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The Modal and Harmonic Analysis of a Cylindrical Horn Designed for an Airborne Ultrasonic Dryer

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ABSTRACT

The extensive radiating surface of horn is an imperative part for amplifying the ultrasonic airborne waves generated for drying of foodstuffs. However, the radiation of high-intensity waves may lead to frequency shift, interaction of modes and reduced efficiency of ultrasonic system. Furthermore, the excitations may also be harmful to the transducer, horn and the joints. In this study, a finite element model is developed for the dynamic analysis of a cylindrical horn. In the modal analysis, the natural frequency of horn is found to be 19,498 Hz which is very close to the working frequency of the ultrasonic generator i.e., 20 kHz. At this frequency, the amplitude of non-beneficial longitudinal mode is negligible compared with that in main radial mode shape and there is no risk for interaction of modes. According to the harmonic analysis, the generated von Mises stresses in the horn are much lower than the endurance limit of constructing material and the horn can safely radiates the waves. The experimental data measured by a laser vibrometer are also used for validating the results of numerical simulation.

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INTRODUCTION

The convective hot-air drying of food materials is a main process which is widely used for preserving the beneficial characteristics and nutritional compounds. In a pure convective dryer, increasing the temperature of drying air is the only way for accelerating the drying process and reducing the required energy. However, the results of several studies showed that the drying process at higher air temperatures may induce deteriorative and undesirable changes in the products and should be avoided (see e.g., (Branisa et al., 2017; Khaloahmadi et al., 2023 Mahanom et al., 1999; Shi et al., 2017)). In recent years, finding safe and non-thermal drying methods was a great challenge for researchers. High-intense radiation of ultrasonic waves with 20 kHz frequency on the drying materials creates clean and non-contaminating action and results in higher quality products (Namjoo, Moradi, Dibagar, Taghvaei, et al., 2022; Namjoo, Moradi, & Niakousari, 2022). The fast cyclic contractions and expansions generated in this process, stimulates the removal of moisture content from the bulk to the skin surface of material (Huang et al., 2020; Namjoo, Moradi, Niakousari, et al., 2022).

The common basic components used in all ultrasonic system are a generator and a piezoelectric transducer. The ultrasonic generator is mainly a transformer that converts the supplied electrical power (220V, 50Hz) into an output voltage 800-1000 V and frequency of 20,000 Hz. The transducer has a sandwich structure with two masses as face sheets, and a set of piezoelectric layers in the core. The delivered electrical power to this component is directly converted into the acoustic waves with output amplitudes between 0.001 to 0.1 mm (Rao & Kalyankar, 2014; Wang et al., 2018).

In a combined convective/ultrasonic drying there is an air gap between the transducer and drying chamber and the generated high-intensity waves are indirectly radiated on the samples. Therefore, this process is commonly referred as airborne ultrasonic drying. It is noted that the air and other similar gaseous media have low specific acoustic impedance and absorb high amount of supplied ultrasonic power. Therefore, the generated acoustic waves for the indirect ultrasonic radiation should be amplified for more effective impedance matching between the air medium and the transducer. To meet these requirements, an extensive radiative surface known as acoustic horn is attach to the transducer for boosting the waves (see Figure 1). Depending on the field of application, the horns are constructed in different shapes and profiles including stepped-plates, grooved-plates and cylindrical configuration (Gallego-Juarez, 2010).

The cylindrical horns which are widely used in drying systems, permit for effective control on distribution of acoustic beams on the samples and provide a concentrated ultrasonic impact on the drying chamber (Asfaram et al., 2019; Ghanbarian et al., 2020; Namjoo, Moradi, Dibagar, Taghvaei, et al., 2022; Namjoo, Moradi, Niakousari, et al., 2022). In this way, a great improvement is achieved and the convective drying may be conducted at lower air temperatures. Therefore, the hybrid convective/ultrasound drying systems are highly recommended for drying of foods with heat-sensitive nutritional compounds (Namjoo, Moradi, Dibagar, & Niakousari, 2022). The main challenge in development of ultrasonic systems is proper design of components, especially the acoustic horn. The natural frequency of horn should have an appropriate value within a frequency range produced by the ultrasonic generator. Furthermore, the horn must provide a proper impedance matching with the air as the intermediate medium. It is noted that the horn directly receives the excitations of transducer and undergoes a variety of cyclic stresses and strains. Therefore, it is imperative to design the horn with proper load-carrying capacity.

In order to design the acoustic horns various analytical and numerical methods have been implemented in the literature (Huang et al., 2013; Rani & Rudramoorthy, 2013). The finite element method (FEM) is recognized as a robust tool with widespread use for evaluating the profiles and geometrical aspects in various considered applications. Based on FEM, Akbari et al. (Akbari et al., 2008) conducted a geometrical analysis for determining the convenient dimensions and contours of horns. The shape of conical horns was optimized by Lee et al. (Gang, 1991; Moon et al., 2003) and the presented design was validated with the experimental data. In a FEM model, Cardani and Locas (2002) examined the possible contribution of adding extra slots to the front face of block rectangular horns to the increase in amplitude of vibrations. Cretu (2005) studied the effect of change in the cross-sectional dimensions of horn in variation of harmonic frequency of vibrations. Nad (2010) evaluated the half-wave and full-wave modes of horns with different shapes and showed that for each specific application, convenient geometrical properties may be prescribed for the horns. Kumar et al. 2018 proposed an efficient FEM-based tool for design of simple and complex ultrasonic horns for stir welding process.

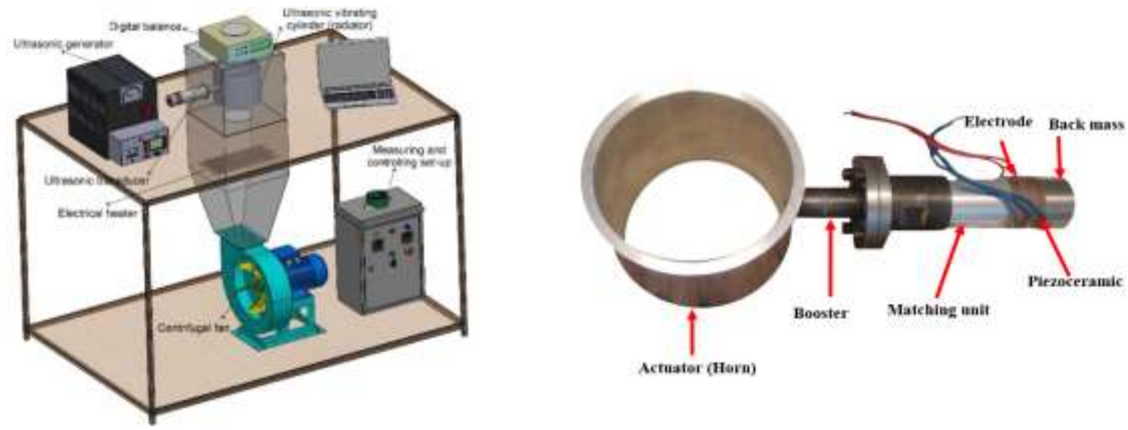
The ultrasonic systems have been successfully applied for welding, cutting, polishing and machining of light weight metallic alloys. However, the drying of food materials has been recognized as another field of application for the ultrasound technology. With some minor modifications in the dimensions of components, the available ultrasonic systems were later used in the hybrid drying systems. However, the ultrasonic dryers operate at certain range of frequencies that may be different from those in the other engineering applications. Therefore, for proper tuning the natural frequency of oscillating components and avoiding the harmful interaction of modes shapes, an exclusive vibrational analysis seems to be required. To the best knowledge of authors, no study has been specifically developed for the application of ultrasound in the field of foods drying. To fill this gap, a convenient numerical model is prepared by the commercial SolidWorks 2022 as a powerful FEM-based tool for modal and harmonic investigations, and finding the optimized configuration of the acoustic cylindrical horns. The results are used for manufacturing a main sample which was experimentally investigated by a laser vibrometer for estimating the validity of the proposed design.

MATERIALS AND METHODS

Horn design

The acoustic horn is a main part in the airborne ultrasonic system which amplifies the mechanical vibrations generated by the transducer, and imposes an imperative contributing effect on the drying process. The insufficient amplitude transformation by the horn may reduce the amplitude of waves generated by the piezoelectric transducer and diminished the performance of drying system (Seah et al., 1993). In this regard, the geometrical dimensions of horn should be properly determined for tuning its resonance frequency within the range of frequencies produced by the ultrasonic generator. It is noted that the majority of ultrasonic drying systems radiate the waves with frequencies around 20 kHz. Therefore, the dimensional design of components should provide a proper performance for this range of frequencies. Garcia-Perez et al. (2006) developed a cylindrical horn with 320 mm and internal dimension 100 mm, while the dimensions of the horn used by Bantle et al. (2011) was 200 and 300 mm in internal diameter and height, respectively. Both the systems were reported to have satisfactory performance in drying of orange peel (Garcia-Perez et al., 2012), strawberries (Gamboa-Santos et al., 2014), persimmon (Cárcel et al., 2007) , clipfish (Bantle & Hanssler, 2013) and peas (Bantle & Eikevik, 2011).

In all of these works, the main attention was focused to tune one of natural frequencies of cylindrical horn on the vicinity of main working frequency of the system. However, it is noted that the total mode shape corresponding to each natural frequency has a tendency to shake the cylindrical body in both longitudinal and radial directions, but only the radial mode shape is desired and should be increased in amplifying process. The other mode shape (i.e., longitudinal) has no effect on the samples placed inside the drying chamber (see Figure 1) and should be minimized for reducing the risk of harmful interaction of mode shapes. In the present study theses important matters are also included a more convenient geometrical design for the acoustic drying horn.



(a) (b)

Figure 1. (a). A schematic view of the developed ultrasound-assisted air dryer and (b). Real photo of cylindrical actuator and piezoelectric sandwich ultrasonic transducers.

In addition to the proper geometrical design, suitable material should also be chosen for manufacturing of acoustic horns. The selection of material is intensely dependent on the purpose of application. For high-stress applications such as friction stir welding (FSW), hardened steel or titanium are used (Kumar et al., 2018), while for low-stress sonication used in drying process, aluminum may be efficiently used for avoiding the acoustic loss and fatigue failure (Nad', 2010). In this study, the aluminum alloy 7075-T6 with the mechanical properties given in Table 1, is selected as the constructing material.

The conventional analytical methods for design of horns leads to a formidable mathematical procedure and may not be used for the complex horns. Fortunately, the FEM approach can easily include the configuration, material and boundary conditions and therefore has been widely used for the design of acoustic horns (Chhabra et al., 2016; El-Hofy, 2018; Kumar, 2016; Kumar et al., 2018). Furthermore, the fluctuation of displacements and the working stresses during the service time can also be predicted by the FEM approach (Rani & Rudramoorthy, 2013).

Table 1. Properties of horn Material used for simulation

Property	Value	Unite
Elastic Modulus	7.2e+10	N/m^2
Poisson's Ratio	0.33	N/A
Shear Modulus	2.69e+10	N/m^2
Mass Density	2810	kg/m^3
Tensile Strength	570000000	N/m^2
Compressive Strength		N/m^2
Yield Strength	505000000	N/m^2
Thermal Expansion Coefficient	2.36e-05	/K
Thermal Conductivity	130	W/(m·K)
Specific Heat	960	J/(kg·K)
Material Damping Ratio	N/A	

Based on the discussed considerations, the geometrical dimensions of the horn used by Namjoo et al. (2022) were modified and new values shown in Figure 2, were selected for the configuration of the horn. Then the FEM-based vibration analysis was used to evaluate the effectiveness of proposed design.

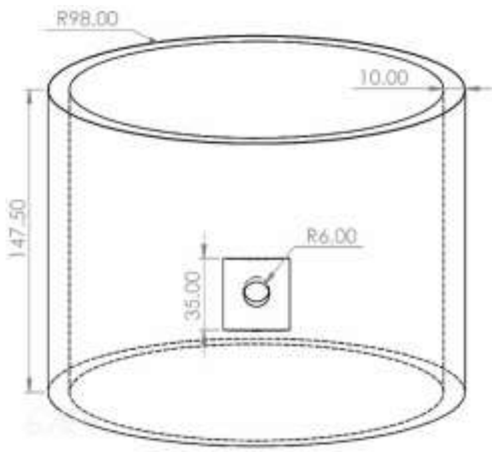


Figure 2. The geometry and configuration of the designed acoustic horn.

RESULTS AND DISCUSSION

The Modal analysis of horn

The modal analysis tries to evaluate the eigenvalues and eigenvectors of the structural element. In special case of an ultrasonic horn, the eigenvalues are known as natural frequencies and the eigenvectors are the mode shapes. It is noted that any arbitrary shaking form of horn comprises of various mode shapes determined within the frequency range of system. So, the modal analysis is an essential step for predicting the dynamic response of horn, especially when a spectrum of frequencies or combined harmonic loads is applied to the system. The modal displacements of the vibrating horn may take place in radial direction, longitudinal direction or combination of both. Thus, for achieving acceptable and accurate predictions high enough number of nodes and elements should be employed for meshing the domain of problem. However, the excess use of nodes just increases the computational cost and may not provide sufficient contribution to the numerical analysis.

In order to find the optimum number of nodes and elements, nine different meshes with various densities of nodes are used for discretization of domain. The detail of each mesh is given in Table 2. As can be seen in Figure 3, by increasing the number of nodes the completion time of analysis drastically increases but for the nodes more than 54,555 the results conveniently converge to the final value and there is no need to use extra density of nodes. For better understanding the effect of mesh density on the accuracy, the variation of three natural frequencies with different discretizing meshes is illustrated in Figure 3. It is evident that the sparse meshes could not provide accurate results, but by using 32,338 elements the results properly converge to the final asymptotic values. One of three values obtained for the natural frequency is 19,498 Hz. This value is close to the frequency of ultrasonic wave generator (20,000 Hz) and it can be concluded that the dimensions in Figure 2 are properly selected for the horn.

Table 2. The various meshes used for the convergence study

	1	2	3	4	5	6	7	8	9
Max Element Size (mm)	19.02	16.40	13.79	10.94	8.92	7.61	6.18	5.35	4.76
Min Element Size (mm)	6.34	5.46	4.60	3.65	2.97	2.54	2.06	1.78	1.59
Total nodes	3834	4918	6789	10704	23366	33870	54555	75606	119160
Total elements	1826	2361	3284	5270	13446	19969	32338	45399	75331
Maximum Aspect Ratio	12.07	10.67	8.83	8.28	9.48	7.31	6.95	6.73	6.70
Mesh completion time (sec)	12	18	50	93	123	157	254	303	360

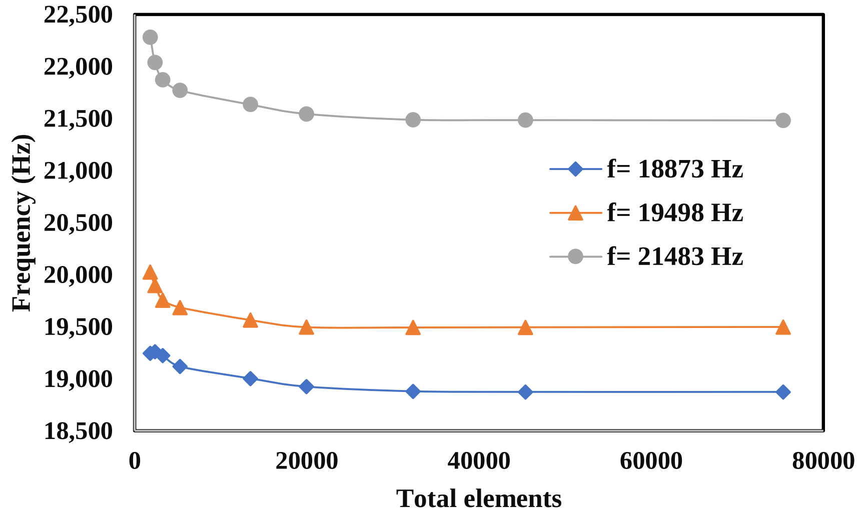


Figure 3. Variation in the results of modal analysis with the number of nodes

Based on characteristics of the waves generated by the ultrasonic system, the modal analysis is conducted in the frequency range 15-25 kHz to find all mode shapes of the horn near the working frequency of ultrasonic generator. It is noted that the general vibrating configuration of horn is comprised of radial and longitudinal mode shapes. In the radial mode the oscillations of material particles around their equilibrium position are directed along with the propagation of generated waves, while in the longitudinal mode the vibration of horn is perpendicular to the direction of ultrasonic waves. As a result, just the radial vibrating movement of horn helps the drying process inside the drying chamber. On the other hand, by diminishing the amplitude of longitudinal oscillations, the risk of failure in the horn, transducer and joints significantly reduces.

Figure 4 illustrates the total, radial and longitudinal mode shapes of the horn at the resonance frequency of 19,498 Hz. It is noted from Figures 4b and 4c that the ratio of undesirable longitudinal amplitude to the radial amplitude in this frequency is 9.97 %. It should be noticed that the horn has also several other natural frequencies in the range of exciting frequencies produced by the ultrasonic system. The close-range distance between the natural frequencies may lead to interaction radial and longitudinal of mode shapes and inflict harmful effects on the safety of system and should be avoided. According to the results of modal analysis, the two other natural frequencies in the vicinity of main resonance frequency (i.e., 19,498 Hz) are 18,873 and 21,483 Hz. Fortunately, due to the proper design of horn there is a large gap between these two frequencies and the main resonance value (19,498 Hz). This greatly reduces the risk of frequency shift, interaction of modes and unsafe performance of ultrasonic system during the working hours.

The piezoelectric transducer can produce forced vibrations at various frequencies. Therefore, it is necessary to investigate various mode shapes for adjacent natural frequencies. The radial and longitudinal modes of vibrations for the natural frequencies 18,873 and 21,483 Hz are depicted in Figures 5 and 6, respectively. According to Figure 5, at the frequency 18,873 Hz the ratio of longitudinal amplitude to the radial amplitude is found to be 53.17 %, while at the ratio increases to 55.54 % at frequency 21,483 Hz. In these cases, the undesirable longitudinal oscillations have an intense effect on the total mode shape of vibrations. So, the components of system, including the links and junctions endure considerable extra cyclic stresses during the working hours.

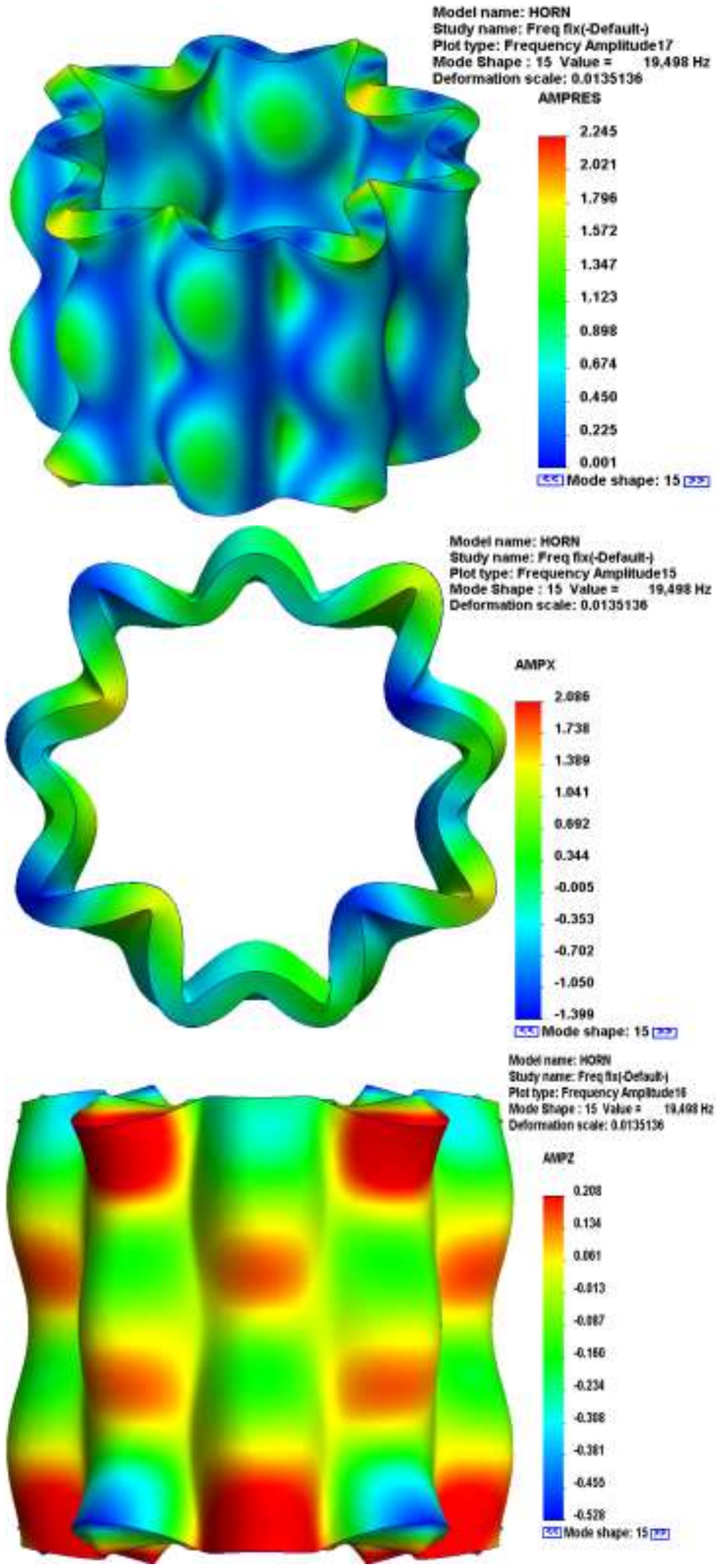


Figure 4. The mode shapes of the horn at the frequency of 19,498 Hz: a) total mode shape, b) radial mode shape and c) the longitudinal mode shape

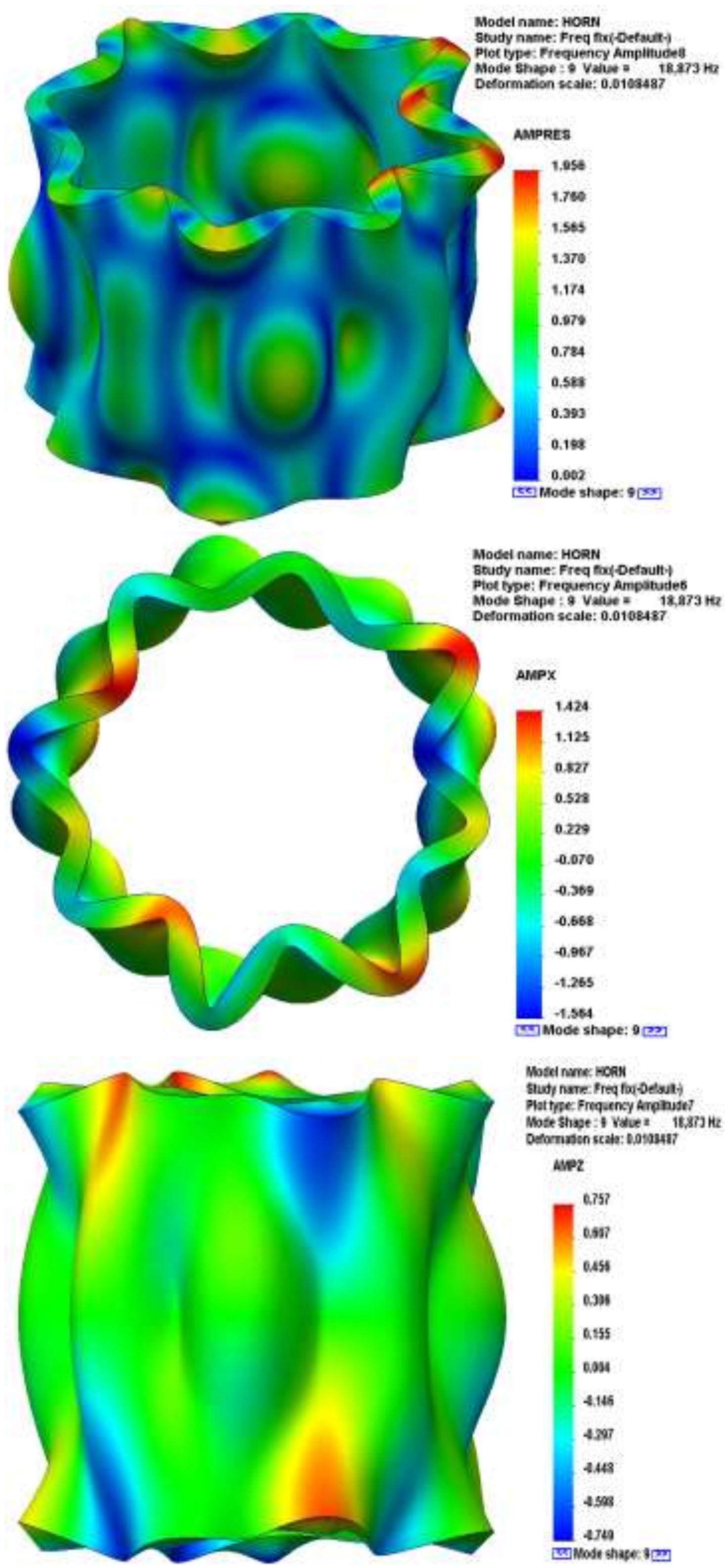


Figure 5. The mode shapes of the horn at the frequency of 18,873 Hz a) total mode shape, b) radial mode shape and c) the longitudinal mode shape

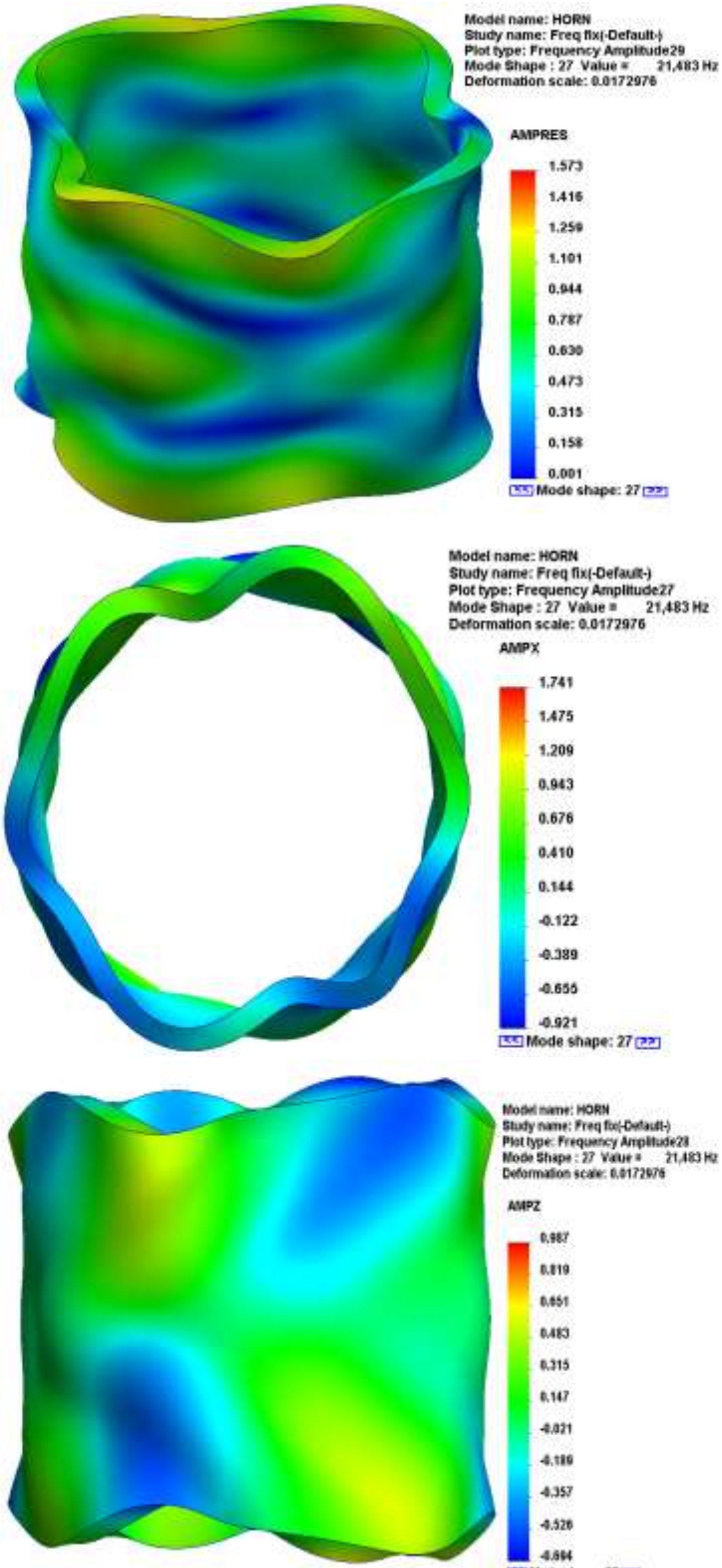


Figure 6. The mode shapes of the horn at the frequency of 21,483 Hz a) total mode shape, b) radial mode shape and c) the longitudinal mode shape

As mentioned earlier, the resonance frequency of the horn is found to be 19,498 Hz which has a sight deviation from the nominal frequency of generated waves by the system (20,000 Hz). In order to achieve a more accurate tuning the natural frequency on the exciting frequency of ultrasonic system, a further adjustment is required for the dimensions of horn. In this regard, the thickness of horn is reduced to 9.85 mm and the modal analysis is repeated for the new configuration. It is noted that the obtained natural frequency for this case increases to 20,077 Hz and the corresponding total mode shape takes a more desired shape (see Figure 7). So, the proposed modification provides more improvements in the dynamic characteristics of the design, and the horn with such dimensional aspects is investigated in the next section by the harmonic analysis.

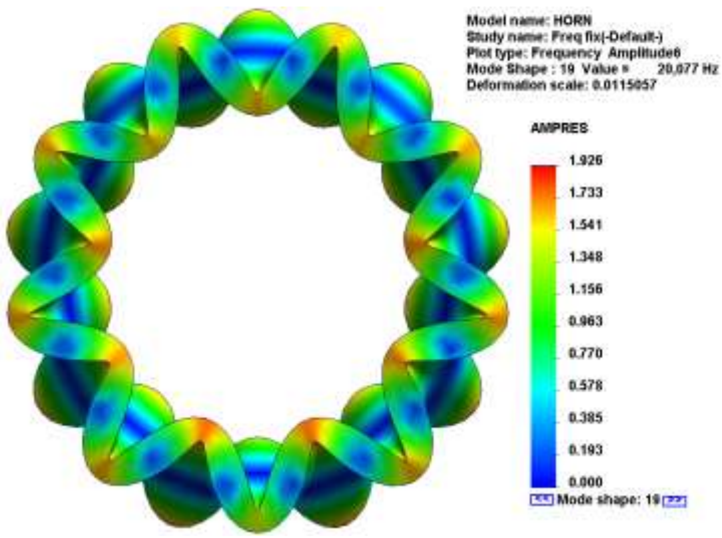


Figure 7. The modal analysis of acoustic horn with modified dimensional design

The harmonic analysis of horn

During the full load operation of the ultrasonic generator, the transducer converts the electrical power to the mechanical waves with the amplitude of $0.9\mu m$. However, in airborne radiation of ultrasonic waves significant part of supplied power is absorbed by the gaseous medium and it is necessary to amplify the acoustic radiations for providing more contribution to the drying applications. For this purpose, the horn which is directly fastened to the transducer, should increase the amplitude of generated waves. By applying proper boundary conditions for the problem, the harmonic analysis is conducted for evaluating the displacement of horn caused by the external excitations. The external exciting load in this study is assumed to be harmonic with span of frequencies varying between 18,000 to 21,000 Hz. This domain is divided into 20 steps and the values are used for finding the amplitude of forced vibration of horn. In this way, the frequency with highest amplitude can be readily determined. The results of harmonic analysis for various exciting frequencies are shown in Figure 8. It can be observed that the maximum amplitude occurs at 19,452 Hz which is very close to the natural frequency found from the modal analysis.

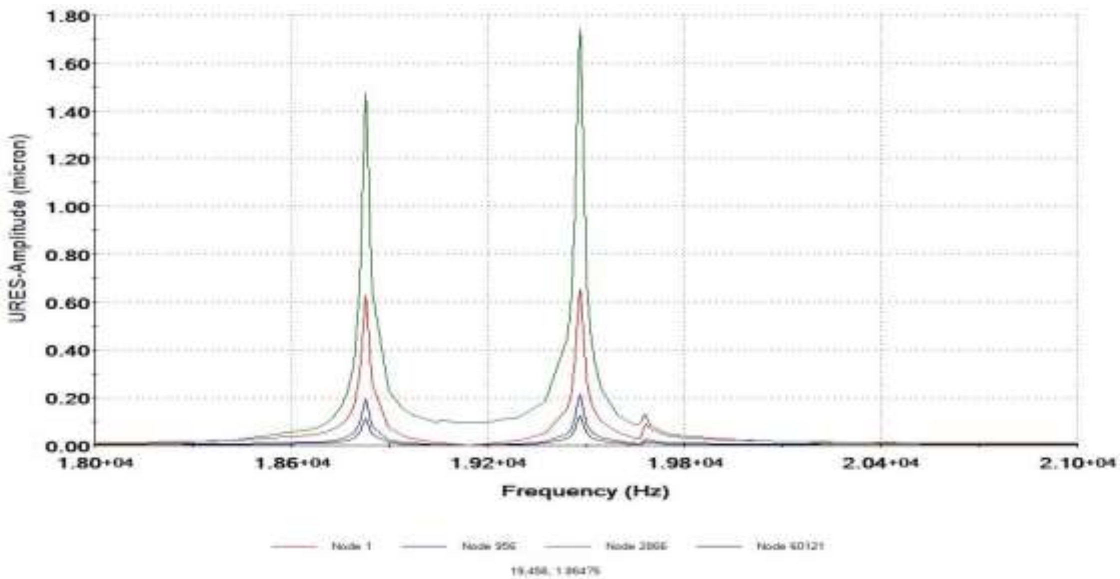


Figure 8. The response of acoustic horn for various frequencies of applied harmonic loads.

For the exciting frequency of 19,452 Hz, the contours of vibrations induced in the horn are illustrated in Figure 9. It is noted from Figure 9b that the maximum amplitude of radial oscillations is $1.642\mu m$ which happens at the upper and lower edges of the horn. The amplitude of vibrations in the longitudinal direction is shown in Figure 9c. As can be seen, the magnitude of non-radial displacements is not sizeable in comparison with the primary radial movements. This is an indicative of well dimensional design considered for the horn. So, the horn propagates the ultrasonic waves in the desired direction, and the harmful dynamic effects on the components of system, including the horn itself, are effectively avoided. The variation in the amplitude of vibration for several check points selected along the height of horn, is depicted in Figure 10. It is evident that the amplitude of oscillations at the middle height of horn near to the region where the transducer is attached, is much lower than the maximum value (i.e., $1.727\mu m$ in Figure 9a).

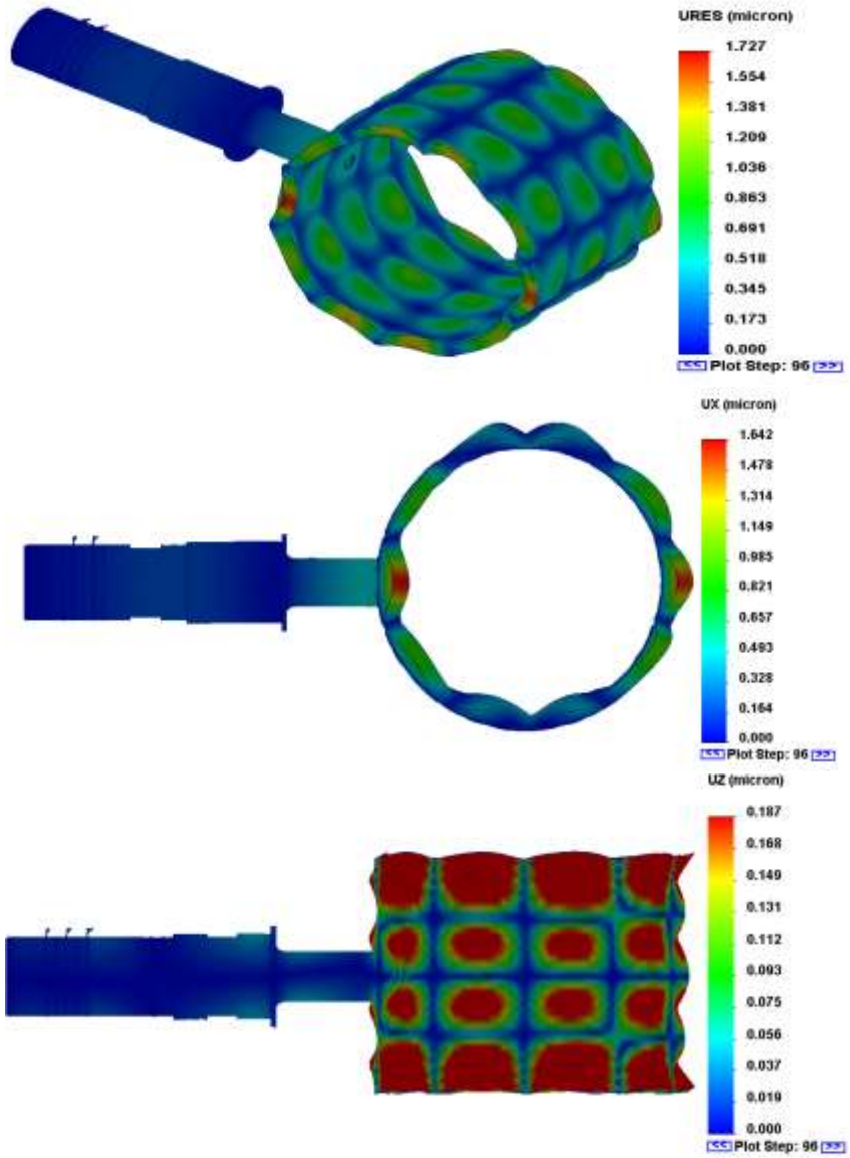


Figure 9. The contour of displacements induced inside the horn at the frequency of 19,452 Hz.

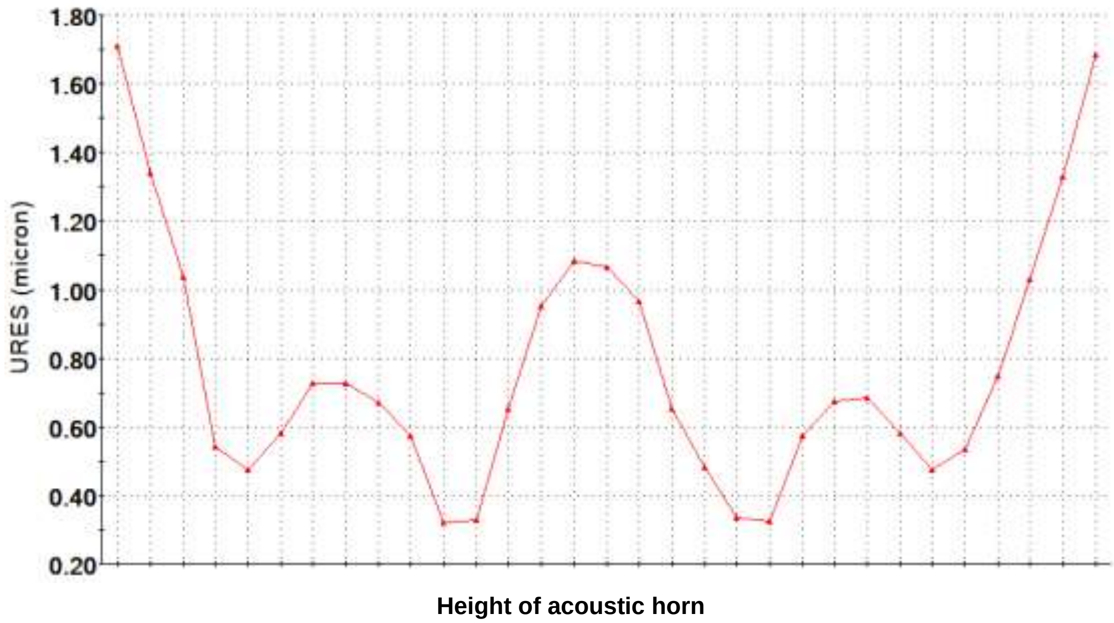


Figure 10. Variation of displacement amplitude along the height of excited acoustic horn.

It is noted that when an acoustic horn operates at high power and for long duration, some cyclic stresses are created inside its domain which may lead to the fatigue failure of the body. So, other than the exact tuning of natural frequencies and mode shapes, achieving a proper stress-withstanding capacity is a main concern for transferring the ultrasonic power to the surrounding gas medium. According to the experimental observations, the endurance limit of aluminum is approximately 30 MPa, and for the dynamic stresses below this value, the loads may be applied without facing the danger of fatigue failure (Naď, 2010). The evaluated values of von Mises stresses for the exciting frequency of 19,452 Hz are shown in Figure 11. As can be seen, the maximum stress is 4.437 MPa which occurs at the connecting point of transducer to the horn. Fortunately, the extreme value of stress is less than the endurance limit and it is concluded that the horn can safely transfer high ultrasonic powers in full loading condition.

Validating the results of FEM analysis

In this section, we aim to evaluate the validity of the simulated results obtained for the dynamic oscillations of the acoustic horn. To this end, an experimental setup (see Figure 12) is prepared and a laser vibrometer (Polytec OFV-505) is employed for measuring the vibrational movements of the horn. The laser vibrometer is focused on the middle height of the horn opposite to the point where the transducer is attached to the horn, and then collects the data. The recorded data is analyzed in MATLAB and then visualized by LabView software. For a frequency rage 0-100 kHz and delivering 100 % ultrasonic power to the horn, the results for amplitude are evaluated and given in Figure 13. It is noted that the amplitude has an abrupt jump at a frequency near to 19,800 Hz. This is the primary natural frequency of the system found from the experimental analysis. Interestingly, this value is very close to the results of FEM model and is a clear fact for demonstrating the reliability and robustness of conducted numerical study presented for dynamic analysis of acoustic horns

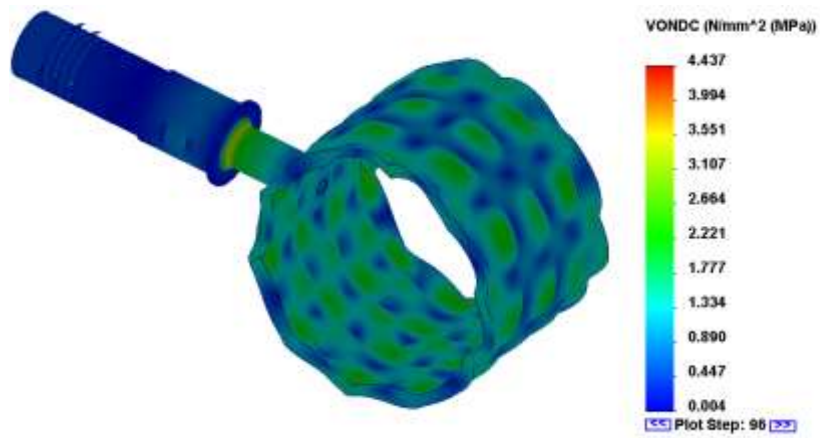


Figure 11. The von Mises stresses induced inside the horn at the frequency of 19,846 Hz.

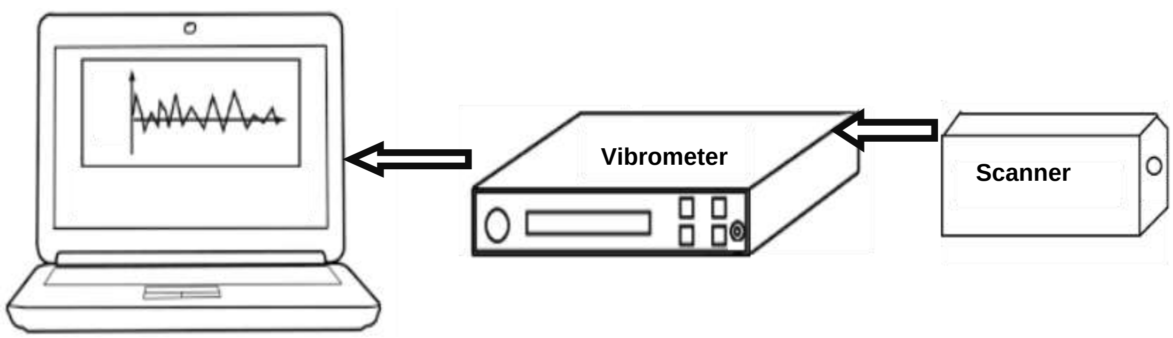


Figure 12. The experimental setup for evaluating the dynamic characteristics of the system

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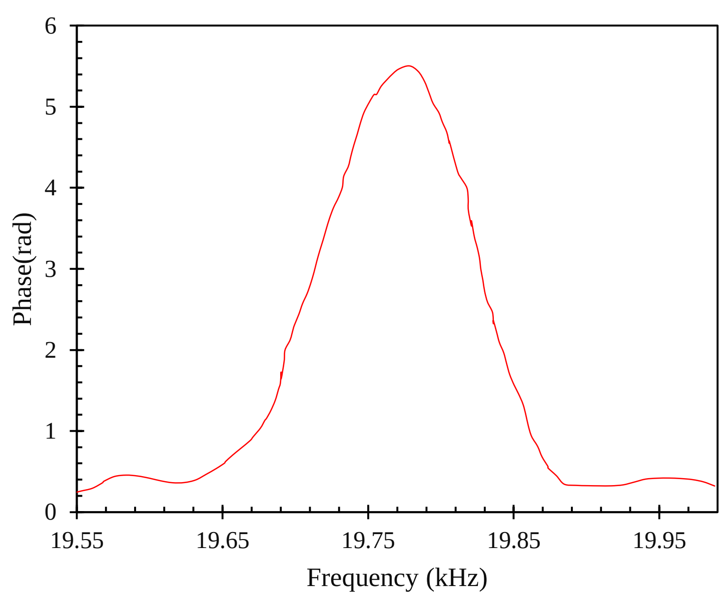

Figure 13. Variation of amplitude with the frequency of excitations obtained from the experimental analysis.

CONCLUSIONS

An airborne ultrasonic dryer with a cylindrical horn is considered in this study. Using the Solidworks FEM-software a numerical model is developed for the analysis of dynamic behavior of the horn subjected to the ultrasonic excitation. Several mode shapes in the working frequency of the system are found by the modal analysis and it is found that due to the proper design of horn, there is a large gap between the natural frequencies. This reduces the risk of harmful interaction of modes during the working hours of the system. The harmonic analysis of horn evidently shows that the horn can conveniently intensify the amplitude of vibrations generated by the piezoelectric transducer in the desired radial direction and produces negligible longitudinal oscillations. It is also noted that the magnitude of cyclic stresses, computed from the dynamic investigation, is much lower than the endurance limit of the constructing material and the horn can safely radiate the ultrasonic waves on the drying samples. In an experimental analysis, the main resonance frequency of the horn is measured and it was found that the FEM model can accurately predict the dynamic behavior of the system.

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