot{y}+\ddot{z}$, is obtained.

The acceleration signal of the platform having the required PSD in Fig. 1 can be generated by the following procedure. If $G_P(f)$ denotes the PSD of a signal at frequency f and Δf a narrow frequency bandwidth around f, $G_P(f)\Delta f$ is equal to the mean square value of the signal in the frequency band [5]. If the signal in the band is assumed to be a sinusoid with a single frequency f and amplitude A, the mean square value is equal to $A^2/2$. Hence we obtain the following equations.

$$G_P(f)\Delta f = \frac{A^2}{2} \tag{14}$$

$$A = \sqrt{2G_P(f)\Delta f} \tag{15}$$

Therefore, the signal in the frequency band is represented as $A\sin(2\pi ft+\phi)$ with arbitrary value of ϕ. Then the whole signal is obtained by summing the signals in each narrow frequency band. The resulted signal becomes the acceleration time signal of the platform. The time signal generated in this way is shown in Fig. 4. a_{P12} and a_{P34}, which represent the unweighted rms values of this time signal between frequencies f_1 and f_2, and f_3 and f_4, were equal to 1.6258 and 1.3300, respectively. These values are within the tolerances in Eqs. (8) and (9), and very close to the target values, 1.63 and 1.33. The PSD of the generated signal was calculated using a spectral analysis function in the MATLAB signal processing toolbox and is shown along with tolerances in Fig. 5.

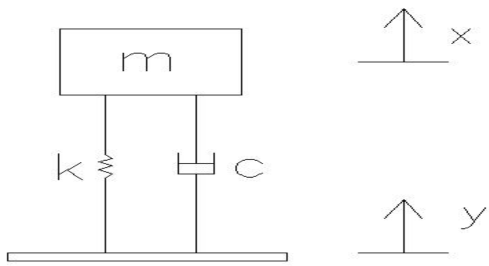

Fig. 3. Model of a seat composed of a mass, a spring, and a damper.

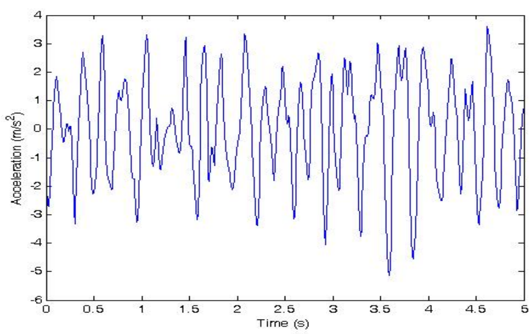

Fig. 4. Acceleration time signal of the platform.

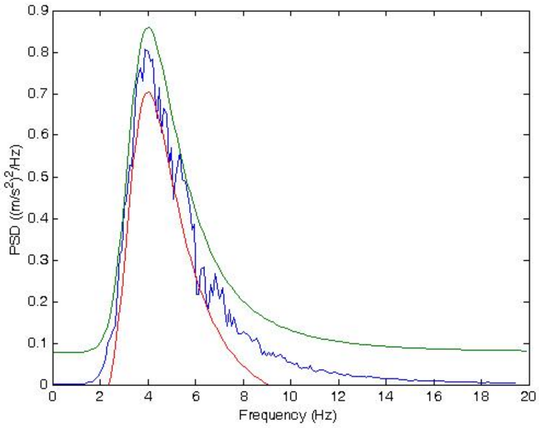

Fig. 5. PSD of the generated signal with the tolerances.

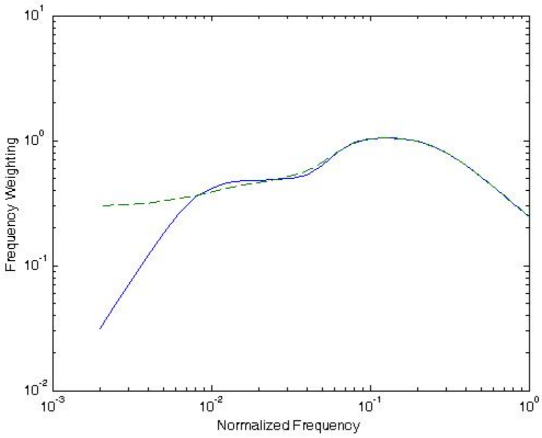

Fig. 6. Comparison of the designed frequency weighting with the target.

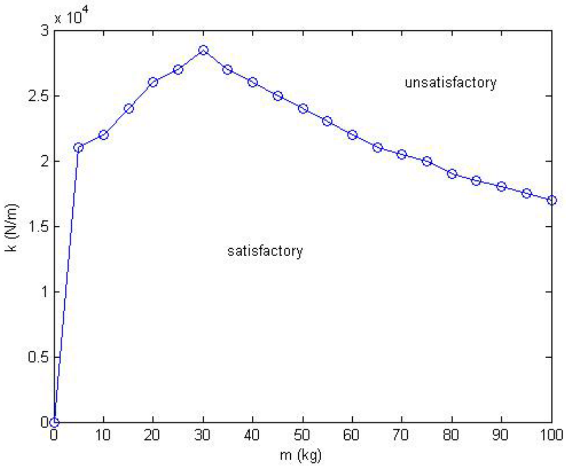

Fig. 7. Parameter values which meet the requirements of the ISO 7096.

A filter which has the frequency weighting curve in Fig. 2 was designed using the 'yulewalk' function of the MATLAB signal processing toolbox. The frequency weighting of the designed curve with the filter order equal to 22 is compared with the target frequency weighting in Fig. 6. Two curves in the figure show discrepancy below the normalized frequency 0.01, which corresponds to 0.5 Hz when the sampling frequency is equal to 100 Hz and the Nyquist frequency 50 Hz. However, the frequency contents of the generated signal below this frequency are negligible, it is believed that this discrepancy does not affect the results and the designed filter is satisfactory.

The SEAT factor in Eq. (10) is calculated by passing the acceleration signals of the platform and the seat through the above designed frequency weighting filter. The mass of a light person was set to 53.5 kg, while that of a heavy person to 100.5 kg. These mass values are the median values of the ranges for a light and a heavy person, respectively.

The developed program was run for parameter values of m_{SEAT} = 20 kg, k = 20,000 N/m, and c = 1000 Ns/m. The transmissibility at resonance, $H(f_r)$, was equal to 1.7029, meeting the standard. The SEAT factors for a light person and a heavy person were 0.7320 and 0.4623, respectively. Since both the factors are less than 0.9, they meet the standard. Therefore, the seat with these parameters is satisfactory according to ISO 7096. A series of run of the program were executed with c fixed to 1000 Ns/m and various values for m_{SEAT} and k. The results are shown in Fig. 7.

4. Conclusions

A simulation program has been developed to decide whether an operator seat for earth-moving machinery meets the requirements imposed by ISO 7096. An operator seat is assumed as a linear system composed of a mass, a spring, and a damper mounted on a platform. The program evaluates the transmissibility at resonance, and the SEAT factors for a light person and a heavy person. The evaluated performance is then compared with the requirements of the standard. Even if the developed program is restricted for input spectral class EM9 and linear systems, the program can be extended easily to other input spectral classes and non-linear systems. When extended to non-linear systems, only the part for solving the equation of motion is required to be modified. The developed program can be utilized effectively in designing a new operator seat.

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