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Ultrasonic peening: Dynamics of multi-stricker impact tool

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Introduction

Ultrasonic peening is a promising technology for improving the reliability of welded joints, especially their fatigue resistance. This method employs a multi-striker hand tool with free needle-like intermediate strikers placed in a gap between the ultrasonic vibrator and a specimen [1]. The ultrasonic peening technology is a logical continuation of well-known techniques for surface plastic deformation. The fatigue tests [1,2] showed that ultrasonic peening is the most efficient technique for a post-weld treatment as compared to shot peening, hammer peening, TIG-dressing, deep rolling [3], etc.

Despite the fact that the mechanism of the ultrasonic peening is well understood [1], some important features remain unclear. In particular, there is contradictory information about the frequency of oscillation of the striker in the gap: oscillation at the carrier frequency, subharmonic oscillations and non-periodic (random) impacts were observed [2].

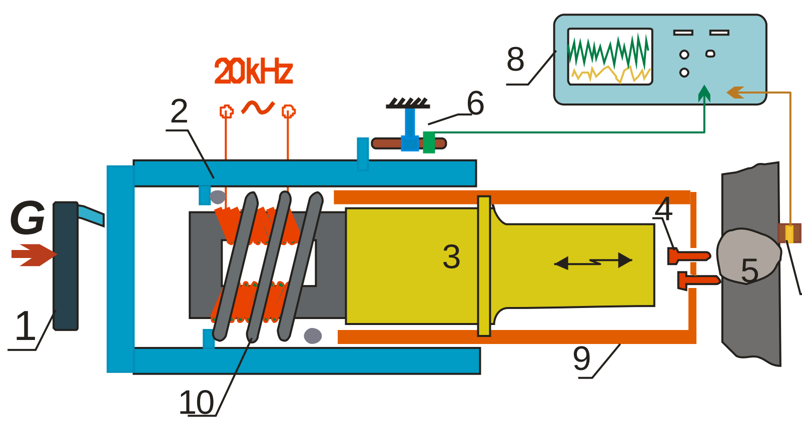

Fig.1. Ultrasonic tool scheme: 1 –pressing weight, 2- tool casing, 3-ultrasonic transducer, 4 - intermediate striker, 5- weld, 6 -displacement gauge , 7 – piezoelectric sensor, 8 – oscilloscope, 9 – holder, 10 – spring.

Furthermore, there is no complete knowledge on the impact interaction of the vibrating striker with the treated material. Periodic continuous (without detachment) contact between the striker and the treated surface and non-periodic interrupted contact with rebound of the striker from the workpiece are described in paper [1]. The efficiency of processing is analyzed depending on the nature of the contact interaction, but the conditions for creating and keeping a particular impact mode have not been clarified. There are no direct measurements or reliable calculations of the dynamic characteristics of the high-frequency treatment (the values of the impact force or velocity, impact momentum or kinetic energy, the impact duration, etc.). These shortcomings, as well as the light weight of the striker, the oscillations invisible to the naked eye, the low weight and dimensions of the ultrasonic hand tool raise doubts about the applicability of the ultrasonic peening method primarily for the massive and large-scale welded elements. Based on the studies so far, a deeper understanding of the dynamics of ultrasonic impact tool with the intermediate striker can be achieved through the use of the advanced methods of the theory of vibro-impact systems [4,5]. In particular, the application of these methods shows that oscillations of the transducer and the striker can occur both in phase and in antiphase [5]. In the first case, the tip of the transducer oscillates in continuous contact with the rear end of the striker and this mode is ineffective. The excitation of antiphase oscillation requires a special tuning of the drive frequency of the ultrasonic transducer.

Objectives

The objectives of this work is to develop a dynamic model of the ultrasonic impact tool with taking into consideration the results of preliminary research [5]; to investigate the conditions of existence and stability of vibro-impact modes; to compare predictions of the theory with experimental observations; to calculate the value of impact stresses induced in the treated material using analytical methods of the theory of vibro-impact systems [4]; to describe a dynamic behavior of the ultrasonic impact tool in the framework of the presented model.

Methodology

1. Experiment

One of the features of the vibro-impact systems is displacement of the mass center into a position of dynamical equilibrium when operating [4]. The value of so called “a dynamic shift” or “dynamic drift” is either permanent or slowly varying and directly connected to dynamic parameters of the system (vibration amplitude, impact force, etc.).

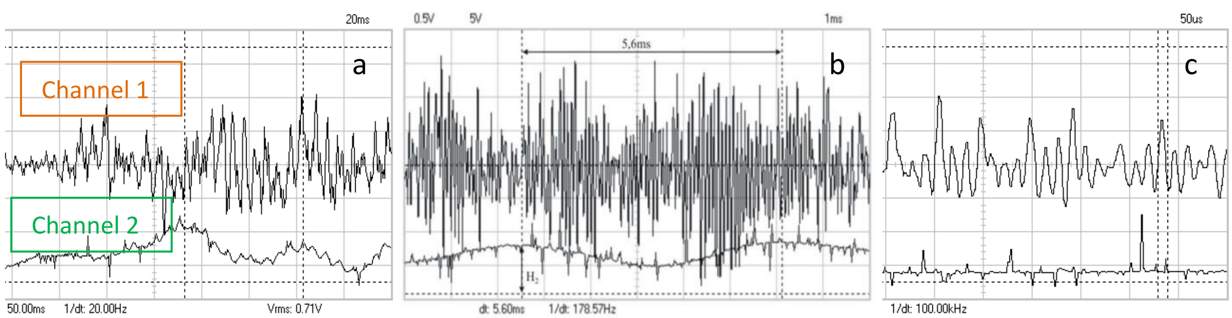

Fig. 2. Oscilloscope pictures of impact force (Channel 1) and tool casing drift (Channel 2). X-axe: time for scaning rate 20ms (a), 1ms(b) and 50 μs(c) per the grid division. Y-axe: voltage 0.5V and 5V per the grid division for Channel 1and Channel 2 consequently.

To study relationships between the dynamic gap value and the vibro-impact mode parameters, the dynamic drift of the tool casing and the impact pulses were measured simultaneously. High-frequency impact pulses were measured by the calibrated piezoelectric sensor. The dynamic drift was measured by the linear displacement gauge. Both signals were recorded with the use of a two-channel digital storage oscilloscope. Scan-pictures of Fig.2 shows that the multiple impacts occur with the frequency exactly equal to the frequency of ultrasonic actuator. The duration of single impact can be estimated as one fifth of ultrasonic period. The tool casing performs two superimposed almost harmonic oscillations with frequencies of order 20Hz and 180Hz. These slow biharmonic oscillations are synchronously accompanied with the periodic change in a height of impact peaks.

2. Theory

2.1 Impact interaction

Consider the dynamic model of the ultrasonic tool as a two-mass system with two impact joints. The interaction of the ultrasonic transducer with the rear end of the striker can be described as a head-on collision of the rod with the effective mass M_1 to the body with a concentrated mass M_2.

Absolutely elastic approximation is acceptable because of the high hardness of the contacting surfaces. Then the postimpact velocities of the tip of the transducer V_1^- and the striker V_2^- are connected with the preimpact velocities V_1^+ and V_2^+ by the relations [4]:

$$V_1^+ = V_1^- - \frac{M_2}{M_1 + M_2}(V_1^- - V_2^-) \ , \quad (1)$$

$$V_2^+ = V_2^- - \frac{M_1}{M_1 + M_2}(V_1^- - V_2^-) \, . \quad (2)$$

The impact of the striker upon the surface of the specimen is considered inelastic and is described by the velocity recovery factor R:

$$V_2^+ = -RV_2^- \ . \quad (3)$$

Consider a steady-state vibro-impact mode with continuous slow oscillations of the ultrasonic tool. According to the experimental observation, the motion of transducer can be described as a superposition of low-frequency oscillations of the transducer as a single unit inside the tool-casing and forced ultrasonic oscillation of the transducer tip:

$$x(t) = (c_1 \cos 2\pi\Omega t + c_2 \sin 2\pi\Omega t) \cdot \exp(-\chi\Omega t) + a\cos(2\pi f t + \varphi) \, , \quad (4)$$

where Ω and χ are the frequency and damping coefficient of the transducer's natural oscillations; a, f, and ϕ are the amplitude, frequency and phase shift of the ultrasonic oscillation.

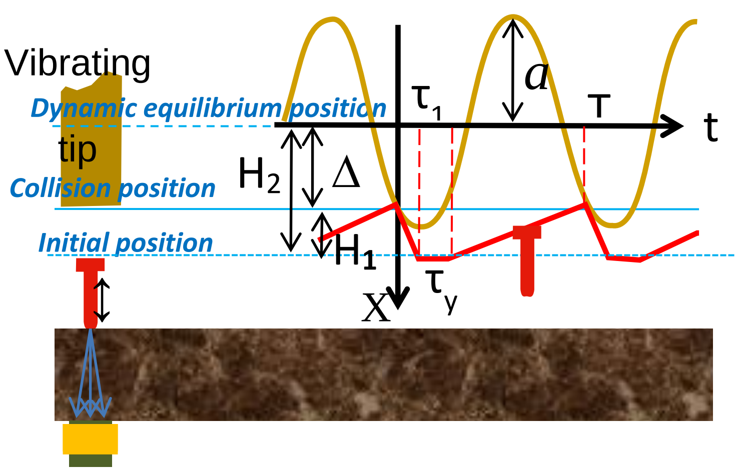

Fig.3. Scheme of vibration of transducer tip and reciprocal motion of striker.

Within a short period of time equal to the period T of ultrasound, the tool-casing stays practically immobile at a position of dynamic equilibrium, which is located above the initial one (before the ultrasound will be switched on) at a height of:

$$H_2 = \Delta + H_1 \ , \quad (5)$$

where Δ is the dynamic shift of the transducer subjected to the high-frequency collisions with the moving back striker; H_1 is the dynamic gap between the transducer tip and the rear end of striker or, that is the same, between the front end of striker and the workpiece surface. During the reciprocating motion the striker passes the same distance in forward and revers directions, therefore:

$$H_1 = V_2^{+} \tau_1 = -V_2^{-} (T - \tau_1 - \tau_y), \qquad (6)$$

where τ_y is the impact duration, τ_1 is the time interval between the collision of the striker with the transducer tip and the next impact upon the specimen (See Fig.3). The momentum conservation theorem can be applied to the ultrasonic transducer in the state of dynamic equilibrium:

$$GT = M_1 (V_1^{-} - V_1^{+}), \qquad (7)$$

where G is the static pressing force. A system of five linear equations (l),(2),(3),(6) and (7) for five unknowns $V_1^{-}, V_1^{+}, V_2^{-}, V_2^{+}$ and τ_1 has a single solution. Here are the expressions for two of them:

$$V_1^{-} = \frac{GT}{M_2}\left(\frac{R}{1+R} + \frac{M_1 + M_2}{2M_1}\right), \; V_2^{+} = \frac{1+R}{R}\left(V_1^{-} - \frac{GT(M_1 + M_2)}{2M_1 M_2}\right) \quad . \qquad (8)$$

Since after the collision both the tip of the transducer and the recoiled striker move in the same positive direction along the X axis, both variables V_1^{-} and V_2^{+} must be required to be positive.

2.2 Steady-state mode

During the steady-state mode collisions of the transducer tip with the rebounded striker occur again and again at a certain permanent point with the coordinate $x=\Delta$ between the position of dynamic equilibrium and the surface of the specimen (Fig.3). According to the method of "fitting" boundary conditions, the requirement of periodicity for repeated collisions should be written in the form [4]:

$$t = 0, \; x = \Delta, \; \dot{x} = V_1^{+}; \quad t = T, \; x = \Delta, \; \dot{x} = V_1^{-} \quad . \qquad (9)$$

Substituting the law of motion represented by Eq. (4) into Eq. (9) we obtain the expression for the dynamic shift Δ of the ultrasonic transducer:

$$\Delta = DV_1^{-} / f + \sqrt{a^2 - (BDV_1^{-} / f)^2} \; . \qquad (10)$$

Coefficients B and D depend on design parameters of the ultrasonic tool [5] and processing parameters G, R and τ_y .

2.3 Range of existence and stability

The range of existence of real solutions for the dynamic shift is a continuous set of values of the collision velocity V_1^{-} bounded by two inequalities arising from Eqs. (8) and (10):

$$\frac{GT(M_1+M_2)}{2M_1 M_2} \le V_1^{-} \le \left|\frac{af}{BD}\right| \quad . \qquad (11)$$

Thus, the steady-state oscillation with the impact repetition frequency equal to the ultrasonic frequency can be initiated only if the velocity V_1^- of the transducer tip vibration at the moment of collision with the striker exceeds the specific threshold value, which explicitly depends on the pressing force G. The oscillatory stability of the obtained solutions was verified within the framework of the Schur's criterion of asymptotic stability [4] (not presented in this paper). It is found, that all vibro-impact modes with the collision velocities V_1^- lying inside the range of existence are stable. The vibro-impact mode losses the stability of oscillations at the upper and lower boundaries of the range of existence, where the strict equalities in Eq. (11) are satisfied.

2.4 Impact stress

Assuming that the impacting elements are deformed in accordance with the Hertz law [4], substitute the velocity V_2^+ of impact upon the workpiece into the known expressions for the impact force F and the radius y of the contact spot:

$$F = q^{2/5}(M_2 V_2^{+2})^{3/5}, \qquad y = \left(2\sqrt{2}F/q\right)^{1/3} \cdot \left(r_{st}^{-1} + r_{wd}^{-1}\right)^{-1/2}, \tag{12}$$

where the factor q depends on the Young's modulus and the Poisson's ratio of the material being treated; r_{st} and r_{wd} are the radii of curvature of the contact surfaces of the striker and the specimen. The value of impact stresses generated in a metal can be calculated as the ratio of the magnitude of the impact force to the area of the contact spot:

$$\sigma = F/\pi y^2. \tag{13}$$

Numerical estimations for the Inconel alloy (4mm butt-welded plate) with the use of the ultrasonic tool design parameters [5] show, that inside the local spot with the area of about 1mm^2 the impact stresses reach the yield strength σ_0 of the metal (See Fig.4). Thus, the redistribution and relaxation of internal residual stresses is caused by the micro-plastic deformation of the surface layers of the metal.

Results and analysis

The Eq. (10) establishes the interrelation between the dynamic shift Δ and the collision velocity V_1^- for each steady-state vibro-impact mode at the carrier frequency. The collision velocity of the transducer tip V_1^- determines the impact velocity of striker V_2^+ (See Eq.(8)), which in turn, determines the impact force F (See Eq. (12)) and the total displacement of the tool-casing H_2 (See Eqs. (5) and (6)). The dependence of the steady-state mode parameters on the dimensionless collision velocity $v = V_1^-/af$ within the range of existence and stability is shown in Fig.4. The calculation was performed using MATCAD 15 and substituting the experimentally measured parameters (f≈20kHz, a≈20±2μm, Ω≈180Hz, τ_y≈10μs, R=0.65) and design parameters of the ultrasonic tool [5] with the automatic frequency control.

It is seen that at the lower boundary of existence the dynamic shift of transducer is equal to the total displacement of the tool-casing, each of which is equal to the amplitude of the transducer tip vibration: $H_2/a = \Delta/a = 1$. It follows from equation (5) that $H_1 = 0$, i.e. The gap is closed and the intermediate striker has no free space for motion. This means that the striker stays on the specimen's surface, and the vibrating tip of the transducer periodically touches the rear end of the immobile striker. This kind of the impact interaction was named in [1] as «ultrasonic periodic impact». It is known [4], that the exact equality of dynamic shift to the oscillation amplitude is exclusively fulfilled for the approximation of the impact upon an absolutely hard material.

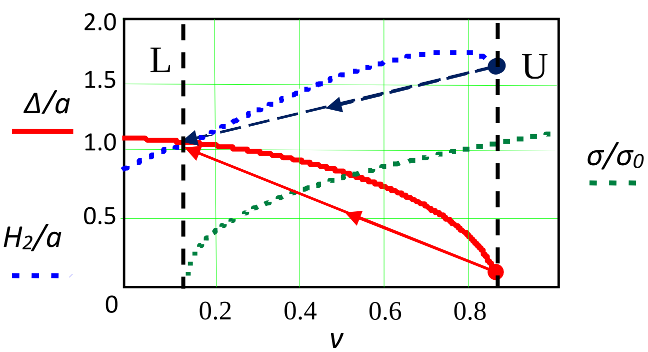

Fig.4. Dependence of dynamic parameters on the dimensionless denote the lower and upper boundaries of the range of existence. The arrows illustrate the change in values due to the transition of the vibro-impact system from the upper unstable state to the lower one.

Therefore we can conclude, that in the case of the «ultrasonic periodic impact» the deformation of the specimen surface is negligibly small.

At the upper boundary of the range of existence (right side of Eq. (11)) the value of the dynamic shift of the transducer Δ is equal to zero (See Fig.4). According to Eq. (5), this means that the width of the dynamic gap is equal to the displacement of the tool-casing and is estimated by one and a half of the amplitude: $H_1 = H_2 \approx 1.5a$. In the framework of the method of harmonic linearization [4], the reciprocal motion of the intermediate striker should be approximated as a harmonic oscillation at ultrasonic frequency with an amplitude equal to half the gap width: H_1 / 2. Thus, a gradual increase in the dynamic gap should be interpreted as a phenomenon of pulling-up the oscillation amplitude of the striker. This is a well-known feature of vibro-impact systems [4]. It should be noted, that at the upper boundary of existence the oscillation velocity of the transducer tip reaches its maximum at the moment of collision with the striker. So, the single impact pulse, which is transmitted from the ultrasonic tool to the workpiece, has the highest value compared with any other stable vibro-impact mode.

Conclusion

Main features of ultrasonic impact processing are explained and numerically estimated in the frames the proposed vibro-impact model. The periodic change in the amplitude of the impact pulses is explained by the alternation of the pulling-up of the oscillation amplitude of the striker and subsequent loss of stability of the oscillations. Two modes of the contact interaction - continuous (without detachment) contact and vibro-impact (periodically interrupted) contact - are described as two different solutions of the same dynamic equations for the upper and lower limits of the oscillation stability zone. The frequency of impacts is equal to the frequency of ultrasound, and the induced impact stresses reach the yield strength of the material being processed.

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