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ADJUSTMENT OF SEAT SUSPENSION DAMPING

J. Blekta*, J. Petříček*, J. Mevald*, A. Lufinka**

Abstract: *Now a day, many types of seat used in working machines are constructed for vibroizolation in vertical direction. Seat vibrations in longitudinal and lateral directions are very unsuitable, because seat mechanism is not designed for this type of excitation. This paper shows one of the suitable seat suspension to horizontal seat vibrations minimizing and seat motion damping adjusting.*

Keywords: ***Vibrations, damping, optimization, simulation, multibody.***

1. Introduction

Thanks horizontal vibrations, seat mechanism contact parts are abraded after some time and clearances in guide ways come up. The lateral vibrations increase passive resistance of mechanism also. This is very unpleasant, because in dependence of excitation intensity, passive resistances are changing and damping of whole seat mechanism is changing together.

2. Seat measurement

To verify expectations mentioned above, laboratory measurement with real seat was realized in Hydrodynamic laboratory of Technical University of Liberec. Seat Actimo M by Grammer company was measured - see Fig. 1.

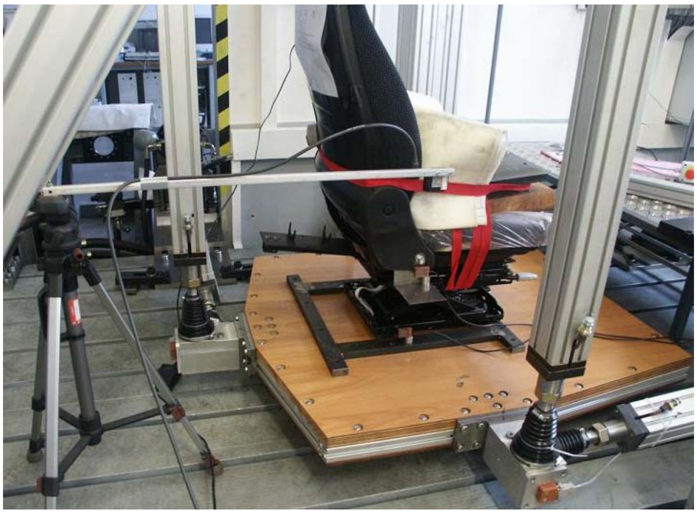

Fig. 1: Seat measurement.

Seat were excited by three signals of 5 mm, 10 mm and 15 mm amplitude. Each of these signals contain harmonics sub-signals of 0.5 Hz, 0.75 Hz, 1 Hz, 1.25 Hz, 1.5 Hz, 2 Hz, 2.5 Hz, 3 Hz, 3.5 Hz, 4 Hz and 5 Hz frequencies. Each frequency was represented by 10 cycles. Seat was excited by this signal triads in longitudinal (x), lateral (y) and vertical (z) direction separately. Measurement was realized with passive mass of 40 kg, 60 kg and 80 kg weight and also with active mass - human of 80

* Ing. Jiří Blekta, Ph.D., Ing. Jiří Petříček and assoc. prof. Ing. Josef Mevald, CSc.: Department of Applied Mechanics, Technical University of Liberec, Studentská 2; 461 00, Liberec; CZ, e-mails: jiri.blekta@tul.cz, jiri.petricek@tul.cz, josef.mevald@tul.cz

** Ing. Aleš Lufinka, Ph.D.: Department of Design of Machines Elements and Mechanisms, Technical University of Liberec, Studentská 2; 461 00, Liberec; CZ, e-mail: ales.lufinka@tul.cz

kg weight. In this case, only signal of 5 mm amplitude was used, because norm limits for test with human body have not to be exceeded.

Seat squab position in relevant direction was measured by laser sensor. Position of platform board desk was measured by LVDT sensors built in hydraulic motors. On seat squab and platform desk accelerometers were mounted to measure of acceleration of these parts.

In the Figs. 2, 3 and 4 bellow dependence of transmission ratios of seat squab displacements in *x*, *y* and *z* directions for 80 kg passive weight on excitation frequency are shown. Figs. 5, 6 and 7 show the same dependence for active weight.

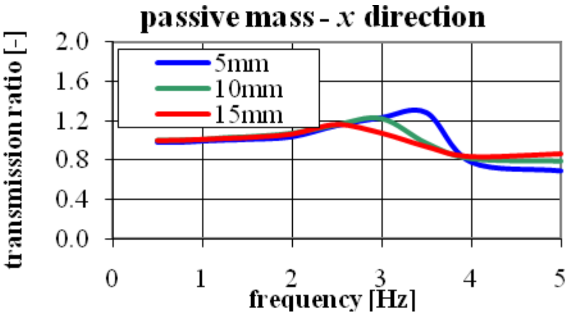

Fig. 2: Trans. ratios - passive mass, x direct.

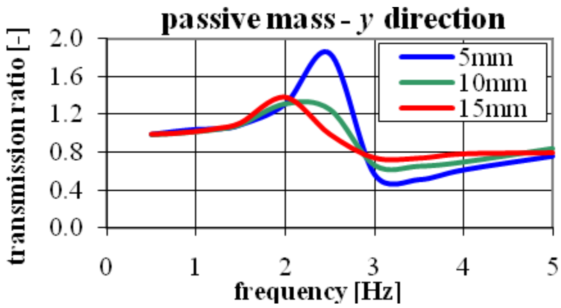

Fig. 3: Trans. ratios - passive mass, y direct.

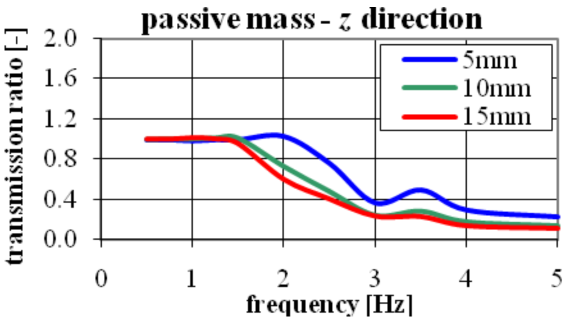

Fig. 4: Trans. ratios - passive mass, z direct.

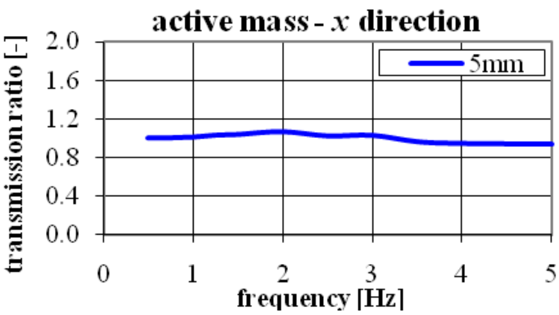

Fig. 5: Trans. ratios - active mass, x direct.

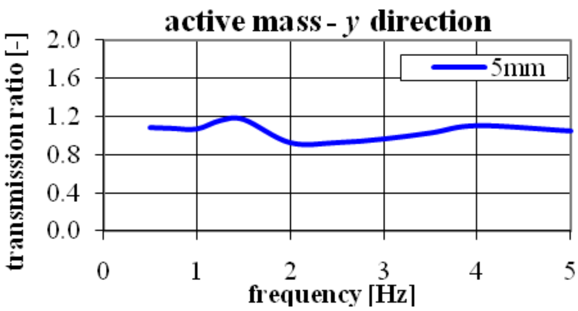

Fig. 6: Trans. ratios - active mass, y direct.

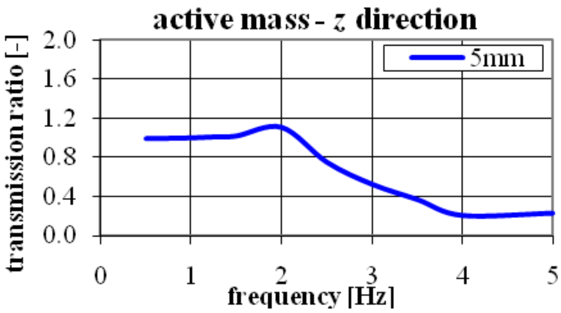

Fig. 7: Trans. ratios - active mass, z direct.

From figures above follows, that exciting signal is reduced for most of frequencies in case of vertical direction. This can´t to be said for *x* and *y* directions however. Vibration transfer in *x* direction is a little bit better than in *y* one (better stiffness of seat mechanism in *x* direction). Peak of 5 mm curve in Fig. 3 is done by comparable value of exciting signal amplitude and seat mechanism clearances.

From reasons mentioned above is very suitable to minimize seat vibrations in horizontal directions. One possibility how to do it is thinking about a proper seat suspension.

Suitable seat suspension can be realized by mechanism of parallelogram type. It can be e.g. desk hanged by three arms on a frame of working machine. This is shown in Fig. 8. Desk with mounted seat has 3 degrees of freedom - translational movement in longitudinal and lateral direction and rotation about vertical axis. By proper parameters setting (length of arms, desk movement damping) a good vibroizolation in horizontal directions can be achieved.

3. Application of parallelogram seat suspension

This type of seat suspension was applied in Doly Nástup Tušimice, where vibroizolation of colliery machine driver was requested (Blekta et al., 2010). Cabin with driver seat is placed inside of cabin frame. Cabin frame is excited by motion in general direction (by mining of bucket wheel). Originally, cabin was mounted to frame by rigid joints. These results in the very intensive driver vibrations, root mean square values (RMS) of seat acceleration in horizontal directions were measured about 0.1 g during c. 50 s of measured data.

A new conception of cabin suspension is shown in Fig. 9. This conception is based on cabin motion unlocking in two horizontal direction (x and y) and one cabin torsion motion.

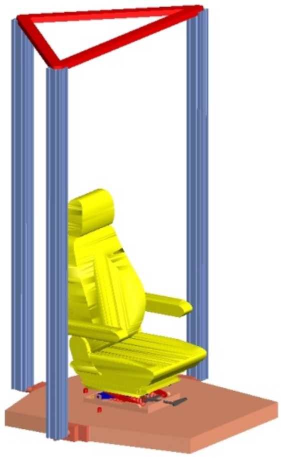

Fig. 8: A new conception of seat suspension.

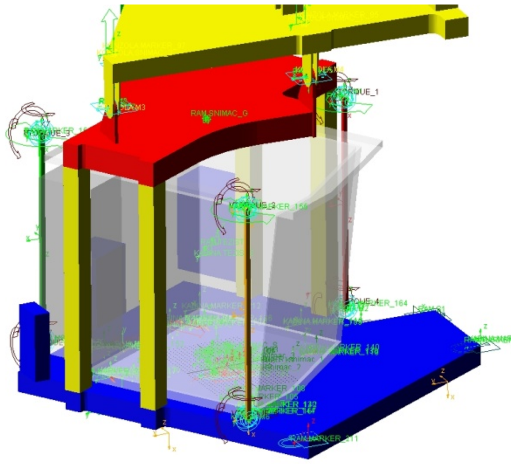

Fig. 9: A new cabin suspension.

4. Optimal cabin motion damping

Ideally, cabin movement could not be damped. Damping makes worse vibration transmition between cabin and cabin frame. In other side, zero damping makes unpleasant cabin pitching when driver get on the cabin floor. Another unpleasant pitching makes random impulses, when bucket-wheel mines in hard subsoil.

From reasons mentioned above there was constructed friction damper (between cabin frame and cabin floor). This damper is shown in Fig. 10. One of the most important advantage of friction damper is damping value independent upon cabin position in the cabin frame (in the cases, that machine don´t work in ideally horizontal position). Damper is made from a friction desk pressed by spring to the cabin floor. By spring preload friction force value can be changed. Spring preload can be easily set by four screws.

Optimal cabin motion damping was computed from model in MSC.ADAMS software. It was set on the base of the seat squab acceleration RMS value. Model was excited by signal measured on real machine. In Fig. 11 dependence between seat squab acceleration RMS value and spring preload is shown.

Fig. 10: Friction damper.

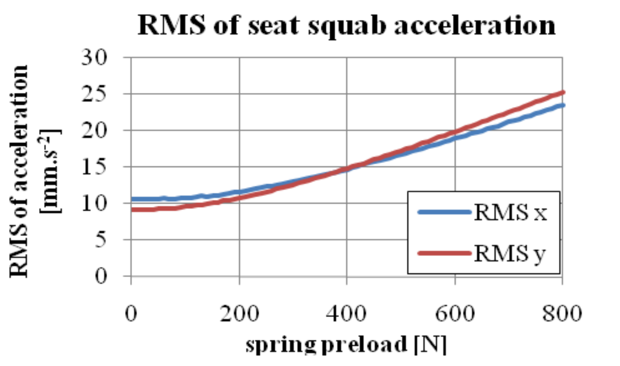

Fig. 11: RMS of seat squab acceleration vs. preload.

For driver getting on board simulation we used force impulse of T = 0.5 s duration. The value of force is done by momentum, which is passed to cabin. For human weight m = 80 kg and his velocity v = 2.5 m.s^{-2} resultant force is

$$F = \frac{m \cdot v}{T} = \frac{80 \cdot 2.5}{0.5} = 400 \left[N \right] \tag{1}$$

From a lot of simulations value of optimal spring preload was set to 400 N. Criteria for value setting was chosen by maximum signal oscillations during transient condition excited by above mentioned force impulse and random kinematic impulse, which was inserted in exciting signal measured on real machine (at time 15 s) in lateral and longitudinal direction. Result seat center of mass displacement is shown in Figs. 12 and 13.

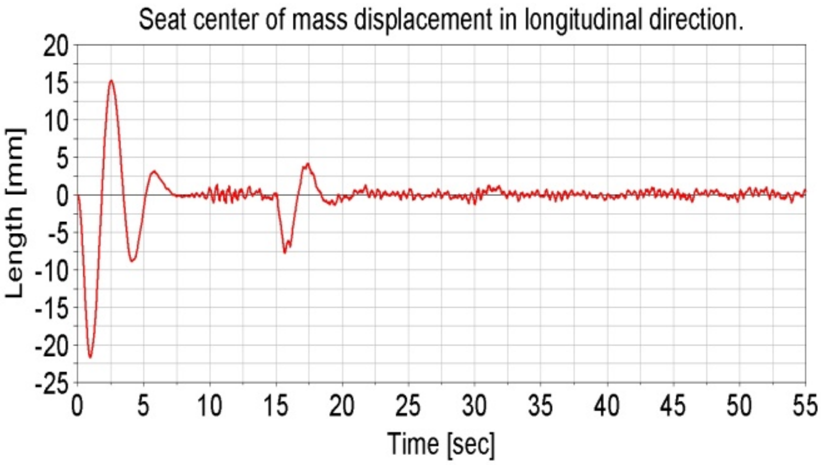

Fig. 12: Seat squab displacement - x.

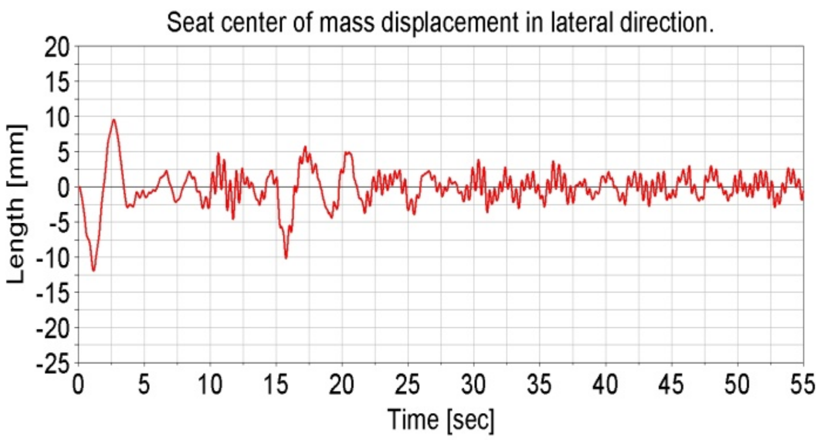

Fig. 13: Seat squab displacement - y.

5. Conclusions

Finally, spring preload was set to 400 N. After one month of machine running RMS of horizontal acceleration of seat central mass was measured c. 0.01 g. It is tenth of origin value.

Acknowledgement

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Reference

Blekta, J., Petříček, J., Mevald, J., Bocko, P., Sivčák, M. (2010) A new cab and seat mount of bucket wheel excavator Schrs 1320, in: Proc. 12th Int. Conf. Applied Mechanics 2010, Technical University of Liberec, Liberec, pp. 13-16.