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OPTIMIZATION OF LOADER CABIN

OPTIMALIZACE KABINY NAKLADAČE

Abstract

Difficult operating conditions at mining put high demands on technical responsibility of mechanical machines and influence the operator of mechanical machines in a negative way as well. The external influence of the operator has a negative impact on health and also on the general working output. The basic negative factors influencing the operator include vibrations, noise, temperature etc. Vibrations cause additional cyclic stress on construction parts of mechanical machines and lower their working life. Therefore it is necessary to mineralized vibrations.

Abstrakt

Náročné provozní podmínky při těžební činnosti kladou vysoké nároky na technickou spolehlivost strojních zařízení a také negativně působí na obsluhu strojních zařízení. Působení vnějších vlivů na obsluhu má negativní vliv na zdraví, ale I na celkový pracovní výkon. Mezi základní negativní faktory působící na obsluhu patří vliv vibrací, hluku, teploty atd. Vibrace způsobují dodatečné cyklické namáhání konstrukčních dílů strojních zařízení a neúměrně snižují jejich životnost, proto je třeba vibrace minimalizovat.

1 INTRODUCTION

In this case the excessive vibration of loader cabin where at working engine speed (1450 rpm.) immoderate increase of vibration was reached were solved. The machine operator complained about the immoderate vibration. That was why we were asked by our customer to take measurements and consequently to prepare the project of cabin optimization.

2 BASIC MACHINE CHARACTERISTICS

The machine is driven by a four-cylinder Diesel engine cooled by water Cummins B 3.3 with the volume 3300 cm^3 and power 60 kW. The free-running engine speed is at 850 rpm (14.2 Hz), working revolutions for the hydraulic systems operating are set to 1450 rpm (24,2 Hz), the reduction of maximum speed is set to 2450 rpm (41 Hz). The cabin of the machine is made as a weldment from profiles and connected with the basic frame by four rubber silent blocks. The turning of the cabin is hydraulic operated with a gear by means of gear ring with an inner tooth system and a pinion.

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3 THE REALISATION OF MEASUREMENT AND PROCESS OF EVALUATION

Simple Vibropen by SKF appeared as an invaluable help before the measurement. By using it we were able to detect measuring places with biggest vibrations very quickly. These places will be measured later again by means of more sophisticated measuring facilities which however demand more time. Regarding the fact that there was accentuated the shortest length of measuring the measuring places had to be reduced to minimum. By means of Vibropen we were able to choose by this simple way suitable measuring places which showed vibrations of the whole cabin. The measuring places are shown in the figure 1. The most interesting measuring was made on the frame of the machine and on the cabin latchet by means of the analyser Adash VA 4 Pro. For the further evaluation of the measured results a programme DDS 2007 by Adash Company was used. At these places the cabin is connected with the frame on four places by means of four rubber silent blocks, see figure 2. The measured values of efficient values are shown in the individual chart num. 1. From the chart it is clear that the vibration values on the cabin latchet on the right side reached even higher values than directly on the frame of the machine. On the right back latchet it is double compared to the vibration value directly on the frame below this place. From these values we can assume that the machine is probably operated in a resonance area or owing to bad or improperly chosen rubber silent blocks the vibrations are spread from one cabin latchet to another.

The next matter of interest is a low size of amplitude on the speed frequency in some cases (its size depends on revolutions) but high amplitude is the second double of the speed frequency, see figure 3. The dominance of the second harmonic frequency is caused by using a four-cylinder Diesel engine where two firings in working cylinder fall to one revolution of a crank shaft. Regarding that it is necessary to count with the fact that the cabin is excited by a speed component part but also its double. The scheme of the spectrum of vibration points on the left side of the cabin is in the figure 6 and for better comparison of the amplitude size at individual frequencies it is also shown in the figure 5. Apparently, the closer to the frame we are so the dominant amplitude becomes the amplitude of vibrations on the second harmonic.

Tab 1: Efficient vibration values on the machine frame and on the cabin latchet in a horizontal direction.

Measuring point	1 (LF)	2 (LB)	3 (RB)	4 (RF)
Efficient values of the vibration velocity on the frame of the machine RMS 10÷1600 Hz [mm/s]	1.7	1.6	1.0	1.2
Efficient values of the vibration velocity on the cabin latchet RMS 10÷1600 Hz [mm/s]	1.4	1.6	2.1	1.3

To confirm or disprove the resonance origin there were made measurements of the starting right on the cabin latchets and on the machine frame, see figure 4. We can assume from the measuring that there is a possible resonance originating in the area of working revolutions and also the increase of vibrations about 1050 a 1800 rpm. The size of vibration amplitude at 1800 rpm is much more smaller than at the previous ones. With a further increase of revolutions it comes to the stabilization of the machine, lowering of vibrations and also to the lowering of the amplitude of the second harmonic component part and finally to the increase of the amplitude on the speed frequency and to its dominance. The highest vibration values appear at 1470 rpm, which are in fact working revolutions of the machine. This high vibration value of the machine influence in a negative way the effective life of the machine and also the working area of the operator.

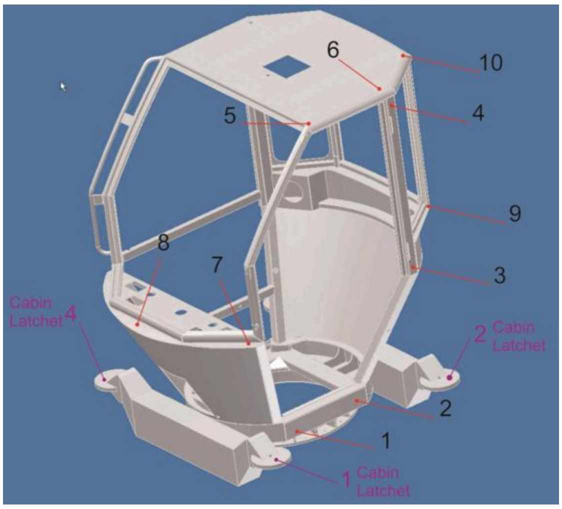


Fig.1 Representation of measuring places on the left side.

Fig.2 Measuring places on the cabin latchet and on the frame.

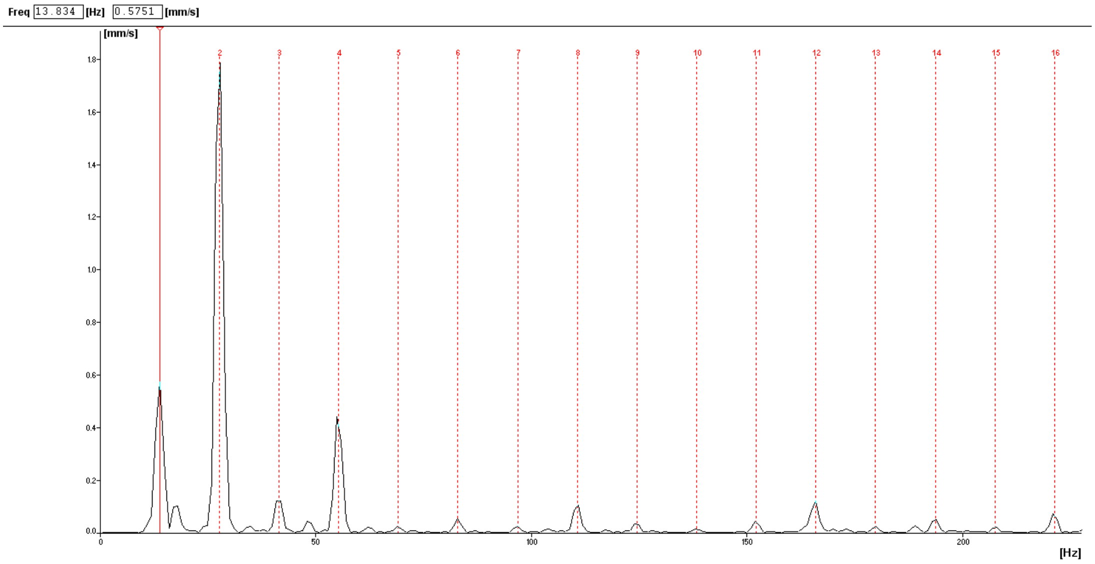


Fig.3 Spectrum of vibration amplitudes on the cabin latchet, the first amplitude is on the speed frequency 1.8 Hz (830 rpm).

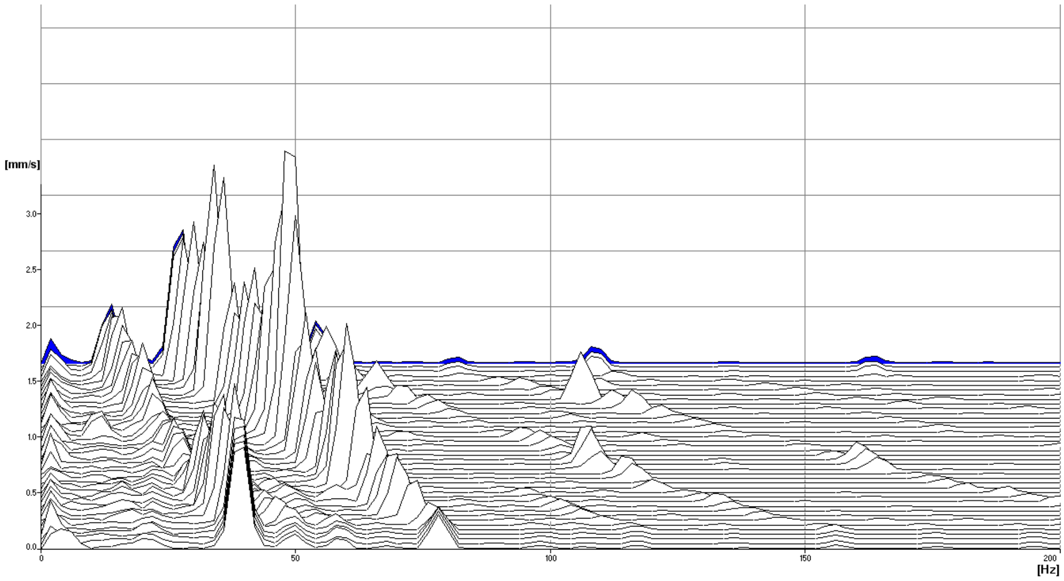


Fig.4 Spectrum of right side cabin starting.

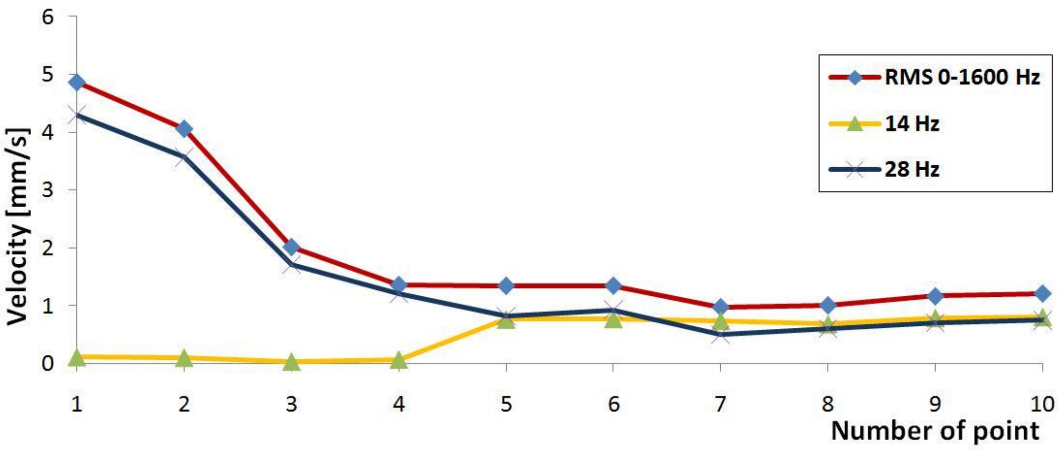


Fig.5 Comparison of the size of velocity amplitude of particular left side points.

4 MODAL ANALYSIS OF THE CABIN

To detect resonance frequencies of the cabin and its further optimization there was created an adapted 3D cabin model and made simulations of modal analysis in the Autodesk Inventor Professional 2010 programme. Regarding high demands on the computer hardware the analysing model had to be adapted and eliminated of all components which did not have any connection with the analysis. The original model (delivered by a company) contained parts with adjustments for welding (gaps, chamfers, etc.). These adjustments had to be eliminated or adapted so that the connection of particular parts could be without gaps and the model conformed to the principles for the programme. Otherwise the software could not be able to cooperate with the model.

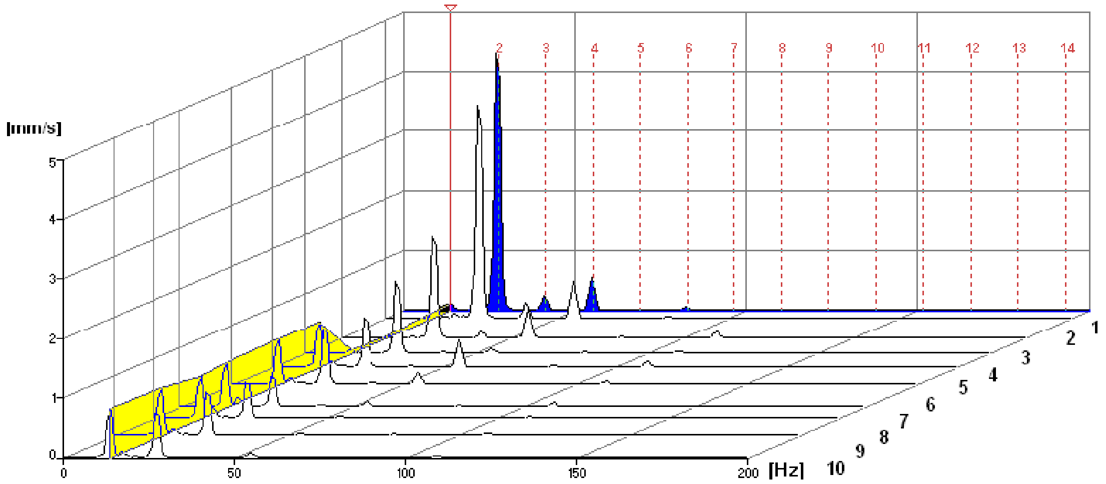


Fig.6 Spectrum of vibration amplitude of the left cabin side.

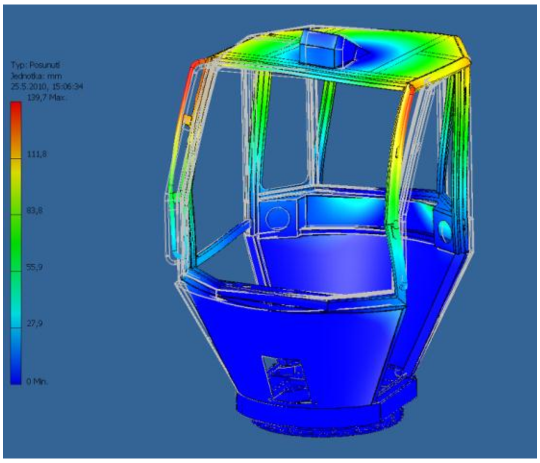
Some component parts are not included in the calculated model. These parts have an influence on the resonance frequency of the cabin such as glass, door, protective frames etc., however their creation with practical features would be practically impossible or they would put immoderate demands on the computer hardware. Therefore the calculated frequencies will not be completely accurate. They will be partially distorted but still they are able to inform us about the resonance features of the cabin. After realization of the adaptations the simulations of modal analysis were made. For completeness´ sake there are shown the first four frequencies of the machine cabin:

- mode 1 22.39 Hz $\cong$ 1340 rpm
- mode 2 27.86 Hz $\cong$ 1670 rpm
- mode 3 54.35 Hz $\cong$ 3260 rpm
- mode 4 70.71 Hz $\cong$ 4240 rpm

It is obvious from the results that the machine works in the area of its own cabin resonance frequency. The representation of the cabin behaviour at resonance modes is shown in the figure 7 and 8. The oscillation shapes are at the modes 1 ÷ 3 very similar therefore it is enough to show only one figure relating to the cabin behaviour.

If we analyse the problem deeper we will realize that the cabin is in the vibration spectrum waken up also by the second harmonic component part (see picture 3 and 4). Then we will find out that the revolutions where the resonance origin might occur are by half smaller. So we can reach the particular modes much more earlier. After the check calculation we will come to the following values of the frequencies and revolutions:

- shifted mode 1 11.2 Hz $\cong$ 670 min^{-1}
- shifted mode 2 13.93 Hz $\cong$ 720 min^{-1}
- shifted mode 3 27.18 Hz $\cong$ 1630 min^{-1}
- shifted mode 4 35.36 Hz $\cong$ 2120 min^{-1}



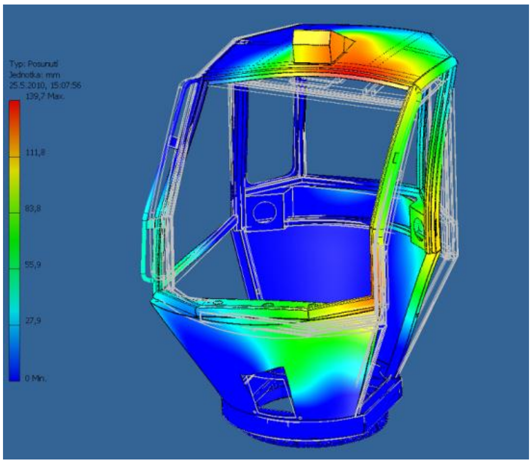


Fig.7 Mode 1÷3 showing the oscillation shapes of the cabin [1].

Fig.8 Mode 4 [1].

The result is a possibility of theoretical origin up to six resonances within the range of the engine. Its origin and demonstration on the cabin construction depends on several factors and their mutual combination. It is important to count with the fact that the cabin is placed on four rubber silent blocks and over them different spectra with various vibration size are transferred. Sometimes the cabin is waken up mainly by the revolution component part, sometimes by its double (the second harmonic). This also depends on the rubber silent blocks, on their attenuation characteristics, on revolutions, on the construction stiffness, etc.

It is also necessary to notice the unusual proximity of mode 2 frequency and the shifted mode 3 frequency, the difference is insignificant. Therefore we assume that near these frequencies it comes to the collision of two resonance modes and to their addition, which is finally displayed by much more significant vibrations in the spectrum than at other modes. After the calculation this value of critical revolutions is approximately 1650 rpm. At the analysis of picture 4 where there is a record of vibrations at starting we will detect the highest amplitude on the value 1470 rpm. When compare the measured and calculated values of revolutions with the highest value of vibrations we will come to the conclusion that these revolution values are very similar with a difference about 180 rpm. We can say that the top in the starting spectrum at the value of 1470 rpm was caused by the combination of the shifted mode 3 and mode 2.

If we further go through the measured starting spectrum we can give the second highest top in the spectrum which originated at the revolutions of 1050 rpm to the revolution mode 1. The difference between the reality and calculation is at this time 290 rpm. To last top in the spectrum whose value is 1800 rpm belongs to the shifted mode 4. All other possibilities are run out as the shifted modes 1 and 2 cannot be found in the spectrum. They are overcome right at the machine start, when free-running revolutions are reached.

5 CONCLUSIONS

By means of the programme Autodesk Inventor Professional 2010 and after the realisation of the modal analysis in this programme there were detected own cabin resonance frequencies which match with slight differences the reality. After analysing it was found out that the machine cabin is operated in several resonance areas, which is an inconvenient state that causes immoderate vibrations. To prevent the cabin resonance the complete rebuilding of the cabin including its placing would be necessary. This rebuilding would require high financial investment. However the company requirement was to design a solution which would be cheap and would improve the present state.

The first and at the same time the simpliest solution is to change the present cabin rubber silent blocks Contitech 212706 for at new type Contitech 210470 which is harder and has at higher frequencies better damping characteristics. Regarding the size the rubber silent blocks are very similar and after the introduction of several construction adaptations they can be replaced.

Next proposal which should lead to the improvement of the present state is a set of particular simple construction adaptations whose aim is to move the resonance frequencies out of the working area (1450 rpm). These adaptations have to push the resonance mode 2 and shifted mode 3 out of the working area and at the same time the resonance mode 1 is not allowed to get to the area. Further suggested construction adaptations are for example to add a reinforcement of the back side, a reinforcement of the top cabin part, an adaptation of the back cabin column and so on. These adaptation should according to new simulations ensure a suitable displacement of the resonance frequencies out of the working area. After their application we will find out if they were right.

At next proposals of a new cabin construction but also of other mechanical machines it is necessary to count with a possibility of resonance origin and to reach the resonance displacement out of the range of machine revolutions by suitable construction adaptations. Otherwise if we neglect the standards we will face the risk of a bad construction arrangement and the origin of further resonances in the working range of the machine, which additionally can cause big complications.

REFERENCES

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