Design and performance analysis of Multi-linear Horn for Ultrasonic machining application using FEM approach

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Abstract

This work reports an ultrasonic horn design and development for improving the performance of ultrasonic machining process in terms of better material removal rate for the same input conditions. The suitable operating frequency range (20 kHz-25 kHz) of the horn is analyzed with the help of finite element approach using the Ansys and for modelling Creo software is used. Typically, the machining tool vibrates around 10-20 μm, but for effective machining requires the more magnitude or intensity (80-200 μm) of vibration within this frequency range. Performance of ultrasonic horn mainly depends on the geometrical aspects and type of material used. In this work, we focus on the ultrasonic horn with an innovative design compared to earlier reported works to obtain the maximum magnification factor within permissible stress limits. The standard criteria “stresses below the permissible limit and mode shapes at different resonance frequencies (modal analysis)” are considered while designing the horn for avoiding the tool and horn failures during machining. The optimum numbers of elements are identified to mitigate computational time based on the optimum mesh criteria. In this study, we have analysed the equivalent stresses, magnification factor (ratio of output to input amplitude) of the horn by harmonic response analysis, which can be used to arrive at the effective machining processes. Finally, the magnification factor is increased within the safe stress limit using our innovative horn design. Newly, designed horn’s performance is compared and validated with the earlier reported literature at a similar input parameters or loading conditions.

Keywords: *Ultrasonic Machining (USM), Multi-linear stepped Horn/Sonotrode, FEA, Horn design, Magnification/Amplification Factor.*

1. Introduction

Ultrasonic machining (USM) is a non-conventional machining process, which is used for machining of materials that has high strength and brittleness. As ever increase in technological trends that demand high strength materials and machining of such hard and brittle materials with conventional machine tools is very difficult. Compared to workpiece the tool material used is soft usually made up of mild steel, copper etc. which vibrates at around 20-30 kHz with amplitude of 15-100 μm. For machining along with tool abrasives are also used in the form of slurry and used during the machining which produces the indentation on the workpiece due to simultaneous tool vibration leading to brittle fracture of the workpiece removing the material. It has low material removal rate. Workpiece need not to be electrically conductive. Ultrasonic machine composed of many components namely transducer, horn, tool, table among this horn or sonotrode or acoustic wave guide is the most important element of USM as it has dominant contribution in the machine’s effective and economic performance [1]. The ultrasonic vibrational energy is utilized in two approaches the first approach, called as an ultrasonic machining, is based on abrasive principle of material removal, the tool is shaped in the exact configuration to be ground in workpiece and it is attached to a vibrating horn. The second approach is based on the conventional machining technologies ultrasonic assisted machining [3]. At transducer end the amplitude is small 10-15 μm which is not sufficient for effective machining, hence to overcome this inadequacy the horn is used to amplify the amplitude at the output or tool end in order to perform proper machining operation [5]. Generally horn is attached to the transducer using threaded studs. The amplification or magnification of vibration energy depends on the shape of the horn. The vibration energy remains constant throughout the horn length which indicates that with the decrease in the cross-section, energy over unit area i.e energy density increases [1]. Hence horn shape is usually narrowed down towards the end, hence large amplitude can be obtained at the tool side end this is the working principle of horn. Several standard horn geometries are also worked out by previous researchers including cylindrical, stepped, conical, exponential horn etc.

In the present work the horn is designed using multi-linear steps which are modification of stepped horn using the finite element method (FEM).

1.1 Traditional Horn Contours

All the traditionally adopted horns have certain advantages also accompanied with some limitations.

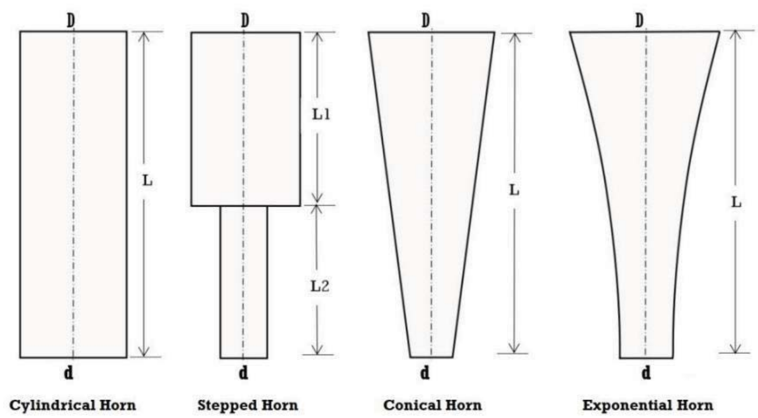

FIG. 1 *Traditional Horn Contours*

Stepped horn has highest magnification factor (M.F) among all the horns, but this merit is outweighed by its stress levels much above safe stress limit. Hence stepped horns are preferred when the amplitude requirements are low on other hand conical and exponential horns have comparatively lower stresses but this advantage is nullified by the lower magnification levels. Cylindrical horn gives no amplification. Also the exponential horn however requires complex setups or CNC machines for its manufacturing which may not be available [4]. Hence there seems to be scope of improvement in simple stepped horn if we manage to overcome these higher stresses which direct our attention towards designing the horn profile close to the stepped horn and that too involving linear profile.

2. Design of ultrasonic Horn

The governing differential equation of horn is developed for evaluating the equilibrium of the forces (Elastic forces and inertial forces) on the infinitesimal length of the horn's profile as shown in Fig.(2)

x = Axial position of horn

$\sigma_x = \sigma(\mathrm{x})$ = Stress at section at distance x (from transducer end)= E× $\frac{\partial u_x}{\partial x}$

$u = u(x,t)$ = Displacement of section at distance x

$\mathrm{A}_x = \mathrm{A}(x)$ = Shape function governing area at any section at distance x

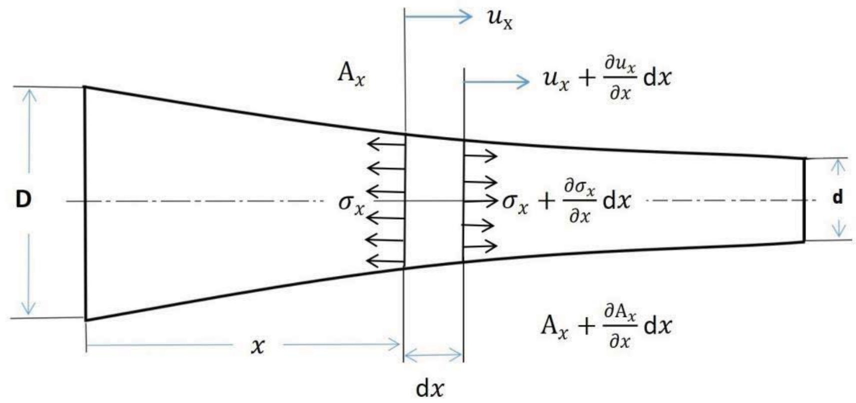

FIG. 2 *Free Body Diagram of Horn*

If there exist any unbalanced force that will produce acceleration,

This unbalanced force (Elastic Force) between two sections at dx distance apart is,

$$\mathrm{F}=(\sigma_x + \frac{\partial \sigma_x}{\partial x}\,\mathrm{d}x\,).(\mathrm{A}_x + \frac{\partial \mathrm{A}_x}{\partial x}\,\mathrm{d}x) - (\sigma_x).(\mathrm{A}_x)$$

$$= \sigma_x.\frac{\partial \mathrm{A}_x}{\partial x}\,\mathrm{d}x + \mathrm{A}_x.\,\frac{\partial \sigma_x}{\partial x}\,\mathrm{d}x$$

And total inertia force = Mass ×Acceleration

$$= \rho \times \mathrm{A}\times\mathrm{dx}\times\partial^2\mathrm{u}/\partial\mathrm{t}^2$$

Where, ρ = density of Horn material.

E = Modulus of Elasticity of horn material.

$\partial^2\mathrm{u}/\partial\mathrm{t}^2$ = Acceleration of horn = f(x,t)

Using Newtons law, equating the Total elastic force and Inertia forces and putting above values we get on solving,

$$\frac{\partial^2 u}{\partial t^2} + \frac{\partial ln(\mathrm{A}_x)}{\partial x} \times \frac{\partial \mathrm{u}_x}{\partial x} + \frac{\omega^2}{\mathrm{C}^2}\,u = 0$$ .. Governing Differential Equation of Horn

Where, $\frac{\partial \mathrm{u}_x}{\partial x}$ = Strain = f(x,t).

C = Acoustic speed in Horn material

And is given by $\mathrm{C} = \sqrt{\frac{E}{\rho}}$

Above differential equation can be solved for specified horn shapes and longitudinal (area) decaying function which involves unknown constants that can be found by using boundary conditions as,

$$\left(\frac{\partial u_x}{\partial x}\right)_{\text{at } x=0} = 0; \qquad \left(\frac{\partial u_x}{\partial x}\right)_{\text{at } x=L} = 0.$$

and u_x at x=0 = u_o Initial excitation displacement

The length of the horn is found by solving the above governing differential equation for specific horn and boundary condition. This horn length is called resonant length of horn and this depends on the working frequency and has no effect on magnification factor [4], and this resonant length of the horn is half the wavelength [7]. The resonant length calculation of conical horn is much difficult and also it does not affect the magnification factor. Hence we chose the horn length as 120 mm, which is close to length of horn used in available literature [4]. And also performance of conical horn is almost same as exponential horn but having added advantage of ease of its manufacturing.

2.1 Design data

While designing the horn, there exist some constraints i.e the input and output diameters (in case of circular cross section) of horn are restricted by the transducer and tool dimensions. The attachment used to hold the horn to the ultrasonic transducer should not be of greater diameter than that of the horn, otherwise the amplitude of the latter will be damped [4]. USM is considered to have transducer diameter of 40 mm and that of output or tool side diameter around 10 mm. These bounds are specified within which it is needed to design the horn profile.

The horn design should not only be responsible for improved performance of ultrasonic machining, at the same time it should be much easier from manufacturing perspective. As the complexities in the horn profile is always accompanied with the requirement of expensive production equipment which increases the cost of horn. The stepped horn profile shows highest magnification factor of 16, so it can be observed that horn must be closer to stepped horn along with linear profile. Hence, the stepped horn with linear profile is introduced in this work. The performance of horn is verified using magnification factor (ζ) within safe stress levels.

$$\zeta = \left|\frac{A_f}{A_0}\right| \qquad \text{…. (1)}$$

Where,

A_f = Amplitude obtained at the horn end

A_0 = Amplitude at input end (Input excitation displacement, 15 μm).

The horn is assumed to be free from tool attachment and it does not affect the horn's natural frequencies [5]. The operating frequency of ultrasonic machining (USM) to be maintained near the machine's natural frequency 23.5 kHz. This natural frequency is an important consideration in the design of horn profile. The magnification factor can be increased, when USM horn operates at machine's natural frequency that leads to better material removal rate. In this horn design studies, displacement 15 μm is applied at transducer end of horn.

2.2 Material Used

Ultrasonic machining can be done at higher frequency (>20 kHz) and hence the horn is subjected to cyclic loading due to horn vibration. The horn material should have better fatigue resistance and high endurance strength. Many such materials are used like Titanium alloys, AISI 4140 alloy steel, cold rolled steel, Al alloys etc., to manufacture the horn [1,2,4]. In this work Titanium grade-5 alloy is used which is also termed as Ti-6Al-4V. The properties of Ti-6Al-4V are shown in the table below:

Table 1 *Properties of Titanium Grade-5 alloy*

Property	Value
Density, ρ	4429 Kg/m^3
Modulus of Elasticity, E	113 GPa
Poisson's ratio, μ	0.345
Endurance strength, σ_e	529 - 566 MPa

The stresses generated in horn material should not exceed its material's endurance strength. But there are several factors involved in modifying this strength i.e surface finish factor, size factor, temperature, reliability factor, load factor etc. Account all these factors to identify a single equivalent factor 'k' and its value is found to be 0.75 according to Marin's equation [1]. Hence allowable endurance strength of horn will be $(\sigma e)_{allowable} = k \times \sigma e = 397$ MPa.

3. Finite element modelling (FEM) of horn

Designing the multi-linear horn profile is quite easy, however its analytical calculations are little difficult. So, the numerical approximation based FEM tool is used to meet the requirement. Analysis is carried out in three stages; a) pre-processing that involves defining the inputs, boundary conditions or constraints, b) solution and c) post-processing in which the results are analysed. MATLAB is used for visualizing the results in graphical form. The horn under investigation is discretized into several numbers of elements and nodes. The CAD model of horn has been designed using PTC Creo parametric 3.0. ANSYS Workbench R2 package used for computing the mode shapes and natural frequencies of the designed horn profile using ANSYS' modal analysis. This information helped to predict the free vibration response of horn at corresponding natural frequencies. Then, displacements (amplitudes) and stress variations along horn profile were observed at 23.65 kHz with help of harmonic analysis. Stepped horn shows the maximum magnification factor of 16 and stress level was greatly exceeding the allowable limit due to sudden change in geometry [4], that has been verified. In this work, our aim is to minimize the stress level simultaneously to obtain the appropriate magnification factor, that should be more than the existing horns' design (conical, exponential horns [2]). The suitable length of each portion of the horn was studied by trial and error method for finding the appropriate lengths. Finally it is found that each portion of horn must be of equal lengths for improving magnification factor of USM horn with in the safe stress level as shown in Fig. 3. A linear profile is introduced in the vicinity of step of stepped horn for designing the multi-linear horn.

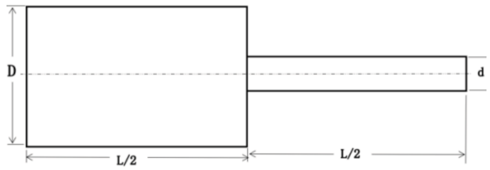

FIG. 3 Case 1

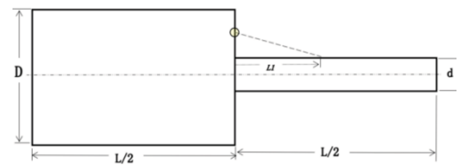

FIG. 4 Case 2

In the portion (of smaller diameter d) length L_1 is varied along the horn profile as shown in Fig. 4 and all these models verified for magnification factor and it is observed at 37 mm (conical part) shows better results within permissible stress limits.

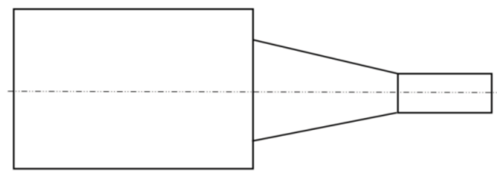

***FIG. 5** Proposed horn design*

4. Result and Analysis

4.1 Modal Analysis

4.1.1 Convergence study based on number of elements

The results obtained using FEM, that are mainly depends upon the number of elements used to discretize the horn geometry. The convergence effect was observed by trial and error method, when the horn profile discretized into 15000 elements or more as shown in Fig. 6 below

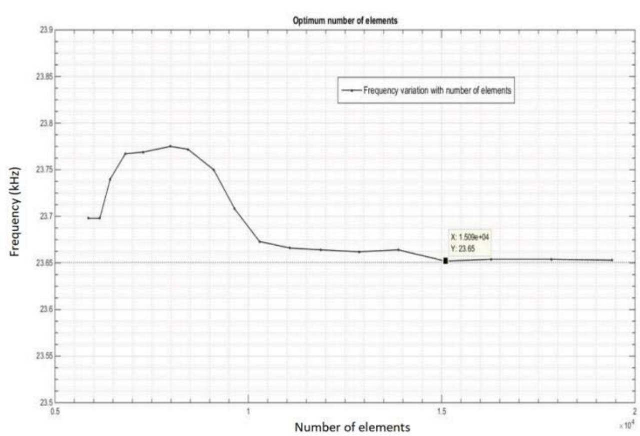

***FIG. 6** Optimum number of elements for steady results*

The obtained horn's natural frequency "23651 Hz" is close to machine's natural frequency, while discretzing the horn geometry into 19403 elements. Typically, many authors reported that ultrasonic machining has natural frequency of 23.5 kHz [1,2,4]. Here, modal analysis and harmonic analysis were carried out using tetrahedral (SOLID187) element. Modal analysis performed to identify the different natural frequencies and for finding the mode shapes of the horn profile at these frequencies and harmonic analysis for obtaining magnification factor and stresses. The horn's performance could be improved such as increasing the magnification factor (within permissible stress limit) based on these parameters.

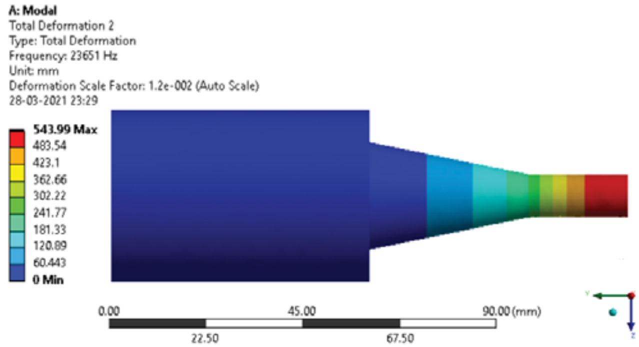

FIG. 7 First mode shape obtained at 23651 Hz

The frequency range 20-25 kHz was given as input in this model, and then three different modes could be observed within this frequency range. Finally, horn's resonant frequency "23651 Hz" is observed which is close to machine's natural frequency and corresponding horn's behaviour is shown in Fig.7.

4.2 Harmonic Analysis

In this harmonic analysis, input frequency 23651 Hz and displacement 15 μm are provided to horn, where it is attached to transducer. Based on this study the magnification factor was found with respect to horn displacement's amplitude. The same study was used to verify the stress distribution at corresponding horn profile. The ratio between the the displacement amplitude of horn end (smaller diameter) and input displacement (15 μm) is considered for finding magnification factor (refer Equation 1). The magnification factor (displacement) variation was obtained along the length of the horn profile as shown in Fig. 8 and 9, the maximum magnification factor obtained is 5.23 and corresponding equivalent (Von-Mises) stress 175.65 MPa was observed and indicated in Fig.10

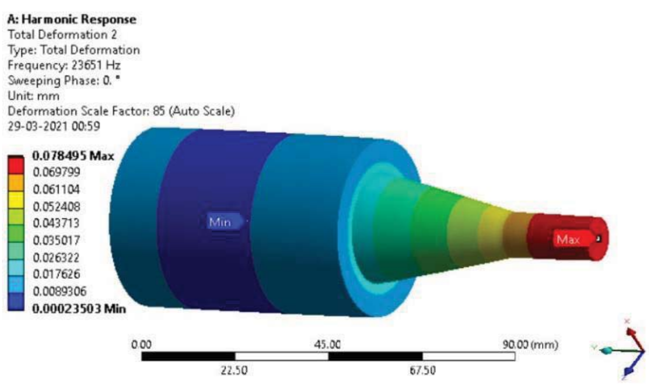

FIG. 8 Amplitude Variation Along the horn length

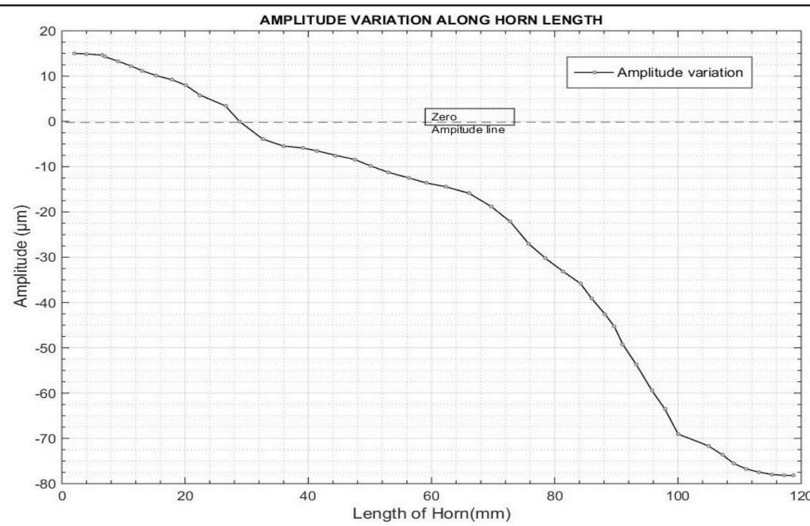

FIG. 9 Graphical form of amplitude Variation

Hence the obtained magnification factor 5.23 which is higher than the existing conical and exponential horns and also obtained stress which is quite lower than those horns.

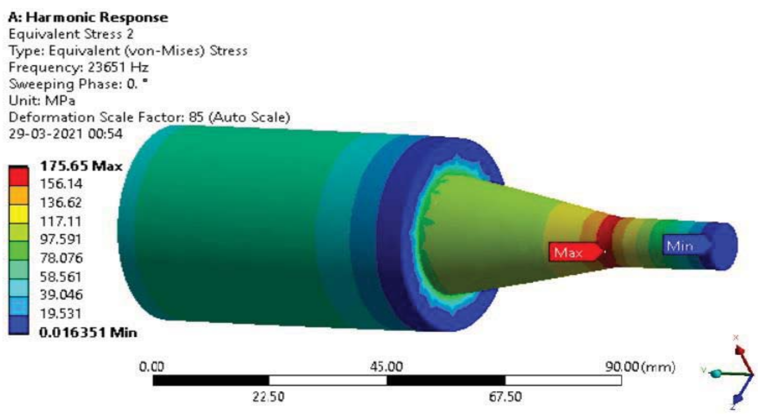

FIG. 10 Stress (Equivalent Von-mises) Distribution

5. Validation of Results

Table 2 Result Validation

Horn Profile	Stepped	Conical	Exponential	Multi-linear Horn
Length (mm)	100.5	124.7	116.4	120
Diameter ratio (D/d)	4	4	4	4
Magnification factor obtained	16	3.39	3.94	5.23
Stresses (MPa)	Much above the permissible limit	188	220	175.65

6. Conclusion

The horn is designed using the FEM approach. The suitable length of each portion of the horn was studied by trial and error method for finding the appropriate lengths. The maximum magnification factor of 5.23 was obtained compared to the earlier reported horn design. The maximum stress 175.65 MPa was observed which much below the allowable stress limit. This horn design shows the better results compared to stepped horn, conical and exponential horns. The maximum stress (critical stress) was induced at near initial horn's step, while using simple stepped horn. The maximum stress is shifted towards the end and got reduced when the multi-linear horn design was introduced. This study has been restricted to single step, even though obtained the maximum M.F within the allowable stress limit. The convergence study is reported to identify optimum number of elements above which the horn's natural frequency matches closely with machine's frequency. By introducing even more number of multi-linear steps, the magnification factor might be increased significantly with slight increment of stress level as well without exceeding its permissible limit. This study will be addressed in future.

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