AN ANALYSIS OF THE DYNAMIC BEHAVIOR AND ACCELERATION SENSITIVITY OF A SAW RESONATOR SUPPORTED BY FLEXIBLE BEAMS

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Abstract: A numerical analysis is performed of the normal acceleration sensitivity and the low frequency dynamic vibration response of a SAW resonator supported by flexible beams of various geometries. The acceleration sensitivity problem is solved using Tiersten's perturbation theory for small fields superposed on a static bias in combination with a very accurate solution to the general two dimensional surface wave equations of Sinha and Tiersten. The static biasing states under normal acceleration are obtained using three dimensional finite element models which are generated and solved using recently developed FEA software. The frequency shift under normal acceleration is computed using a rigorous and precise integration algorithm which allows an analytical surface wave mode shape to be combined with a discrete finite element solution to obtain the first perturbation of the eigen-value. A modal analysis of the SAW substrate and support beam assembly is also performed using the FEA software.

Introduction

The normal acceleration sensitivity and low frequency dynamic behavior of a surface acoustic wave resonator supported by flexible beams is numerically analyzed using recently developed semi-analytical finite element tools [7]. The general perturbation theory of Tiersten [1,2] is used to calculate the frequency shift under normal acceleration when the biasing state is solved using the finite element method and the surface acoustic wave mode shape is solved analytically using the two dimensional theory of Sinha and Tiersten [3,4,5]. The theoretical development of the finite element solution method and the integration algorithm are given in reference [7] and are not listed here. The modeling software is used to study problems of three different geometries and it is shown that each of these configurations leads to different observed behavior.

Numerical Results

The first example considered is that of a rectangular ST-Cut quartz plate with four symmetrically placed beam supports, also ST-Cut quartz, along the edges parallel to the $x_1$ axis. A 360 MHz SAW resonator of the dimensions and characteristics indicated in reference [6] is placed in the center of the plate and the entire assembly is subjected to a unit normal acceleration $(a_2 = 1g, a_1 = a_3 = 0)$. Figure 1 shows the general geometry of the structure with the basic design variables labeled. Figure 2 shows a plotting of the finite element model used for this study with numerical values for the fixed design variables given. The calculations are carried out for various beam lengths and substrate thicknesses to obtain a detailed picture of the static acceleration performance of this particular structure. Since the SAW mode resides primarily in the center of the substrate, it is appropriate to assume that the deformations in this area under gravity induced bending are the important quantities to observe and manipulate during the design process. Figure 3 shows a three dimensional distortion plot of the structure under normal acceleration and shown in Figures 4 and 5 are curves which represent the deflection of the middle plane of the plate along the $x_1$ and $x_3$ axes, respectively. The deflection plots of Figures 4 and 5 are shown for a plate of thickness 0.508 mm with beam lengths of 2, 4, and 6 mm, and the “slices” are taken along the coordinate axes, which conveniently coincide with the center-lines of the plate. In Figure 5, it is observed that for each beam length the curvature in the center of the plate increases slightly, while in Figure 4 it is seen that as the beam length is increased, the curvature at the middle point decreases, tending to flatten along this direction. This suggests a shift in the distribution of biasing stresses and strains which can be controlled by the lengths of the beams. It is observed for this particular case that an apparent zero acceleration sensitivity is achieved at a beam length of approximately 3.5 mm. Figure 6 shows the
computed acceleration sensitivity for this structure for plate thicknesses varying from 0.254 mm to 1.016 mm and beam lengths varying from 1 mm to 8 mm. It is seen that for each thickness an optimal point exists where the normal acceleration sensitivity vanishes.

The second example considers a rectangular plate supported at its corners by flexible beams. Figure 7 shows the general geometrical layout of the structure and Figure 8 shows the model used for the calculations with pertinent dimensions listed. As before, ST-Cut quartz is used throughout and the same 360 MHz SAW resonator [6] is assumed. Figure 9 shows the general three dimensional deformation of a typical structure and Figures 10 and 11 show the deflections of the middle plane along the coordinate axes. These deflection plots are shown for a beam angle of $\theta = 45^\circ$ and for planar aspect ratios of $L_p/W_p = 0.5$, $L_p/W_p = 0.7$, and $L_p/W_p = 1.0$. Once again it is observed that while the $x_3$ flexure characteristics are somewhat constant, the $x_1$ behavior exhibits some variation for the different dimensions. As shown in figure 11, the curvature of the plate along $x_1$ is almost flat at $L_p/W_p = 0.5$ and increases with increasing $L_p/W_p$. It is observed that for this structure a point of aspect ratio compensation occurs at approximately $L_p/W_p = 0.56$. Figure 12 shows the results of normal acceleration sensitivity calculations for beam angles ranging from $20^\circ$ to $70^\circ$ and for planar aspect ratios from 0.3 to 1.8. It is observed that for each beam angle, there is one point of aspect ratio compensation. It is also observed that away from these optimal points, the performance of this structure degrades rapidly with nominal values in parts in $10^8$ and below.

The third example considered is that of a rectangular ST-Cut quartz plate supported by three beams along each of two edges parallel to the $x_1$ axes. The same 360 MHz resonator [6] is again used and is centered with respect to the plate’s major edges. Figure 13 shows the general description of the geometry and Figure 14 shows the details of the model used. The general three dimensional deformation of this structure is shown in Figure 15 and the deflections along the center-lines of the plate are shown in Figures 16 and 17. In this study, the beam separation distance is varied along with the planar aspect ratio. Figure 18 shows the normal acceleration sensitivity versus the beam separation distance for planar aspect ratios of $L_p/W_p = 0.5$, $L_p/W_p = 1.0$, and $L_p/W_p = 1.5$. It is interesting to note that in this case, no optimal point is noticed and the performance is generally poor with nominal values in parts in $10^8$ to $10^7$.

The final example explores the dynamic behavior of the plate-beam structure. Central to the design of the types of structures considered here is the calculation of the fundamental resonance of the overall device. This package resonance defines the limiting dynamic range that the system may operate in. To study the dynamic behavior of these plate structures and package assemblies, an efficient eigen-value solver has been incorporated into the finite element code. Using this eigen-solver, a modal analysis has been carried out for the three-beam model for a planar aspect ratio of 1.0. Figure 19 shows the first three vibration modes of the 3-beam supported structure, and Figure 20 shows a parametric study of beam length versus plate thickness for the fundamental resonance. It has been found that these resonant frequencies are most sensitive to these parameters. Equivalent studies on the other geometries yield similar results. It is observed that for the most desirable beam lengths, i.e. 3.0 mm to 5.0 mm, the fundamental resonant frequency is reduced by approximately one half, generally reaching a value of about 8 KHz to 10 KHz.

Conclusion

The numerical experiments carried out on the beam supported SAW resonator structures reveal some very interesting results. From the first two examples it can be concluded that by varying the geometry and the flexibility of the larger supporting structure it is possible to achieve a theoretical zero normal acceleration sensitivity. This is achieved by proportioning critical dimensions to manipulate the biasing stresses and strains. It can also be concluded that when such precise dimensioning can not be achieved, the performance will generally be poor. The results obtained for the optimal values are dimensionally realistic, as all calculations were carried out for standard wafer thicknesses and the dimensions used are in proportion with typical SAW resonator dimensions. The dynamic performance of these structures appears to be compatible with many consumer applications and some military applications, where higher frequency noise is not present.

References


Figure 1. Geometrical layout for four beam/edge configuration

Figure 2. Finite element mesh and dimensions for the four beam/edge configuration.

Figure 3. Three dimensional deformation plot of the four beam/edge plate under normal acceleration.

Figure 4. Flexural deformation of the middle plane along the x1 direction for four beam/edge configuration.
Figure 5. Flexural deformation of the middle plane along the x3 direction for four beam/edge configuration.

Figure 6. Normal Acceleration sensitivity of four beam/edge configuration as function of plate thickness and support beam length.

Figure 7. Geometrical layout for corner supported configuration

Figure 8. Finite element mesh and dimensions for the corner supported configuration.

Figure 9. Three dimensional deformation plot of the corner supported plate under normal acceleration.

Figure 10. Flexural deformation of the middle plane along the x1 direction for the corner supported configuration.
Figure 11. Flexural deformation of the middle plane along the x3 direction for the corner supported configuration.

Figure 12. Geometrical layout for three beam/edge configuration

Figure 13. Geometrical layout for three beam/edge configuration

Figure 14. Finite element mesh and dimensions for the three beam/edge configuration.

Figure 15. Three dimensional deformation for the three beam/edge configuration under normal acceleration.

Figure 16. Flexural deformation of the middle plane along the x1 direction for three beam/edge configuration
Figure 17. Flexural deformation of the middle plane along the $x_3$ direction for four beam/edge configuration.

Figure 18.

Figure 19. First three modes of the 3-Beam configuration.
Figure 20. Parametric study of the fundamental resonance of the 3-Beam plate as a function of beam length and thickness.