International Journal of Engineering Inventions
e-ISSN: 2278-7461, p-ISSN: 2319-6491
Volume 11, Issue 3 [May.-June. 2022] PP: 331-335

Effect of the operating conditions on a wheel loader ride comfort

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ABSTRACT:

Operating conditions of wheel loader not only affect the economic indicators of the vehicle, but also affect the ride comfort of the vehicle. In order to evaluate their effects on a wheel loader ride comfort, a half-vehicle dynamic model of a wheel loader is established under the different operating conditions. The weighted root-mean-square (r.m.s.) acceleration of the vertical driver's (a_{ws}) according to the international standard ISO 2631-1 (1997) is selected as an objective function for the effect analysis. The obtained results indicate that the operating conditions of the vehicle greatly affect vehicle ride comfort. The study results are the theoretical basis for the optimal design of vehicle suspension system, cab's isolation system and driver seat's suspension system.

Date of Submission: 02-06-2022 Date of Acceptance: 15-06-2022

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1. INTRODUCTION

Wheel loader often work in harsh environments. On the other hand, it is hardly equipped with a suspension system connecting the axle and the chassis. The vibration sources of the vehicle are transmitted to the driver through cab's isolation system and driver's seat. Whole-body vibration (WBV) exposures in front-end loader operators, and evaluated the effects of traction chains and work tasks on their WBV exposures was measured and compared across three different front-end loader tire configurations: (a) stock rubber tires, (b) rubber tires with ladder chains, and (c) rubber tires with basket chains based on a portable data acquisition system collected tri-axial time weighted and raw WBV data per ISO 2631-1 and 2631-5 standards [1]. Vibration accelerations were measured on a compact wheel loader during 11 operations with two drivers and with/without the activated boom suspension system (BSS). Two standards, ISO 2631-1 (1985) and ISO 2631-1 (1997), were used to assess the effect of wheel loader vibration on comfort [2]. Two dynamic models for the loaders either with the application of vibration absorber system or without that of it were proposed to evaluate the effectiveness of vibration absorber system on vehicle ride comfort [3]. A multibody dynamic model of a compact wheel loader includes the sprung and the unsprung masses of the tractor and the mass of the bucket was proposed to survey the natural frequencies and the accelerations of the dynamic system [4]. A multibody simulation model was used to investigate the possibility to reduce driver vibrations by introducing suspended wheel axles [5]. To improve construction machine ride comfort, a six-degree-of-freedom (d.f.) model of the cab supported by hydraulic mounts with quadratic damping was proposed to explore the low-frequency advantages and characteristics of the hydraulic mounts used for vibration isolation of an earth-moving machinery cab [6]. The lumped parameter model of passive hydraulic damping rubber mount (PHDRM) considering the nonlinearity of multi-inertial tracks was established to improve the ride comfort of the cab of construction machinery and reduce the harm of vibration to the driver [7]. The dual behavior hydraulic engine mounts were introduced around 1980 and passed through many analytic and technical improvements. Hydraulic engine mounts were invented as smart isolators to passively produce a soft isolator at low amplitude and a hard isolator at high amplitude [8]. The main aim of this study is to evaluate the effects of the operating conditions on a wheel loader ride comfort. In order to evaluate their effects on a wheel loader ride comfort, a half-vehicle dynamic model of a wheel loader is established under the different operating conditions. The weighted root-mean-square (r.m.s.) acceleration of the vertical driver's (a_{ws}) according to the international standard ISO 2631-1 (1997) is selected as an objective function for the effect analysis.

2. HALF-VEHICLE DYNAMIC MODEL OF A WHEEL LOADER

In order to analyze and evaluate the effect of the operating conditions on a wheel loader ride comfort, a half-vehicle dynamic model of a wheel loader established under the different operating conditions. In this model, m_s, m_c, m_b and m_f are driver's seat mass, cab mass, vehicle body mass; and I_c and I_b - the mass moment of inertia about the pitch axis of the vehicle body and cab respectively; k_s and c_s - the stiffness and damping coefficients of driver's seat suspension system respectively; k_{ti} and c_{ti} - thoses of tires respectively; z_s, z_c and z_b - the vertical displacements of driver's seat mass, cab mass and vehicle body mass respectively; φ_c and φ_b - the pitch angle displacements of cab mass and vehicle body mass, and the mass of a front mounted lifting implement; q_{ti} - road surface roughness excitations; F_{ci} - the vertical forces of cab's mount system; F_g - an equivalent alternative force of a front mounted lifting implement, l_k - the distances (i=1÷2, k=1÷8).

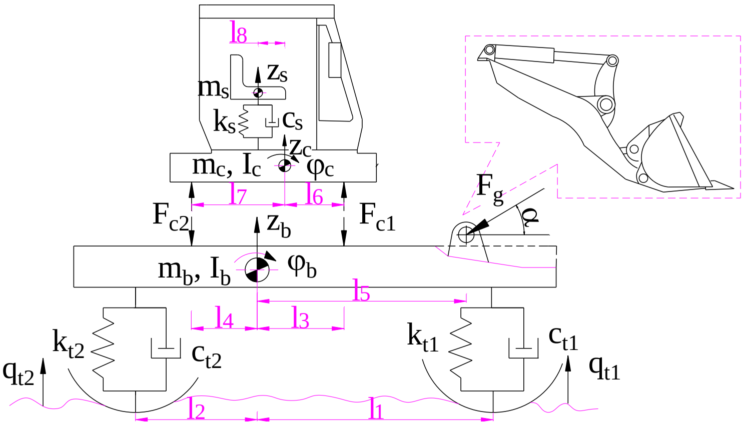


Fig.1. Half-vehicle dynamic model of a wheel loader

The equations of vehicle motion can be formulated in different ways such as Lagrange's equation, Newton-Euler equation, Jourdain's principle. In this study, Newton-Euler equation is chosen to describe the equations of vertical motion of electric vehicle. From quarter-vehicle dynamic model of in-wheel-motor electric vehicle as shown in Fig. 1, the dynamic equations of an electric vehicle are written as follows:

$$\begin{cases} m_s \ddot{z}_s = -F_s \\ m_c \ddot{z}_c = F_s - \sum_{i=1}^{i=2} F_{ci} \\ I_c \ddot{\varphi}_c = \sum_{i=1}^{i=2} (-1)^{i+1} F_{ci} l_{i+5} - F_t l_8 \\ m_b \ddot{z}_b = \sum_{i=1}^{i=2} F_{ci} - \sum_{i=1}^{i=2} F_{ti} + F_g \sin\alpha \\ I_b \ddot{\varphi}_b = \sum_{i=1}^{i=2} (-1)^{i+1} F_{ti} l_i + \sum_{i=1}^{i=2} (-1)^{i} F_{ci} l_{i+2} - F_g \sin\alpha . l_5 \end{cases} \quad (1)$$

where, F_{ti}, F_{ci}, F_s and F_g are the vertical forces of tires, cab's mount system and the equivalent alternative force of a front mounted lifting implement

Road surface excitation: It is described by various mathematical functions. In this study, the random road surface function is selected as the input function of the vehicle dynamic analysis problem. The random road surface roughness of random white noise is used as excitation source waveform for vehicle [1], the random road profile is produced by filtering the white noise using the following mathematical model of the road roughness.

$$\dot{q}(t) + 2\pi f_0 q(t) = 2\pi n_0 \sqrt{G_q(n_0) v(t)} w(t) \quad (2)$$

where, $G_q(n_0)$ is the road roughness coefficient which is defined for typical road classes from A to H according to ISO 8068(1995) [10], n_0 is a reference spatial frequency which is equal to 0.1 m, v(t) is the speed of vehicle; f_0 is a minimal boundary frequency with a value of 0.0628 Hz, n_0 is a reference spatial frequency which is equal to 0.1 m, w(t) is a white noise signal.

3. VEHICLE RIDE COMFORT CRITERIA

The time-domain method can be applied to evaluate the vehicle ride comfort according to ISO 2631-1 (1997) [11], in this study, the vibration evaluation based on the basic evaluation methods including measurements of the weighted root-mean-square (r.m.s.) acceleration defined as

$$a_{wz} = \left[\frac{1}{T} \int_0^T a_z^2(t)\, dt \right]^{1/2} \tag{3}$$

Where, $a_z(t)$ is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurement, s.

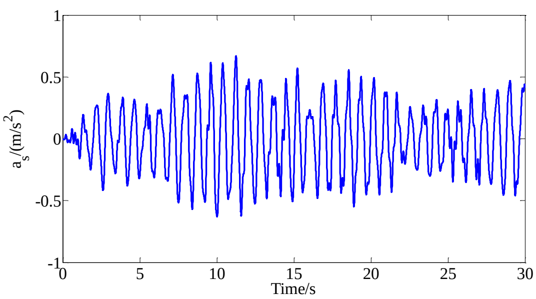
For indications of the likely reactions to various magnitudes of overall vibration in the public transport and vehicle, a synthetic index-called the root-mean-square (RMS) acceleration, a_{wz} can be calculated from formula Eq. (3); besides, the RMS value of the acceleration in vehicle would be compared with the values in Tab.1.

Tab.1: Comfort levels related to aw threshold values [11]

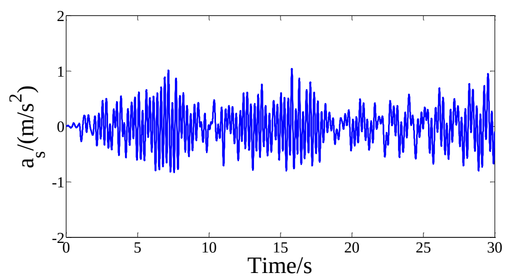
a_w/(m/s^2)	Comfort level
< 0.315	Not uncomfortable
0.315÷0.63	A little uncomfortable
0.5 ÷ 1.0	Fairly uncomfortable
0.8 ÷ 1.6	Uncomfortable
1.25 ÷ 2.5	Very uncomfortable
> 2	Extremely uncomfortable

4. RESULTS AND DISCUSSION

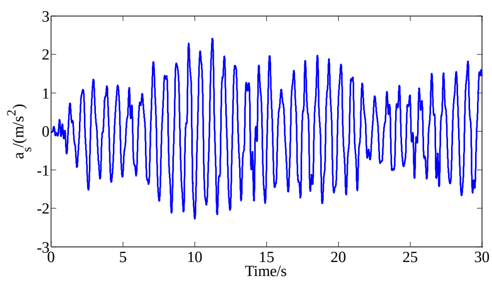
Effect of road surface conditions: The vehicle moving on the different road surface conditions, five road surface conditions from ISO class A road surface to ISO class E road surface according to ISO 8068(1995) are selected for evaluation when vehicle moves at speed of 15 km/h and full load with the design parameters in the reference [12]. The time domain acceleration responses of the vertical driver's seat with variable road surface conditions are shown in Fig.2. From Fig. 2 we see that the peak amplitude values of a_s increase rapidly when vehicle moves on bad road surface. That means that the ride comfort of the wheel load is reduced.



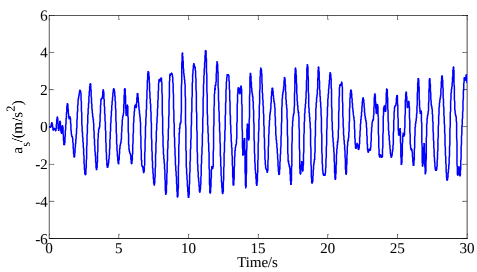
(a) ISO class A



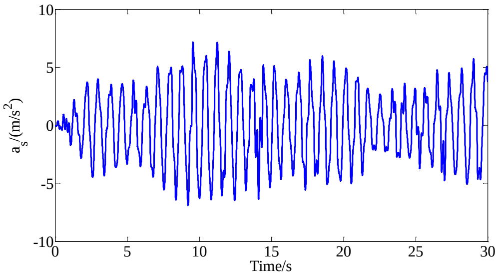
(b) ISO class B



(c) ISO class C



(d) ISO class D



(e) ISO class E

Fig.2. The time domain acceleration responses of the vertical driver's seat with variable road surface conditions

From the results of Fig. 2, we could be determined the values of the root mean square (r.m.s) acceleration responses the vertical driver's seat (a_{ws}) with variable road surface conditions based on the International Standard ISO 2631-1: 1997 which are determined through Eq. (3), as shown in Fig 2. From the results of Fig 3, it shows that the a_{ws} values increase height fast when electric vehicle moves on poor and very poor road surface conditions, especially ISO class D road surface and ISO class E road surface. The a_{ws} values respectively increase 19.97%, 66.64%, 41.00% and 43.00 % when vehicle moves on from ISO class A road surface to ISO class E road surface which make the vehicle comfort worse with bad road surface conditions.

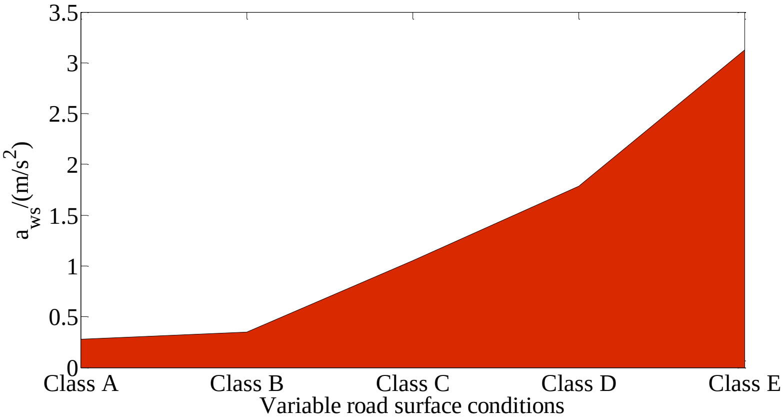


Fig.3. a_{ws} values with variable road surface conditions

Effect of vehicle conditions: The different vehicle speed conditions from 10 km/h to 20 km/h are selected to evaluate the influence of the operating conditions on wheel loader ride comfort when vehicle moves on ISO class D road surface and full load. The a_{ws} values with variable vehicle velocity conditions are shown in Fig.4. From the results Fig 4, we show that the a_{ws} values respectively increase when the value of the electric vehicle speed increases. The a_{ws} values respectively increase 6.96% and 8.09 % when the velocity increases which make the electric vehicle comfort become worse, when the velocity increases.

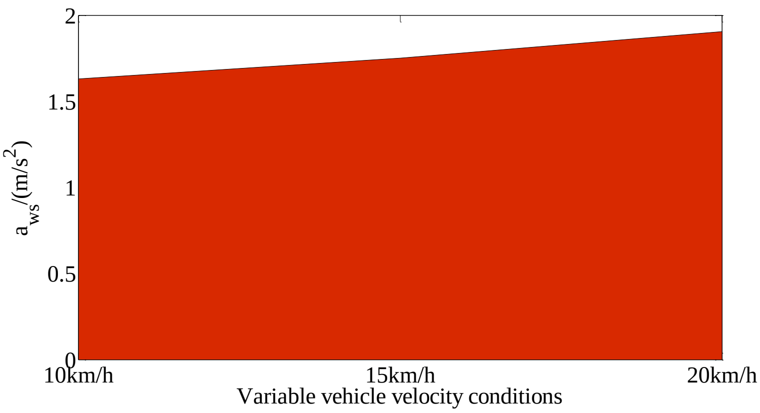


Fig.4. a_{ws} values with variable vehicle velocity conditions

5. CONCLUSION

In this paper, a half-vehicle dynamic model of a wheel loader established under the different operating conditions to evaluate the effect of the operating conditions on a wheel loader ride comfort according to the international standard ISO 2631-1 (1997). The obtained results showed that the peak amplitude values of as increase rapidly when vehicle moves on bad road surface, the a_{ws} values respectively increase 19.97%, 66.64%, 41.00% and 43.00 % when vehicle moves on from ISO class A road surface to ISO class E road surface which make the vehicle comfort worse with bad road surface conditions and the a_{wb} values respectively increase 6.96% and 8.09 % when the velocity increases.

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