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A new method for reliable rotary tool shank use at speeds above the critical speed range

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Abstract

Dynamic tool characteristics affect both the design and the use of rotary tools applied to high-speed cutting. At present, spindle-driven tools commonly operate at speed values below their critical bending speed. Currently, this is the only way to guarantee reliable tool operation. Consequently, the low natural frequency – in particular for tools with a high length-to-diameter ratio – often results in spindle speeds that utilise neither the cutting materials' nor the machine tools' potential. The paper elucidates a method to design tool shanks and tools to be run above the critical speed, as well as its implementation. The reliable and safe operation of these special tool shanks is based on the physical effects of self-centering and self-balancing, which come into play above the critical speed.

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Introduction

The trend towards high-speed milling (HSC for high-speed cutting) is never ending. If we were to succeed in combining the speed values the speed values typical of HSC run with analogously high feed rates, then we could derive substantial benefits from increased efficiency in comparison with conventional milling. However, as a result of increasing speed, even at speeds below the critical value, the unavoidable effect of imbalance becomes the criterion limiting speed. The phenomenon of resonance has to be avoided; this is, in general, achieved by setting the tool's maximal speed at about 60% of the natural frequency's value. Consequently, it is frequently impossible to achieve the speed and feed rate values commonly used for HSC in many cases due to tool design. A high length-to-diameter ratio (L/d ratio) is desirable in relevant ranges of rotary tool application, which offers another challenge. The natural frequency of rotary tools decreases as a function of an increasing length-to-diameter ratio as physical characteristic. This means that the higher the length-to-diameter ratio, the more difficult high-speed milling becomes and the more probable it is that HS milling will no longer be feasible, insofar as we would not succeed raising the speed value above the tool's natural frequency. It is known that rotors running at speeds above the first critical bending natural frequency tend to self-centering [9].

Theoretical considerations and solution

In the operation of milling cutters, large flexural vibrations occur mainly in proximity of the first critical speed and, as a result, the milling cutter usually breaks in the case of resonance. The reason is the unavoidable, albeit very low, asymmetric mass distribution related to the rotary axis. Consequently, the rotary axis "O" and centre of gravity "S" do not coincide, which results in the eccentricity of the centre of gravity. Furthermore, a distance "a" between the

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geometrical tool centre and the rotary axis is unavoidable. This phenomenon does not only result from manufacturing errors of the tool but, for instance, also from angular and radial displacement in the tool clamping. Apart from high tool precision, low tool masses are also striven for in order to keep the natural frequency as high as possible. A reduction in mass can be obtained by designing tools or tool shanks as hollow shaped parts. This approach has previously become common practice in many applications [2, 4, 6]. The hollow shank design only slightly decreases both the moment of inertia and the stiffness in comparison with commensurable full-volume shanks. An increase in revolutions per minutes makes it possible to elevate the cutting speed to an optimal level for any material. This is particularly applicable to small tool diameters used for aluminum or plastic machining. A significant potential for an increase in productivity can arise from elevated feed rate.

Model of the tool shank dynamics

To pass through the resonance range, it is very important to know the trajectory of the tool's centre of gravity. To calculate the trajectory of the tool's centre of gravity, the formulas (1) and (2) were derived [5] based on the equations of the vibratory motions in x- and y- directions:

$$r_w(t) = a\left(1+\frac{\eta^2}{1-\eta^2}\right) + \varepsilon\frac{\eta^2}{1-\eta^2} = a + (a+\varepsilon)\frac{\eta^2}{1-\eta^2} \quad (1)$$

$$r_s(t) = a\frac{1}{1-\eta^2} + \varepsilon\left(1+\frac{\eta^2}{1-\eta^2}\right) = (a+\varepsilon)\frac{1}{1-\eta^2} \quad (2)$$

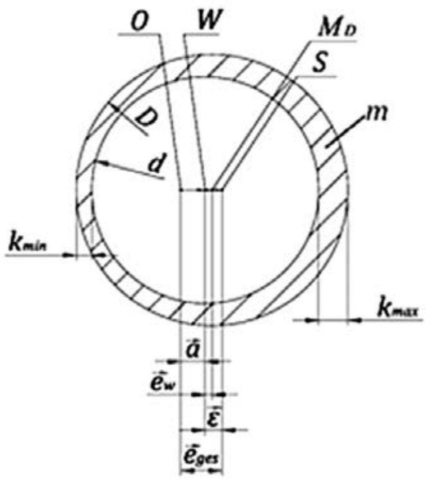
$$r_s(t) = a\frac{1}{1-\eta^2} + \varepsilon\frac{\eta^2}{1-\eta^2} + \varepsilon = a\frac{1}{1-\eta^2} + \varepsilon\left(1+\frac{\eta^2}{1-\eta^2}\right) \quad (3)$$

with $\eta=\Omega/\omega$, ε – distance of imbalance, a – distance between geometric tool centre axis and rotary axis *O*, $r_w(t)$ – distance between shank centre axis *W* and *O*, $r_s(t)$ – distance between centre axis of gravity *S* and *O*, Ω - natural frequency, ω - rotary frequency

In this derivation, the influence of external damping was neglected, since it is very low and thus only of minor importance [8]. The term $(a+\varepsilon)$ in the equations (1) and (2) represents the eccentricity of the dynamical system resulting from form errors and dynamic displacement (e_{ges}). It is very relevant in practice, when balancing the hollow shanks. In other words: if one intends to reduce the vibration amplitude during operation, then the imbalance resulting from the total eccentricity (e_{ges}) should be compensated. Altogether it was discovered that extra long hollow shanks clamped on one side show similar dynamical characteristics to those of super-critical, elastic rotors rigidly mounted on two sides, such as the Laval rotor [5, 8]. Thus a self-stabilisation or self-centering effect of the extra long tool shanks introduced is maintained at speeds above the critical one. At values above the critical speed, the rotors' characteristics can be described as "elastic" [8]. Studies of elastic rotors outline a distinct quiet running of the flexurally elastic rotor when run at values above the critical speed [1, 7]. It has been shown that the eccentric tool centre of gravity – which was caused by inaccuracy in machining and tool clamping - drifts to the geometrical centre line, as a function of increasing values above the critical speed. This phenomenon is called the "self-centering effect" [3].

Self-centering

To achieve an efficient self-centering effect for shanks run at values above the critical speed, one has to consider the correlation among the axis of rotation *O*, the shank centre axis *W and* the centre axis of gravity *S*, or the alteration of their mutual relative position as a function of the speed (Figure 1). When running the tool at values below the critical speed ($\eta<1$), the deflection of the centre axis of gravity *S* is greater than the dislocation of the geometrical shank centre axis *W*, since the centre of gravity is pushed outward due to the application of centrifugal forces. If, however, the range of the operating speed is near the natural frequency, then *W and S* are orthogonal to one another related to the axis of rotation *O*. That is, there exists a phase frequency angel $\alpha=90°$. In this case, the deflection of the shank becomes infinite, if there is no damping. However, a strong damping effect can significantly limit the deflection or motion amplitude. When running the tool at speed values above the critical speed ($\eta>1$), then, at increasing speed, the tool's axis of gravity *S* drifts towards the axis of rotation *O and*, finally, upon reaching the phase frequency angle $\alpha=180°$, the geometric shank centre axis *W* lies outside the centre axis of gravity *S*, whereby the distance is equal to the eccentricity. If the rotational speed is beyond super-critical level ($\eta\gg1$), the centre of gravity approaches the state of static equilibrium, whereby the displacement of the shank centre axis W from the axis of rotation *O* is equal to



m – mass of shank, M_D – centre axis of D, *k – wall thickness*

Figure 1. Position of the shank's centre of gravity S related to the geometrical shank centre axis *O* as a function of the speed range, according to [5]

the total eccentricity (e_{ges}) [9]. The shank now runs much more quietly again than in the proximity of the critical speed. This phenomenon is called “self-centering”.

Self-balancing

To guarantee that the tool passes through the critical speed range safely, the tools must be balanced in advance. Since, this way, not all of inaccuracies affecting imbalance can be compensated, ”internal“ balancing is required in order to bring together the central main axis of inertia and the rotary axis by means of mass redistribution [1, 7]. According to this method, during the operation of the tool, the special compensating weights are re-distributed as a result of the action of centrifugal forces resulting from imbalance. In the case considered, the weight remains in its initial state and is thus able to respond to quickly appearing vibrations and to re-distribute itself. Thanks to this phenomenon, a high damping effect can be achieved. Both solid and liquid materials are used as compensating substances [1, 10]. Various approaches were developed to emplace the compensating weights on the rotary body. Shanks with hollow inner cavities here provide a good opportunity to apply an internally active energy absorber. The chamber size is determined by the volume of the compensating substance, which is obtained from the equation (4) below [11]:

$$m_F = \frac{U\left(1+\sqrt{1-\frac{h_0}{h_K}}\right)}{R_i}\, s \qquad (4)$$

with U – imbalance; h_0 – height of the emplaced compensating substance in system rest position; h_k – balancing chamber height; R_i – inner radius of the balancing chamber; s – factor for safety

As derived from the author‘s own experiments [5], an efficient damping inside the tool shank can be achieved by means of a compensating weight in the form of a dry iron powder with approximately 100 µm grain size.

Experimental and measurement setup

The execution of experiments was aimed at verifying the self-balancing and self-centering effect. For this purpose, an HSC 120 milling machine manufactured by the firm Klink was employed in conjunction with a precision HF spindle (40000 rpm). The parameters of the test tools are summarized in Table 1.

Table 1. Test shank parameters (aspect ratio 27)

Name	Material	D [mm]	d [mm]	Mass [g]	f_e [Hz]
Al 10x1	EN AW-7075	10	8	29	118, 8
Steel 10x2	E355	10	6	110	108,9
HM 10x2	DK 460 UF	10	6	200	116,7

D – outer diameter, d – inner diameter, f_e – first natural frequency value

A safety bearing was employed to carry out comparative investigations with/ without balancing weights in order to prevent possible shank destruction when passing through the natural frequency. The vibration amplitudes of the shank were recorded by means of a laser displacement sensor optoNCDT 2300-50 by the firm Micro-Epsilon Messtechnik GmbH at a 20 kHz scanning frequency. The test stand is shown in Figure 2. The shank end chucked in the spindle collet is located in the safety bearing range. The radial clearance between the shank and the safety bearing was about 1 mm. The shanks were rotated at speeds of 9000 min^{-1} and 18000 min^{-1}, that is at values above the critical speed.

Self-centering effect

Figure 3 shows the typical curve of the measured vibration amplitudes as a function of time. Having passed through the natural frequency (resonance point), the self-centering effect is established at values above the critical ones. The shank shown in Figure 2 with a radial runout of 24 µm had

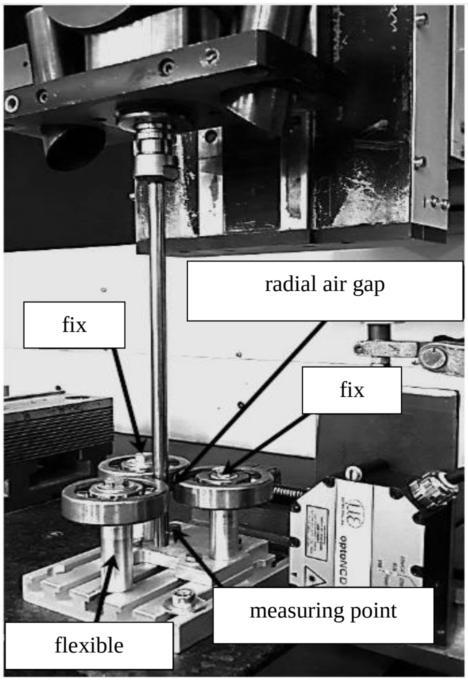


Figure 2. Experimental stand

a twofold vibration amplitude (peak-to-peak value) of about 285 µm at a speed of 9000 min^{-1} and about 182 µm at a speed of 18000 min^{-1}. This reveals that the vibration amplitudes decrease at increasing speed above the critical frequency values.

Self-balancing effect

The shanks were filled with iron powder to investigate how the compensating weight affects the vibration characteristics – both when passing through and exceeding the critical natural frequency value. The test results are summarised in Figure 3. As indicated in Figure 3a, the shank without the compensating weight comes into contact with the safety bearing when passing the critical value. In this range, the vibration amplitude is determined by the clearance between the shank and the safety bearing and amounts to about 1 mm. The other shank filled with the compensating weight passes through the natural frequency range without coming into contact with the bearing, which results from the damping effect. In this case, the twofold vibration amplitude was approximately 0.8 mm and is less than the clearance between the shank and the safety bearing (Figure 3 b)).

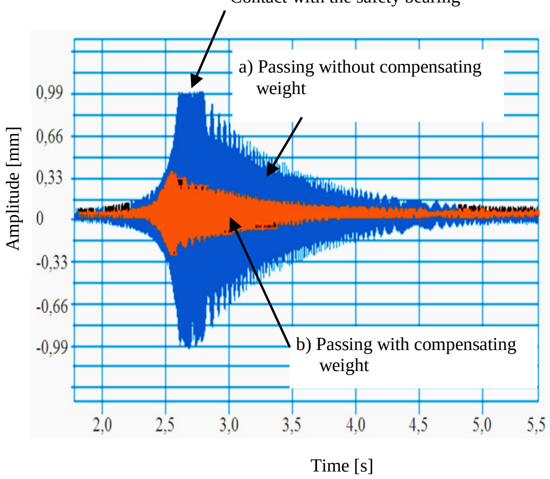


Figure 3. Experiments carried out without (a) and with (b) compensating weight by using the shank "Steel 10x1", (balancing weight: 6g iron powder; radial runout: 9μm)

The feasibility of the proposed tool design in practice was evidenced by means of milling of test parts made of plastic material. At first, the parts were rough-milled using a torus mill equipped with 6 cutting edges, and then finished by means of a ball-headed mill fitted with 6 cutting edges. The milling cutters ran quietly during the tests (Figure 4). The surface of the milled workpiece was smooth.

Figure 4. Tool holder and shank (L/D = 27, D = 10 mm, d = 8 mm) (not shown the cutter head at the right side)

Summary and Outlook

The tool concept described in this paper and the technology used to employ the tool at speed values above the critical frequency indicate that it is possible to implement both self-centering and self-balancing in cutting tools by means of the "elastic" rotor principle. The design, engineered by taking into account the influencing parameters of radial runout, compensating weight and material, has made it possible for the first time to gain knowledge of a tool's characteristics at frequencies above the critical speed range. The advantages of the demonstrated tool arise from its weight and dynamical stability:

- An L/D ratio > 20 is feasible;
- The critical bending speed has been overcome,
- A clearly greater speed range is available;
- The tool may have a lightweight design.
- The potential of machine tool and cutting material for high-speed machining is more fully utilised.

Ongoing research work is focused on the investigation and enhancement of statically and dynamically tool characteristics aimed at an increase in process reliability. Since the disturbances appearing at tool frequencies above the critical speed values can be dangerous for the process itself, the process reliability of these tools is very important. Another direction in development is aimed at implementing the concept described here for the design of long HP cutters with a diameter of approximately 40 to 60 mm to overcome the limiting speeds of tools that are currently used in the market.

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