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Optimization of passive tractor cabin suspension system using ES, PSO and BA

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A survey was made to determine the spring and damper settings of ITM285 tractor's cabin to insure optimal ride and the comfort of the tractor operator. Analysis has been done in terms of root mean square acceleration response (RMSAR) in one-third-octave band and International Standard Organization (ISO). Optimization is performed using particle swarm optimization (PSO), Bee Algorithm (BA) and Evolution Strategy (ES) methods on a 2 DOF modeled for frequencies ranging from 1 to 10 Hz. Optimized parameters for tractor's cabin suspension in line with ISO 2631-1985 showed significant reduction in transmitted vibration as well as improvement in the ride comfort of the tractor operator. Furthermore, among the latter optimization methods, ES was more successful with regard to vibration transmission reduction to tractor operator.

Keywords: Tractor cabin, Transmitted vibration, optimization (PSO, ES and BA)

Introduction

The suspension system of a vehicle is designed to isolate vehicle's passenger from road irregularities and also controls vehicle body attitude for a safe, stable and comfortable ride. Since tractors do not have suspension systems and the vibration levels are high in comparison with other road vehicles (Bovenzi and Betta, 1994), the problem of tractor ride is more crucial. Furthermore, the dominant natural frequencies of the tractor are 1-7 Hz which is within the range of the most critical frequencies of human body. For instance, human trunk and lumbar vertebra have a natural frequency in the range of 4-8 and 4-5 Hz, respectively (Kumar *et al.*, 2001). The backbone is especially sensitive in this frequency range for severe physical damage (Boshuizen *et al.*, 1992). Moreover, there is no attenuation in transmission of frequency in the lower frequency range up to 6 Hz (Chaffin and Andersson, 1990). This kind of vibration can lead to harmful spinal effects which make problems in the blood supply to tissues that result in the destruction of tensile cells. Furthermore, due to harmful effects of

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vibration to the limbs,the joints will be soft and fragile. Meanwhile, whole body vibrations can result in human digestive, fatigue, disorder, increased heart rate followed by increased blood pressure. In addition, the reduction of cognitive ability, as well as vision impairment, causes error in fine tasks and damage to the reproductive system, along with driving-related accidents (Dewangan *et al.*, 2005; Cakmak *et al.*, 2011). Transmitted vibrations are the origin of these health problems to the driver and impairment to farm tasks. These kinds of vibrations are caused by the irregularities of the road or soil profile, or moving elements within the machine or implements. Although, there is an extensive study on this issue however, none of the above studies can be directly applied to the vibration problem. Development of tractor cab suspension systems has been widely discussed over the last 40 years by a number of researchers (Scarlett *et al.*, 2007). Hansson (1995) described optimization of agricultural tractor cab suspension using the evolution method. The studies performed showed that the method is a very useful tool for optimization of cab suspension characteristics. In another study Deprez *et al.* (2005) have investigated the effect of a passive and a semi-active hydropneumatic cabin suspension on the comfort of the drivers. After modelling the hydropneumatic device, the parameters in the model were optimized with respect to objective comfort parameters using a global optimization technique. The parameters in the semi-active control laws were optimized using the same approach. Improvements in comfort values up to 90% were observed. Baumal *et al.* (1998) used genetic algorithm to determine both the active control and passive mechanical parameters of a vehicle suspension system. The objective is to minimize the extreme acceleration of the passenger's seat, subject to constraints representing the required road-holding ability and suspension working space. In another study, Gundogdu (2007) carried out an optimization of a four-degrees-of-freedom quarter car seat and suspension system using genetic algorithms to determine a set of parameters to achieve the best performance of the driver. Comparatively better results were obtained from the optimized system in terms of resonance peaks, crest factor, and vibration dose value. Spelta *et al.* (2011) analyzed the use of semi-active cabin suspension for agricultural vehicle. A magnetorheological damper is located in the cabin suspension with the aim of improving the comfort perceived by passengers. In this paper, the 2 DOF of tractor cabin has been modelled and analyzed. To study the ride comfort of the vehicle, vertical acceleration and pitch angle have been calculated, and, according to Tamboli and Joshi (1999), vertical acceleration is used for optimization of suspension parameters of the tractor's cabin. For this case, different kinds of optimization methods e.g. evolution strategy (ES), bees algorithm (BA) and particle swarm optimization (PSO) have been used to determine the front and rear spring and damper settings of suspension geometry to ensure optimal ride comfort.

Materials and methods

Single differential IT285-2WD tractor model was used in the present study. The reason for choosing this model was the availability of the required information for the simulation and modelling. It is worth noting that the method presented in this study could be applied on each model of tractor if the required information of modelling would be provided.

Simulation and mathematical modelling of tractor cabin

In IT285-2WD tractor, the cabin is attached to the tractor's body at four points, two end points on rear axles and the other two on the front part of gearbox case.
Rear axles excite the end points, and the other two are excited with a ratio of rear and front axle distances. Suppose, $q_1(t)$ and $q_2(t)$ are random excitations at the front and rear axles of the vehicle at instant i, respectively; thus, the displacement of the front point cabin attachments, q(t), can be obtained by (Tamboli and Joshi, 1999):

$$\frac{q(t)-q_2(t)}{q_1(t)-q_2(t)}=\frac{b}{a+b} \qquad (1)$$

Where *a* and *b* are front point cabin attachment distances from the front and rear axles, respectively, as shown in Fig. 1.

Now by taking the effect of the time lag τ between $q_1(t)$ and $q_2(t)$ into account, the values $q_1(t)$ and $q_2(t)$ can be related by introducing a new ratio $\alpha(\tau)$ as (Tamboli and Joshi, 1999):

$$q_2(t+\tau)=\alpha(\tau)q_1)(t) \qquad (2)$$

Using Eqs. (1) and (2), the relationship between front and rear base displacement excitations can be obtained:

$$\frac{q(t)}{q_2(t)}=\bar{\alpha}(\tau)=\frac{b}{a+b}(\alpha(\tau)-1)+1 \qquad (3)$$

Zehsaz et al. (2011) carried out experimental test for single differential IT285-2WD and the value of $\alpha(\tau)$ has been obtained 0.9 using Eq. (3).

Due to the fact that the tractor's cabin has been excited by q(t) and $q_2(t)$, it can considered as a two-degree-of-freedom model and the equation of motion can be written as follows:

$$M\ddot{x} = -(K_1 + K_2)x - (K_2L_2 - K_1L_1)\theta - (C_1 + C_2)\dot{x} - (C_2L_2 - C_1L_1)$$
$$+K_2q_2 + K_1q + C_2\dot{q}_2 - C_1\dot{q} \qquad (4)$$

$$I\ddot{\theta} = -(K_2L_2 - K_1L_1)x - (K_2L_2^{\,2} - K_1L_1^{\,2})\theta - (C_2L_2 - C_1L_1)\dot{x} - (C_2L_2^{\,2} - C_1L_1^{\,2})\dot{\theta}$$
$$+K_2L_2q_2 + K_1L_1q + C_2L_2\dot{q}_2 - C_1L_1\dot{q} \qquad (5)$$

Where M and I are the mass and mass moment of the inertia for the cabin, respectively, K_1 and K_2 are the stiffness coefficients of front and rear, respectively, C_1 and C_2 are the viscous damping coefficients of front and rear respectively, L_1 and L_2 are the distance of mass centre from front and rear axle respectively. Degrees of freedom of the cabin model are the vertical motion and the pitch motion and denoted as x and θ, respectively. Zehsaz *et al.* (2011) conducted experimental measurement on rear axel and compare it with mathematical simulation which has been done based on Eqs. (4) and (5); According to reported results there were a good agreement between experimental and mathematical measures (Fig. 2).

The parameters of the tractor's cabin model which are used in the simulation study are shown in Table 1.

Taking Fourier's transforms and putting the initial conditions as x=0, $\dot{x}$=0 at t=0, Eqs. (4) and (5) can be written in matrix form in frequency domain as:

$$\begin{bmatrix} \mathrm{K}_1+\mathrm{K}_2-\mathrm{M}\omega^2 + i\omega(C_1+C_2) & K_2L_2 - K_1L_1 + i\omega(-C_1L_1 + C_2L_2) & -K_1 - i\omega C_1 \\ K_2L_2 - K_1L_1 + i\omega(-C_1L_1 + C_2L_2) & K_1L_1^{\,2} + K_2L_2^{\,2} - I\omega^2 + i\omega(C_1L_1^{\,2} + C_2L_2^{\,2}) & K_1L_1 + i\omega C_1L_1 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x/q_2 \\ \theta/q_2 \\ q_1/q_2 \end{bmatrix} = \begin{bmatrix} K_2 + i\omega C_2 \\ K_2L_2 + i\omega C_2L_2 \\ \overline{\alpha}(\tau) \end{bmatrix} \qquad (5)$$

The transfer function or frequency response function is defined as the ratio of the output to input (Adachi *et al.*, 1996). From this the required transfer function $H(i\omega) = x/q_2$ can be calculated by inverting the complex matrix 5 and the mean square acceleration response is given by (Tamboli and Joshi, 1999):

$$\mathrm{E}(\overline{\ddot{\mathrm{x}}}^{2}) = \int_0^\infty |H(i\omega)|^2\, \omega^4 G(\omega)\, d\omega \qquad (6)$$

Where $G(\omega)$ is the Power Spectral Density (PSD) of excitation and it is calculated according to Steinwolf (2006) and Zehsaz *et al.* (2011).

Optimization method

In order to carry out optimization, the parameters of a suspension system, K_1, K_2, C_1 and C_2 have been taken into account as design variables. The suspension system parameters are optimized based on vertical transmitted vibration to the driver using evolution strategy (ES), bees algorithm (BA) and particle swarm optimization (PSO) and compare to the results obtained by Zehsaz et al. (2011) (Poli *et al.,* 2007; Gao *et al.,* 2012; Jaindl *et al.,* 2009; Deb, 2001).

Results and discussion

Since ride comfort is a qualitative subject, appropriate criteria should be defined to quantify it. This can be obtained by selecting an objective function. The optimum design of the suspension system from a ride comfort point of view can be achieved by selecting an objective function, Z, and taking the design variables as the front and rear springs and dampers coefficient of suspensions. The objective function Z is defined as follows (Tamboli and Joshi, 1999):

$$\text{Minimize} \sum \max \left[\mathrm{w}_\mathrm{i}[\ddot{\mathrm{X}}_\mathrm{i} - g_i] \right]^2 \qquad (7)$$

where w_i is the weighting factor at ith frequency; $\ddot{\mathrm{X}}_\mathrm{i}$ is the vertical Root-Mean-Square (RMS) acceleration at ith frequency which is calculated in its 1/3 octave band from Eq. (8); g_i is the desired value of RMS vertical acceleration at the ith frequency. The values of w_i and g_i are taken from the ISO 2631-1985 (E), which are given in Table 2 (Anonymous, 1990).

$$\text{RMSAR} = (2\pi)^4 \left[\int_{0.89f}^{1.12f} |H(if)|^2 \; f^4 S_g(f) df \right]^{0.5} \qquad (8)$$

Drivers of agricultural wheeled tractors are exposed to low frequency (<10 Hz) vibration (Koen *et al.,* 2005; Zehsaz *et al.,* 2011). Besides, Khaksar *et al.* (2013) measured whole body vibration of ITM399 tractor operators using power spectral density. According to the results, total weighted acceleration had the maximum value for disking, hay baling, trailer transportation and ploughing operation, respectively. It was concluded that in the three directions and at different operations, except for trailer transportation, maximum amplitude of whole body vibration energy which is transmitted to operator's body through driver's seat takes place at low frequencies (under 5 Hz). Therefore, up to 10 Hz frequency according to

ISO 5008- 2002 was selected (Anonymous, 2002). Using Eqs. (6), (7), and (8), and plugging in g_i and w_i values from Table 2 in the equations, the objective function Z is minimized and the optimum values of it are obtained via an BA ES and PSO approaches. The desired PSD used in the minimization procedure is $S_g(\Omega_0) = 10^{-6}$ which was obtained according to the ISO 5008-2002 and considering the average tractor forward speed as 1.7 m s^{-1}.

In order to carry out optimization, 4-parameter optimization constraints C_1 C_2, K_1 and K_2 are defined according to Zehsaz *et al.* (2011), Adachi *et al.* (1996) and Marsili *et al.* (2002) as follows:

$$\left\{\begin{array}{ll} 5349 \leq K_1 \leq 21397 & \dfrac{N}{m} \\ 7489 \leq K_2 \leq 29956 & \dfrac{N}{m} \\ 298 \leq C_1 \leq 919 & \dfrac{Ns}{m} \\ 417 \leq C_2 \leq 1287 & \dfrac{Ns}{m} \end{array}\right\}$$

The large number of optimized parameters resulted in many local minima being found in the objective function. In numerous cases it is found that repeating optimization results in other optimal parameter vectors. Therefore, the solutions cannot be considered as representing absolute minima. As a result, the optimized values of cabin suspension system of a multi-dimensional problem having the differences in the objective function in no case more than 1.5% for the different solutions are reported as optimum suspension characteristics. These results and the correspond values obtained in Zehsaz *et al.* (2011) study are shown in Table 3.

The tractor cabin suspension was remodelled with the obtained values in Table 3, then the diagram of cabin acceleration and displacement versus time are plotted (Fig. 3, 4). A unit step is used as the input to excite the system. As illustrated in Fig. 3 (a, b, c, d) tractor cabin stops vibrating after 3.0125, 3.0625, 2.8125 and 5.0625 seconds, respectively. Therefore, the values obtained for tractor cabin using ES optimization method exerts less force to the tractor operator due to less acceleration oscillation time. As a result, less damage would be applied to the operator and consequently more driving comfort could be provided.

According to the displacement diagram of tractor cabin Fig.4 (a, b c and d), the displacement oscillation of cabin using ES and PSO is dissipated faster then displacement oscillation in BA and Zehsaz *et al.* (2011) method. This indicates that the ES and PSO methods are more successful than the BA and proposed method in Zehsaz *et al.* (2011) study.

In order to analysis vibrations of tractor cabin more carefully and quantitatively, rise time, peak time, settling time and max. overshoot of tractor cabin displacement are calculated (Ogata, 2001) (Table 4).

According to Table 4, obtained rise time of tractor cabin using PSO method is lower then the other optimization methods. This shows that designed system using PSO values can response to the road irregularities faster than the other methods. Conversely, settling time and maximum overshoot obtained by ES method are less than the other methods. Therefore, using ES optimized values to design tractor cabin's suspension system results in better ride comfort comparison to PSO, BA and reported values in Zehsaz *et al*. (2011).

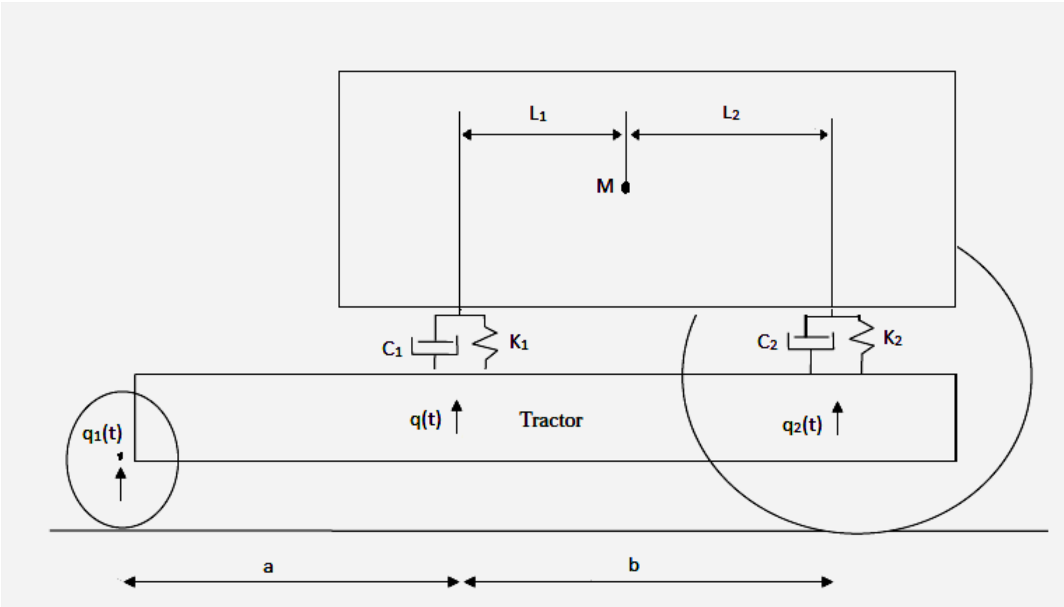


Fig. 1: Two- DOF half model of tractor cabin (Displacement excitations of front axle $q_1(t)$, rear axle $q_2(t)$ and front base of cabin $q(t)$)

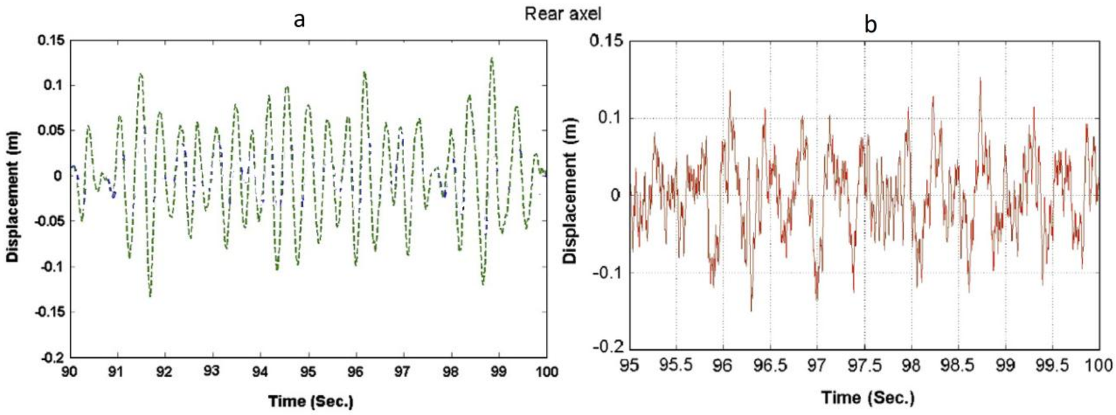
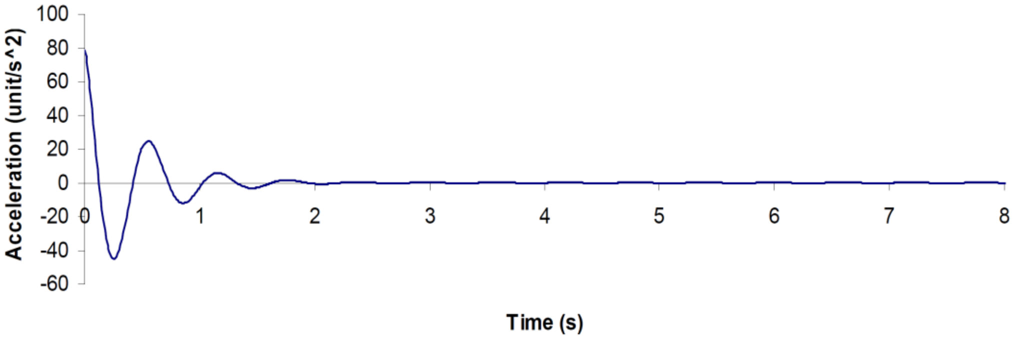
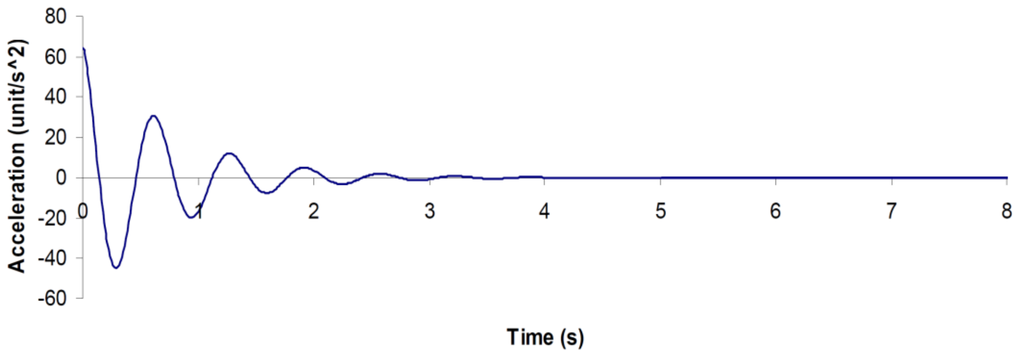


Fig. 2: Displacement measurement on traction test road at high 1st speed: (a) simulation and (b) rear axle (Zehsaz et al., 2011).

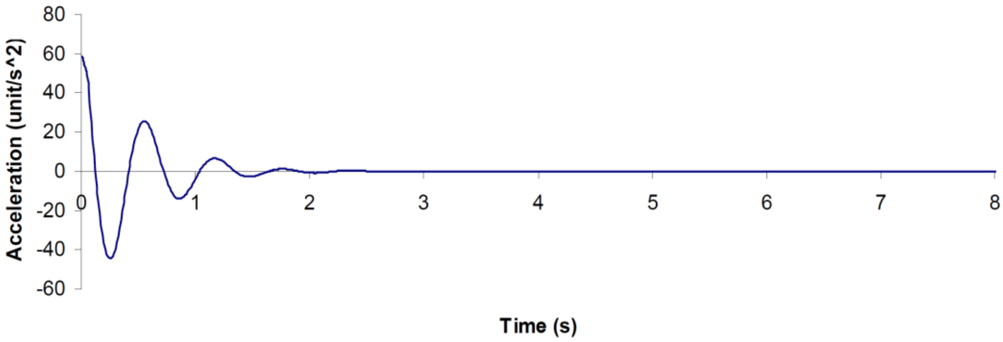
(a)


Acceleration (unit/s^2)
100
80
60
40
20
0
-20
-40
-60
0
1
2
3
4
5
6
7
8
Time (s)

(b)


Acceleration (unit/s^2)
80
60
40
20
0
-20
-40
-60
0
1
2
3
4
5
6
7
8
Time (s)

(c)


Acceleration (unit/s^2)
80
60
40
20
0
-20
-40
-60
0
1
2
3
4
5
6
7
8
Time (s)

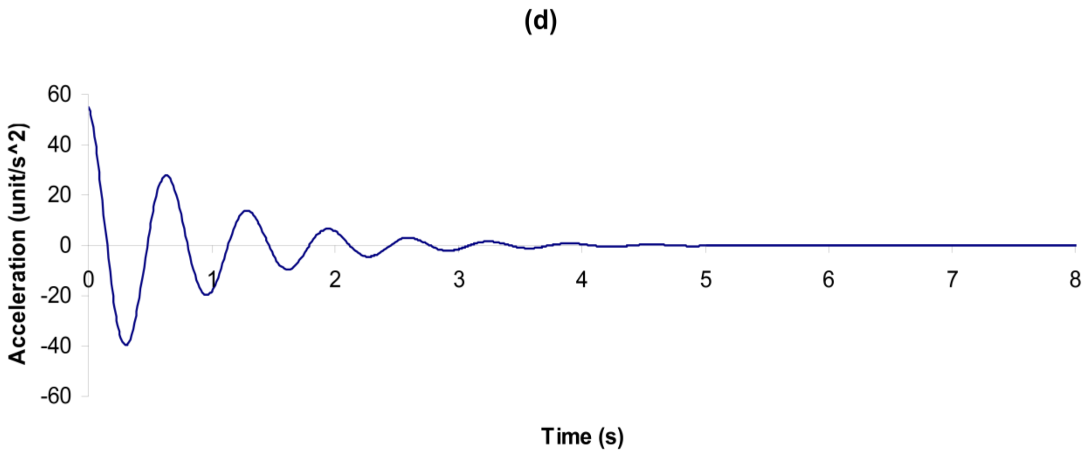
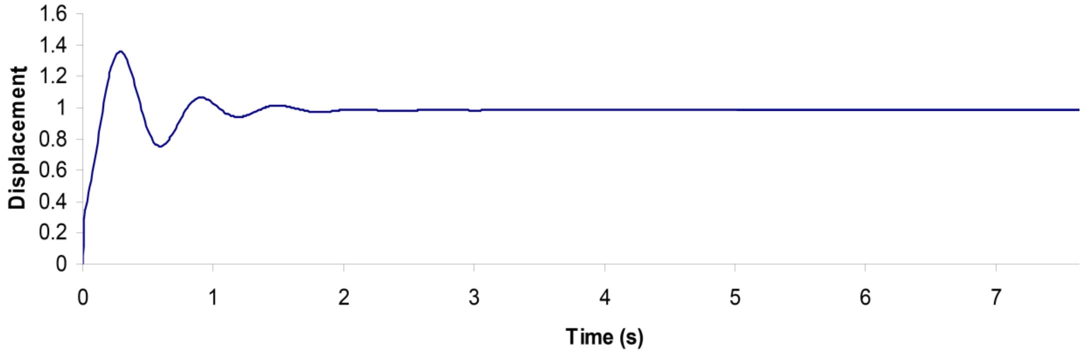
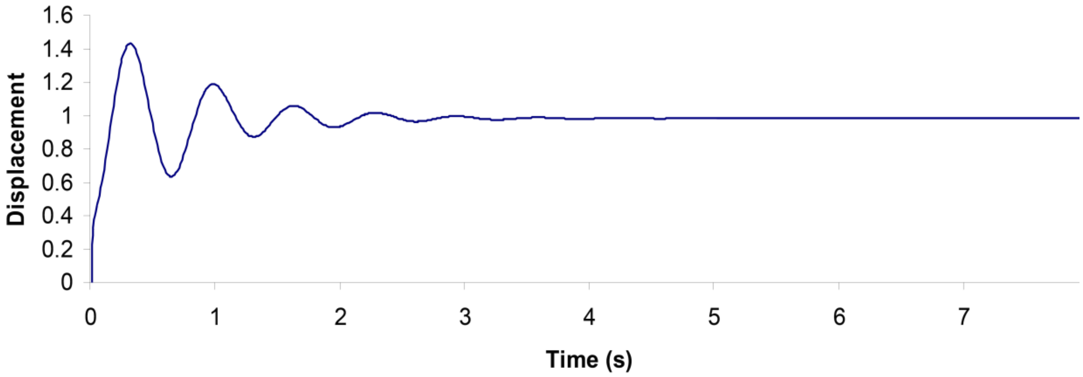


Fig. 3: Vertical acceleration of the tractor cabin suspension optimized using PSO (a), BA (b) ES (C) and reported in Zehsaz *et al.* (2011) study (d)

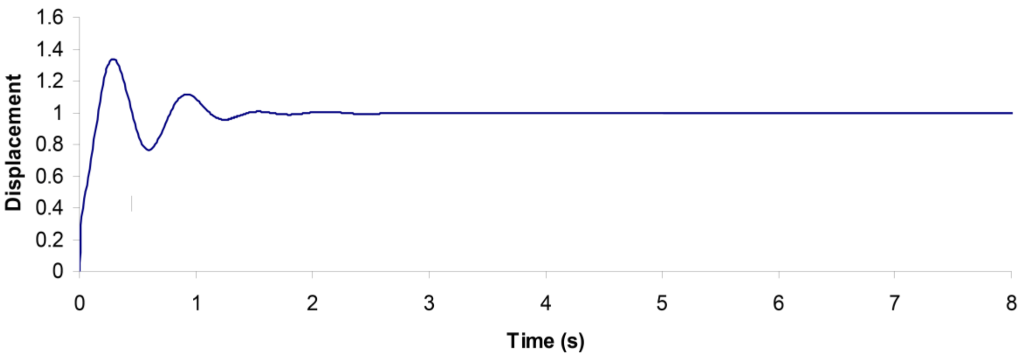
(a)


Displacement
Time (s)
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
0
1
2
3
4
5
6
7

(b)


Displacement
Time (s)
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
0
1
2
3
4
5
6
7

(c)


Displacement
Time (s)
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
0
1
2
3
4
5
6
7
8

(d)

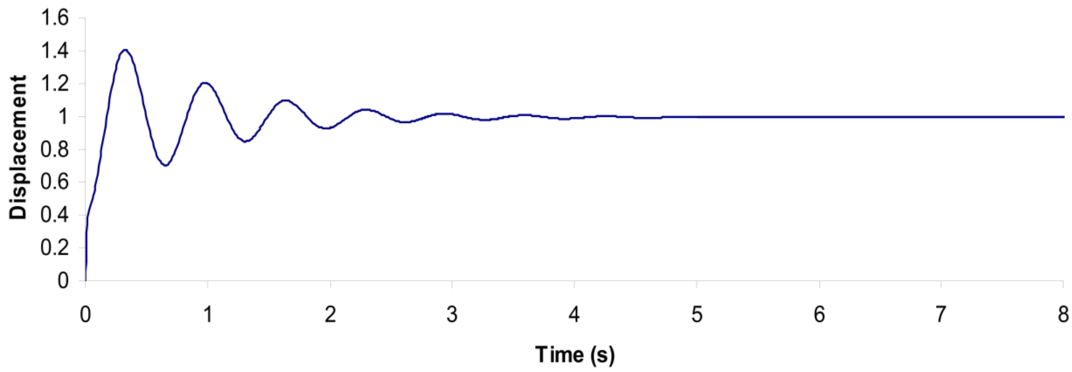


Fig. 4: displacement of the tractor cabin suspension optimized using PSO (a), BA (b) ES (C) and reported in Zehsaz et al (2011) study (d)

Table 1. parameters of the tractor cabin, ITM285-2WD model

M (kg)	I (kg.m^2)	L_1 (m)	L_2 (m)
325.2	196.4	0.7	0.5

Table 2. Values of g_i and W_i according to ISO 2631-1985 (E) based on 1 min duration

Frequency (Hz)	1	1.25	1.6	2	2.5	3.15	4	5	6.3	8	10
RMSAR (m s^{-2})	5.6	5	4.5	5	3.55	3.15	2.8	2.8	2.8	2.8	3.55
Weighting factor	0.5	0.56	0.63	0.71	0.8	0.9	1	1	1	1	0.8

Note: (Tamboli & Joshi, 1999)

Table 3. Optimum values of cabin suspension system obtained from the optimization process

C_2 (Ns/m)	C_1 (Ns/m)	K_2 (N/m)	K_1 (N/m)	Optimization Method
850	943	26199	3927	PSO
524	776	21585	5439	BA
819	730	22530	7742	ES
417	298	17898	12469	Zehsaz *et al.* (2011)

Table 4. Values of the rise time, peak time, settling time and max. overshoot

Max. overshoot (%)	Settling time (s)	Peak time (s)	Rise time (s)	Optimization Method
0.3571	2.4500	0.2875	**0.1250**	PSO
0.4301	3.9125	0.3125	0.1375	BA
0.3384	**1.3625**	0.2875	0.1375	ES
0.4068	2.725	0.3250	0.1500	Zehsaz *et al.* (2011)

Conclusion

In the performed studies, the suspension element characteristics is optimized using ES, PSO and BA. The methods were equally well suited for optimizations of tractor's cabin suspension system. The results showed that a suspension the parameters obtained for the tractor cab suspension (C1 and C2, K1 and K2) using ES and PSO techniques, reduction of the transmitted vibration to the driver. Furthermore, shorter rise and settling time, as well as lower maximum overshoot acquired using ES and PSO techniques, enhanced driver comfort and lowered transmitted force to him/her due to lower acceleration oscillation time. As a result, the suspension optimized in this study is, logically, reduced damage to the driver's torso due to the lower oscillation time both in displacement and acceleration.

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