Available online at www.sciencedirect.com
ScienceDirect
Journal of Sound and Vibration 308 (2007) 805–814
JOURNAL OF SOUND AND VIBRATION
www.elsevier.com/locate/jsvi

Vibration excitation and energy transfer during ultrasonically assisted drilling

V.I. Babitsky[a,*], V.K. Astashev[b], A. Meadows[a]

[a] Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, UK
[b] Institute of Machine Studies, Russian Academy of Sciences, Moscow, Russia

Accepted 21 March 2007
The peer review of this article was organised by the Guest Editor
Available online 30 May 2007

Abstract

Successful application of ultrasonically assisted drilling needs dynamic matching of the transducer with the drill bit considered as a continuous system loaded by the nonlinear processing load. When using standard tools this leads to the compatible choice of the transducer and accurate matching of the transducer and tool. The principal dynamical features of this matching are considered. Optimal position of excitation cross section of the drill bit, which depends on the relationship between elasto-dissipative characteristics of the transducer, the drill bit and the work load, is found in general analytical form. The optimal matching preserves the resonant tuning of the transducer and compensates the additional energy losses in the drill bit and processing. This produces also an amplification of vibration amplitude. The effect is achieved through the generation and maintenance of a *nonlinear resonant mode* of vibration and by active matching of the oscillating system with the dynamic loads imposed by the cutting process with the help of the intelligent electronic feedback circuitry.

A prototype of an ultrasonic drilling system has been designed, manufactured. and tested. Improvements of machining characteristics due to superposition of ultrasonic vibration are demonstrated. Substantial improvements in the cutting performance of drill bits lead to benefits in drilling performance, which include faster penetration rates, reduction of tool wear, improvements in the surface finish, roundness and straightness of holes and, in ductile materials, the reduction or even complete elimination of burrs on both the entrance and exit faces of plates. The reduction in the reactive force experienced also causes greatly reduced deformation when drilling through thin, flexible plates and helps to alleviate delamination hazard.

1. Formulation and analysis of dynamical problem

Ultrasonically assisted drilling takes place when ultrasonic vibration is superimposed onto the relative cutting motion between a drill bit and the workpiece being drilled. Usually this is achieved by excitation of the drill vibration either torsional [1] or axial [2,3]. The latter took place in majority of applications and will be mainly analysed below. A reduction in the cutting forces, an increase in the penetration speed, and elimination of burrs are among the main benefits of drilling with ultrasonic assistance.

*Corresponding author. Tel.: +44 0 1509 227503; fax: +44 0 1509 227502.
E-mail address: V.I.Babitsky@lboro.ac.uk (V.I. Babitsky).

0022-460X/$ - see front matter
doi:10.1016/j.jsv.2007.03.064

Effective vibration of the drill bit is achieved when it is used as a wave guide for amplification of vibration amplitude. Only in drilling small holes with diameters below ~3 mm a drill can be treated as a small lamp mass attached to the transducer. With an increase in diameters of holes a successful application of ultrasonically assisted drilling needs acoustical matching of the drill bit as *a continuous system* with a transducer. Effectively, the drill bit has to be treated as an additional step of the wave guide. When using the standard tools this leads to the compatible choice of the transducer and accurate coupling of transducer and tool.

Let us analyse the principal dynamical features of this coupling. We propose that a drill bit can be considered as a rod-type wave guide (see Fig. 1) of longitudinal vibration with a length l and its cross section with coordinate $x = s$ is connected rigidly with the transducer. The end section of the drill bit with a coordinate $x = l$ is interacted with a material being drilled.

Resonant tuning of the ultrasonic vibratory system impose a strong filtering effect on the regime of vibration. Due to it, a longitudinal vibration of arbitrary cross section with a coordinate x of the drill can be described with a high accuracy by a harmonic function

$$u_x(t) = \tilde{a}_x \exp \mathrm{j}\omega t = a_x \exp \mathrm{j}(\omega t - \varphi_x), \tag{1}$$

where a_x, ω and φ_x are an amplitude, an angular frequency and an initial phase of vibration, $\mathrm{j} = \sqrt{-1}$.

An axial load of the drill bit from the drilling process we describe with a help of a nonlinear dynamic characteristic $f(u_l, \dot{u}_l)$ of tool–workpiece interaction, which is generally ill-defined function of cutting end vibration $u_l(t)$. Under stationary conditions of drilling the load can be transformed with a help of harmonic linearisation as follows [4,5]:

$$f_l = f(u_l, \dot{u}_l) \approx [k(a_l) + \mathrm{j}\omega b(a_l)]u_l + P(a_l), \tag{2}$$

where $P(a_1)$ is the permanent component of the technological load under ultrasonic vibration (*vibro-induced force*), $k(a_1)$, $b(a_1)$ are the coefficients characterising the equivalent elastic and dissipative components of the load. Due to nonlinearity of the tool–workpiece contact interaction the coefficients of linearisation depend on parameters of vibration and are calculated according to the formulas [6] based on Fourier series:

$$\begin{aligned} P(a_l) &= \frac{1}{T}\int_0^T f[u_l(t), \dot{u}_l(t)]\mathrm{d}t, \\ k(a_l) &= \frac{2}{Ta_l}\int_0^T f[u_l(t), \dot{u}_l(t)]\cos\omega t\,\mathrm{d}t, \\ b(a_l) &= -\frac{2}{Ta_l\omega}\int_0^T f[u_l(t), \dot{u}_l(t)]\sin\omega t\,\mathrm{d}t, \end{aligned} \tag{3}$$

where $T = 2\pi/\omega$ is the period of the tool vibration. Dependence of the linearisation coefficients on the parameters of cutting end vibration a_l and ω reflects the nonlinear influence of a machining process on the cutting system.

The vibro-induced force $P(a_1)$ acts as an average permanent static load on the feed drive and it is equalised by the drive. That is why in the following analysis of vibration excitation we will use the dynamic component

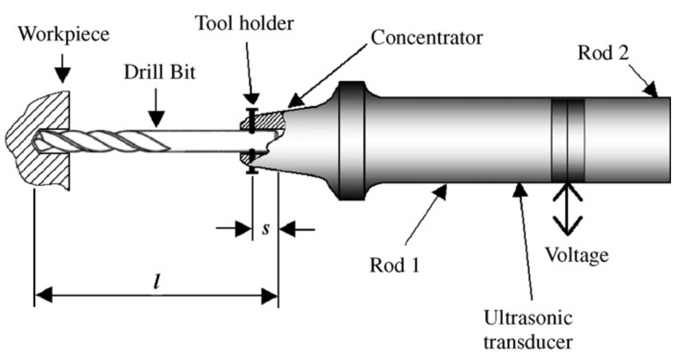

Fig. 1. Dynamical model of transducer–drill bit coupling.

of axial force only. After solution of the dynamic problem and calculation of amplitude a_l, the vibro-induced force can be calculated from the first expression of Eq. (3).

An interaction of the drill bit and transducer can be described by an axial force acting in the cross section $x = s$ as follows:

$$f_s(t) = \tilde{F}_s \exp \mathrm{j}\omega t = F_s \exp \mathrm{j}\omega t.$$

We describe vibration of the drill bit as a wave guide with a help of dynamic compliances (receptances) $L_{kx}(\mathrm{j}\omega)$ $(k = l, s)$ coupling a displacement $u_x(t)$ of a cross section x with the forces acting in cross sections l, s:

$$u_x(t) = L_{sx}(\mathrm{j}\omega)\tilde{F}_s \exp \mathrm{j}\omega t - L_{lx}(\mathrm{j}\omega)[k(a_l) + b(a_l)]u_l(t). \tag{4}$$

According to Eq. (1), we have

$$\exp \mathrm{j}\omega t = \frac{u_l}{a_l} \exp \mathrm{j}\varphi_l, \quad u_x = u_l \frac{a_x}{a_l} \exp \mathrm{j}(\varphi_l - \varphi_x). \tag{5}$$

Substituting Eq. (5) in Eq. (4) we have for $x = s$ and $x = l$ correspondingly:

$$a_s = \frac{F_s}{W_{ss}(\mathrm{j}\omega)} \exp \mathrm{j}\varphi_s - \frac{a_l[k(a_l) + \mathrm{j}\omega b(a_l)]}{W_{ls}(\mathrm{j}\omega)} \exp \mathrm{j}(\varphi_s - \varphi_l), \tag{6}$$

$$a_l W(a_l, \mathrm{j}\omega) = F_s \exp \mathrm{j}\varphi_l. \tag{7}$$

Here

$$W(a_l, \mathrm{j}\omega) = [W_{ll}(\mathrm{j}\omega) + k(a_l) + \mathrm{j}\omega b(a_l)] \frac{W_{sl}(\mathrm{j}\omega)}{W_{ll}(\mathrm{j}\omega)},$$

$W_{kx}(\mathrm{j}\omega) = L_{kx}^{-1}(\mathrm{j}\omega)$ is a dynamic stiffness of the drill bit coupling a force $f_k(t)$ of a cross section k with the displacement $u_x(t)$acting in cross section x.

Separating in Eq. (7) a real and an imaginary component we find the expressions for amplitude a_l and phase φ_l for the cutting end of the drill bit ($x = l$):

$$a_l = \frac{F_s}{|W(a_l, \mathrm{j}\omega)|} = \frac{F_s}{\sqrt{[\mathrm{Re}\, W(a_l, \mathrm{j}\omega)]^2 + [\mathrm{Im}\, W(a_l, \mathrm{j}\omega)]^2}}, \tag{8}$$

$$\cos \varphi_l = \frac{a_l}{F_s} \mathrm{Re}\ W(a_l, \mathrm{j}\omega), \quad \sin \varphi_l = \frac{a_l}{F_s} \mathrm{Im}\ W(a_l, \mathrm{j}\omega). \tag{9}$$

Let us present the dynamic stiffness $W_{sx}(\mathrm{j}\omega)$in the following form:

$$W_{sx}(\mathrm{j}\omega) = U_{sx}(\omega) + \mathrm{j}V_{sx}(\omega). \tag{10}$$

In the following consideration the parameters $V_{sx}(\omega)$ and $b(a_l)$ which characterise dissipation in the drill bit structure and the drilling dissipative load are treated as the small ones and all transformations are limited by variables of the first order of magnitude.

Eq. (8) defines amplitude–frequency characteristic of the drill bit excited by harmonic force (3) applied at the cross section $x = s$. Maximum amplitude is achieved under a condition $\mathrm{Re}\, W(a_l, \mathrm{j}\omega) = 0$, which with account of Eq. (10) can be presented as follows:

$$U_{ll}(\omega) + k(a_l) = 0. \tag{11}$$

Under condition (11) any amplitude a_l is achieved with a minimal force F_s. Hence, Eq. (11) is a condition of the best matching of the transducer and the loaded drill bit.

Let us rewrite an expanded expression (7) for amplitude a_l:

$$a_l = \frac{W_{ll}}{W_{sl}(W_{ll} + k + \mathrm{j}\omega b)} F_s e^{\mathrm{j}\varphi_s}.$$

With account of it the expression (6) will be as follows:

$$a_s = \frac{W_{sl}^2(W_{ll} + k + j\omega b) - W_{ll}(k + j\omega b)W_{ss}}{W_{ss}W_{sl}^2(W_{ll} + k + j\omega b)} F_s \exp j\varphi_s$$

and as a result

$$\frac{a_l}{a_s} = \left| \frac{W_{ll}W_{ss}W_{sl}}{W_{sl}^2(W_{ll} + k + j\omega b) - W_{ll}(k + j\omega b)W_{ss}} \right|. \tag{12}$$

Taking Eq. (10) into consideration and treating V and b as the small values we will find approximately $|W_{sx}(j\omega)| \approx |U_{sx}(\omega)|$. Then neglecting by small values and taking Eq. (11) into account we have

$$K_s = \frac{a_l}{a_s} = \left| \frac{U_{sl}(\omega)}{U_{ll}(\omega)} \right|. \tag{13}$$

We will call it as *an amplification ratio* from an exciter to a drilling section of the bit.

The coefficient K_s is not influenced by the load and for each magnitude of a_s it defines an axial amplitude of the cutting edge of the drill bit under the best matching (11) with the transducer.

2. Loading of a transducer with a drill bit

Connection of the drill bit with the transducer leads to additional loading of the latter. Suppose that unloaded transducer vibrates in its connecting cross section according to equation $u^*(t) = a^* \exp j(\omega t - \varphi)$. A connection with the loaded drill bit leads to arrival of an additional force $-f_s(t)$ applied to the transducer. As a result its vibration $u(t)$ will be modified as follows:

$$u(t) = u^*(t) - \frac{f_s(t)}{W_n(j\omega)}, \tag{14}$$

where $W_n(j\omega) = U_n(\omega) + jV_n(\omega)$ is a dynamic stiffness of the transducer in the cross section of connection with the drill bit.

With account of Eqs. (7), (10), (13) we can rewrite the expression for $f_s(t)$ as follows:

$$f_s(t) = \dot{u}_s(t)K_s^2[V_{ll}(\omega) + \omega b(a_l)]. \tag{15}$$

Ultrasonic transducer is a resonant electro-acoustical device. Its resonant tuning corresponds to condition $\mathrm{Re}\, W_n(j\omega) = U_n(\omega) = 0$. Taking into consideration that in the connecting cross section of the transducer and the drill bit $u(t) = u_s(t)$, we have from Eqs. (14), (15) for the loaded transducer:

$$u(t) = u^*(t)\left[1 + \frac{V_{ll}(\omega) + \omega b(a_l)}{V_n(\omega)} K_s^2\right]^{-1}. \tag{16}$$

It follows from Eqs. (16) that connecting of the matched drill bit with the transducer does not change the resonant tuning of the latter. Its vibration amplitude is defined by relationship of damping characteristics of the transducer, the drill bit and the load as well as an axial coordinate of the excitation.

From Eq. (13) with account of Eq. (16) we find the amplitude of vibration for the cutting end of the loaded bit:

$$a_l = a^*\left[\frac{1}{K_s} + \frac{V_{ll}(\omega) + \omega b(a_l)}{V_n(\omega)} K_s\right]^{-1}. \tag{17}$$

As it follows from Eq. (17) an amplification of amplitude of drill bit vibration can be achieved only when dissipation in transducer will be comparable or exceed the consumption of energy in the drilling process.

Let us suppose that the drill bit can be modelled as a rod with unified cross section area S. The dynamic stiffness $W(j\omega)$ and proper operators of dynamic compliances (receptances), which are needed, can be found from solution of the boundary problem for steady-state waves in one-dimensional wave guides [7]. The equation of longitudinal vibration of the wave-guide is

$$\rho \frac{\partial^2 u(x,t)}{\partial t^2} = \tilde{E}S \frac{\partial^2 u(x,t)}{\partial x^2}, \tag{18}$$

where $u(x, t)$ is a motion of the cross section x, $\tilde{E} = E(1 + j\psi/2\pi)$, E is the elastic modulus, ψ is a coefficient of inner damping in a material, ρ is the density of the material.

Let $x = s$ be a coordinate of the application of the exciting force $f_s(t) = \tilde{F}_s e^{j\omega t}$. For a wave guide of length l with free ends the boundary conditions are

$$\left.\frac{\partial u(x,t)}{\partial x}\right|_{x=l} = 0, \quad \tilde{E}S_x \left.\frac{\partial u(x,t)}{\partial x}\right|_{x=s} = \tilde{F}_s e^{j\omega t}. \tag{19}$$

Considering a harmonic vibration of the wave guide $u(x,t) = \tilde{a}_x e^{j\omega t} = a_x e^{j(\omega t - \varphi_x)}$, where a_x, φ_x are the amplitude and phase of the vibration of a cross section x, we have instead of Eqs. (18) and (19)

$$\begin{aligned} \tilde{E}\tilde{a}_x^{''} + \rho\omega^2 \tilde{a}_x = 0, \\ \tilde{a}_x^{'}\big|_{x=l} = 0, \quad \tilde{E}S\tilde{a}_x^{'}\big|_{x=s} = \tilde{F}_s. \end{aligned} \tag{20}$$

Taking into account that $L_{sx}(j\omega) = \tilde{a}_x/\tilde{F}_s$ we have from Eq. (20)

$$L_{sx}(j\omega) = W^{-1}(j\omega) = \frac{\tilde{a}_x}{\tilde{F}_s} = (-1)^{s/l} \frac{\lambda}{\omega^2 \rho S} \frac{e^{\lambda x} + e^{-\lambda x} e^{2\lambda(l-s)}}{e^{\lambda s} - e^{-\lambda s} e^{2\lambda(l-s)}}, \tag{21}$$

where $\lambda = j\omega\sqrt{\rho/\tilde{E}} = j\omega/c\sqrt{1 + j\psi/2\pi}$, $\quad c = \sqrt{E/\rho}$ is sound speed in the rod material.

In the following we assume that dissipation ratio is a small value ($\psi \ll 1$), and we preserve the linear terms only in Taylor expansion of the terms containing ψ. Based on it we will get the following receptances for the further application

$$L_{sx}(j\omega) = \begin{cases} -\dfrac{\lambda}{\rho \cdot S\omega^2} \cdot \dfrac{\text{ch}\,\lambda x\,\text{ch}\,\lambda(l-s)}{\text{sh}\lambda l}, & 0 \le x \le s, \\ -\dfrac{\lambda}{\rho \cdot S\omega^2} \cdot \dfrac{\text{ch}\,\lambda(l-x)\text{ch}\,\lambda s}{\text{sh}\,\lambda l}, & s \le x \le l, \end{cases} \tag{22}$$

where $\lambda \approx (j + \psi/4\pi)\frac{\omega}{c}$. From (22) we find dynamic stiffnesses for Eqs. (11), (13):

$$U_{ll} = -\frac{ES\zeta}{l} \frac{\sin\zeta}{\cos\zeta}, \tag{23}$$

$$U_{sl} = -\frac{ES\zeta}{l} \frac{\sin\zeta}{\cos\zeta\frac{s}{l}}, \tag{24}$$

where $\zeta = \omega l/c$.

While running without load (idle regime) we have from Eqs. (11), (23) a condition of ideal matching:

$$\zeta = n\pi \quad (n = 1, 2, \ldots). \tag{25}$$

It defines the natural frequencies of a drill bit. As it is physically clear, the condition does not depend on position of the contact cross section of the drill bit with the transducer. For the amplification ratio (13) with account of Eqs. (24)–(25) we have for the case:

$$K_s = \left|\cos^{-1}\pi n\frac{s}{l}.\right| \tag{26}$$

It follows from Eq. (26) that the amplification ratio changes its value from $K_s = 1$, when the excitation is applied in an antinodal cross section of a standing wave ($s/l = m/n, \quad m = 0, 1, \ldots, n$), to $K_s = \infty$, when approaching a nodal cross section ($s/l = (2q-1)/n, \quad q = 1, 2, \ldots$).

Let us analyse a proper vibration of the cutting cross section (17). In absence of dissipation and load the force of interaction $f_s(t) = 0$ and connection of the matched drill bit with the transducer does not influence the latter. Dissipation of energy in the drill and due to the drilling process has to compensated by the transducer.

This changes its vibration amplitude a^* and amplitude of the cutting edge a_l. For the further estimation this relationship, we find from Eq. (22) a dissipative component of the dynamic stiffness:

$$V_{ll} = \frac{ES}{l}\frac{\psi\zeta}{4\pi} \quad \frac{\zeta - 0.5\ \sin 2\zeta}{\cos^2\zeta}. \tag{27}$$

Substituting Eqs. (25)–(27) into Eq. (17), we have when $b = 0$:

$$a_l = a^*\left[\left|\cos \pi n\frac{s}{l}\right| + \frac{ES}{4lV_n(\omega)}\frac{\pi\psi\zeta n^2}{\left|\cos \pi n\frac{s}{l}\right|}\right]^{-1}. \tag{28}$$

It follows from Eq. (28) that when application of excitation approaches the nodal points $a_l \to 0$ due to increase of a transducer dissipative loading. This means that there exists some optimal position of excitation cross section which depends on relationship between elasto-dissipative characteristics of the transducer and the drill bit. The optimal coupling preserves the tuning of the transducer and compensates the additional energy losses in drill bit and from processing load. Additionally it produces an amplification of vibration amplitude.

3. Experimental drilling

A prototype of an ultrasonic drilling system has been designed and manufactured. The system employs an autoresonant control system described in Ref. [9] to stabilise the drilling process.

Experiments with ultrasonically assisted drilling were conducted on a lathe as it shown in Fig. 2. Transducer was held in the three jaw chuck of the lathe through the intermediate bush and was energised by means of the slip ring assembly fitted to the hollow shaft of the lathe at the end remote from the chuck. Once firmly clamped, the working end of concentrator was drilled in situ so that the hole, used for securing drills to the transducer, coincided perfectly with the rotational axis of the lathe. Samples to be drilled were clumped in a vice with a vertical traverse that was rigidly attached to the saddle of the lathe or held in a simple holder attached to the load cell, which could also be positioned on the saddle as an alternative. The usual tool holder assembly with its traverse having been previously removed.

This arrangement allowed rows of holes to be drilled in quick succession. The saddle of the lathe can be power driven in a line parallel to the lathe axis and the tool post mounting stage can be similarly power driven in a lane at right angles to this for facing.

A wide variety of materials was drilled. It was found universally that the application of ultrasonic vibration significantly increased the penetration rate of the drills. In some cases, the penetration rate was increased by a factor of four depending on the rotational speed. Fig. 3 shows a typical example of penetration speed when ultrasonic vibration is switched on and off (denotes as “US on” and “US off”, respectively) while drilling in

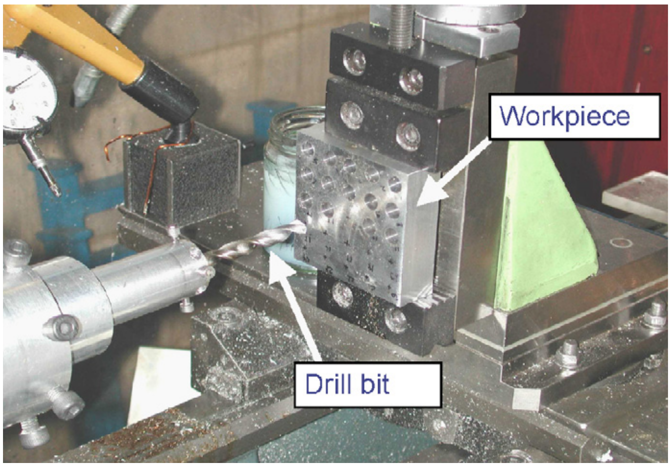

Fig. 2. Experimental set-up of ultrasonically assisted drilling.

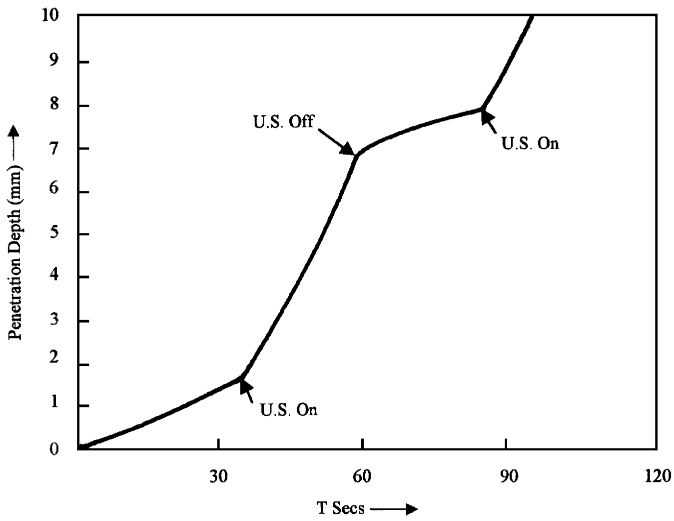

Fig. 3. Speed of penetration with and without of ultrasonic vibration.

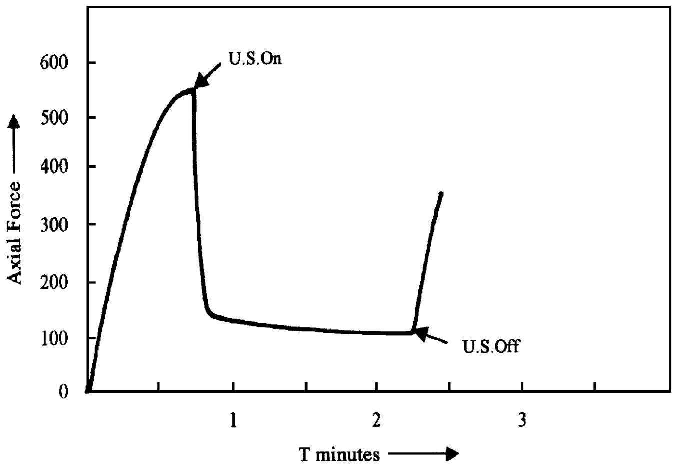

Fig. 4. Force reduction due to superposition of ultrasonic vibration.

titanium with constant force. The drilling is conducted with a standard twist drill having diameter $\varnothing$ 3 mm and rotation speed $n = 125$ rev/min. The axial force $P = 127.5$ N. Longitudinal ultrasonic vibration had frequency ~20 kHz and amplitude of a cutting edge a_l ~10 μm.

The wear state of the drill is obviously an important parameter in determining the penetration rate. Examples of this have been recorded where, owing to drill wear, the drill has been unable to penetrate the material except when ultrasonically excited.

The axial reactive force on the drill is usually reduced significantly by the application of ultrasonic vibration. Fig. 4 gives an example of force reduction while drilling with constant feed rate 0.051 mm/rev in 3 mm thick plate of a carbon fibre/resin composite material using a standard twist drill mounted on a longitudinal transducer. The parameters of longitudinal vibration are as indicated above. Cutting was started with no excitation at first to establish the steady level of force and then the ultrasonic excitation was switched on. As it apparent from the figure there is a dramatic reduction in the reactive force when the drill is excited ultrasonically demonstrating that the cutting efficiency is substantially improved. In some drilling experiments, the reduction in the reactive force for given rotational speed and feed rates can as high as 90%.

This leads to significant benefit in drilling performance. These include increase of drill bit lifetime, improvements in the surface finish, roundness and straightness of holes. In ductile materials such as aluminium, copper and mild steel, the elimination of burrs on both the entrance and exit faces of plates takes place.

Fig. 5 demonstrates the exit faces when drilling aluminium. Upper row of the holes corresponds to conventional drilling, the lower one — to superposition of ultrasonics. In the case of drilling through brittle materials, the reduction of stress with superposition of ultrasonic vibration yields large reductions in the size of cratering on the exit face.

The reduction in the reactive force experienced also causes greatly reduces deformation when drilling through thin, flexible plates. With superimposing of ultrasonic vibration it was possible, for example, to drill 6 mm holes cleanly through thin aluminium strips (0.25 mm thick) supported just at the ends (see Fig. 6a).

Attempt to drill through such strips conventionally only resulted in severe deformation of the strip and no penetration (Fig. 6b).

A sample of a 3 mm thick plate made from a special purpose composite material comprising silicon carbide particles dispersed in an aluminium matrix has been successfully drilled using an ultrasonically excited

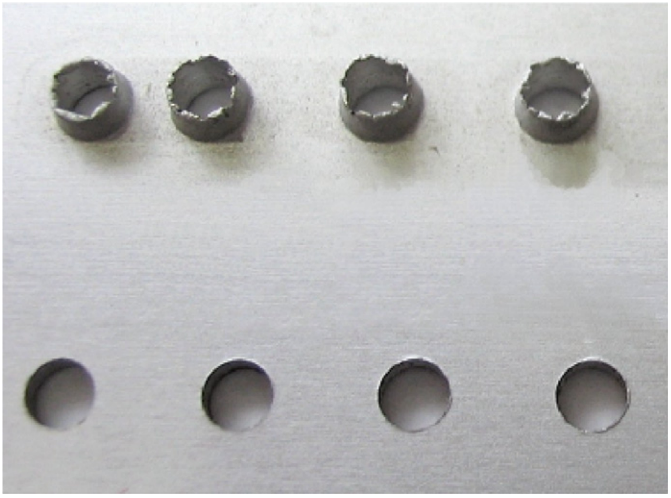

Fig. 5. Comparison in drilling of aluminium.

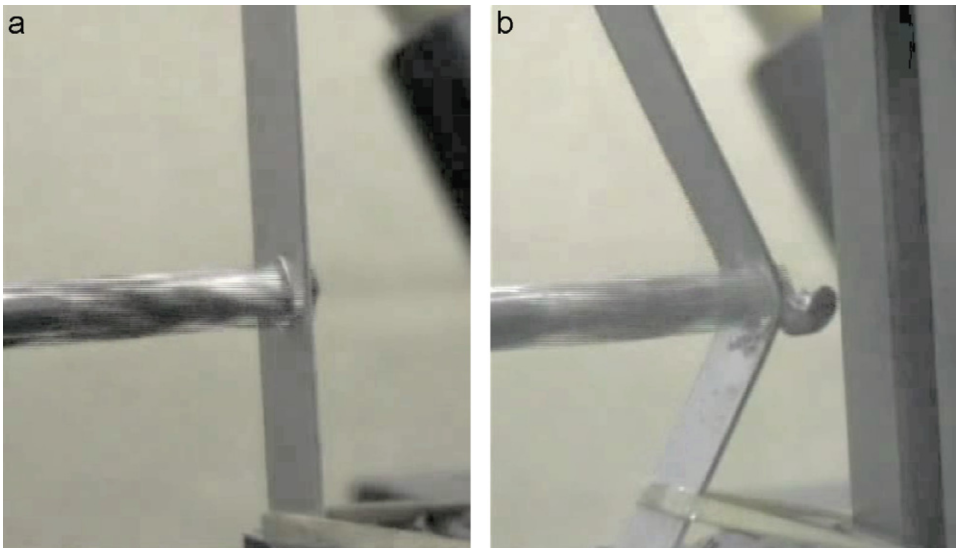

Fig. 6. Comparison in drilling of flexible aluminium plates: (a) ultrasonically assisted drilling, (b) conventional drilling.

diamond impregnated cutter which was substituted for the conventional twist drills used for the previously described drilling trials in metals. The composite material under test proved to be very brittle and extremely abrasive. Attempts to drill through the sample with a new solid carbide drill produced a very ragged hole with a huge exit crater. The drill was severely blunted to such a degree that further attempts to drill completely through the plate failed. In contrast, the material proved to be very easy to drill with the diamond impregnated cutter excited ultrasonically and rotated at only a very slow speed typically 40–125 rev/min, and there was very little exit damage apparent. Diamond drills for penetrating such materials would usually be rotated at very high speeds in excess 20,000 rev/min and would require the use of very expensive, special purpose machine tools.

Reduction in axial force with longitudinal excitation was not observed, however, in the case of drilling through the glass. For this material a special purpose torsional transducer was developed. Fig. 7 shows an experimental set-up for drilling with torsional transducer. A significant reduction in the axial force was measured when drilling glass with torsional transducer. A beneficial effects that, when such a drill is allowed to proceed through the thickness until it emerges on the opposite side, it is found that the area of damage surrounding the exit point of the drill is considerably reduced. In the case of conventional drilling at the same feed rate it is usually found that a substantial area surrounding the exit point of the drill is severely damaged and a large crater is formed. This occurs because, before breakthrough, the stresses surrounding the hole are very large and eventually the fracture strength of the material is exceeded before the tip of the drill has reached the geometric boundary. The greatly reduced damage surrounding the exit point of the drill is independent verification of the substantial reduction in cutting forces resulting from the superposition of ultrasonic vibration on the usual motion of the drill.

Fig. 8 demonstrates a comparison in glass milling with a conventional slot drill. The upper part of the picture demonstrates an unsuccessful attempt of conventional milling of the slot, the lower part is an example of slot received after superimposing of ultrasonic vibration. All the other parameters of machining regime were the same in both cases.

The beneficial effect of vibration on the cutting process in glass has also been illustrated by measurements of the axial reactive force and penetration rates on drills that were not excited externally, which reveal large differences when drilling with and without a lubricant. It has been observed that the presence of a lubricant such as water or oil greatly increased the axial reactive force on the drill because the cutting process can be virtually halted and no penetration is then possible. A laser vibrometer was used to measure the natural vibration of the drill in the two cases when it was found that a dry drill developed a high level of vibration as it was penetrating and cutting the glass but, in presence of a lubricant, the natural vibration of the drill was no longer excited and cutting ceased. The higher level of friction in the dry case appears to induce vibration by a stick-slip process which, once initiated, is sustained by the natural resonances in the drill and the maintenance of physical contact by being continuously advanced into the material.

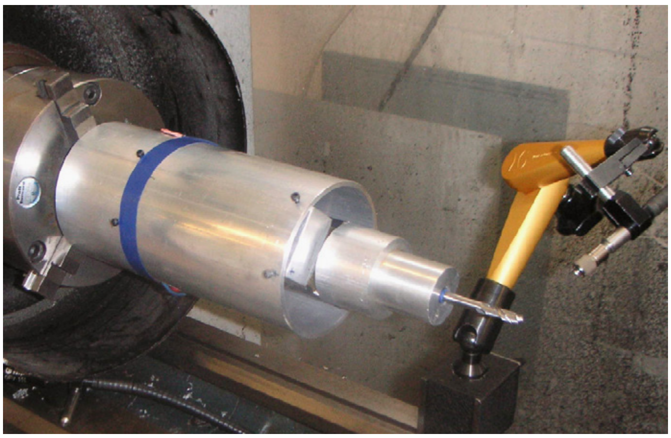

Fig. 7. Experimental set-up of drilling with a torsional transducer.

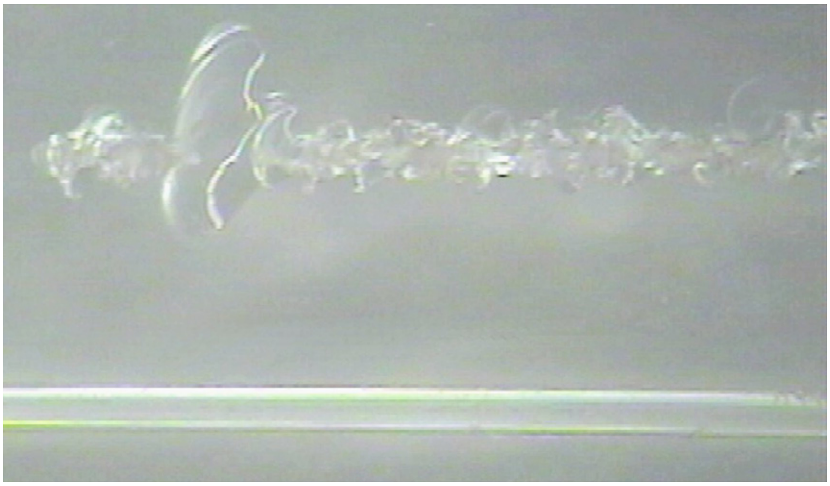

Fig. 8. Comparison in glass milling with a conventional slot drill.

Opposite to the glass drilling, no reduction in axial force was observed when metals were drilled using torsional vibration.

4. Conclusions

The superimposition of longitudinal vibration at ultrasonic frequencies onto the normal rotary motion of drilling bits has a profound effect on their cutting performance in a wide range of materials. A universal effect of the application of the superimposed ultrasonic vibration is the substantial reduction in the cutting forces resulting from the greatly increased cutting efficiency produced by the ultrasonic vibration.

A successful application of the technology needs an intensive and stable supply of ultrasonic energy into the machining zone in conditions of nonlinear and unpredictably variable cutting loads. Matching of the dynamical qualities of the transducer and drill bit with the dynamic loads imposed by the cutting process will help in stable generation and maintenance of an intensive nonlinear resonant mode of vibration, which can be accomplished by the autoresonant control system [8,9].

Acknowledgments

V.K. Astashev thanks the Russian Foundation for Basic Research for support (Project 05-08-50183).

References

[1] J. Kumabe, *Vibratory Cutting*, Jikkyo Publisher, Tokyo, 1979 (in Japanese).
[2] J. Devine, Ultrasonically assisted metal removal, American Society for Metals, Fabrication of Composite Materials: Source Book, 1985.
[3] R. Neugebauer, A. Stoll, Ultrasonic application in drilling, *Journal of Materials Processing Technology* 149 (2004).
[4] V.K. Astashev, Calculation of rod concentrators with a nonlinear load, *Soviet Physics-Acoustics* 27 (6) (1981).
[5] V.K. Astashev, V.I. Babitsky, Ultrasonic cutting as nonlinear (vibro-impact) process, *Ultrasonics* 36 (1–5) (1998).
[6] V.K. Astashev, V.I. Babitsky, M.Z. Kolovsky, *Dynamics and Control of Machines*, Springer, Berlin, 2000.
[7] V.I. Babitsky, *Theory of Vibro-impact Systems and Applications*, Springer, Berlin, 1998 (Revised translation from Russian, Nauka, Moscow, 1978).
[8] V.I. Babitsky, V.K. Astashev, A.H. Kalashnikov, Autoresonant control of nonlinear mode in ultrasonic transducer for machining applications, *Ultrasonics* 42 (1–9) (2004).
[9] V.I. Babitsky, A.N. Kalashnikov, F.V. Molodtsov, Autoresonant control of ultrasonically assisted cutting, *Mechatronics* 14 (1) (2004).