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Dynamic Perturbation Characterizations And Regressive Energy Profiling In High Velocity Machining

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ABSTRACT

Studies have shown that high velocity machining operations are significantly characterized by vibration potentials and dynamic transients largely connected to varying instabilities and departure from process constraints. The challenge of increased performance indices of production schedules and the rising competition in quality defined products and services places enormous demand on the research and development unit to evolve technological initiatives and designs that would support and sustain the reduction or elimination of transient perturbations and vibration levels in high velocity machining conditions. Thus, vibration level minimization or elimination is therefore a function of stability limits which define the cutting loads assignable to tools with respect to damping of vibration before its arrival at tool ends. The investigation established a nexus between capitulated energy quotients (CEQs) under vibration and transient perturbations; and the efficiency of machines. By this finding, the paper indicated that the resolution of the problems of clamp load deviation indices (due to vibrations) can significantly enhance dimensional accuracy, technical and commercial viability of items manufactured under high velocity machining regimes.

Keywords: *capitulated energy quotients, clamp load deviation, particulate force, quantized regressive energy profiling, transient vibration fluxes, structural imbalance*

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I. INTRODUCTION

The paper intends to bring to relevance the critical fact that joint stiffness and damping capacity of machines should effectively be designed to counter incidental energy transients and perturbations. This means that in order to increase vibration resistance of a machine tool structure as to cut down on perturbation tendencies, damping capacity must be enhanced. This paper is therefore focused on the mechanics of these perturbation energies resulting from process instabilities and how these energies impact a retrogressive energy sequence on the general performance of the machines.

Further, in addition to dissecting the energy and relative quotients the paper shall also resolve energy values of various forms of joints that are encountered in machines. The reason being that these joints serve, both as energy transmission links and exit points. Thus, a good understanding of joints and jointing dynamics is critical to energy profiling and distribution matrices. It has been observed that the total damping in a machine tool structure is greater than that of the sum of the internal materials damping of individual structural member [1]. The design implication of this development is that the predominant nature of mechanical joints as the investigations would show includes fixed and moving links such as, welded joints, bolted joints, sideways bearings, etc.

It is imperative to state that process transients and perturbations are fundamental means of energy transformation in machining centers, studies have indicated that in order to minimize their effects, proper attention should be channeled to mechanical joint analysis; this is due to the fact that joints damping assumes between 70% -90% of total damping in a machine tool structure [2], [3]. In view of this finding three energy dissipation mechanisms have been observed for bolted joints namely, macroslip, microslip, and cyclic plastic deformation of the contacting asperities [1].

Thus, the study of mechanical joints is necessary to the understanding of the various forms of energy progression and advances in the areas of lubrication, structural health monitoring, smart washers, belt damping and higher rivet squeezing forces. This paper thus aligns with the experiment of Mascarenas et all [4] who studied joint integrity using impedance based sensor, that transmitted data wirelessly and can also be powered wirelessly. The sensor was designed and integrated into the joints and by it, data was acquired based on changes in electrical impedance sent out by high-frequency ultra sonic pulses, which this papers views as *transient vibration fluxes* capable of electromagnetic characteristics. Though, their experiment was highly rewarding, suffice it to mention that it did not address the energy dissipation question with respect to the effect of these identified dynamic transients and perturbations resulting various degrees of vibration to which a machine's efficiency is dependent. Thus, dynamic transients and perturbations generate ultra-high frequency waves due to frictional relativities of contacting surfaces.

As the paper would show, these waves are the direct results of energy transformation processes from the various force centers in the machine structure and substructural units. In furtherance of this argument, this condition would be more appreciated under a spectrum analyzer where the frequency and the magnitude of microscopic motions are shown. Thus, the use of accelerometer for sensing of motion implies the translation of motion (rotary or linear) into an electrical signals; which are picked by the spectrum analyzer and translated into dynamical sequential signatures with properly defined particulate dimensions that can be easily associated with a particular type of motion. This paper shall further show that dynamic transients and perturbations can result from:

i) structural imbalance of core machine components resulting from manufacturing defects or maintenance errors.
ii) misalignment and shaft run out due to angular re-organization of shaft during assembly or while in operation.
iii) wear as consequential result of age and operation. This causes relative deformity that rubs the equipment of its logical operational sequence.
iv) looseness which occurs unconsciously and targeted at jointing points, i.e. either welded, bolted or otherwise.
v) leveling and gravitation which has to do with the level placement of the equipment thus providing appropriate gravitational capacity to maintain logical operations.

In view of these identified factors, this paper posits that vibration initiators give rise to particulate energy components with consequential frictional and geometric considerations. This imply that the irregularities identified above, results forces and their forcing functions which aggregately exit from their *source centers* or points as *capitulated energy quotients* (CEQs) of *varying degrees of freedom* (VDOF). Thus, the value of VDOF becomes a critical issue when such energy in the form of multi-directionally distributed waves destinate at bolted or welded joints; thus resulting loosening which this paper views as *regressive energy profiling* in the sense that, upon impact at the joints, these energy values if higher than the aggregated stiffness factor S_{AF} (of the joint), has the capacity to break the binding energy which keeps the welded joint or break the stable force binding the bolts and the nuts. It should be noted that when such impact results, the joint stiffness becomes susceptible to further flow of energy in the form of high amplitude waves. The focus of this paper therefore is to analyze the frictional and geometrical characteristics of this joint cleaving energy and its effects on the stiffness factor of the joints in such a manner as to show that *stiffness regressivity* is a correlate of perturbation forcing functions.

II. PERTURBATION INITIATORS AND THE DIRECTIONAL ENERGY FACTOR

In the view of this paper, perturbation initiators with respect to high speed machining have been identified above. These vibration generators could be viewed as regressive energy source points (i.e. where vibrations portend inimical force components to the system under consideration). It should be noted that a machine is designed to perform in a sequential and logical manner [5], and that, its resultant functionality is dependent on this behavioral sequence of output which also determine and define its relative operational stability and reliability [6]. On the other hand let us assume that the progressive energy source centers or points associated with the machine of interest during operation falls out of the logical sequence as designed and thus generates regressive energy projected (in the form of distinctive wave forms) to the joints, where joint stiffness may be attacked by:

(i) forces opposite and greater in magnitude to the binding force or stiffness factor of the joint, thus resulting instant failure due to shatter.
(ii) forces opposite but of lesser magnitude to the stiffness factor or binding energy, thus resulting increasing reduction in the binding energy and consequent graduated loosening within machine operability regime.

It should further be noted that this regressive energy (force) unlock the bolted joints (in the case of bolt and nut joints) or cracks the welded points (in the case of welded joints); thereby exposing the joints to further attacking actions as the machine continues to operate at elevated speed over a prolonged time. The effect of this action on the machine is a drop in its efficiency and consequently, illogical operation; resulting energy exit in converted forms of machine noise and heat. In view of the foregoing, this paper shall consider this regressive energy as a *quantized particulate force* (QPF) transmitted by energy waves in the sinusoidal form with amplitude and frequency defined by wave function, $\psi(x,t)$, pursuant to structured space wave mechanics and variability for which uniformity shall be imposed as a basis for inferential assessment and analysis.

Impliedly, and for the sake of the position in this paper, regressive energy would be viewed as a particle with definable and measurable characteristics. This means that, if the regressive energy, W_{reg} , is a particle subjected to a derivative wave function incumbent on a lattice phase space condition, then, the core of this systemic functionality of transmitted *QPF* is represented by the metallic lattices of the machine integral parts serving as the conduction field for transfer of this energy from source to target. Thus, if a particle whose mobility is defined by the wave function $\psi(x,t)$ exists in a lattice phase space, then the probability of finding that particle in the region $(x, x + dx)$ at time t is:

$$|\psi(x,t)|^2 dx \qquad (1)$$

It should be noted that while equation (1) is definitive for a single particle, it is necessary to posit that this particle shall be deemed to be a representative element and as such normalized to unity by virtue of the expression:

$$\int_{-\infty}^{+\infty} |\psi(x,t)|^2 dx = 1 \qquad (2)$$

Thus, equation (2) supposes that the total probability of finding the particle within the metallic lattice structure of the machine parts must add up to unity (i.e.1). This harmonization thus implies that W_{reg} exists within the localized constraint of a lattice space.

Further, it is important to keep in mind that the particle of interest, W_{reg} is a regressive energy factor which implies that the wave function must be non-existent as $x \rightarrow \mp\infty$. The meaning of this position is that the energy entity x must exist within the bounds of certainty; otherwise the area integral factor $\int$ would not be a infinite and indefinable. This paper therefore posit that this condition of the wave function imposes a value on the energy of the particle and as such can be described as a quantized particle within the area integral limits devoid of $\mp\infty$ extrapolation.

Thus, energy particle quantization in the view of this paper asserts that, if the particle has any value, then, that value is a discrete and tangible value capable of delivering force to achieve an objective as would be seen latter.

In furtherance of the foregoing argument, it is important to point that although the regressive particle posses the following energy and momentum characteristic;

$$\psi(x,t) = \begin{array}{l} A\sin(kx - wt) \\ A\cos(kx - wt) \\ Ae^{i(kx-wt)} \end{array} \qquad (3)$$

describing this particle within the integral framework of $|\psi(x,t)|^2$, implies that the integral is infinite, presupposing the improbable condition of existence. However, this particle cannot be said to be a non-existent entity when it is a force that could be used to do work. Sufficient therefore is the position of this paper that the wave function circumscribing the characteristics of this particle possesses the same amplitude everywhere in the phase space, meaning that the particle is equally likely to be found anywhere within the lattice space or structure of the metal, composing the machine. Having established the universal equi-potential energy quantization possibility of the particle, suffice it to say that this normalized function is not only unitary but also exist within finitely defined region of space, bounded by $a < x < b$ and expressed as;

$$\int_a^b |\psi(x,t)|^2 dx \qquad (4)$$

Thus, the limits imposed on equation (4) imply a boundary condition which confines our particle of interest to equi-potential matrix arrangement, definitive of source to destination. Where source imply the vibration initiators and destination is the joints from where energy is exited.

III. ANALYSIS OF BOLT LOOSENING UNDER STRUCTURAL PERTURBATION

It has been established that vibration energies are perturbation factors to joints which have been identified as exit points to the generated energies. This implies that the energy propelled particle E_{pp} is impacted upon by the operator E_n, whose expression holds that:

$$E_n = \frac{\pi n^2 h^2}{2mL^2} \qquad (5)$$

It should be noted that equation (5) defines the impact relativity and hence becomes the fundamental carrier of the force required to attack the clamp load or stiffness factor of the joint. It should be borne in mind that the E_{pp} consequence is a wave propagation directed at the joint using the medium of the lattice structure of the metal. It has been reported by Hartman[5] who identified that two wave patterns are noticeable through ultrasonic wave analysis of the loosening behavior of bolted joints. As would be shown in the resultant Figure 1, pursuant to his finding, the E_{pp} under the influence of the E_n constitutes force fields in the nature of *longitudinal waves* maintained in the same direction as the bolt and nut ***x-x*** sectional axis of coupling and another wave formation in the nature of *transversal waves* which swings the E_{pp} in a direction perpendicular to the axis of the bolt.

The culmination of these *geometrics of point mechanics* is a force opposite and higher (or lesser) in value to the fastening energy at the point of highest binding stress within the internal interface and lining of the interlaced bolt and nut threads. It should be noted that these twin waves travel at different velocities resulting Hartman's finding that, for standard steel longitudinal waves are propagated at a velocity of 5,900 m/s and for transverse waves 3,200 m/s. The implication of this is that the amplitude of these individual waves varies with the forcing functions of the vibration initiators, as explained below.

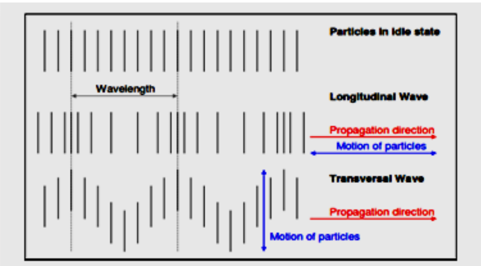

Figure 1: ultra sonic waves in clamp load measurements indicating vibrational transients within operability regimes of high velocity machining

Pursuant to the foregoing, this paper is of the firm view that although the speed ratio of the two Hartman identified ultrasonic waves is about 1:8, their aggregated effect in the relativity of force and space, $\boldsymbol{f = m_{Epp} \times a_{E\psi}}$, is the cleaving or breaking of the binding energy profile of the joint. It should further be noted that while m_{Epp} defines the quantifiable value of the force operator, $\boldsymbol{E_n}$; E_{pp} resolves the energy mobility factor, $\boldsymbol{a_{E\psi}}$; thus describing the aggregated acceleration of the ***ScrÖdinger*** *particle*, whose one-dimensional characteristics is defined as:

$$E\psi = -\hbar^2/2m(d^2\psi/dx^2) + V(x)\psi, \psi = \psi(x) \qquad (6)$$

Thus, while $V(x)\psi$ defines the potential energy propelling the particle, the Hamiltonian factor H, expressed as, $\{-\hbar^2/2m\,(d^2\psi/dx^2)\}$ defines the energy expectation value being propelled for which the resultant impact on the joint over a period of time amounts to bolt loosening effect referred to in this paper, as *clamp load deviation indices*. However, the degree of this deviation is also dependent on the combined stiffness factor of the joint components and the interrelationships of their various process variables. To determine this stiffness, *k*, of any axially loaded member, the following expression applies:

$$k = AE/\lambda \qquad (7)$$

Where A = loaded cross – sectional area
E = Young's modulus
λ = loaded length (of bolt)

It should thus be noted that the ratio of joint member stiffness to bolt stiffness is a critical factor in determining *incident perturbation values* capable of discharging clamp load restrictions. Hence such ratio can be maximized using the expression:

$$\frac{A_j}{A_b}, \quad \frac{E_j}{E_b}, \quad \frac{\lambda_b}{\lambda_b} \qquad (8)$$

which invariably imply that combined joint stiffness is a monumental consideration and directly contributory to the strength of the joint.

In view of the foregoing, assume that the identified structural perturbation requires that the particle $\boldsymbol{E_{pp}}$ shall be induced by $\boldsymbol{E_n}$ under canonical vibrations and based on "optical amplitude function", which has a defined frequency component, then a time evaluated constraint results a phase consequence of:

$$\varphi = e^{2\lambda ivt}\,\psi(x,y,z) \qquad (9)$$

while the amplitude – function is determinable using the "amplitude equation".

$$\frac{\partial^2\psi}{\partial x^2} + \frac{\partial^2\psi}{\partial y^2} + \frac{\partial^2\psi}{\partial z^2} + \frac{4\lambda^2}{\lambda^2}\,\psi = 0 \qquad (10)$$

Pursuant to equations (9) and (10) above, suffice it to posit that amplitude–function is a relative application to the momentum expected on $\boldsymbol{E_{pp}}$ and only reflect a perturbation condition that is characteristically defined by:

$$U = Ae^{\,i2\lambda\,(vt-\,\varphi(x,y,z)} \qquad (11)$$

here:
U = perturbation characterization relativity
A = amplitude
V = frequency
φ= phase angle and coordinate canonicity transformation

It should be noted that a wave under U impact as indicated in equation (11) would possess a surface best described in 3-D relativity and expressed canonically as; $\varphi(x,y,z)$=Constant, implying that the amplitude of the wave evens out and maintains that uniformity throughout all surfaces of transit and impact. The result of this uniformity imposition is better described by de Broglie's wavelength [7] expressed as:

$$1/\lambda^2 = \frac{2m\,(E\text{-}v)}{h^2} = \frac{M^2v^2}{h^2} \qquad (12)$$

and obligated on equation (10), thereby resulting a definition for the amplitude of the progressive energy waves. The outcome of this subjugation is the distinctive wave prototype relativity;

$$\frac{\partial^2\psi}{\partial x^2} + \frac{\partial^2\psi}{\partial y^2} + \frac{\partial^2\psi}{\partial z^2} + \frac{8\lambda^2}{\lambda^2}(E\text{-}V)\psi = 0 \qquad (13)$$

Hence equation 13 is the ***ScrÖdinger's*** differential relativity, deterministic of the behavior of particulate oriented perturbation; where the particles in attempt to transfer energy carried by the propagated waves vibrationally oscillate in their mean positions. It is instructive to note that this micro-scale particulate perturbation due to energy transmission, portend some distinguishable level of thermal consequence as its conduction of energy causes state space excitation which results heat dissipation due to energy hysteresis and loses. This is a attested to, by the characteristic temperature rise during machine operation.

Further, the loss of vibration energy by conversion to heat also compensate for the thermodynamic balance achieved by virtue of the fact that rising temperature resulting from frictional forces and surface resistance offered to the flow of energy by the particle causes structural realignments due to linear metallic expansivity [8] which cusses the expansion of the joints upon temperature rise and drops when the temperature forcing function is removed. It is important to state that this situation also impacts on the bolt–nut or bolt-tapped hole threaded interface; as the occasioned expansion and contraction results self-loosening tendencies by exposing the clamp load value to micro-structural depletion due to direct impact of heat on the frictional variables and tensors.

IV. PERTURBATION SUSCEPTIBILITY ASSESSMENT OF BOLTED MACHINE JOINTS

The vibrations of machine parts have been acknowledged to flow from perturbations originating from unstable disoriented illogical machine operations and behavior. This implies that vibration is the result of non-sequential dynamics; thus ensuing irregular performance of machine parts during operability phases of machine life. The evaluation of this susceptibility regime of bolted joint utilized the experimental results for vibration resistance using the DIN 65151 test bed [9] as shown in Figure 2 below:

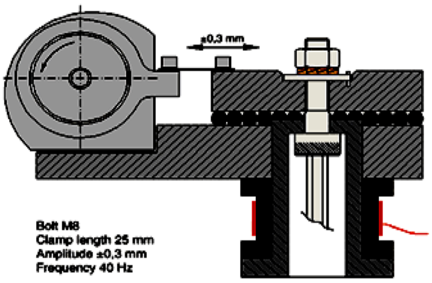

Figure 2. Junker DIN 65151 test bed (Courtesy: AME-Lab, Niger Delta University)

The Junker DIN 65151 vibration tests was used to evaluate the stability of bolted joints under high frequency and amplitude displacement conditions reminiscent of vibration of machines during high velocity machining. As could be seen in Figure 2, the bolt is fitted into the vibration axis of the test bed. The bolted joints is further subjected to transverse movement while the tension, frequency, amplitude of wave and other variables are measured from a load cell computer laptop device connected to the setup and observed pattern analyzed in Figure 1.

As stated in the reported experiments [5], [9]. Figure 3 below, indicates a test of various bolts (5 in number) from different manufacturers, under the same conditions. Though this experiment did not address the core issues of this paper, it nevertheless established the practical results of the mathematical relativities of wave particle dynamics under energy transmission conditions and this agrees with the experimental results of Nord-Lock[11] as shown in the graph below.

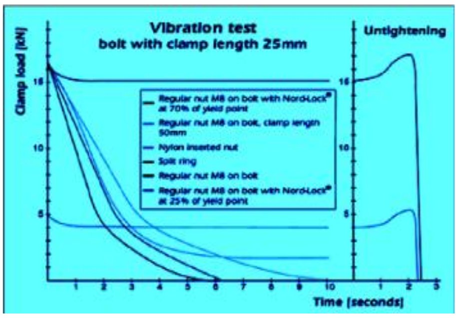

Figure 3: results of bolt preload failure evaluation for five bolts of same size from different manufacturers

Thus, Figure 3 indicate that all the bolt types that are vulnerable to vibration and transient instabilities failed the vibration resistance test and this is due to high and moderate velocity transverse movement of the Junker lever. It should be noted that these failure tendencies results from transverse forces to the bolt and nut coupling. The upper line in Figure 3 indicates bolt coupled with specialized washers embedded with interlacing cam face and rough externals which creates stiffness between the bolt and nut, and the joint members.

This investigation found that this line is evident of the fact that the vibration initiated values of E_n in the case of this specialized washer could not decimate the binding force of the clamp load, but for the other type of bolts without the specialized vibration resistant washers, it was observed that clamp load failed within 9 minutes of vibration exposure under 2000 load cycles.

V. TEST PARAMETERS AND ANALYSIS

Further, in a another complimentary assessment for the determination of breaking forces for preload of an M12 bolt under Junker test, parameters were chosen based on the believe that the design parameters have the greatest resistant impact on bolt loosening. The test was conducted according to standard DIN 65151 on a M12 bolted joint. The clamping force required was generated by turning of the nut until the fastener set was preloaded to the desired load i.e. initial preload of 7,900 lbs and upon conversion to force value, the result indicated 35,140N. The fastener set was tested at a speed range of 750-2000 CPM. The amplitude of vibration was defined at ±0.1mm [11], [12], and failure point as the Junker vibrated was recorded at 28,112.7N. The following table resulted from this assessment:

Table 1: Bolt preload failure parameters

Parameters (M12 bolt)	Preload (in lbs)	Preload (in N)
Frequency (Hz)	10.0	10.0
Initial Preload*	7,900 Lbs.	35,140.9N
Amplitude	0.2mm	0.2mm ±0.1mm
Failure point*	6,320 lbs	28,112.7N
Amp. variability	±0.1mm	±0.1mm
Cycle runs (range)	750-2000 CPM	750-2000 CPM

VI. DISCUSSIONS AND COMMENTS

It has been stated earlier that vibration is the result of illogical operations giving rise to instability, thus making the bolt coupling or welded joint possible exit point for the generated energy that travels in wavelike form through metallic lattice of the material. In view of the foregoing, it is imperative to state that this energy conversion and transmission condition has a direct relativity to the efficiency of the machines.

The relevance of this test to this paper is that; during the complementary support evaluation at the Advanced Manufacturing Engineering Laboratory (AME-Lab) of the Niger Delta University, the transverse force impacted on the bolt coupling resulted vibration of the bolt and as could be seen on the computer monitor, scattered sinusoidal waveform was noticed which indicate that in addition to the perturbations of the system, certain optical transients with electromagnetic consequences accompanied the waves and can be isolated and measured using appropriate sensors, and metering devices.

A careful observation of the table above indicate the fact that in a typical case study, initial bolt preload of 7,900lbs (35,140.9N) was the force required to engage a bolted joint and satisfy an efficient machine operability conditions. Upon application of vibration and perturbation, resulting from earlier identified factors, generated energy is transported to the bolted joints as have been explained in this paper. Thus, within a measured time of less than 9 minutes, a failure point was reached at 6,320lbs (28,112.7N). This implies that within the boundary area defined by equation (4); the operator, E_n in equation (5) impacts a one-dimensional E_{pp} with quantifiable energy value m_{Epp} and mobility factor of, $a_{E\psi}$; and these are controlled by the phenomenal ***ScrÖdinger*** particle transport characterization, $E\psi$.

Further, it should be noted that, since $E\psi$ implies energy profiling and acceleration within the lattice space, and $\psi = \psi(x)$; then a continuous unstable operability condition due to increasing vibrations results continuous impact on a force build-up basis which varies based on selected process parameters. Thus, a point is reached, as in the case above, where a specific force value as defined in equation (5) where a resultant E_n cleaves the binding energy of the joint and further results joint loosening and graduated failure. The systemic consequence of this condition is the increasing illogical operation which in turn redefines the designed *process constants* by converting them into *process variables*; the consequential and cumulative effect being poor quality of finished products due to dimensional inconsistencies and economic waste.

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