Unexpected Vibrations of Relatively Simple Cutting Machine Mechanism

P. Šidlof, Z. Braier, P. Klouček and J. Ondrášek

Abstract Unexpected vibrations and resonance with frequency three times the rpm were detected in a relatively simple cutting machine during process of inertia force reduction (balancing). A knife of the cutting machine is actuated by two identical crank mechanisms at the ends of a crossbeam. The vibrations induced inertial forces larger than the considerable machine imbalance. The crank length to connecting rod length ratio was small enough that the corresponding excitation kinematic acceleration overtone component was practically negligible. The vibration characteristics consisting mainly of the drive shaft gyratory vibrations was determined based on measurements of forces transferred to floor, accelerations of the mechanisms and on a CAD model calculation involving the mechanism main members compliances. Improvements were proposed and tested upon the results with focus on vibrations elimination.

Keywords Measurement · Unexpected vibrations and resonance · Balancing improvement · Inertial forces · Simple cutting machine

1 Introduction

During test measurement of inertia force balancing on a relatively simple cutting machine resonance vibrations with high amplitudes were detected. Vibration frequency was three times the drive shaft rotation frequency of the crank mechanism.

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The machine was used in practice only at frequencies above or below resonance rpms, since it proved not to work well in this bandwidth.

After the machine was balanced, the resonance frequency increased thus shifting into the frequency used in practice. Although amplitudes of resonance vibrations decreased significantly compared to the unbalanced situation, the resonance caused undesirable increase in the noise.

The problem was solved by measurement and calculation. To measure the original state of the machine and the balanced state, the machine was settled on dynamometers for measurement of forces acting to the ground and equipped with accelerometers measuring the movement of the cutting knife. A laser triangulation displacement sensor to ascertain deflection of the drive shaft was also used. All measurements were done as the function of rotation angle of the drive shaft. Incremental rotary encoder was connected to the end of the drive shaft and measured also the angular velocity of the shaft. Calculations of resonance frequencies were carried out on a model of the machine by use of Adams software [1].

2 Measurements and Calculations

Figure 1 shows a schema of the machine placed on multicomponent dynamometers measuring forces acting on the floor. The electromotor powered from a frequency converter drives the drive shaft via a V-belt. The drive shaft is mounted in two bearings. The drive shaft carries a relatively heavy adjustable eccentric placed to the right of the shaft's centre. The eccentric powers a light feeding mechanism, which is not depicted. Flies and cranks are placed on the overhanging ends of the shaft. By means of long connecting rods these power the crossbeam with the cutting knife fastened in the centre. The crossbeam is guided by two sleeve bearings and is symmetrical. The driving crank mechanisms have very long connecting rods and thus stroke of the cutting knife is almost purely sinusoidal.

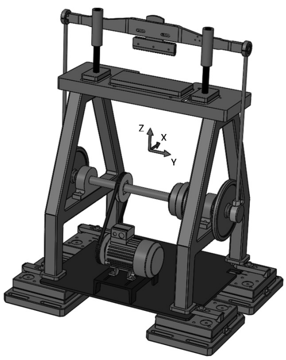

Fig. 1 Schema of the cutting machine settled on multicomponent force platforms

2.1 Measurement Before and After Balancing of the Machine

Four 3-component piezoelectric dynamometers Kistler with excellent dynamic properties (stiffness in the vertical direction ≈2 kN/μm) were used to measure the forces acting to the floor. Sensors of vertical acceleration of the cutter crossbeam were placed near the sleeve bearings. Incremental encoder of angle and angular speed was attached to the drive shaft.

Figures 2 and 3 show the amplitude multispectrum of the measured horizontal force *X* and vertical force *Z* before balancing of the machine in idle run. The forces were evaluated as a sum of corresponding components of forces measured in all the four dynamometers. There are high amplitudes of the fundamental frequency (first harmonic) in spectrum of the *X* force, which are caused by rotational imbalance of the machine. Spectrum of the *Z* force shows high amplitudes in the third harmonic with a maximum at 1350 rpm. The plotted spectrum of the cutter acceleration is very similar to the spectrum of the *Z* force with the exception of the first harmonic, only amplitudes of higher than third harmonic are relatively lower.

Figure 4 shows the course of the *Z* force during resonance depicted depending on the angle of the shaft. The diagram starts at the top dead centre of the cutting knife. The first harmonic is significantly phase-shifted due to unbalanced centrifugal forces. With respect to the geometry of the crank mechanism the ideal force *Z* (without

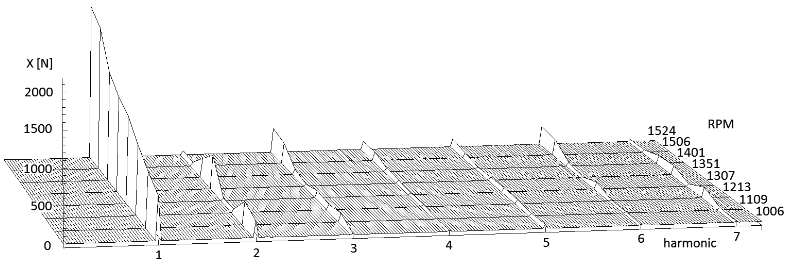

Fig. 2 Amplitude multispectrum of the resulting *X* [N] force before balancing of the machine

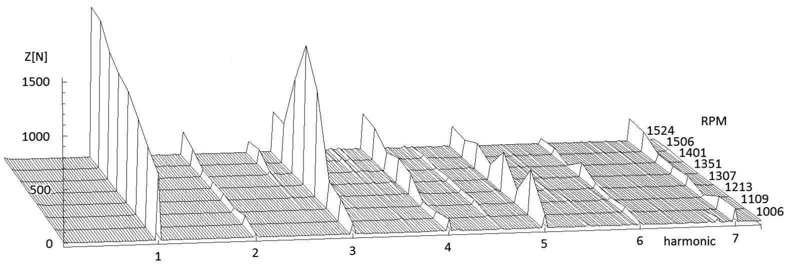

Fig. 3 Amplitude multispectrum of the resulting *Z* [N] force before balancing of the machine

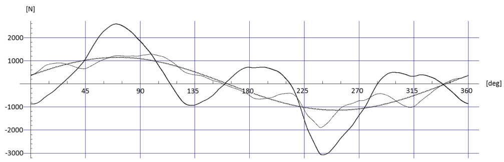

Fig. 4 Resonance force *Z* (*thick line*), force *Z* with filtered-out third harmonic (*thin line*) and first harmonic component of the force (*dotted*) before balancing at 1351 rpm

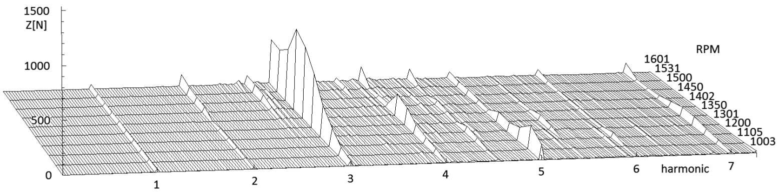

Fig. 5 Multispectrum of the resulting force *Z* [N] after balancing of the machine

effects of flexibility and backlash) should correspond to almost exactly its measured first harmonic, which has an amplitude of 1030 N. The real force, however, has the peak value almost three times higher. The main culprit here is the third harmonic with amplitude of 1370 N, which ideally should not be exceeded at all.

The balancing process include lightening of the adjustable eccentric on the drive shaft, centrifugal forces were balanced and a mechanism with contra-rotating balancing eccentrics was used to balance the vertical inertial force of the crank mechanism. Having employed these adjustments we carried out our measurements again. The horizontal *X* force practically disappeared. On the vertical inertial force *Z* the first harmonic dropped to almost zero (see Fig. 5), but resonance on the third harmonic remained with a peak at 1450 rpm. Although the maximum amplitude dropped to 780 N (by 42 %) the resonance frequency was now shifted to the range of operation rpms of the machine. Figure 6 depicts the *Z* force during resonance. To clarify an approximate calculation was carried out.

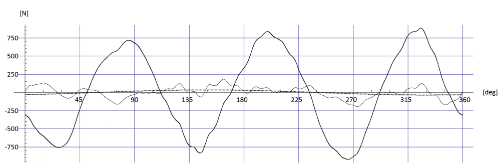

Fig. 6 Resonance force *Z* (*thick line*), force *Z* with filtered-out third harmonic (*thin line*) and first harmonic component of the force (*dotted*) at 1450 rpm after balancing

2.2 Calculation of Resonance Vibrations of the Mechanism

With respect to the complicated effects of flexibility and gyroscopic moments calculation of resonant frequencies of the mechanism was carried out in the Adams software. As input we used the CAD model of the mechanism, where the drive shaft with flywheels, the connecting rods and the cross beam with the cutting knife were considered flexible. First the resonant frequencies with the shaft set at top dead centre of the cutting knife were determined. The corresponding resonant frequency was calculated at 74.6 Hz and main deformations on the mechanism consisted mainly in simple bending of the drive shaft in the vertical plane. Resonant frequency in rotation was determined by use of an approximation approach calculating run-up to a constant angular speed of the drive shaft. For the vibrations to appear a very low modal dumping had to be set in the model. Resonant frequency was found at 72.6 Hz at 1452 rpm, but with amplitudes ten times lower than in reality.

Figure 7 shows the calculated movement of the shaft axes in the place of the connecting rod axes in vertical direction and Fig. 8 shows this process with filtered-out first harmonic. Movement of the shaft axis in the middle of machine is very similar but has an opposite sign and is somewhat larger. The shaft axis moves to a lower extent also in the horizontal direction.

The derived deflection of the shaft depicted in coordinate system fixed to the shaft with z axis in direction of the crank is on Fig. 9. Vectors indicating the concrete deflection have attached a number describing the corresponding angle of the shaft rotation. During one rotation of the shaft the curve runs twice.

Deflection of the shaft without the first harmonic is depicted on Fig. 10. The curve, again, runs twice during one rotation of the shaft.

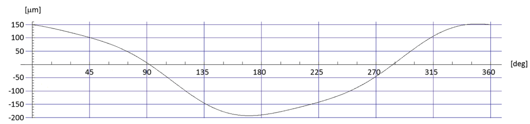

Fig. 7 Calculated movement of the shaft axes in the place of the connecting rod axes in z direction (vertical) at 1452 rpm

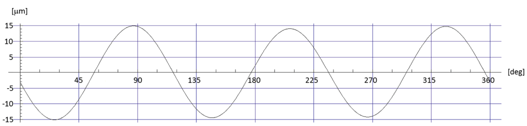

Fig. 8 Calculated movement of the shaft axes in the place of the connecting rod axes in z direction without the first harmonic at 1452 rpm

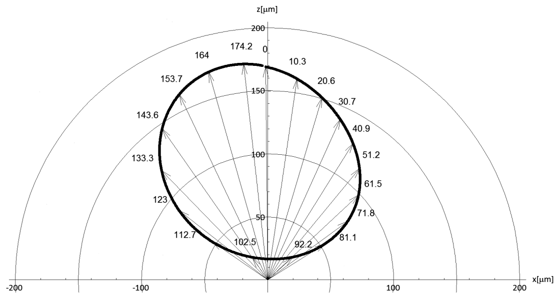

Fig. 9 Deflection of the end of the shaft at 1452 rpm. The *z* axis is in the direction of the crank

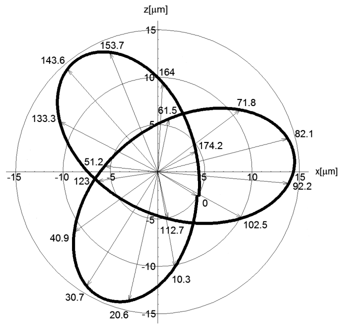

Fig. 10 Deflection of the end of the shaft without the first harmonic at 1452 rpm. The *z* axis is in the direction of the crank

3 Conclusions

The unexpectedly strong vibrations of the cutting machine mechanism are probably not caused by linear effects, since on a linearized computational model it shows insignificant amplitudes. The vibrations are not random, since they appear also on other similar machines. The measured unevenness of the angular speed of the drive shaft, which might cause the vibrations [2, 3] is very low (below 0.2 %). Also torsional vibration of the shaft is very low. No significant corresponding resonance amplitudes were found in the frame and the base plate with the electromotor and beam of the cutting knife bending vibrations are very low. Products of inertia of imperfectly balanced flywheels, nonlinear effects of backlash in bearings, their non-linear stiffness and solid friction in guiding of the cutting knife in dead centres should not have any significant effect on the vibration with third harmonic [2, 3].

To eliminate vibrations increase of diameter of the shaft between the bearings from the original 35–42 mm had been proposed. The calculated resonance frequency thus increased to 97 Hz at 1895 rpm. Although the calculated frequency for the shaft at 35 mm corresponded very well with the measurement, a resonance appeared on the mechanism with third harmonic at 1600 rpm (80 Hz) with the *Z* force amplitude of 560 N during tests on a 42 mm shaft.

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