International Journal of Latest Engineering Research and Applications (IJLERA) ISSN: 2455-7137

Volume – 01, Issue – 04, July – 2016, PP – 13-20

GEOMETRIC NON-PARAMETERIZATION CONSEQUENCES OF TANGENTIALLY DEFECTIVE CONVOLUTIONS IN MULTI-CUTTING TOOLING OPERATIONS

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Abstract: Machining operations set limits of dimensional accuracy, surface quality and surface integrity of work pieces. The ability of any machining system to set these limitations depends on, among others, thermal conductivity of the cutting tools and workpiece, static and dynamic stiffness and vibration of the machine system. In metal cutting, having a knowledge of the effect of asynchronous vibrational frequencies and cutting temperatures at those frequencies on the tool and workpiece is important as excess temperature gradient between workpiece and cutting tool damages the geometry of either or both of them. The simulation method adopted here utilizes the structural conditions of the spindle in terms of its ability to withstand varying imposed vibration signatures. Consequently, results of the simulations show that low stiffness gives unsatisfactory part deformation or relative displacement between parts, thus resulting machined components with low geometric conformations and compromised quality standards. Further investigations indicate that forced vibrations equal to or very close to the machine tool *eigen-value* of frequency lead to serious vibrations and hence bad surface finishing. Therefore, adequate damping procedure is necessary to alleviate or eliminate the effects of vibrations during cutting operations, thus reducing or eliminating the resulting energy convolutions that are impacted on the finished products.

Key terms:non-parameterization, convolutions, energy balance, energy envelope, sequestration, orientation errors, functional elements, operational variabilities and transients

Introduction

Geometric non-parameterizations of work pieces originate from tool malfunctioning in single and multi-cutting tooling operations. These deviations are a consequence of defective tooling operations. The errors such as deviation from straightness and squareness originate from each component that deviates in 6 degrees[1]. Instructively, a predictive modelling tool that sequestrated and optimized a defined *z-axis* component in a relatedly distinct space within a known Cartesian coordinate system has been developed [2].

This model was used to illustrate the importance of geometrical non-alignment of output signatures of cutting tools on finished products. It utilized the three dynamical translational positionsof***x, y, z*** and matches these with the three angular misalignments of $R_x(\alpha)$, $R_y(\beta)$, and $R_z(\gamma)$ within matrixial structures of the Eulerian angles.

The result is indicated in the pitch yaw and roll, which evoke positioning and orientation errors. As an assessment of quality conditions in manufacturing, measuring instruments such as laserinterferometer, electronic spirit levels andcapacitancegages can be used to decipher the magnitude of these errors[3] in addition to modern industrial position scanners. In this regard, the importance of integrating scanners, contact and non-contact sensors into the manufacturing process as a means of real-time quality assessment and definition has been reinforced in many studies[4].

Further, it has been observed that the contributory effects of operational *variabilities and transients* which are dynamic in nature, impose undefined structural conditions on cutting tools and this is more appreciated in multi-cutting machining operations[5]. It should be noted that a multi-cutting machine tool structure as in Fig.1[1]has a supporting frame base on which is connected moveable components and other functional parts. These functional parts or elements create serial or parallel kinematic structure as shown in Fig 1. However, it should be noted that the sequence of logical operations form the basis of applicability and operability of manufacturing designs.

As have been notably observed and more particularly the view that appropriate kinematic structures should form machine tool guide way systems[6], it is imperative to assert that in both cases, the difference of the kinematics is that while in the serial structure, each axis can be moved independently, two or more drives have to be controlled for a linear movement in parallel kinematic structure. The functional elements ofaccuracy contribute to the overall capability to fix and correctly orient the cutting tool and workpiece. Derivatively, geometric and kinematic accuracy in cutting tool positioning is determined to a large extent by its control system.

However, with the advent of the idea of geometric non-parameterization, this capability becomes a dependent factor of machine components vibrations and transients which are consequences of energy imbalance attributable to non-parametric convolutions resulting from defective tangential conditions of gearing forcing functions and impacted on the finished goods as signature indicators of the occasioned vibrations and transients. Thus, where effective system control is compromised in this identified way, poor and unacceptable product finishing results.

In view of the foregoing, this paper considered the issue of indexing errors along the lines of its position on geometric non-parameterization. Thus, indexing error results impact loading ranging from less severe to quite harsh error, depending on the degree and apportionment which has accordingly been averred to be the most unfavorable scenario[7]. Thus, the linear distribution of indexing of errors, which is a consequence of the accumulation of small consecutive indexing errors has been investigated and to indicate that indexing errors occur when there is either, a mismatch between the linear motion of the machine bed and the indexing rotation of the workpiece chuck or a deflections from the cutting forces[7].

Indexing error therefore, is that remarkable divergence from fixated conditions of the process constants and their generative variables defined by allowable limits [2]. The result of this condition, is in this paper referred to as tangential energy convolution. In order to avoid this tangential energy convolution caused by the defective gears, modification on the tooth profiles can be done, in which case, tip relief, root relief or a combination of both are all approved ways of modification. This paper aims to look into the consequences of defective cutting tool operations on machine system energy balancing condition, with reference to these variables.

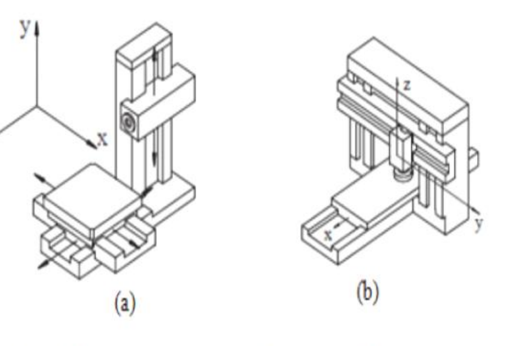

Fig.1. Kinematic scheme for a 3-axis (a) serial and (b) parallel kinematic structured machine tool

Geometric imbalance due to static stiffness.

In determination of this energy dissipation consequence due to geometrically non-parameterized gear-tooth profiles and subsequent resulting cutting operations, structured evaluation of multi-cutting condition is necessary. In this regard, deflection proclivities of multi-purpose cutting tools can also be encountered as a result of impacts from static loads which cause displacement and misalignments of guiding elements. Forces initiating or creating such loads are, for instance, the static part of the cutting force, weight of moving machine parts and weight of workpiece. Contact stiffness between components and structural stiffness of each single part, both highly determines stiffness of machine tool. Stiffness in the circumstance, is introduced to limit the possibilities of misalignments which characteristically results energy imbalance. Instructively, a joint has no stiffness until the connecting surfaces are in firm contact[3]; structurally, this is expressed as;

$$K_{(x)} = \begin{cases} 0 \; if \; \Delta x > 0 \\ k \; if \; \Delta x = 0 \end{cases} \qquad (1)$$

Where k is the stiffness factor with the defined constraint of $\Delta x > 0$ and $\Delta x = 0$ and Δx is the measure of its variability within acceptable limits of tolerance. It should be noted that a symmetrical condition exist, where equation (1) sets the limits of applicability on equation (2) to (16) such that the relevance of equation (1) applies to limit the extent of allowable variability of any components or parts and this is reflected on all the stages of equation (2) to equation (16).

Thus, low stiffness yields substandard geometric deformation or relative displacement between components, consequently, giving machined parts low geometric quality resulting critical energy imbalances. Thus, energy losses in form of heat, sounds, and other remote transients is relatively quantifiable and addresses the resultant reduction in performance of the system. As can be seen, Fig. 2 shows the effectiveness of high static stiffness[1]. The desired final thickness of the machined part is represented by $\boldsymbol{x}$, required chip thickness to be removed is represented by $\alpha_{max}/\alpha_{min}$, actual removed chip thickness due to static deflection is represented by $\alpha'_{\max}/\alpha'_{\min}$. it should be noted that this is different from $\alpha_{\max}/\alpha_{\min}$. The resulting workpiece thickness variation on the machined part is $\boldsymbol{x_{max}/x_{min}}$.

In order to find the deviation ratio resulting from misalignments occasioned by vibrations and related transients of failed gear tooth profiles, the following equations will be systematically analyzed with the view to establishing their process capability relevance in anticipation of an appropriate performance predictive measures and mechanism. Thus, a characteristic simulative approach to this mechanism could be used as the basis for *stiffness and variability assessments* of the spindle-chuck relativity. Hence, a simulation link connecting equation (2) to (16) below is indicative of a line upon line predictive sequence that gives the system the logical orientation required to impart the production standard and quality on the finished product. The implication of the foregoing is that any form of obstruction in the line upon line arrangement as stated above, structurally deforms the process capability of the production regime. This obstruction therefore represents the initiated vibrations, which directly impacts on the components and converts the energy loss into undefinable structural complexities on the surface of the workpiece. Thus, equation (2) to equation (16) characteristic sets the standard operational engine of components process behavior.

$$K_{static}=\frac{Fstatic}{x} \rightarrow static\ stiffness\left[\frac{N}{m}\right] \quad (2)$$

$$K_{cp}=\frac{Fstatic}{\alpha} \rightarrow process\ stiffness\ \left[\frac{N}{m}\right] \quad (3)$$

$$K_{cp}=K_c.\ f\,R \implies Fstatic = Kc.fR.\alpha \quad (4)$$

$$\Delta = \alpha_{max} - \alpha_{min} Original\ waviness$$

$$x + a_{max} = x_{max} + \alpha'_{max} \quad (5)$$

$$x + \alpha_{min} = x_{min} + \alpha'_{min} \quad (6)$$

$$\Delta = x_{max} - x_{min} + \alpha'_{max} - \alpha'_{min} \quad (7)$$

$$\Delta = \delta + \alpha'_{max} - \alpha'_{min} \quad (8)$$

Where δ is the final deviation, this gives:

$$F_{static}^{max} = K_{\text{cp}} \cdot \alpha'_{\max} \quad (9)$$

$$F_{static}^{min} = K_{\text{cp}} \cdot \alpha'_{\min} \quad (10)$$

$$F_{static}^{max} = K_{\text{static}} \cdot x_{\max} \quad (11)$$

$$F_{static}^{min} = K_{\text{static}} \cdot x_{\min} \quad (12)$$

$$K_{\text{cp}} \cdot \alpha'_{\max} = K_{\text{static}} \cdot x_{\max} \quad (13)$$

$$K_{\text{cp}} \cdot \alpha'_{\min} = K_{\text{static}} \cdot x_{\min} \quad (14)$$

$$K_{\text{cp}}(\alpha'_{\max} - \alpha'_{\min}) = K_{\text{static}}\,(x_{\max} - x_{\min}) \quad (15)$$

$$(\alpha'_{\max} - \alpha'_{\min}) = [K_{\text{static}} / K_{\text{cp}}]\ \delta$$

$$\Delta = \delta + [K_{\text{static}} / K_{\text{cp}}]\ \delta$$

$$\frac{\delta}{\Delta} = \frac{K_{cp}}{K_{cp} + K_{static}} \quad (16)$$

As stated earlier, the measure of stiffness of the *spindle-chuck coupling* define the extent of accuracy and assignable quality index. Thus, equation (16) shows, in terms of deviation over original waviness or variability, that increasing stiffness will result to a desired geometric accuracy on the workpiece. It should be noted that the result of the extreme limit of this stiffness is not within the evaluation of this paper.

Assessment of the Effectiveness of Dynamic Stiffness

This paper is in agreement with previous findings that dynamic stiffness is a property of vibrating mass, static stiffness and damping of machining system[7]. The capability of converting mechanical energy of a vibrating motion into other forms of energy such as heat is known as damping. Damping prevents or decreases the intensity of vibration in a mechanical system. When machining with a rotating cutting tool, the teeth repeatedly engage and disengage in the workpiece.

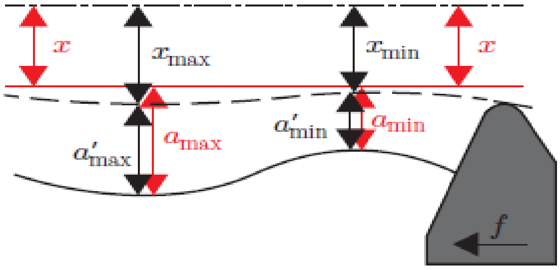

Fig. 2. Impact of static loading/cutting on a workpiece with varying nominal thickness

Hence, the kinetic energy of the impacting mass is converted into potential energy in the form of elastic deformation of the tooth. This continuous entering and exiting of the teeth creates impact forces with inadequate damping, causes tool vibration and as a result produces distorted surface finishing. In terms of energy balance, this deformation creates a window of energy exit which have been seen to be in forms of sound, heat, and many other transients.

Geometry imbalance due to vibrations in machining systems

One limiting factor for precision in mass production is vibration[2]. Some of the effects of vibrations in machining are excessive tool wear, undulated surface finishing and increased risk of tool chipping. Further, during machining, *free vibrations* occur due to disturbances in the processwhich are traceable to various degrees of impurity (inclusion in the workpiece or the breaking of built up edge from the tool, etc.)[7].

However, this vibration quickly decays back to stationary conditions. In this regard, forced vibrations also occurs intermittently in machining due to iterating entry and exit of the cutting teeth thus generating a fluctuating force that sets the cutting tool or work in motion. Since unstable machining is one of the major limitations for high speed machining which consequently shortens the tool life and hampers the surface quality required to be produced; measures must be adequate to take care of machining conditions that can likely result losses in geometricity.

As a result of the energy convolution occasioned by tooth profile, a simulation experiment of the spindle was conducted and this indicated consequent impact of energy imbalance from defective tooth profiles, thus, an identification of different energy flow modes of vibration of spindle imparted on tool during machining process was been carried out using simulations based on Solidworks software. The result is shown in Table 1 and graphical illustrations from figures 4 to 7.

Table 1. Vibrating frequency and Period of tool spindle.

Mode No	Frequency (Rad/Sec)	Frequency (Hertz)	Period (Seconds)
1	14995	2366.5	0.00041902
2	14997	2386.8	0.00041897
3	35522	5653.6	0.00017688
4	48109	7656.8	0.0001306
5	48115	7657.8	0.00013059

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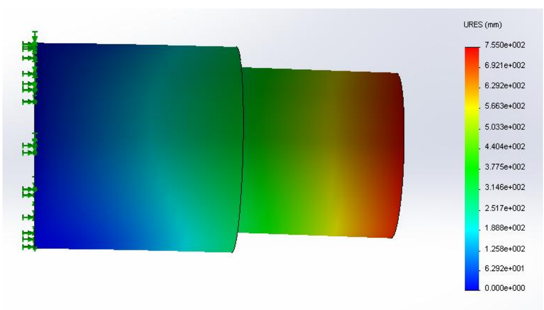

Fig 3.Shape and deflection of cutting tool in the 1st mode

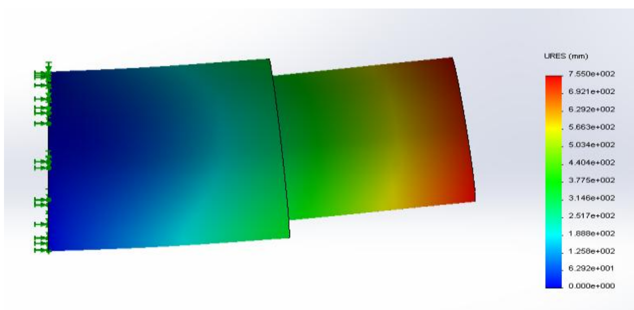

Fig. 4.Shape and deflection of cutting tool spindle in the 2nd mode

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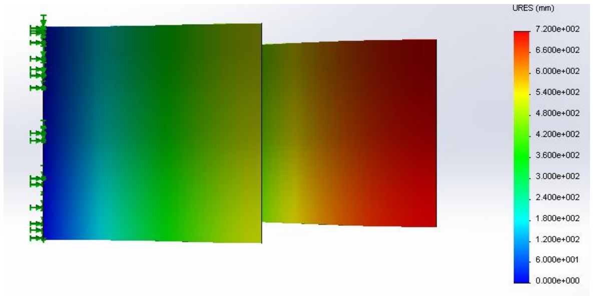


Fig. 5. Shape and deflection of cutting tool spindle in the 3rd mode

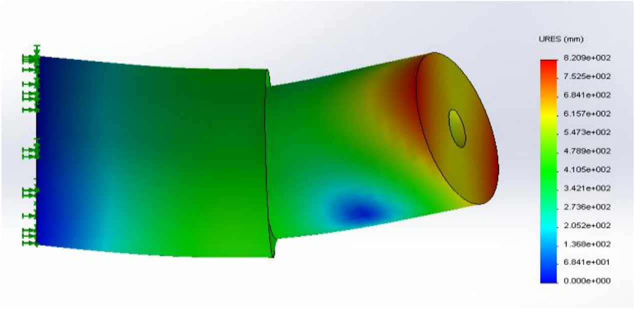


Fig. 6. Shape and deflection of cutting tool spindle in the

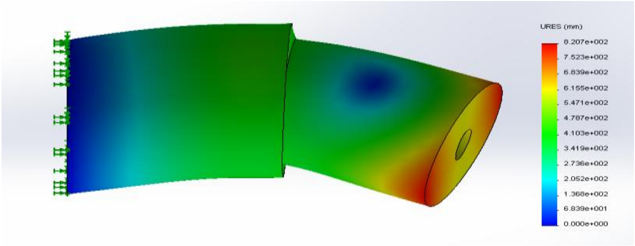
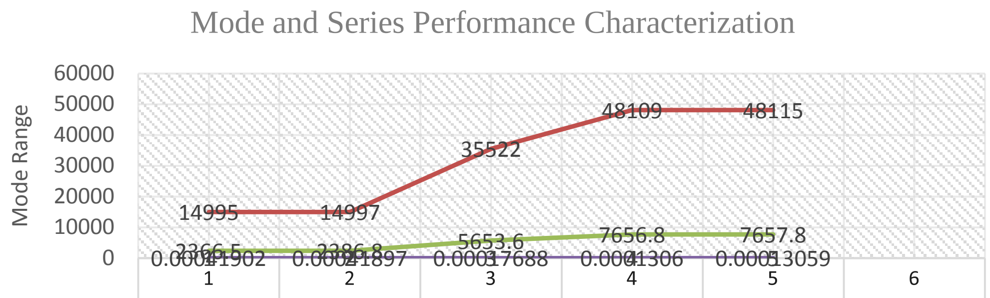


Fig 7 shape and deflection of cutting tool spindle in the 5thmode



	1	2	3	4	5	6
Series1	1	2	3	4	5	
Series2	14995	14997	35522	48109	48115	
Series3	2366.5	2386.8	5653.6	7656.8	7657.8	
Series4	0.00041902	0.00041897	0.00017688	0.0001306	0.00013059	

Machine operational series in amplitude and time

Fig. 8. Four points plus time performance characterization

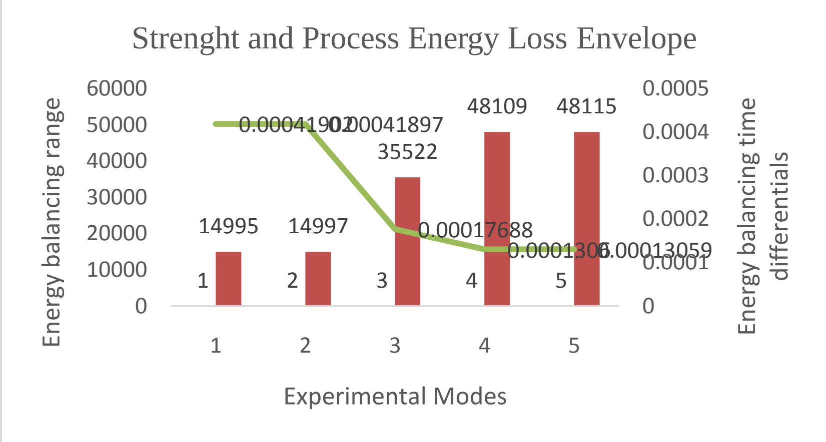


Fig. 9. Process energy balancing and loss envelope

Discussions

Dynamics of Mode 1 Operational Ratios

It should be noted that in Fig 3 the spindle orientation condition equilibrated between amplitude ration x/y_{static} and frequency ratio ω/ω_n in such a characteristic manner as to create spindle operational stability conditions. Thus Fig 3 indicates equilibrium vibration displacements resulting from stable frequency ratio values and amplitude ratio values. In this regard, the forcing function of component energy input operates at a reasonable appropriate damping conditions, and in such a situation, spindle *assymetricity* is evenly horizontal, indicating compliance with dimensional accuracies.

Dynamics of Mode 2 Operational Ratios

In mode 2 (as shown in Table 1) vibrating frequency increased by 2 rad/sec (i.e. 14997 rad/rec) resulting a corresponding increment in vibrating frequency by 20.3Hz. The essence of this change is to indicate that spindle orientation losses momentary assymetricity and this occurred within (0.000005 sec) $5x10^{-6}$ sec, implying that an increase in spindle velocity causes a corresponding increment in vibration frequency and as such, attained new amplitude in $5x10^{-6}$ sec. Thus, by the simulated sequence of Fig 4, assymetricity orientation occurs in the direction of +ve y-axis indicating a significant loss of dimensional accuracy on the finished product.

Dynamics of Mode 3 Operational Ratios

Mode 3 indicates an astronomical increase in spindle velocity thus resulting vibration damping conditions and standard asymmetric orientations as could be seen in Fig 5. Thus, mode 3 has similar structural orientation as mode 1 implying equilibrium conditions in the amplitude ratio values and the frequency ratio values. Thus, operational relationships between x/y_{static} and ω/ω_nin mode 1 and 3 is a unitary value of 6.33**:**6.28 i.e. implying that an approximation of 6.3: 6.3 i.e. 1:1 is applicable in the circumstance. Structurally, this means that spindle orientation at mode 1 is the same as mode 3, and maintains a steady characteristic at both modes.

Dynamics of Modes 4 and 5 Operational Ratios

The 4^{th} and 5^{th} mode deflections in spindle orientation indicate same characteristics but in different directions. However, mode 4 and 5 simulations defines the same orientation but in different directions, thus maintaining a structural component of maximum +ve and –ve amplitudes at higher frequencies achieved within the same time range for both modes. The impact of this alternative amplitude frequency condition of the dual-mode is a *steady surface derogatory pattern* of $1x10^{-7}$sec interval at high velocity and frequency machining (i.e. high speed machining) conditions. Thus, surface finishing under these machining regimes resulted poor dimensional accuracies which in turn negatively impacted on product quality. The foregoing implies that where a machine system has a high internal damping ratio, there will be a reducing sequence in the vibrational amplitude.

Incidental contributory conditions and findings

1. Spindle Rigidity

As could be seen in Fig 8, differences in ranges of the mode implies variations in rigidity profiling and its stability is essential in maintaining dimensional accuracy of the machinedsurface.It should be noted that under vibration conditions, the tool shifts into or out of the cut planes of the material, this repetitive action impliesaccumulation of staticdeflections and results *slack fits* and *joints differentials*. In this regard, the essence of rigidity is the maintenance of surfacefinishintegrity and quality, avoiding even the microscopic marks, void linesand crack initiators made by elastic vibrations and free play of loose fits andbacklashes.

In the control of vibration, rigidity of the part and cutting tool can make the difference between success and failure of the machining operation. Thus, increasing mass reduces vibration amplitude and resonant frequency, while dampening reduces amplitude by dissipating vibratory energy as frictional heat, as the green color trend in Fig. 8 indicates. Sinceeach part of the cutting system (i.e. the machine, the fixture, the tool and the workpiece) can affect the mode and amount of vibration, most should be madeoversize and firmly supported.

2. Component Strength and Fatigue Profiling

It should be noted that in Fig. 9 a rise in component machining efficiency resulted a decrease in energy loss. Thus, the strength of each member can be considered separately and related to the magnitude and application of the forces it will transmit, if this strength is not a critical factor of the design process, then it becomes a measure of failure, as the green line indicates, a decrease in energy balancing with drop in time. Thus, aggregate component strengthsshould clearly besufficient to prevent breakage or deformation beyond the elastic limit when the operation is performed correctly. This implies that the designer must also consider overloads anddamage that may be encountered, providing abundant strength wherever economically possible. In this regard, Fig 6 and Fig 7 i.e. 4^{th} and 5^{th} modes depicts material fatigue profiling conditions resulting the corresponding rise in energy balance of Fig. 9.

Conclusion

During metal cutting operation, vibration is encountered which characteristically redefines the multi-tooth cutting conditions by introducing parametric imbalances resulting energy convolutions, this condition thus imply also that a considerable quantity of the machine energy is transferred into heat through plastic deformation of the workpiece surface, the friction of the chip on the tool face and the friction between the tool and the work piece. The possible consequences of vibrational frictions and cutting temperatures on the produced surface are dimensional inaccuracies due to thermal distortion and expansionthat results contraction during and after cutting.

When this occurs, surface damage by oxidation, rapid corrosion, burning and induction of tensile residual stresses, includingmicro cracks are found at the surface and subsurface regions[8].Forced vibrations equivalent to a *machine tool eigen-frequency*thus lead to critical vibrations and surface deterioration of workpiece geometry[9]. In order to minimize these forced vibrations and their effects, the paper thus finds and suggests that the number of the teeth in operation should be raised concurrently or improve the damping strength. Since significant damping is very effective in reducing vibrations at resonance frequencies, improved damping is recommended to lower forced vibrations.

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