ISSN 1392-2114 ULTRAGARSAS, Nr.2(59). 2006.

Calculations for diagnostics of influence of grinding wheel and electric motor unbalance on grinding wheel wear at cylindrical grinding

A. H. Marcinkevičius

Vilnius Gediminas Technical University

Abstract

Different aspects of grinding, its stability and arising of self excited vibrations are investigated by many authors. Part of them researched the phenomenon in practice at grinding, the other theoretically and proposed the explanation of grinding stability from different points of view. In our paper successive development of self excited vibrations resulted by initial unbalance of grinding wheel and rotor of electrical motor of a wheel spindle drive is examined. The grinding wheel is dressed in an open loop elastic system of a machine tool while and run-out of its surface according to axis of center pins of grinder is cut off. The grinding is performed in a closed loop elastic system which stiffness differs from an open loop system and it results in some run out of the wheel surface at grinding according to the workpiece. This run out is summing with run out resulted by unbalance of an electrical motor of a wheel drive. Complex periodical oscillations with run out frequencies are excited resulting propagation of the waveness on the workpiece and on the wheel. All it at summing result the impacts in the system which leads to self excited vibrations with natural frequency of some more weak element in a system. Calculation equations are proposed for evaluation of the oscillations and diagnostics of grinding wheel wear.

Key words: grinding wheel wear, unbalance of grinding wheel and drive motor, vibration propagation and development, calculation equations.

Introduction

Common for most of grinding operations is that the grinding wheel wear is accompanied by vibro- propagation of a surface waviness on the part being ground and on the wheel and of vibration increase in the system which in time transforms to resonance vibrations. Such vibrations are named “self excited vibrations” or “chatter vibrations”. Surface waviness of the ground part is undesirable or inadmissible and for that reason it is necessary to redress grinding wheel before it looses its cutting ability. It takes additional time and abrasive waste, so the means are searched to improve grinding possibilities of the system. In purpose to achieve this it is necessary to examine and diagnose the process of provocation of self excited vibrations.

Grinding wheel wear and propagation of self excited vibrations was researched by many authors. Some of them have tested mainly the process of waviness propagation in grinding without deeper theoretical analysis of the phenomenon, others analyzed the phenomenon on the ground of vibration theory stating that self excited vibrations are propagated by non linearity of the system though mechanism of such a phenomenon was not fully explained.

Phenomenon of self - excited vibrations or “chatter vibrations” is common for most of metal cutting operations, though their origin is differently interpreted by many researchers. Without vast investigation of the problem it is possible to assert that there are two different explanations of the “self excited vibrations”. The first is that vibrations are excited in the system unstable from the point of view of vibrations. The systems are examined as ones with “negative damping” [1] in which damping coefficient *c* is negative in the homogeneous equation of an oscillatory motion

$$m\ddot{x} + c\dot{x} + kx = 0 \ . \qquad (1)$$

There *m* and *k* are the mass and stiffness coefficients, while $\ddot{x}, \dot{x}$ and x are the displacement, velocity and acceleration of vibrations accordingly. Other authors [2, 3, 4, etc.] reject it and explain the propagation of these vibrations due to regenerative effects at which, e.g. at turning, for some random effects the cutting force changes and cuts changed chip thickness at (i-1)-st revolution of the wokpiece leaving not constant allowance for the i-th revolution. It results further not constancy of cutting force and vibration propagation. Stability charts were derived showing system stability boundaries depending on system stiffness and process parameters. Practically all the cutting systems, so the grinding ones may be of this type: with decrease of system stiffness and increase of cutting rates the system becomes unstable and intensive vibrations generate in a very short period of time in which an amplitude of vibrations increase many times. Although with change of cutting rates or system stiffness the system transforms to stable one.

Different phenomenon is observed in grinding systems in which vibrations are propagated and maintained by grinding wheel wear. If the stiffness of the grinding system is large the “self excited” or chatter vibrations excite not instantly but after a long period of time, after corresponding wear of the grinding wheel. Though some of researchers consider them as principally unstable systems, it is not the case. These systems as any other have stability borders. When working with a newly dressed grinding wheel the machining is performed with forced vibrations which amplitude depend on excitation force value while the frequency is equal to that of the excitation force. And only after longer time of grinding the vibrations with some one natural frequency of the system are generated, and their amplitude is increasing in time with wheel wear. After redressing of the grinding wheel the work is fulfilled without such vibrations till grinding wheel again wears to some value. We assert that such a grinding system is not unstable or “with negative damping” and it not transforms to it by grinding wheel wear, but the state of the grinding wheel is so changed in time that it generates the excitation force with frequency close to some one natural frequency of the system, mostly sensitive in existing grinding conditions. Vibration amplitudes in stated conditions grow only with increase of grinding wheel wear. Also the fact is confirmed [5] that cubic boron nitride wheels may be manufactured using new technology which enable to work without wave generation and redressing through the life of the wheel. It confirms that the grinding systems are not "unstable" by their origin but their stability depend on the state of the grinding wheel. The increased amplitude of oscillations is common for all "stable" systems if excitation force approaches to some natural frequency of the system. Also the common for almost all grinding systems is that after the wheel dress the forced vibrations coinciding with some natural frequency of the system are absent. In other case it would be impossible to achieve good quality of the parts being ground. For that reason the interest arises to investigate the regularities of such wheel wear which creates the excitation force with frequency, close or equal to some one natural frequency of the system.

Some authors consider that the frequency of "self excited vibrations" is equal either to the natural frequency of the part being ground, or of the grinding wheel. Although in practice oscillations are generated with frequency of the most sensitive unit of the system. In many grinding systems the self frequency of the part or grinding wheel vibrations is comparably high and two phenomenon show itself at these frequencies: first of all, large damping energy is created at higher frequencies, damping coefficients increase for it; the second, at high frequencies the so named "wave cut" [6] on the part may show itself. It also increases damping in the system and the generative effect decrease.

Among big number of investigations we did not find the explanation how at grinding with a newly dressed grinding wheel the process for some long period of time is stable and only later it becomes "not stable" and how the oscillations are generated with the frequency of most sensitive unit of the system. Because the mechanism of such vibrations was not examined in full, our paper is assigned to it. Fine cylindrical plunge grinding is analyzed in the paper. Experimental and theoretical research of the problem were fulfilled.

Experimental and analytical research of forced vibrations in grinding and their influence on grinding wheel wear and stability of grinding process

A significant question concerning machine tool vibrations is the intervibrations of different mechanisms, excited by external sources and by machine tool work. The problem is what influence the machine tool design exerts on vibrations. The main influence on results of cylindrical grinding have vibrations in radial direction between the workpiece and the grinding wheel. Amplitude of these vibrations depends on value of acting excitation force, stiffness and damping properties of a machine tool design. The properties can be defined by experimental and theoretical research. Four stage experimental research was fulfilled. At the first stage elastic deflections in connection of separate masses to which the machine tool was divided for calculation of vibrations were researched at loading the machine tool by static force. Static stiffness of connections in a system was defined. At the second stage the oscillation amplitudes were examined by using a special aperture for loading of a system with harmonic excitation force of changed frequency and magnitude. Resonance frequencies of machine tool vibrations and amplitude amplification coefficients depending on excitation frequency were found. At the third stage frequency and oscillation amplitudes actuated at idling of a machine tool were measured, and at the fourth stage the cutting force, oscillation amplitudes, and their frequency were defined at grinding.

Experimental results of the first two stages of experiments were compared with theoretical calculations, in which two types of damping (or friction) forces were accepted: external (or proportional to vibration speed, so named "proportional damping") and internal, not proportional to speed (or hysteresis) damping. Calculation equation of flexible system motion expressed in matrix - vector form with complex numbers is:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\left(\mathbf{1} + \mathbf{i}\boldsymbol{\delta}/\pi\right)\mathbf{x} = \mathbf{F}\mathbf{e}^{i\cdot\omega\cdot t}\,. \qquad (2)$$

There **M, C**, and **K** are, respectively, mass, damping and stiffness matrices; **x** is complex amplitudes; δ is logarithmic decrement of vibrations, **F** is harmonic exciting force; ω is excitation frequency.

Knowledge of masses, stiffness, exciting force and displacement amplitudes enable to define by calculations damping properties of the system. Self-frequencies of dynamic system were searched solving homogenous equations of type (2), with the exception that the right side of that equation equal zero. Steady state amplitude - frequency characteristics were defined not only by experiments but also analytically by solving this equation by the method of complex amplitudes.

The third stage measurements showed that at idling there acted two main oscillations excitement factors: unbalance of a grinding wheel and of an electric motor rotor which drives the grinding spindle. Different revolving frequencies of these elements (32 r/s of the wheel and 50 r/s of the rotor) create total periodically changing oscillation amplitude. Knowledge of measured oscillation frequencies and amplitudes and experimentally found magnification factors of oscillation amplitude enables to define by equation (2) approximate value of harmonic centrifugal force (F_c) created by rotation of the grinding wheel and rotor of electric motor.

The fourth stage of experiments enable to define the grinding force, its change in time, and the moment of beginning of self exited vibrations in grinding.

One can see that idling oscillations between center pins and wheel head of the grinder created by centrifugal forces of unbalanced grinding wheel and electrical motor are in an open loop elastic system because the wheel does not contact with the workpiece being ground. At wheel dress from a table or tailstock of the grinder the dressing force is very small and it does not change the open loop and oscillation character. Centrifugal force (F_c) and resulting elastic displacement of wheel axis will not influent on roundness of a wheel because direction of this force will leave the same at every revolution of the wheel. Only the wheel axis will displace according to spindle axis to value straight proportional to wheel centrifugal force and contrary proportional to stiffness of spindle bearing. The accuracy of cross-section form of a wheel will depend only on revolving accuracy of the wheel spindle because the oscillation of electric motor's rotor centrifugal force does not coincide with revolving frequency of a wheel. At machining the centrifugal force (F_c) will not change and if the stiffness of the system would not change the displacement of a wheel axis resulted by centrifugal force of the wheel unbalance would not influent on grinding stability. Only centrifugal force of rotor revolutions would influent on it. But at machining the loop of the elastic system is closed because the wheel contacts with the workpiece. Stiffness of the closed loop system changes, so changes the oscillation amplitudes forced by centrifugal force, say, to value Δ_0. It creates additional excitation force in grinding. Value of the changed force can be defined in such an order. The radial component of the grinding force (response force F_r) depending on forced vibrations in radial direction with amplitude Δ_o may be found from equation

$$F_r = (\Delta_0 - \Delta_r \cos\varphi) k_w b_s v_p \,, \qquad (3)$$

where Δ_r is deflection of the system depending on the response force F_r, φ is phase angle between amplitudes Δ_0 and Δ_r; k_w is grinding force coefficient N/mm^{3}s^{-1} showing dependence of grinding force on volume of material (in mm^3) cut from the workpiece by 1 mm width of a grinding wheel [6], b_s is width of the surface being ground, v_p is revolving speed of the workpiece mm/s. Phase angle φ is defined by solving Eq.2.

Because

$$F_r = \Delta_r k_s \,, \qquad (4)$$

where k_s is stiffness of the system in radial direction between the grinding wheel and the shaft at oscillation frequency, by substituting F_r from Eq.4 to Eq.3 one can find that

$$\Delta_r = \frac{\Delta_o k_w b_s v_p}{k_s + k_w b_s v_p \cos\varphi} \,. \qquad (5)$$

By calculation of k_s, φ and amplitude Δ_0 for every forced frequency response amplitude Δ_r, the wave of allowance left on the shaft for every frequency is found. The wave left at the every second revolution of the shaft will force additional amplitude of vibrations Δ_a which on the ground of Eq.3-5 is equal

$$\Delta_a = \frac{(\Delta_r - \Delta_{r\,\min}) k_w b_s v_p}{k_s} \,, \qquad (6)$$

where $\Delta_{r\,min}$ is the minimal magnitude found by Eq.5 from different magnitudes of Δ_r.

Amplitudes Δ_a will lag to the phase angle φ behind the amplitudes of allowance variation. Getting the linear system all oscillations on the principle of superposition will sum algebraically and will result periodical oscillations. These oscillations will influent on non-constancy of cutting force and grinding wheel wear. The non-constancy of the wear may be found in such a way. The force F_r apart of Eq.4 may be expressed in a form $F_r = \Delta_c k_w b_s v_p$, where Δ_c is the allowance being cut from the workpiece for the response force. On the ground of it and from Eq.4, 5 it is seen that allowance Δ_c is equal

$$\Delta_c = \frac{\Delta_0 k_s}{k_s + k_w b_s v_p} \,. \qquad (7)$$

Allowance being cut from the workpiece by grinding will depend on Δ_c and Δ_a. Non-constancy of the wheel wear influenced by variation of allowance being cut may be evaluated by the grinding ratio k_g mm^3/mm^3 which shows what volume of material (in mm^3) is cut by 1 mm^3 of wasted grinding wheel material. So the wheel wear (wheel radius variation in cutting) at any of its revolution may be expressed by equation

$$\Delta_g(t) = (\Delta_c \cos(\omega t + \varphi_c) + (\Delta_a \cos(\omega t + \varphi_a))\frac{d_w}{d_g k_g} \,, \qquad (8)$$

here d_w and d_g are diameters of the workpiece and the grinding wheel accordingly.

Common wear in time will be a sum of wear amplitudes at every revolution of the wheel. The wear will act on work dynamics in the same manner as allowance amplitudes Δ_r and $\Delta_{r\ min}$ expressed by Eq.6. Common oscillations in grinding will depend on the three variations Δ_r, Δ_a and Δ_g.

If in the system would act only periodical variations Δ_r, Δ_a and Δ_g, common oscillation waves would sum in the sequence without impacts. But for the reason that at one revolution of the shaft the grinding wheel will does not all, but fractional number of revolutions, the sequence of allowance waves at every revolution of the shaft abruptly change to height Δ_w producing a force impulse in the system. The impulse time is not infinitely small but has finite limits depending on angle Δ_ϕ which the shaft must turn in purpose that the wheel would cut into it to depth Δ_w. Angle Δ_ϕ is calculated by equation

$$\Delta_\phi = \arccos\frac{(R_1 + \Delta_w)^2 + (R_1 + 2R_2)R_1}{2(R_1 + \Delta_w)(R_1 + R_2)} - \arccos\frac{R_1^2 + (R_1 + 2R_2)R_1}{2R_1(R_1 + R_2)} \,, \qquad (9)$$

here R_1 and R_2 are radius of the shaft and the grinding wheel accordingly.

The impact time T and speed change $\Delta\dot{x}$ of inertia elements for impacts can be calculated on the ground of Eq.2 by dependencies:

$$T = \Delta_\phi / \omega \,; \quad \ddot{x} = \Delta\dot{x} / \Delta_T \,; \qquad (10)$$

$$\Delta\dot{x} = [F(t) - C\dot{x} + K(1 + i\delta/\pi)x]\Delta_T / M^{-1} \,, \qquad (11)$$

here ω is shaft revolving frequency, Δ_T is small interval of time T, M is a matrix of inertia elements.

Impact forces influent on grinding stability and wheel wear. So by using the derived equations the knowledge of initial conditions enable to diagnose grinding stability and grinding wheel wear.

Conclusion

Forced vibrations in the grinding system are analyzed and equations are proposed for calculation how initial work conditions influent on propagation of waves on the workpiece and on the grinding wheel and how these waves change in time and influent on stability of the grinding process. It enables to diagnose the wear of the grinding wheel and its change in time.

References

1. **Shabana A.** Theory of vibration: an introduction. New York: Springer. 1996. P. 338.
2. **Deshpande N., Fofana M. S.** Nonlinear regenerative chatter in turning. Robotics and Computer Integrated Manufacturing. 2001. Vol. 17. Issues 1-2. P. 107-112.
3. **Trmal G., Kalisher H.** Some aspects of unbalance in the wheel-spindle assembly of cylindrical grinding machines. Annals of the CIRP. 1972. Vol. 21/1. P. 89-90.
4. **Biera J., Vinolas J., Nieto F. J.** Time-domain dynamic modeling of the external Plunge grinding process. International Journal of Machine Tools and Manufacture. 1997. Vol. 37-11. P. 1555-1572.
5. **Hogan B. J.** Precision makes grinding vital. Manufacturing Engineering, 2001. Vol. 126. No. 2. P. 1-3.
6. **Marcinkevičius A.-H.** Stabilization of grinding processes and machine tool work. Vilnius. Technika. 1995. 310 p.

A. H Marcinkevičius

Apvaliojo šlifavimo disko ir elektros variklio disbalanso įtakos disko dilimui diagnostikos skaičiavimai

Reziumė

Daugelis autorių tyrė šlifavimą, jo stabilumą ir savaiminį virpesių susižadinimą. Dalis reiškinį tyrė praktiškai - šlifuodami, kiti – teoriškai ir pasiūlė įvairiausių šlifavimo stabilumo paaiškinimų. Mūsų straipsnyje tiriamas palaipsnis savaiminis virpesių susižadinimas, susijęs su šlifavimo disko ir suklio variklio rotoriaus disbalansu. Šlifavimo diskas galandamas atvirojoje tampriojoje staklių sistemos grandinėje: čia disko paviršiaus radialinis mušimas staklių centrų atžvilgiu pašalinamas. Šlifuojama uždarojoje tampriojoje sistemoje, kurios standumas skiriasi nuo atvirosios grandinės sistemos, dėl to šlifuojant gaunamas disko paviršius radialinis mušimas apdirbamosios detalės atžvilgiu. Šis mušimas susideda su disko pavaros elektros variklio mušimu dėl jo disbalanso. Sužadinami sudėtiniai periodiniai mušimų dažnio virpesiai, dėl kurių detalės ir disko paviršius darosi banguotas. Visam tam susidėjus, sistemoje susidaro smūginės apkrovos, dėl kurių kyla savaime susižadinantys jautriausio sistemos nario savųjų dažnių virpesiai. Pasiūlytos virpesių įvertinimo ir šlifavimo disko dilimo diagnostikos skaičiavimų lygtys.

Pateikta spaudai 2006 04 04